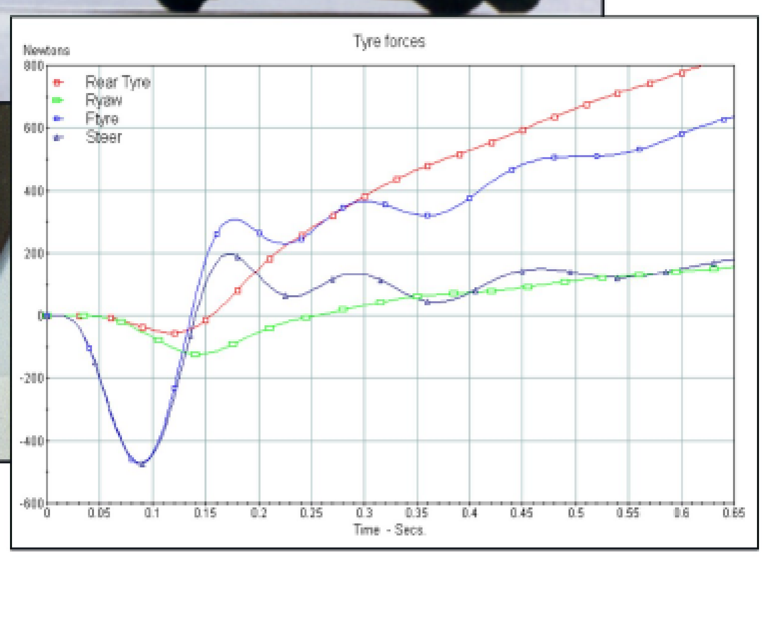
# MOTORCYCLE HANDLING AND CHASSIS DESIGN

## the art and science




Tony Foale

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**WARNING – Important safety and legal notice.**

Building and/or modifying motorcycle chassis or parts is a serious undertaking and the consequences of mechanical failure can result in serious injury. Some of the practices and designs outlined in this book may be dangerous unless implemented by skilled and experienced people who have qualifications in engineering, design and fabrication. The author has tried to ensure the accuracy of the contents of this book, but some text, of necessity, represents the opinion and experience of the author and is not guaranteed to be fact. Therefore, the book should be read whilst mindful of the potential risks of motorcycle modification, both in racing and other use.

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# Foreword

The motorcycle is a complex system that has long defied full analysis. For a very long time, motorcycle handing was hardly even considered a subject. Engines, whose performance could be measured in "objective" terms, therefore received the lion's share of development. Engine development moved rapidly ahead of chassis, suspension, and tires, creating a succession of design crises that required new thought for their solution. Examples might be Rex McCandless's twin-loop swingarm chassis of 1950, Tony Mills's wide, belted Dunlop Daytona tire of 1974, and the present-day elaborations of Antonio Cobas's large-section aluminium twin-beam chassis of the early 1980s. In each case, motorcycle performance had ceased to advance because of specific problems that could not be solved by traditional means.

In general, the innovations that have broken these deadlocks have been creations of practical persons, not of theorists. The role of theory in motorcycle design has, if anything, suffered at the hands of history, for the strange forkless creations of ELF, Fior, and Bimota have come and gone without solving any actual problem.

Yet motorcycle performance is at present again deadlocked, with no sunny uplands of easy progress in sight. As motorcycles lean over farther on their wonderful tires, their suspensions turn sideways, at a large angle to the bumps they are designed to absorb. As engine and brake torque is applied, motorcycles short enough to turn quickly, and tall enough for adequate cornering clearance suddenly lift the front or rear wheel, limiting maximum rates of acceleration and deceleration. While autos present 100% of the width of their tires to the pavement, the motorcycle offers only 1/3 of tread width at a time, severely limiting cornering grip. To make motorcycles steer well, front tires must be of modest section, while rears, to apply engine power, must be large. With the forward CG position necessary for rapid acceleration, a powerful motorcycle must therefore overload its small front tire in cornering, while under-using its larger rear. The result is that as a machine's power increases, its corner speed must decrease.

Racing is the environment in which these problems hurt worst, and from which solutions have most often come. Racing has, however, evolved from a sport into a conservative business. The practical men of racing are now too busy loading and unloading their beautifully painted transport trucks to have much time for innovation. The theoreticians remain, as ever, divorced from practicality, often ignorant of the real problems motorcycles confront.

Yet the infinite refinement of the piston internal combustion engine did not create the gas turbine - only a careful consideration of theoretical heat engine cycles could make that leap. Therefore the practical and theoretical sides need each other - but they have had little dialogue thus far.

This book is a valuable step toward that dialog. Tony Foale's first book was almost entirely practical, and has been deservedly widely read. He is a man who can control a weld puddle and twist safety wire. He also knows that refinement within existing thought must ultimately reach a dead end. This has forced him to learn to walk with one foot upon practicalities and the other upon theory. This new book is the result. Read on.

***Kevin Cameron**.* Technical editor Cycle World magazine. March 2002.

# Acknowledgements

It is quite usual that the author of a book has various people to thank for providing help in its preparation. In my case I have hundreds to thank. Prior to publication in book form, preview versions of the manuscript were made available on CDROM, in different stages of completion. In total about 250 CDs were distributed, and the feedback and notification of errors from a sizable proportion of those readers has proved to be an invaluable aid.

The numbers make it impossible to name everyone, but you know who you are – thanks a lot, you made the job much easier.

Another source of aid came from those who have supplied information, expert proof reading or contributed ideas for topics without which this book would have been the poorer. This group is small enough to thank individually and it gives me pleasure to be able to so. They are, in alphabetical order:

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# Preface

The book “Mototorcycle Chassis Design” was first published in 1984 and was subsequently reprinted several times without under-going change. Although out of print for over 12 years or so, I know from personal inquiries that there is still considerable demand for a book on this subject. A new book was obviously well overdue, although much of the original material is as current today as it always has been. After all, the laws of Newtonian Physics tend to be stable over time.

During the nearly two decades since the original book, the motorcycle chassis has undergone gradual evolutionary change and there is no doubt that handling in general has improved. In the 1970s. the main emphasis was on ever more powerful engines being fitted into flexible tubular frames unable to provide a reasonable level of handling or stability. Thankfully that has generally changed. Forks, frames and swing-arms have become much more rigid, and in some cases lighter as well, at least at the sport bike end of the market. The change to radial ply tyres has been of the utmost importance to this process of change. Despite the prophecies of many commentators the front suspension of choice is still the telescopic fork, although generally much improved. For any number of reasons manufacturers have been reluctant to experiment with other forms in the marketplace. This probably has more to do with the product liability lawyers than it has to do with the engineers. There have been two notable exceptions amongst the major manufacturers. Although now out of production, Yamaha marketed the GTS with a suspension design based on the work of James Parker. BMW changed over completely to the “Telelever” system, similar in principle to the design used by the British Saxon concern.

I’ve had considerable feedback from readers of that first book and I’ve done my best to incorporate the many suggestions. Although greatly enlarged, most of the original subject matter remains. Many topics have under gone revision to improve clarity or remove ambiguity. Material has be added which explores in more depth those subjects which were only briefly mentioned in the original book, mostly due to publishing space constraints. An example of this is the description of initiating a turn, this topic is central to an understanding of motorcycle behaviour. However, it was then covered only briefly, the content on this subject is considerably enhanced in the current book. Completely new chapters have been added on various topics that just weren’t in the original. For example: tyres, aerodynamics, the important subject of anti-squat and a case study of improving a standard production frame for racing.

Since the first book was published the sport of motorcycling in all forms has become much more technical and so in order to do the subject justice this book has had to become more technical also. Reviewers of the previous book praised the lack of drawn out explanations, I have tried to maintain this characteristic where possible, but within the need for coverage in greater depth. This book is not intended as a handbook for chassis setup etc. rather it is an attempt to provide the reader with the background knowledge of how and why motorcycles react in the way that they do. An understanding at this level will however, equip the reader to undertake his own design, modifications or setup with greater confidence. The acquisition of knowledge is rarely easy and requires commitment, any book is purely a passive aid and the benefit to each reader will depend on the effort put into it. It is probably best to initially read it through quickly , ignoring some of the detail to get an overall view and then to re-read it to gain a more in-depth appreciation of the subject. It is also recommended that the reader looks at some of the appendices for background information, prior to tackling the main text. In particular appendices 2,3 and 4.

There are a wide range of technical topics discussed within a relatively small book and so in some cases a prior knowledge of the basics has had to be taken for granted. Naturally some parts of the general text are more technical than others, but there should be little problem for any interested enthusiast in gaining a better understanding of the principles involved. It is not necessary to understand every last detail to derive benefit.

To cover the subject adequately it is impossible to completely avoid mathematics, I have tried to keep this as simple as possible. The level of mathematics used is deliberately kept at a level below that requiring a knowledge of calculus, in the hope that the book will be of use to the widest range of readers. A multitude of diagrams and graphs from both data logging and computer simulation have been used to demonstrate various phenomenon without a great number of formulae.

Even in this age of much greater technical understanding, there are still many aspects of design and handling setup that can better be described as art more than science. Hence, the book title has been changed to reflect this. All engineering design is the art of compromise, the best bike is the one whose designer has achieved the best overall compromise for the intended purpose, whether that be racing or commuting. We often hear that competition machines are built with no compromises, in fact the opposite is true. Highly focused machines such as racers are probably subject to the biggest compromises of all. Throughout the book I have tried to emphasize the conflicting requirements that always compromise any design or setup decision. Nowhere is this more evident than when selecting suspension characteristics, this is demonstrated at every race meeting where much time is spent making minute adjustments to achieve the “optimum” setup.

Many points in the text are illustrated with example photographs. It has been a policy to use older examples where possible to acquaint younger readers with some of these machines and also to demonstrate that much of what is regarded as being new has in fact been around for a considerable period. Most readers will in any case be familiar with photos of modern examples from the general motorcycle press.

The first book was co-authored by Vic Willoughby, undeniably the doyen of motorcycle technical journalists. When I was a teenager (many years ago) I would read his weekly articles many times over and there’s no doubt that these played a great part in the motivation for me to start designing and making my own chassis. Many years later I was privileged enough for him to write articles describing some of my work. We became friends and I was honoured when he agreed to help when I approached him with the idea for the original book. He was in his retirement then but still had enormous energy for the task. Unfortunately, Vic passed away in November 2000 and I have undertaken this new book solo, and so must take sole responsibility for any errors.

I would however, like to dedicate this book to Vic. without whom the original would never have passed the idea stage.

***Tony Foale**,* Spain

March 2002

**Dedicated to the memory of Vic Willoughby.**

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# 1 Function and history

## Some basic definitions

Before getting into much detail we need to consider some definitions of terms that are often banded about loosely and misunderstood as a consequence.

### Handling

By this, we mean the ease, style and feel with which the motorcycle does our bidding. It depends mainly on overall geometry, chassis stiffness, weight and its distribution, tyre type and size. It may come as a surprise to some people to learn that the rider has a major influence on the handling characteristics of a motorcycle. Rider responses have a large effect on the overall interaction of the dynamic forces that control the motion of the machine.

### Roadholding

This means the ability of the machine, through its tyres, to maintain contact with the road. It depends mainly on tyre type and size, suspension characteristics, weight and its distribution, and stiffness between the wheels to maintain their correct relationship to one another. In the days of relatively narrow tyres, roadholding and handling generally went hand-in-hand, indeed, the terms were used interchangeably. However, nowadays the requirements are sometimes contradictory and a compromise must be struck, depending on the intended use of the machine.

A big enemy of tyre grip and hence roadholding is dynamic variation in the vertical load at the road interface, there are many factors that contribute to such variation and we shall see that suspension parameters are important as a means of providing control over this aspect.

### Stability

There are many types of stability or instability that can influence a motorcycle. There’s balance stability, aerodynamic stability etc. Formal definitions of stability in control systems exist but they are too involved for a book of this nature, although we’ll look at these aspects a bit closer in a later chapter. For our present purposes we mean:

The ability to maintain the intended manoeuvre (i.e. continue in a straight line or round a corner) without an inherent tendency to deviate from our chosen path. This implicitly includes the absence of wobbles and weaves.

The ability to revert to the intended manoeuvre when temporarily disturbed by external forces (e.g. bumps, cross winds and so on).

Handling, roadholding and stability are affected by many parameters and the interaction between them. The subject is complex but not magic, and – judging from some chassis designs – has not always been well understood. However, relatively simple laws of physics are always obeyed. This book will try to remove the mysteries and consider the main parameters involved and study their various effects. It must be emphasized that there is much cross-coupling between these effects – there is no ’correct’ combination, no ’perfect’ design. Any motorcycle embodies several essential compromises.

## Linear and angular motions

If we are to study the behaviour of any type of vehicle we first need to consider just how it can move. The linear motions are easy to visualize, firstly the machine can move in a forward direction and the engine and brakes are responsible for controlling this. Road undulations and hills cause motion in a vertical direction and sidewinds can result in sideways movement. It is the angular motions that are somewhat less familiar to most people. The overall angular movements can be completely described by considering the motions about three separate axis. These axis are at right angles to one another and are known as roll, pitch and yaw.

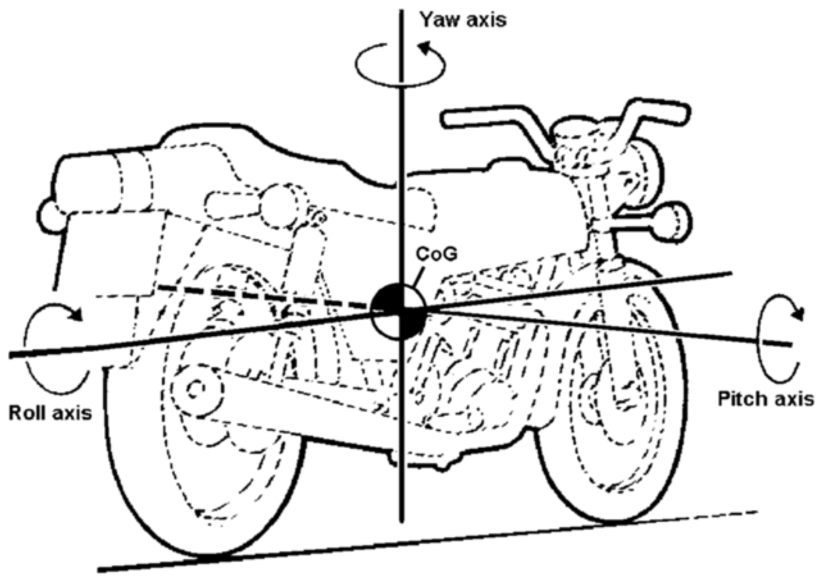



Fig. 1.1 **Showing the three principal axis of rotation.**

**Yaw is the angular motion about a vertical axis.**

**The pitch axis is horizontal and passes sideways through the bike.**

**The roll axis is also horizontal and is orientated fore and aft.**

Roll is probably the most familiar of the three and is the most obvious motion that occurs when we lean the bike over for cornering. Fig. 1.1 shows the roll axis passing through the CoG. However, as we shall see later the location of this axis depends on our frame of reference.

Yaw is the movement about a vertical axis and occurs as we steer around a bend, it can also be caused by various disturbances such as sidewinds.

Pitch is the motion about an horizontal axis that passes sideways through the machine, we get this under braking and acceleration, as well as from road irregularities.

Due to the large roll angles involved with cornering, the pitch and yaw axis of the machine move relative to the global vertical and horizontal coordinates. For this reason it is important to be careful when specifying the axis system that we are using. There are several such systems that are used in vehicle analysis but for our purposes the two most important will be the machine coordinates and earth coordinates, initially defined in terms of the original direction of travel, before performing some manoeuvre.

# Function

The functions of a motorcycle frame are of two basic types: static and dynamic. In the static sense the frame has to support the weight of the rider or riders, the engine and transmission, and the necessary accessories such as fuel and oil tanks. Although less obvious, the frame's dynamic function is critically important. In conjunction with the rest of the rolling chassis (i.e. suspension and wheels) it must provide precise steering, good roadholding, handling and comfort.

For precise steering the frame must resist twisting and bending sufficiently to keep the wheels in their proper relationship to one another regardless of the considerable loads imposed by power transmission, bumps, cornering and braking. By proper relationship we mean that the steering axis must remain in the same plane as the rear wheel, so as to maintain the designed steering geometry in all conditions without interference from frame distortion.

Clearly, however, no steering system can be effective while the wheels are airborne, hence the importance of good suspension, especially at the front. Good handling implies that little physical effort should be required for the machine to do our bidding, so avoiding rider fatigue. (This requirement is largely a function of centre-of-gravity height, overall weight, stiffness, steering geometry, tyre sizes and the moments of inertia of the wheels and overall machine/rider combination.)

Comfort is important to minimize rider fatigue also, and requires the suspension to absorb bumps without jarring the rider or setting up a pitching motion. All these criteria the frame has to fulfil for the expected life of the machine, without deterioration or failure and without the need for undue maintenance.

It should be borne in mind, however, that all design is a compromise. In any particular example, the precise nature of the compromise will be governed by the use for which the machine is intended, the materials available and the price the customer is prepared to pay.

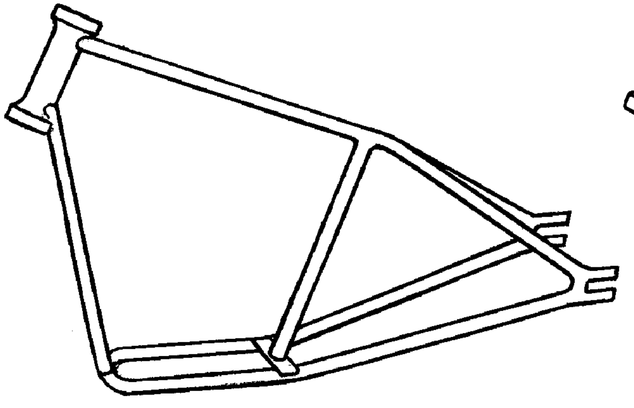
# History

## Tubular frames

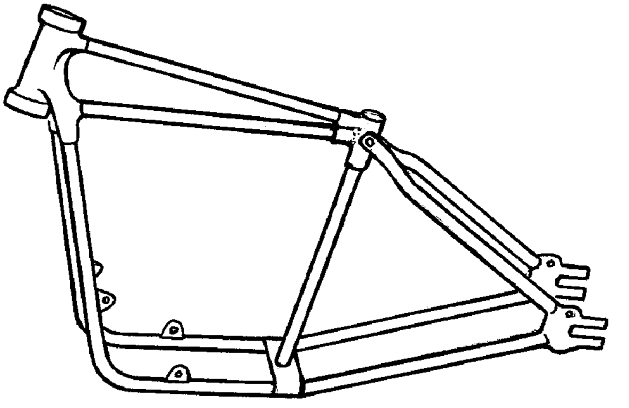
Over the years designers have been repeatedly criticized for their seeming reluctance to depart from the diamond-pattern frame inherited from the pedal cycle. However, since the earliest motorcycles were virtually pushbikes with small low-powered engines attached at various places, that was the logical frame type to adopt, particularly so long as pedal assistance was required.

Until the general adoption of rear springing several decades later, this diamond ancestry (including its brazed-lug construction) was discernible in most frame designs. This was hardly surprising as the frame's depth suited the tall single-cylinder engines that were popular for so long. In any case the motorcycle, like the pedal cycle, was after all a single-track vehicle in which the use of an inclined steering head was a convenient way to provide the front-wheel trail necessary for automatic straight-line stability.

Once pedals were discarded, the frame with the closest resemblance to the ancestral pedal type was the simple diamond pattern in which the engine's crankcase replaced the cycle's bottom bracket to span the lower ends of the front tube and seat tube. For many years both before and after the first world war this type of frame was the overwhelming choice of the established manufacturers. An earlier variant of the diamond pattern was the single-loop frame in which the front tube and seat tube were bent from a single length, which passed underneath the engine. An improvement on both was the cradle frame. In this, the bottom ends of the single front and seat tubes were spaced farther apart and rigidly connected by a brazed-in engine cradle, from the rear of which the tubes reached upward to the wheel-spindle lugs.



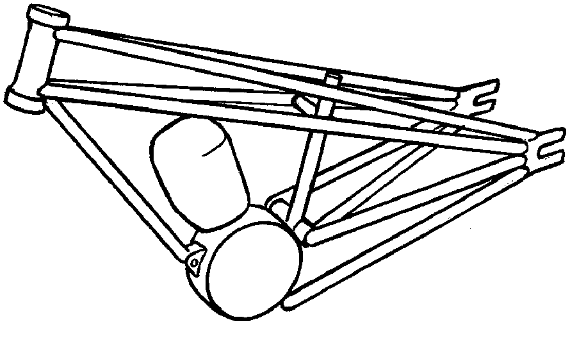
**Cradle frame, successor to the diamond bicycle pattern . The cradle tubes are extended rearward to the wheel spindle lugs.**



**In the duplex cradle frame the cradle tubes are also extended upward to the steering head.**

A straightforward development of this layout was the duplex cradle frame, in which the cradle tubes were continued upward to the steering-head lug as well as to the rear spindle lugs. Both types of cradle frame suited the upright single-cylinder engine by providing room for the narrow crankcase to be slung very low, engines with wider crankcases had to be mounted higher, so raising the centre of gravity.

In the design of these early frames, torsional and lateral stiffness seem to have been given a low priority. However, there were some commendable efforts between the wars to ensure the all-important torsional and lateral stiffness through triangulation of the frame structure. In the Cotton, the four long tubes connecting the steering head directly to the rear spindle lugs were triangulated in both plan view and elevation, and the machine was renowned for its excellent steering.

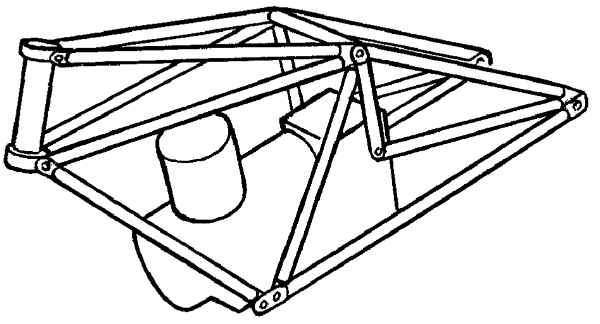


**Triangulated in both plan and elevation, the straight-tube Cotton frame was renowned for its steering.**

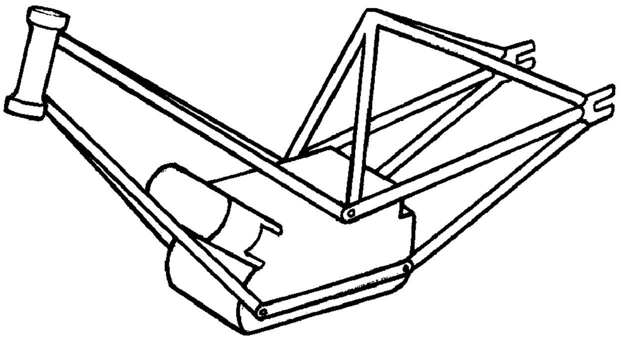
A few attempts were made to stiffen the support of the steering head by incorporating it in one end of a cast H-section frame member, this type of structure replaced the front down tube in the Greeves and the top tube in some BSAs.

**A 1960s. picture of the author racing a Greeves Silverstone with cast aluminium H-section front down member, incorporating the steering head. The under-frame was an open channel section fabricated from steel sheet, a bent steel tube backbone completed the frame loop.**

Bolted-up from straight tubes (for easy repair) and relying on the power plant for some of its stiffness, the Francis-Barnett was fully triangulated from the steering head to the saddle, though the rear end was triangulated only in the vertical plane. Another frame to depend on the engine for part of its stiffness was the open Scott. In this the rear end was triangulated fully but the steering head only laterally – which was much the more important plane from the steering viewpoint. The front brake of the day was hardly powerful enough to tilt the head significantly in the fore-and-aft plane; and even if it did, that would not have impaired the steering nearly so much as would twisting the head sideways. The Scott too earned an enviable reputation for its steering.



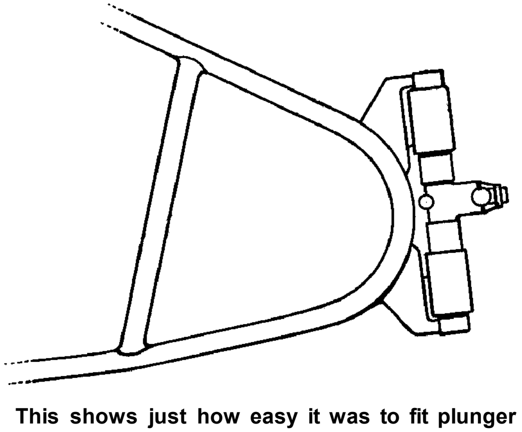
**Fully triangulated at the front but only vertically at the rear, this Francis-Barnett frame could be easily repaired by renewing any of the bolted-in tubes.**



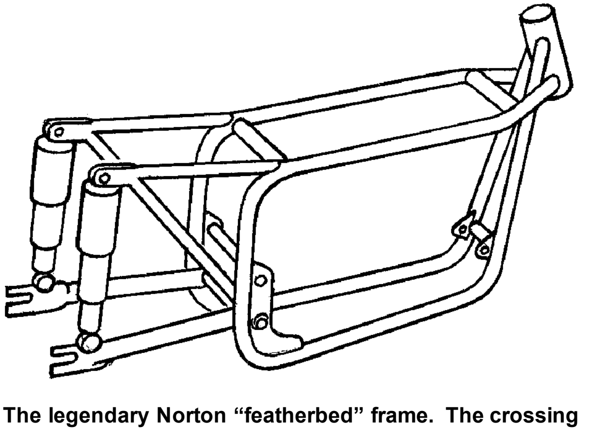
**Triangulated fully at the rear but only laterally at the front, this early Scott frame relied on the engine for some of its stiffness.**

When "plunger" sprung rear ends began to take over from the unsprung variety, many manufacturers simply opened up the rear part of their cradle frames to accommodate the spring units. In most cases only a thin wheel spindle held the two spring units together and so with many designs there was a lot of differential movement between the two sides, resulting in the rear wheel twisting out of line with the rest of the machine. Often, despite the welcome increase in comfort and sometimes in roadholding, there was a deterioration in handling and stability compared to their unsprung forebears. In fact the Norton plunger sprung frame earned the nickname of "garden gate". Its general handling was thought to be akin to riding that particular piece of garden furniture.

A revolution was started in 1950 when the works Norton racers were supported by the McCandless Bros. designed "featherbed" frame. It is hard to over-estimate the influence that this design has had on subsequent chassis development.



**This shows just how easy it was to fit plunger springing in place of the then standard unsprung rigid rear end. This design reduced the amount of re-tooling required.**



**The legendary Norton "featherbed" frame. The crossing over of the tubes at the steering head facilitated the use of a flat-bottom tank but impaired stiffness. A head steady was used which connected the steering head and top of the engine to overcome this.**

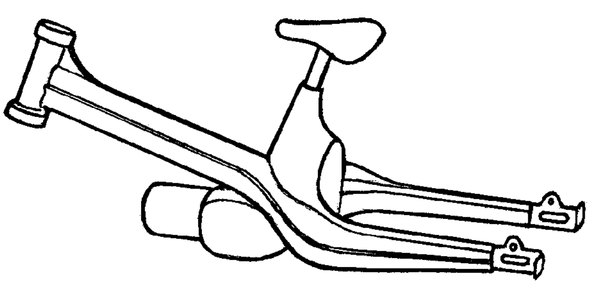
Even now, over half a century later, many current designs still show a direct lineage back to it. The enormous improvement over its predecessor, hinted at by its nickname, probably owed less to any one design feature than to a combination of several. Its duplex-loop layout had mediocre structural efficiency but provided adequate (though not exceptional) stiffness and the general layout was such as to give a fairly even weight distribution and a relatively low centre of gravity (considering the upright position of the cylinder). The front fork was one of the more robust telescopics of the period, and the steering geometry provided light, responsive handling. As with many landmark developments the featherbed's success probably owes much to being the right product at the right time rather than having any overwhelming technical superiority.

Generally speaking steel has been the most used material for tubular frames although both titanium and aluminium have been used. BSA tried titanium in the 1960s. for their works moto-X bikes, and the 1980s. saw various makers using aluminium alloys for both road and racing frames.

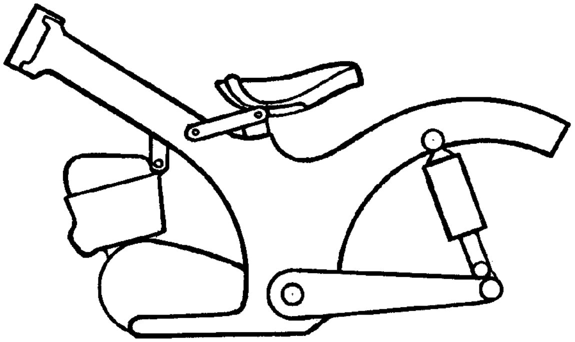
## Beams

An entirely different approach to the problem of achieving adequate resistance to twisting and bending is to use a large-diameter tube as the main frame member, thus combining a high degree of stiffness with simplicity and light weight. Provided it is of sufficient section, the tube does not necessarily have to be circular, though this is the best shape for torsional stiffness. Indeed, when the NSU Quickly popularised this type of frame at the start of the moped boom in the early 1950s the tube – or beam, to use another name – was made from left and right half-pressings seam-welded together, giving an approximately oval section.

Clearly, however, a plain beam could not connect the steering head directly to the rear-wheel spindle as did the top four tubes in the Cotton frame. Hence it was bifurcated at the rear to accommodate the wheel, and the resulting open channel section of the two arms was closed by welding-in a U-shape strip to restore strength. Welding the beam from two halves in this way made it possible to incorporate any necessary curvature in a vertical plane. NSU used curved beam frames not only on their moped but also on their Max roadster, their 250 cc world championship-winning Sportmax catalogue racing single, the works racing 250 cc Rennmax twin and 125 cc Rennfox single.



**In this unsprung beam frame for a moped the open channel section of the rear fork arms is strengthened by a welded-in U-shape strip.**

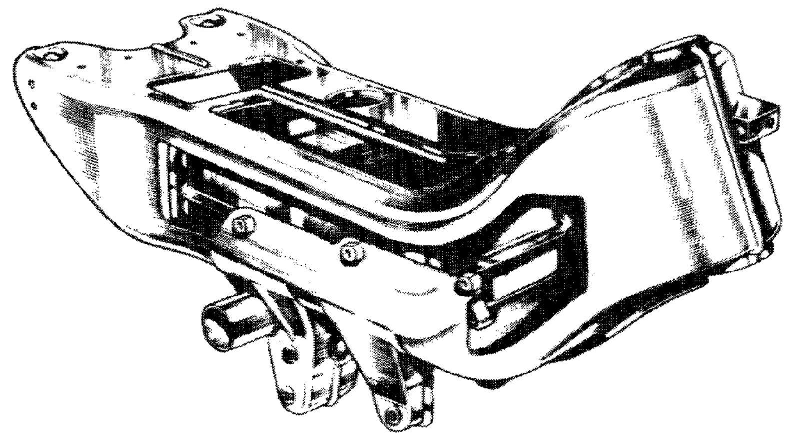


**Welded from left and right hand pressings, this NSU beam frame was shaped to accommodate pivoted fork rear springing and to support the engine at the top and rear.**

Pivoted-fork or swinging-arm rear springing removed the need to bifurcate the rear end of a frame beam because with this type of suspension it is the swing-arm pivot, not the wheel spindle, to which the steering head has to be stiffly connected. (Naturally, the swing-arm itself should continue the torsional and lateral stiffness back to the spindle.) In the NSU Max and racers, the pressed-steel beam was curved downward at the rear to make a direct connection from steering head to rear pivot.


**Winner of the world 250 cc championship in 1953, this NSU Rennmax twin had a curved beam frame welded from left and right halves** (MCW)

Because of its extremely large cross-sectional area (sufficient to accommodate a separate 2½ gallon fuel tank inside), the Ariel Leader (and Arrow) frame was probably the stiffest and most outstanding of the beam type. Predictably when pressed into racing service, its steering proved well up to the extra demands.

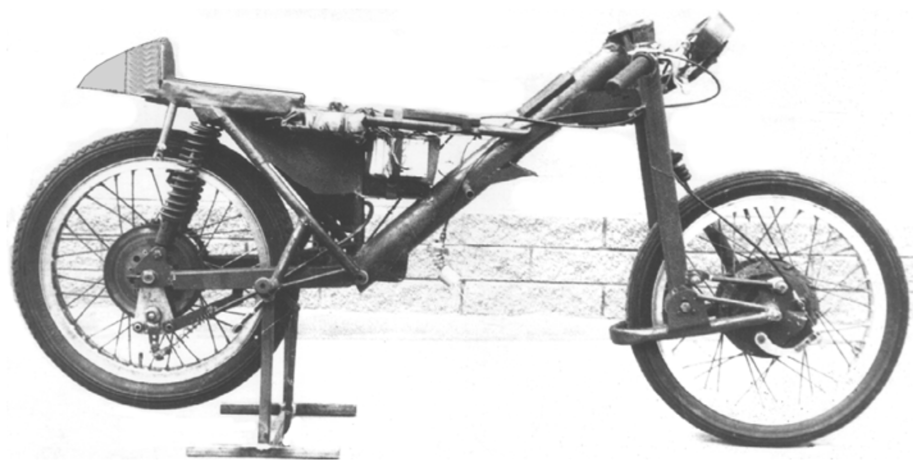


**The Ariel Leader (and Arrow) frame derived great stiffness from the large cross-sectional area of the beam, which enclosed the 11 litre fuel tank.**

**To make a direct connection between steering head and rear fork pivot, the tubular beam on this early 125 cc Honda GP racer was curved through some 90 degrees.** (Salmond)

Other frames – such as that on the early grand-prix Hondas and some Reynolds one-offs – used a large-diameter circular tube, similarly curved, to achieve the same effect. The Honda frame, however, like its duplex successor on the grand-prix fours of the early 1960s, was structurally incomplete without the engine, which was attached at the cylinder head and gearbox.

Making a direct connection with a straight tube is usually impracticable, even with a flat-single engine (although the first frame to be constructed by the author in the 1960s. achieved it). Nonetheless, with that type of engine or say, a sloping parallel-twin two-stroke, the rear end of the tube can be brought close to the fork pivot, as it was in the Foale frame for a Yamaha TZ350. In that case, the 50 mm. gap was bridged by a small welded-in box section.



**The first chassis built by the author in the early 1960s for a small 2-stroke 125cc engine. Features a 76 mm. diameter backbone frame connecting the rear fork pivot directly to the steering head.**

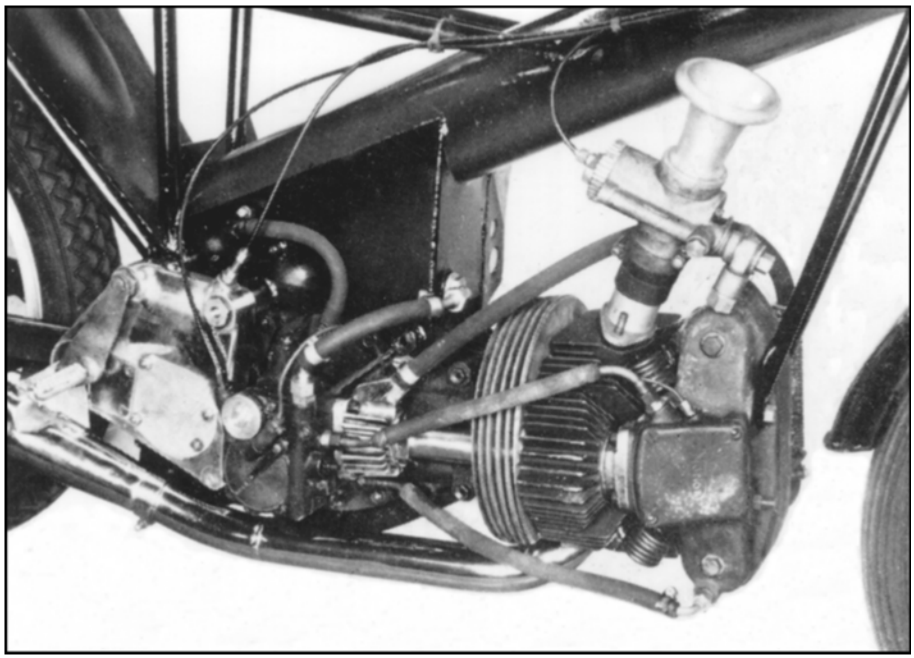
**Note also the leading link forks with Greeves type rubber pivot bushes, which also act as the springs.**

**An early 1970s frame by the author for 250 and 350 cc racing TZ Yamahas. The straight tubular beam is connected to the rear-fork pivot by a box section and stiffened at the steering head by a folded gusset.**

**The front down tubes are there solely to support the engine weight and are bolted at the top to the main frame. Such a construction makes for easy engine installation and removal.**

On a Reynolds frame for a 250cc Moto Guzzi flat single however, the gap was appreciably greater and was spanned by a channel-section light-alloy fabrication, bolted through two cross-tubes welded into the main tube (which doubled as an oil tank) and the box-section sump welded to its underside.

A substantially similar layout was used on Norton's unfinished Moto Guzzi-inspired experimental 500 cc flat single in the mid-1950s. In that case the oil was contained in a 114 mm.-diameter main tube and a welded-on underslung box that also supported the crankcase, while the fork pivot was bolted between a light-alloy gearbox plate on the left and an aluminium casting on the engine.



**Believed to have now been completed and residing in the Sammy Miller museum, this Norton frame was being prepared for 1956 but was never finished by the factory. Oil was contained in the backbone and the underslung box, which supported the crankcase. Surprising, this box section was not used to support the swinging fork pivot, which was held between light alloy plates.**

(MCW)

When a tall, bulky engine (such as a 1-litre twin-cam four abreast) has to be accommodated, an even bigger gap has to be spanned. A self-defeating scheme adopted by some frame builders was to bridge the gap with a pair of bolted-on light-alloy plates, which could make nonsense of the tube's torsional stiffness, depending on detail design.

An alternative arrangement was incorporated in the Foale frame for Honda and Kawasaki fours, in which a pair of tubular triangles splayed out from the rear of the tube to the sides of the fork pivot, so providing good support in both planes.

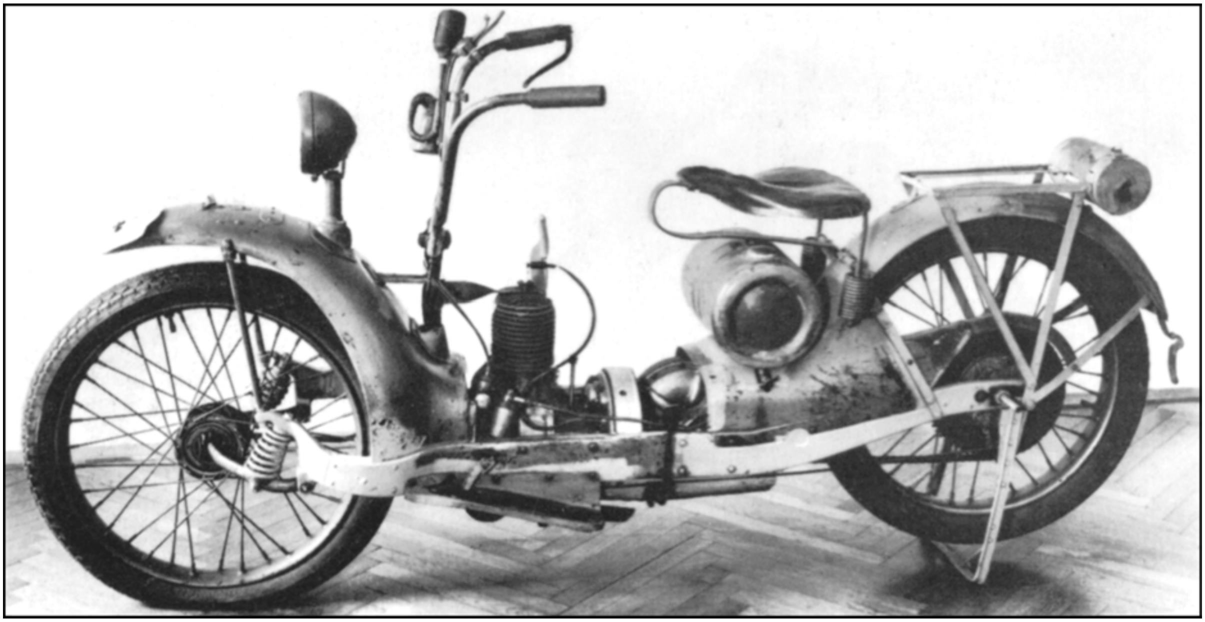
Another approach to the problem of accommodating a large engine is to split the beam around each side, thus we come to the "twin spar" frame initially popularised in the 1980s. These frames have been made in both steel and aluminium, but aluminium is currently the material of choice. Designed and constructed properly this frame type can be made quite stiff.

**Note how the head stock is supported by the two side beams of this 1997 NSR 250 racing Honda. Frame material is aluminium alloy.**

## Ner-a-Car

Although its chassis comprised two full-length, channel-section sides in pressed steel, cross-braced front and rear, the Ner-a-Car of the 1920s defies classification with the beam-type frames if only because it lacked a conventional steering head to connect to the rear-wheel spindle. Indeed, the steering kingpin was set in the middle of the front axle and housed within the front hub – hence the term: *hub-centre steering*. The axle itself was horizontal, shaped like a U (closed end forward) and pivoted in lugs protruding downward from the chassis sides, with stiff coil springs providing a short suspension travel, which thus varied the kingpin inclination. The chassis members were bowed outward at the front for tyre clearance on full lock, which was nonetheless severely restricted.

Although the chassis' resistance to bending in a horizontal plane must have been high, its torsional stiffness was doubtful. The Ner-a-Car's quite exceptional stability most likely stemmed from its ultra-low centre of gravity, allied to an uncommonly long wheelbase – 1500 mm. for the unsprung version, 1740 mm. with quarter-elliptic, pivoted-fork rear springing, and its hub-centre steering, which is often more tolerant of torsional flexure.



**The 1920s Ner-a-Car chassis comprised left and right channel-section pressings, cross-braced front and rear. The engine was slung low and hub-centre steering was a standard feature.**

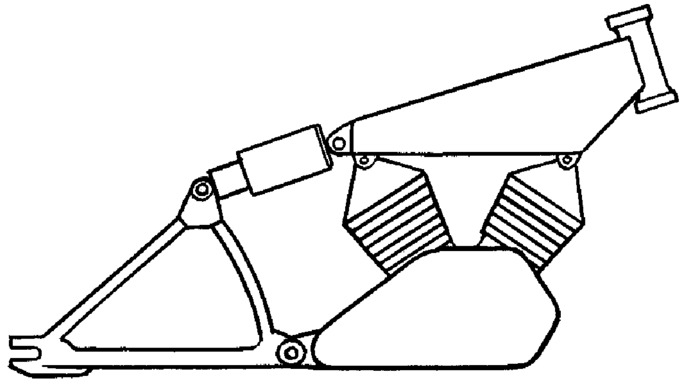

**This Foale frame used a large backbone above the engine, with triangulated support for the swing-arm pivot.**

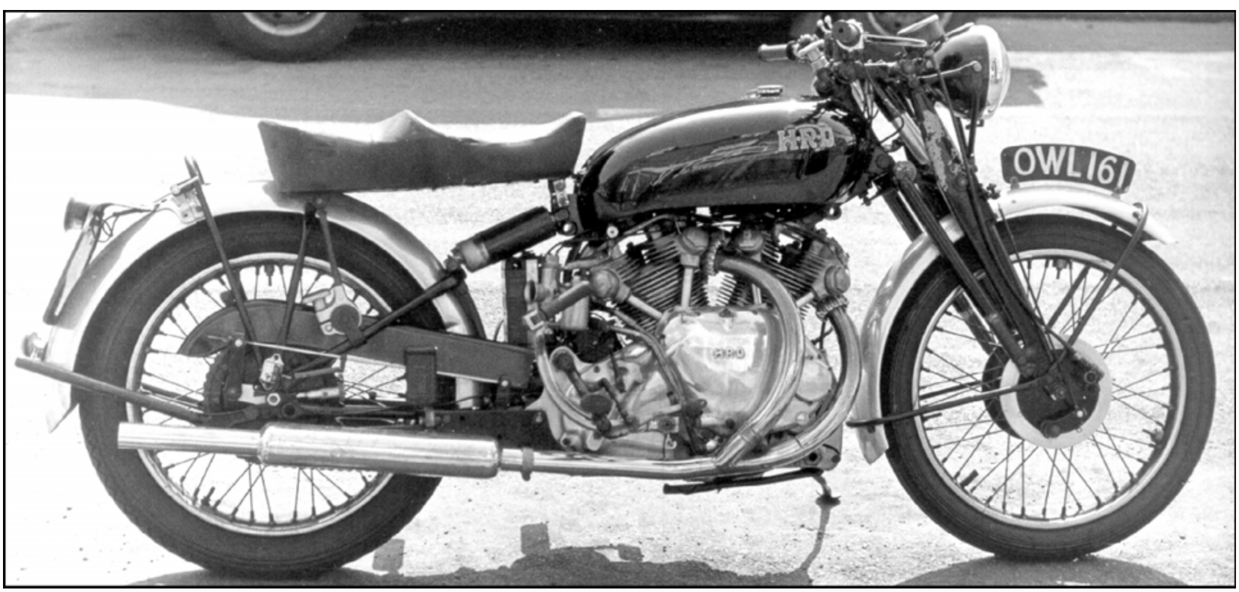
## Stressed engine

One of the earliest examples of the use of the engine as an integral part of the frame was the P&M (later the Panther). In this the very tall cylinder did duty as a front down tube but the cylinder barrel and head were relieved of tensile stresses by long bolts extending from the cylinder head to the crankcase, or long U-bolts reaching down to loop under the main-bearing housings. Altogether more renowned and successful was the layout of the post-war Vincent V-twins, in which a rectangular-section backbone, welded from sheet metal in the form of a three litre oil tank, was bolted to the steering head and both cylinder heads. The triangulated rear fork was pivoted behind the gearbox, while the spring units were anchored to the rear end of the backbone.

**Diagrammatic layout of the stressed-engine principle immortalized by the Series B, C and D Vincent-HRD big twins. The rectangular-section beam connecting the steering head to the cylinder heads contained the engine oil.**

**The rear swinging fork was triangulated and it pivoted from two aluminium plates bolted to the rear of the gearbox.**






**A series C Vincent-HRD Rapide with hydraulically damped Girdraulic front fork and twin rear shock absorbers.**

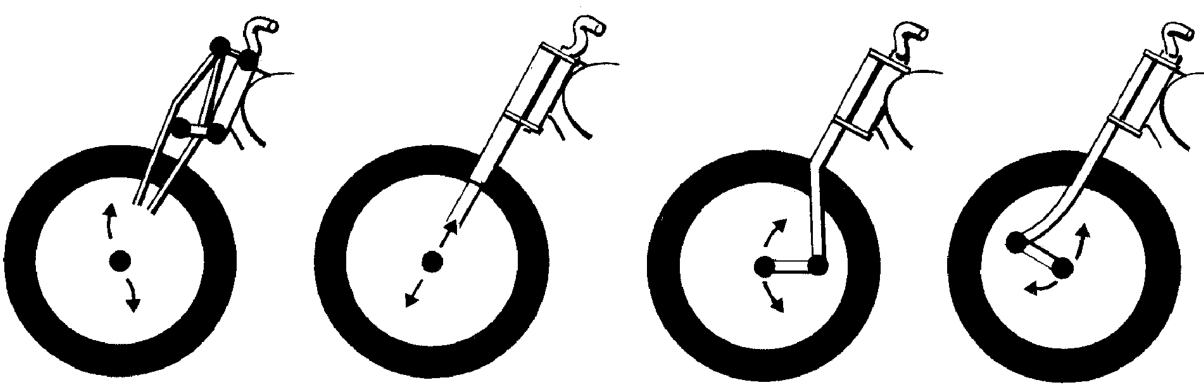
Despite widespread use of this concept with racing cars it is a form of construction which has been largely ignored by the bike world. There have been a few notable exceptions however, in addition to those mentioned. The successful 1950s. racing Guzzis ( single cylinder and V8 ) had the rear swing-arm pivot cast integrally with the engine/gearbox casing, as have later models from Ducati and BMW. All of these still needed a relatively conventional frame to complete the structure.

On the other hand the Norton/Cosworth engine was designed from the outset to be the main structural member, hardly surprising considering Cosworth's pedigree with building car racing engines. As with some others the swing-arm pivot was cast in the engine casings but only a minimal triangulated structure was needed to provide a rigid connection to the steering head, simple frame work to the rear for rider support was all that was needed to complete the machine.

## Front suspension

The first end of a motorcycle to be sprung was the front, for steering reasons. Following considerable variety in the early days, the first type of fork to achieve almost universal adoption was the girder, first with side springs then with one spring in front of the steering head. This spring was usually, though not always, of barrel shape to give a progressive rate and easy end fixing. Amply stiff torsionally, many girder forks nonetheless lacked intrinsic lateral stiffness, though some notable attempts were made to improve this.

Both Rudge and Vincent (on the Girdraulic fork) stiffened the link assemblies by forging the links integrally with the spindle housing (the Girdraulic also had forged light-alloy fork blades). The Webb fork fitted to some early KTT Velocettes had the girders triangulated laterally by tubes joining the middle of the bottom spindle housing to lugs out- board of the fork ends.



**Girder, telescopic, leading-link and trailing-link front forks, showing the path travelled by the wheel spindle throughout suspension movement.**

After a long reign, the girder fork eventually gave way to the hydraulically damped telescopic type, which is still so popular today, when BMW demonstrated the latter's superior characteristics in GP. racing from 1935 onward. Compared to the girder fork, the telescopic required no adjustment or routine lubrication, provided longer wheel travel, gave substantially constant trail under most conditions (except for nose-diving under braking, when the trail reduces) and had much superior damping characteristics.

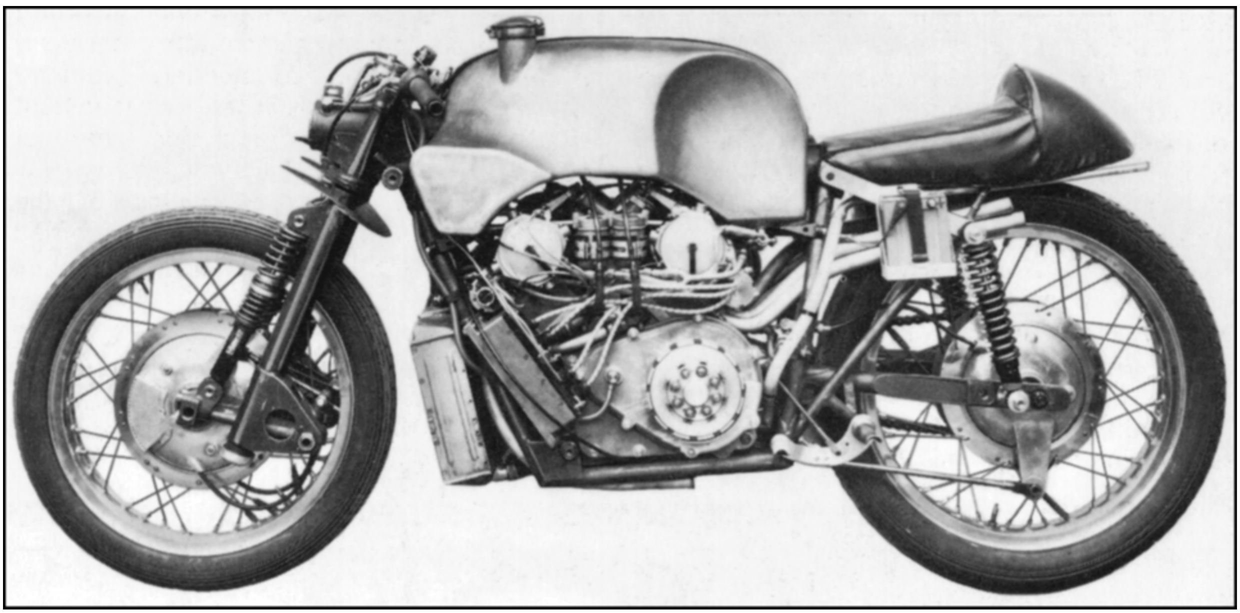
With the exception of snubber springs, pioneered on the works Norton racers of the late 1930s, most girder forks relied on friction damping. Unfortunately, the characteristics were opposite to those required – i.e. resistance to initial movement (stiction) was too stiff and it reduced considerably once movement started. By contrast, hydraulic damping is automatically related to the rate of wheel deflection, and unlike friction damping, it does not have to give the same resistance in both directions, hence it can easily be tailored to most requirements.

Yet BMW were by no means first with a telescopic front fork. Several decades earlier Scott employed the principle, though without damping. Nevertheless, their fork was well engineered, with the sliders firmly braced by being brazed into a lug above the wheel (where they operated a central spring) and with the bottom bushes well supported, latterly by diamond-shaped girders.

**Manfred Schweiger's 1929 Scott TT Replica, showing one of the earliest telescopic front forks, in which the bottom bushes were supported by diamond-shaped girders. The lug at the top of the sliders actuated a central spring.** (Margrholz**)**

Although the hydraulically damped telescopic fork quickly matched the earlier girder in being almost universally adopted, its instant popularity probably owed more to its neat appearance, low production cost and lack of maintenance than to its dynamic qualities. Despite its superiority to the girder pattern many examples lacked sufficient torsional and lateral stiffness, especially for racing. Its hydraulic damping, although a great advance on the friction type, was not so refined as that in the proprietary struts supplied by specialist concerns for rear springing. In the monocoque Norton on which he won the 1973 Formula 750TT, Peter Williams showed what could be achieved here by incorporating Koni damper units in the legs of a telescopic fork. For series production, however, the extra cost might well have been considered unacceptable. More recently we have the so called cartridge forks based on the same idea.

Where cost had a lower priority – i.e. in grand-prix racing – or where a manufacturer of roadsters set more store by high quality than low price and sleek looks, some engineers rejected the telescopic fork for its dynamic and structural shortcomings and preferred the leading-link variety. These offered greater torsional and lateral stiffness, lower unsprung weight, better damping (through proprietary struts) and a choice of steering geometry between virtually constant trail and constant wheelbase, depending on the inclination of the links.



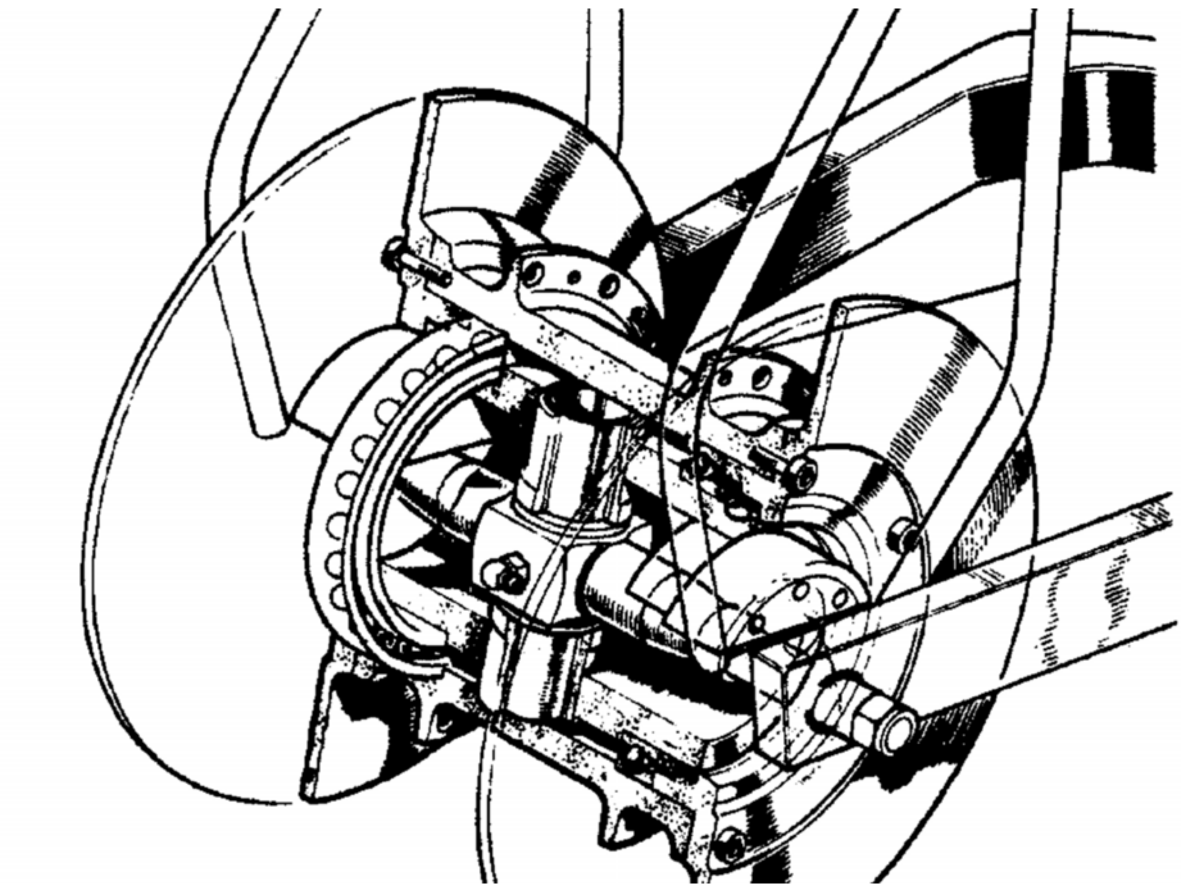
**One of the most famous examples of leading link front forks – that of the 1956 version of the 500 cc Moto Guzzi V-eight grand-prix racer. The externally mounted suspension struts allowed easy setup.** (MCW)

For world-championship racing at the highest level, both NSU and DKW discarded telescopic forks in favour of leading links while BMW changed to an Earles-type pivoted fork. In the same context, Moto Guzzi never seriously considered anything but leading links, and neither did Ken Sprayson on the special frames that he built under the Reynolds Tube banner. For both road and racing use, Greeves made a similar choice as did Cotton and others. Two methods have been used to combine lightness with rigidity. Some designs (e.g. Greeves) united the left and right links with a tubular loop round the back of the wheel. Moto Guzzi, however, whose racing machines were unsurpassed for handling, achieved ample stiffness without a loop by using a large-diameter (hollow) wheel spindle secured in the links by wide clamps, each with four nuts.

**This 1956 Senior TT shot of Walter Zeller shows the Earles-type leading link front fork on the racing BMW (Renn-Sport). The brake-shoe back-plate was fixed to the left fork arm to provide a high degree of anti-dive. Note also the telescopic steering damper, unusual for the period.** (MCW)

Given the same inclination of the links, a trailing-link fork may have similar characteristics to those of a leading-link, except for higher steering inertia. Yet the difficulty of making a sound and neat job of its design discouraged most manufacturers from trying it. Ariel were the most notable exception, using trailing links on the Leader and Arrow, both of which were noted for impeccable handling.

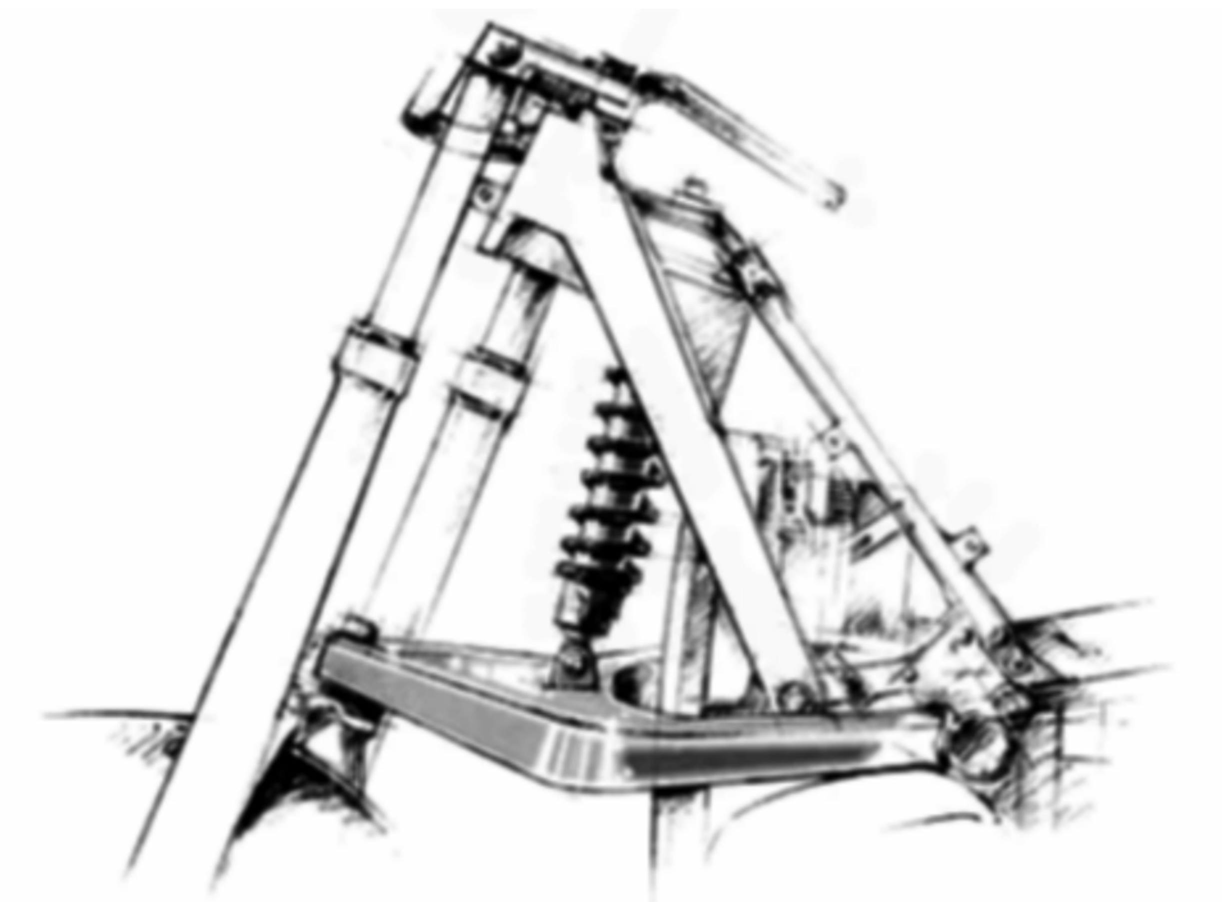
A variant of the leading-link fork was the Ernie Earles type, with long links combined into an integral fork pivoted behind the wheel. BMW standardized this arrangement for practically two decades between abandoning their original telescopic fork and introducing a long-travel version in 1969. Essentially though, their Earles-type fork was a compromise for solo and sidecar duty. Although its lack of sliding friction was a boon in sidecar cornering, its high steering inertia blunted steering response in a solo. As an aid to this dual purpose, BMW provided a choice of two pivot mounting locations so that trail could be easily set for solo or sidecar use.



**Basic layout of the Difazio system. The wheel hub spins on large diameter bearings, and the non-rotating axle is integral with the steering "king-pin". A forward facing swinging fork connects the axle to the main chassis, whilst the upper frame-work controls steering and takes care of braking reaction. (**Motorcycle Mechanics)

Another class of front suspension does away with the high mounted head-stock altogether. Many of these fall into a category loosely known as "hub-centre steering". The early Ner-a-Car was so fitted and in the 1960s the British engineer Jack Difazio revived some small interest and produced a limited number of his own design. A decade later much publicity was given to a series of French built racers, financed by the Elf fuel company and generally known simply as the Elf. These used a link system not unlike the double wishbone design used on many cars, except that the link pivots where behind the wheel rather that to the side. These Elf machines increased awareness amongst motorcyclists and designers of the possibilities of alternatives to the then ubiquitous telescopic fork.

Many years ago the OEC factory produced a design that had no directly defined steering axis, it featured a series of links which determined a "virtual" steering axis, the position of which varied with steering angle. This machine was often praised for its good steering.



**Sketch of the BMW "telelever". The forward facing link pivots at the rear and connects to the unsprung part of the front forks, suspension loads are thence fed into the suspension unit. The upper mounting of the non-sliding stanchions allows for the back and forth movement necessary in this design.**

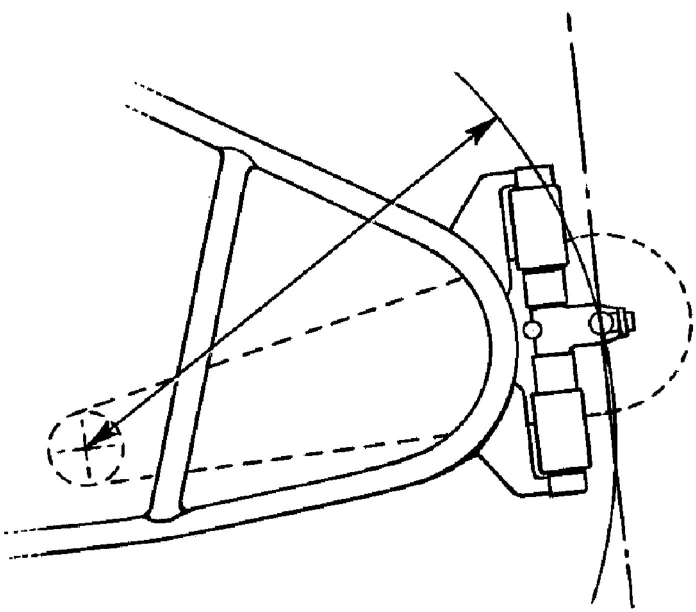
Today (2002) there are two mainstream manufacturers that have recently had standard models fitted with an alternative system.

BMW have their “telelever” system based on the British Saxon design. This largely retains the neat appearance of telescopic forks whilst giving better support to the wheel and some degree of control over braking dive. This does not fall into the general hub-centre steered category. Yamaha had just one model, the GTS, that used a double link system based on the work of the American designer James Parker. The GTS was sold in Europe between 1993 and 1999.

## Rear suspension

In general adoption, rear springing lagged several decades behind the sprung front fork, largely because of the long dominance of the rigid frame in road racing and the inevitable poor handling of some early bolt-on conversion kits.

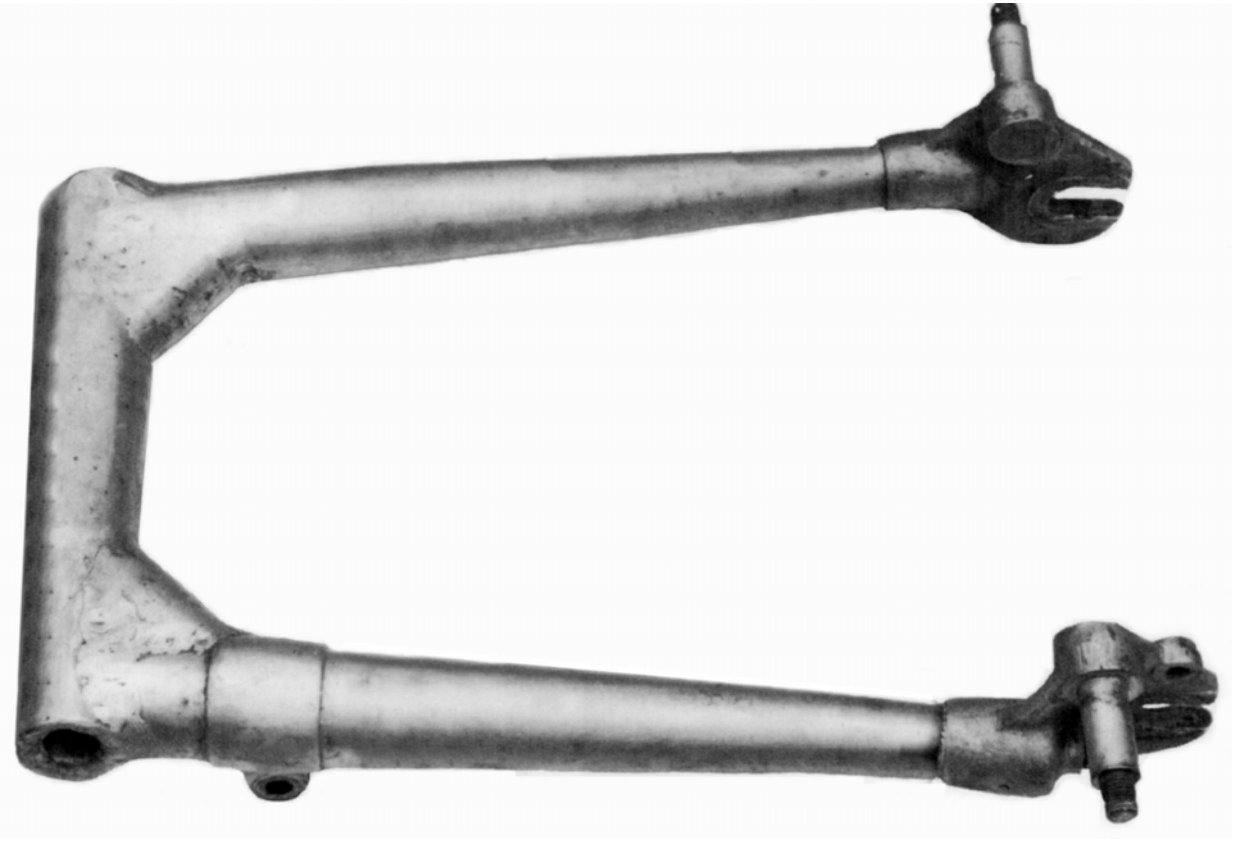
Plunger springing was first to gain wide acceptance – partly because of racing successes by BMW and Norton, and partly because, for the manufacturer, it was the easiest type to which existing rigid frames could be adapted. However, its limitations were clear from the start. Firstly, the incorporation of spring boxes at the rear destroyed the triangulation of the seatstays and chainstays, thus permitting each side to flex independently in a vertical plane, even to the extent of fatigue failure. Secondly, resistance to wheel tilting depended also on very stiff clamping of the spindle in the sliders (as it does with telescopic front forks and some leading links). Finally, the straight-line vertical movement of the wheel tightened the chain considerably at the extremes of travel, so limiting the range of wheel movement and necessitating a slack chain setting in the static load position.



**With plunger-type rear suspension the straight-line movement of the wheel tightens the chain at the extremes of travel, thus setting a limit to total movement and giving a slack chain at static load.**

With such notable exceptions as AJS/Matchless, most manufacturers found their existing rigid frames unsuitable for adaptation to pivoted-fork or swing-arm rear springing. Even so, that form of suspension was soon recognized as being superior to the plunger variety on all other counts. Indeed, it preceded plungers, with NSU and Indian providing examples early in the century, and with Vincent-HRD standardizing their famous triangulated pivoted-fork layout from the outset in 1928 until production ceased in 1955. Vincent obtained great strength and rigidity by triangulating the arms of the fork, mounting it on a very wide pivot and using preloaded taper-roller bearings to obviate play.

Moto Guzzi also chose to triangulate their swing-arm when they introduced rear springing in 1935 and scored spectacular victories in the Senior and Lightweight TTs with Stanley Woods aboard. Later on, however, they changed to a plain fork welded from very large diameter tubing and claimed it to be equally stiff torsionally and stiffer laterally (because the earlier fork was triangulated only vertically, not laterally). Another swing-arm designed to provide ample stiffness without triangulation was introduced by Velocette on the works racers of the mid-1930s and standardized on the Mark VIII KTT in 1939. In this, the arms comprised taper-diameter, taper-gauge tubes.



**In a bid to provide ample stiffness without triangulation the Velocette swing-arm was made from taper-section, taper-gauge tubing.**

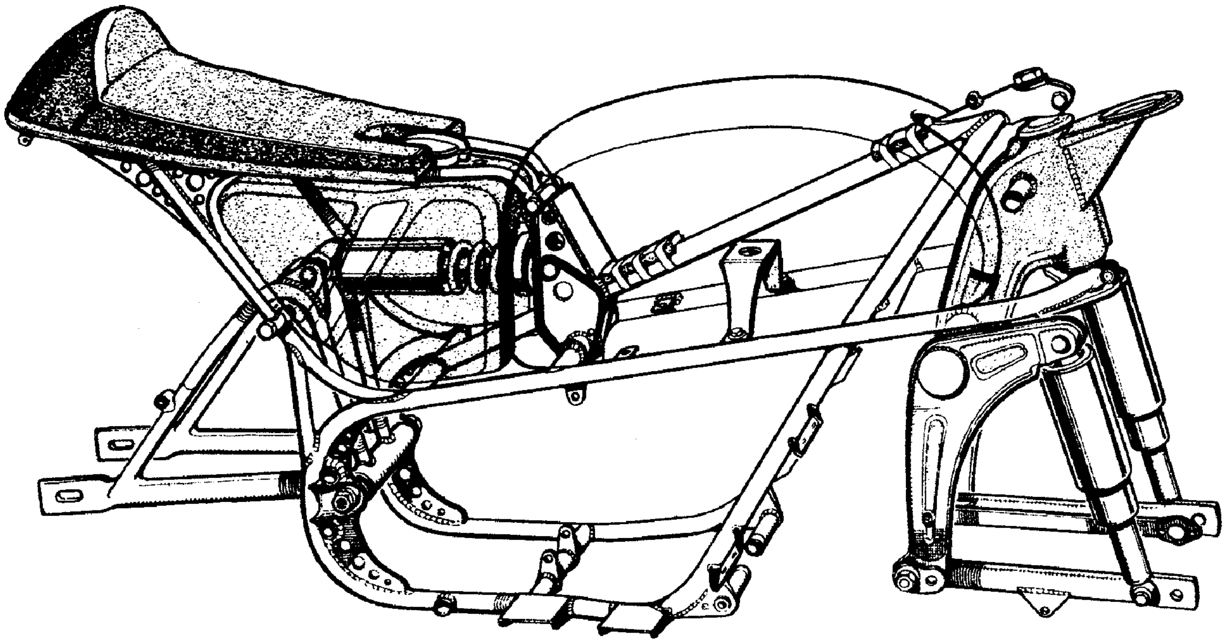
Many plain swing-arms lacked adequate torsional stiffness and this gave rise to a vogue for precisely matched left and right suspension struts to minimize one of the causes of twisting. Girling actually sold so-called matched pairs as an after-market fitment option.

**This view of the so-called garden-gate Manx Norton shows how the plunger type rear springing destroyed the triangulation of the chainstays and seatstays in the unsprung frame that it replaced. At the front, wheel location was by means of the highly regarded "Roadholder" hydraulically damped telescopic forks.**

Probably the most renowned frame with a plain swing-arm was the so-called Norton featherbed introduced on the works racers in 1950. Even though, by current standards, the swing-arm and frame stiffness was relatively low it was still an enormous improvement over its predecessor the plunger-sprung "garden gate".

There have been various permutations on rear fork types and spring struts. A fork triangulated above pivot level lends itself to a single strut actuated by the top apex of the fork, as on the experimental 250 cc BSA grand prix racer built by Doug Hele in 1952.

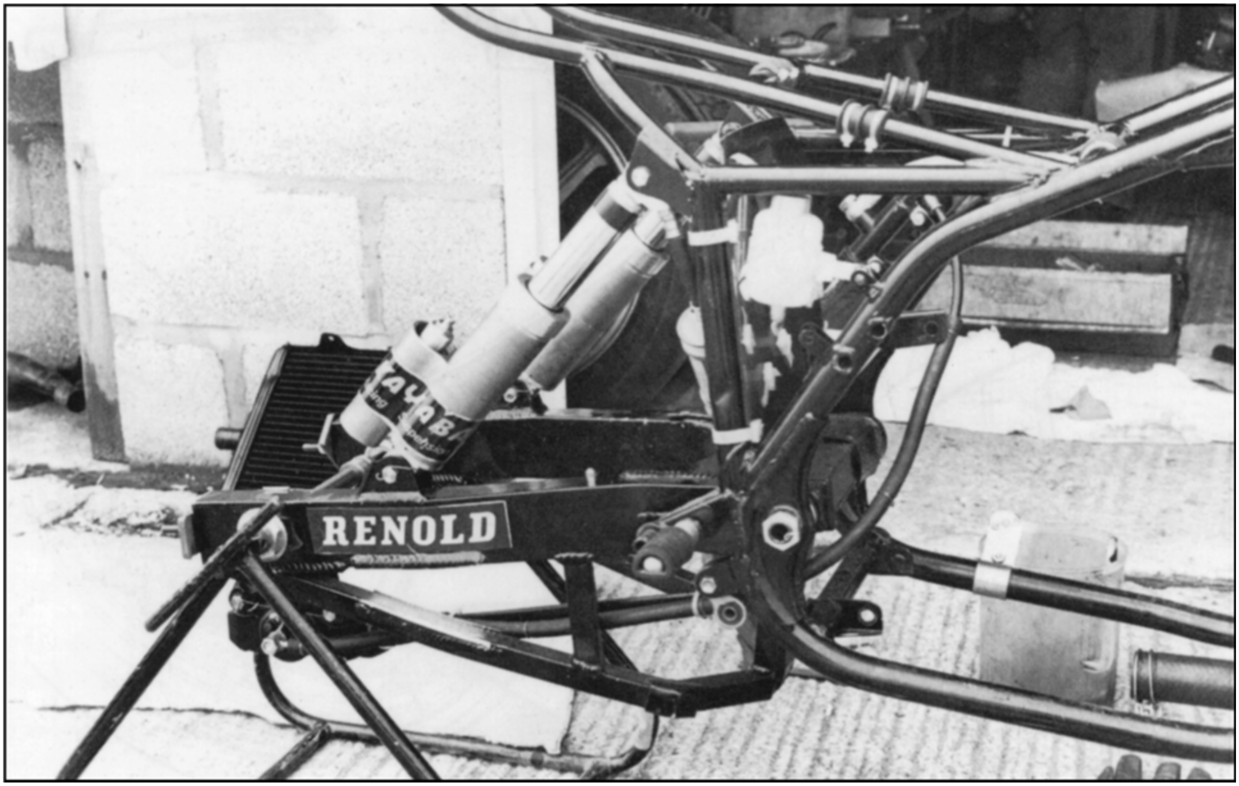
For their Series D twins in 1955 Vincent switched to a single Armstrong strut in place of the previous two spring boxes and separate damper under the seat. Nearly two decades later Yamaha revived the-system, which then acquired the fancy names of cantilever and monoshock. Moto Guzzi's original sprung fork was triangulated below pivot level to keep the weight low down. At first it was linked to a pair of long horizontal spring boxes flanking the rear wheel; later on these gave way to a single strut beneath the engine, anchored at the rear and compressed from the front by a rod passing through the spring. The latter arrangement was retained when the triangulation was abandoned.



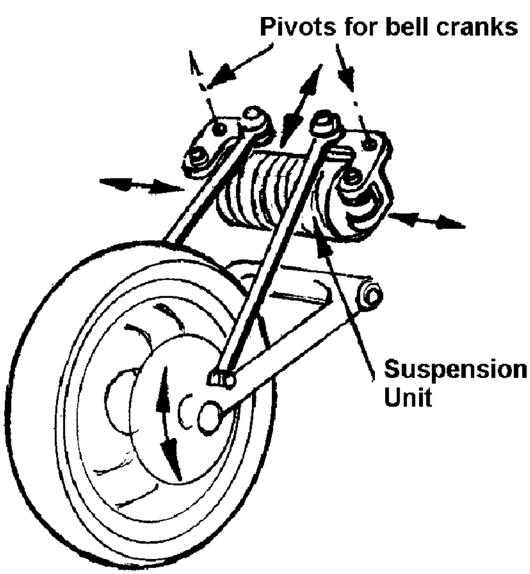
**On the Doug Hele designed 1952 BSA 250 cc grand-prix racer, the top apex of the triangulated rear fork actuated a horizontal suspension strut in a tunnel through the oil tank. Front fork was long leading link. Note the unusual steered head stock arrangement.** (MCW)

For several years on the RG500 racer, Suzuki also triangulated the rear fork below pivot level, yet retained a pair of struts above the fork. Then, on their 250 cc grand-prix twins in the mid-1970s, Kawasaki started a racing trend towards rocker-arm rear suspension, a trend which has continued to the present and spread to sports road machines also (though the system was already used in motocross). In the original layout the single suspension strut was installed vertically behind the gearbox and anchored to the frame at the bottom end. Fork movement was transmitted to the top of the strut through a short rocker arm linked by an A-bracket to points midway along the swing-arm. In a development of the system for the Kawasaki KR500 square four, the fork was triangulated above pivot level and its apex connected to the rocker by a short upright link, the strut was no longer anchored to the main frame but to the fork itself, just behind and the below the pivot.

In both cases the chief aim was to achieve a progressively stiffer resistance to wheel deflection (while using a constant-rate spring) so combining sensitivity to small bumps with an increasing check for larger ones. Although the resistance offered by the spring itself was directly proportional to its own deflection, the effective rate, measured at the wheel spindle, depended on the angles in the linkage. Though the principle was widely adopted, the geometry was sometimes suspect, with the effects of some changes in linkage angles tending to cancel out. Some designers eschewed complex linkages and relied instead on standard suspension dampers but with variable-rate springs – e.g. either two different springs end to end or one spring wound to different pitches.



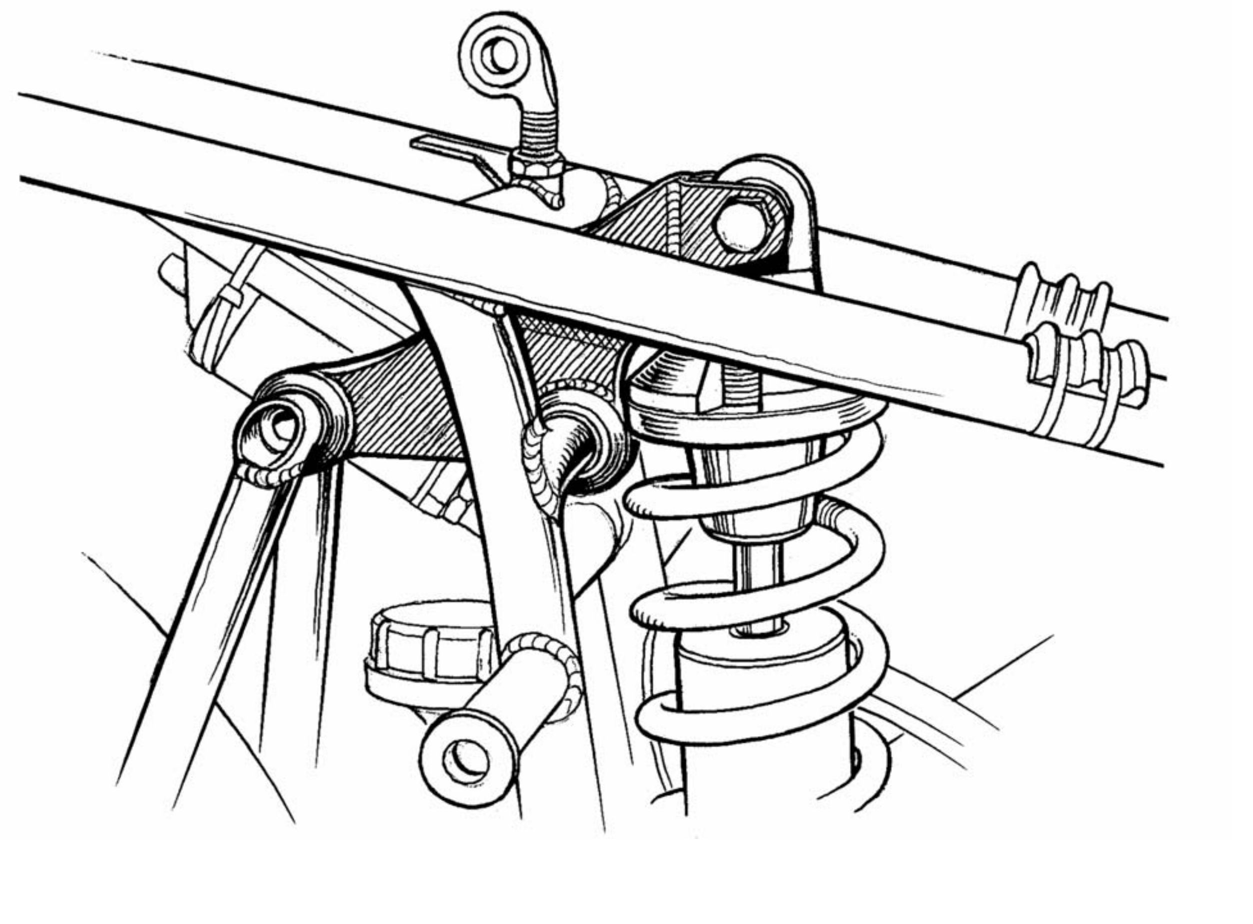

**A fairly modern example of a rear fork braced below pivot level, on the Suzuki RG500 racer. Unlike the Moto Guzzi layout of this type, two gas suspension struts are fitted, in the conventional position.**




**Space saving rear-springing layout on Yamaha's OW61 GP four cylinder racer. The suspension strut was squeezed, concertina fashion, by two bell-cranks linked to the swinging-arm.** (MCN)

Perhaps the most ingenious of the rocker systems was that on Yamaha's OW61 grand prix four. In this, the suspension strut was mounted transversely and squeezed from both ends simultaneously by bell-cranks linked to the top corner of the triangulated fork. The main advantage of this layout seems to be fore-and-aft space saving.

Since that time virtually all GP racing machinery and top of the range road bikes with sporting pretensions have used some form of rocker arm and linkage rear suspension. In some cases there are space saving considerations at work allied to the need (real or fashion) for progressive rate springing. The manufacturer's marketing departments seem to work overtime, dreaming up fancy names for each variant on the same theme, often claiming benefits of dubious validity.



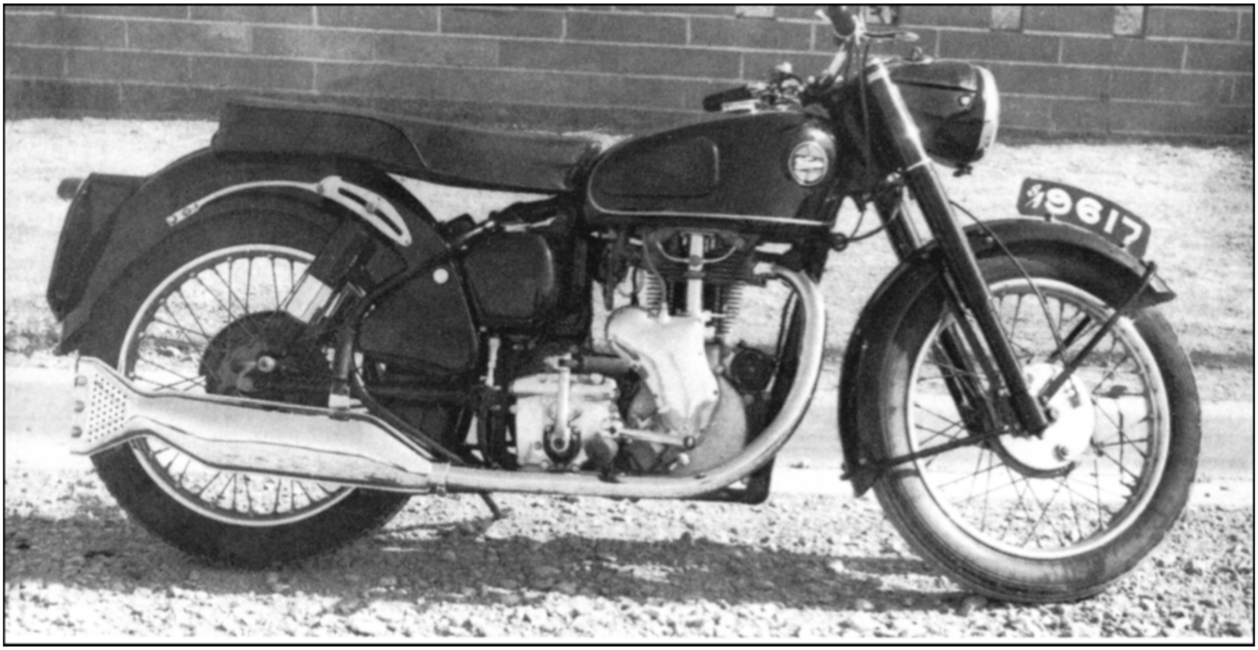
**One of the earliest rocker-arm rear-suspension layouts, on Kork Ballington's Kawasaki GP machine. The A-bracket on the left was attached to the pivoted fork while the suspension strut was anchored to the frame at the bottom.** (Watts)

## Spring types

Generally, coil springs made from special alloys of steel have been the overwhelming choice of designers. On telescopic forks coil springs were once usually fitted around the outside of the fork stanchions but now it is standard practice to use smaller diameter coils inside the stanchions. At the rear the springs are fitted around the outside of the damper tube.

From time to time rubber and air have been used as the springing medium instead of steel. Rubber in tension, in the form of bands, has been used in both telescopic and leading-link forks. Alf Hagon used simple rubber bands on grass track and sprinting bikes, adjustment was a simple matter of adding or removing rubber bands. On some Greeves machines, rubber was used in shear in the form of bonded bushes at the pivots of the fork links.

Although rubber can be endowed with varying degrees of inherent damping, this was never sufficient to preclude the use of separate friction or hydraulic dampers. A significant advantage of air springing (as also of rubber) is its progressive rate, which can be demonstrated by operating a tyre pump with the outlet sealed by one finger. A possible disadvantage, for really extreme conditions, is that the air heats up, so increasing its pressure, hence its effective rate. Velocette introduced this form of springing, made by Dowty, on their works racing machines in the mid-1930s, subsequently extending its use to the rear end of the KTT in 1939 and the front end of some of their post-war roadsters. In production, leakage of both air and damping oil was a problem. Improvements in sealing materials and surface finishes enabled the Japanese to revive air springing in the 1970 / 80s, first as an auxiliary to steel springs, then on its own. On the whole, however, single or multi-rate coil springs remain the first choice for most designers. The French Fournales units being a notable exception.

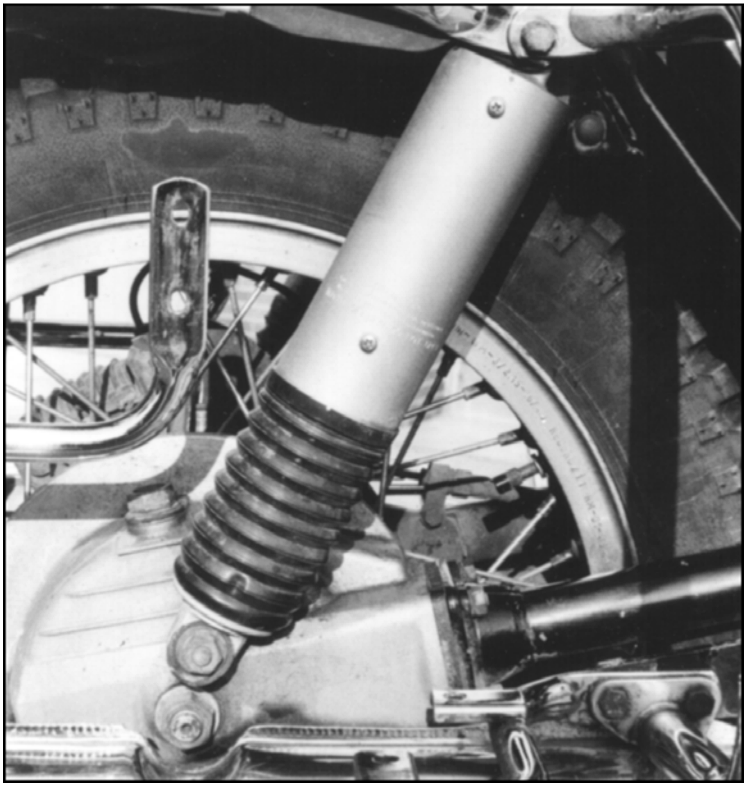



**Velocette used slotted top anchorages on the rear suspension, as a means of compensating for different loads, this altered both springing and damping effective rates. Wheel movement was also increased on the soft setting.**

## Load Compensation

Providing ready compensation for different weights of rider and the added weight of a pillion passenger has always bedevilled the design of rear springing for roadsters because the load on the springs may easily be increased by 50 or even 100 per cent. Under conditions of varying load, multi-rate springing, however achieved, may be an advantage. The most common traditional solution to the problem has been to provide a stepped cam adjustment for the preload on the springs. By this means the attitude of the machine can be preserved for different loads and full travel retained. But no single spring rate can be satisfactory, in any case, only BMW and MZ provided a built-in hand adjustment for the pre-load and few riders bothered to use the C-spanner otherwise necessary.

A much better form of compensation was that designed by Phil Irving for Velocette. In this, the top anchorage for the struts was slotted so that their inclination could be varied to suit the load by slackening two handwheels. Thus both the effective spring rate and the damping, measured at the wheel spindle, were adjustable, but in later production versions a spanner was required and so the adjustment was used less than it deserved to be. A consequence of this scheme is that less wheel movement is available on the hard setting than the soft. A later solution to this problem was the Boge Nivomat self-levelling strut first offered on BMW roadsters in the early 1970s.



**Boge Nivomat installation on an 850 cc Moto Guzzi California. Regardless of load, this automatically keeps the ride height and suspension travel constant.**

# 2 Tyres

It seems incredible that just two small contact patches of rubber, can support our machines and manage to deliver large amounts of power to the road, whilst at the same time supporting cornering forces at least as much as the weight of the bike and rider. As such the tyres exert perhaps the single most important influence over general handling characteristics, so it seems appropriate to study their characteristics before the other various aspects of chassis design.

When Newton first expounded to the world his theories of mechanics, no doubt he had on his mind, things other than the interaction of motorcycle tyres with the road surface. Never-the-less his suppositions are equally valid for this situation. In particular his third law which states, “For every force there is an equal and opposite force to resist it.” or to put it another more familiar way “Action and reaction are equal and opposite.”

Relating this to tyre action, means that when the tyre is pushing on the road then the road is pushing back equally hard on the tyre. This applies equally well regardless of whether we are looking at supporting the weight of the bike (vertical force) or resisting cornering, braking or driving loads (horizontal forces).

What this particular law of Newton does not concern itself with, is which force is the originating one nor indeed does it matter for many purposes of analysis. However, as a guide to the understanding of some physical systems it is often useful to mentally separate the action from the reaction, as we shall see.

The forces that occur between the ground and the tyres determine so much the behaviour of our machines, but they are so often taken for granted. Tyres really perform such a multitude of different tasks and their apparent simplicity hides the degree of engineering sophistication that goes into their design and fabrication. Initially pneumatic tyres were fitted to improve comfort and reduce loads on the wheels. Even with modern suspension systems it is still the tyres that provide the first line of defence for absorbing road shocks.

To explore carcass construction, tread compound and tread pattern in great detail is beyond the scope of this book. Rather we are concerned here with some basic principles and their effects on handling characteristics.

## Weight support

The most obvious function of the tyre is to support the weight of the machine, whether upright or leaning over in a corner. However, the actual mechanism by which the air pressure and tyre passes the wheel load to the road is often misunderstood. Consider fig. 2.1, this sketch represents a slice through the bottom of a rim and tyre of unit thickness with an inflation pressure of **P**. The left hand side shows the wheel unloaded and the right hand side shows it supporting the weight **F**. When loaded the tyre is compressed vertically and the width increases as shown, perhaps surprisingly the internal air pressure does not change significantly with load, the internal volume is little changed.

At the widest section **($X_1$)** of the unloaded tyre the internal half width is **$W_1$**, and so the force normal to this section due to the internal pressure is simply **$2.P.W_1$** . This force acts upwards towards the wheel rim, but as the pressure and tyre width are evenly distributed around the circumference the overall effect is completely balanced. This force also has to be resisted by an equal tension **(T)** in the tyre carcass.

The loaded tyre has a half width of $\mathbf{W_2}$ at its widest section $\mathbf{(X_2)}$ and so the normal force is $\mathbf{2.P.W_2}$. Therefore, the extra force over this section, when loaded, is $\mathbf{2.P.(W_2 - W_1)}$ but as the tyre is only widened over a small portion of the bottom part of the circumference, this force supports the load **F.**

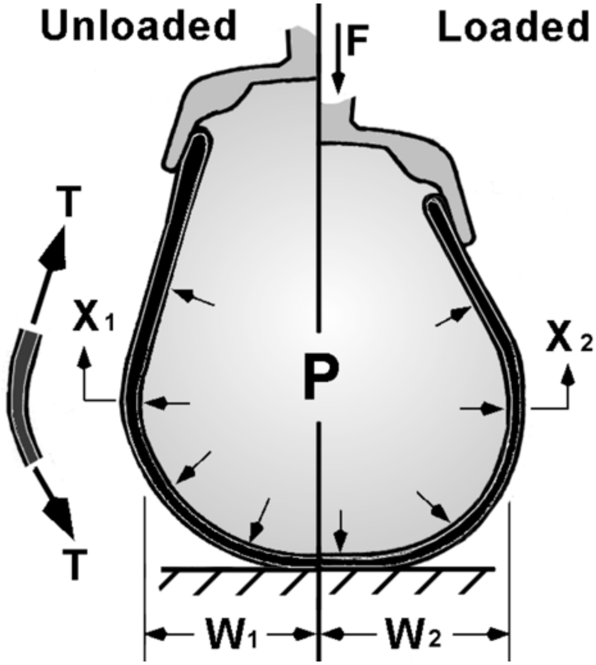



Fig. 2.1 **The left hand side shows half of an inflated but unloaded tyre, a tension (T) is created in the carcass by the internal pressure. To the right, the compressed and widened shape of the loaded tyre is shown.**

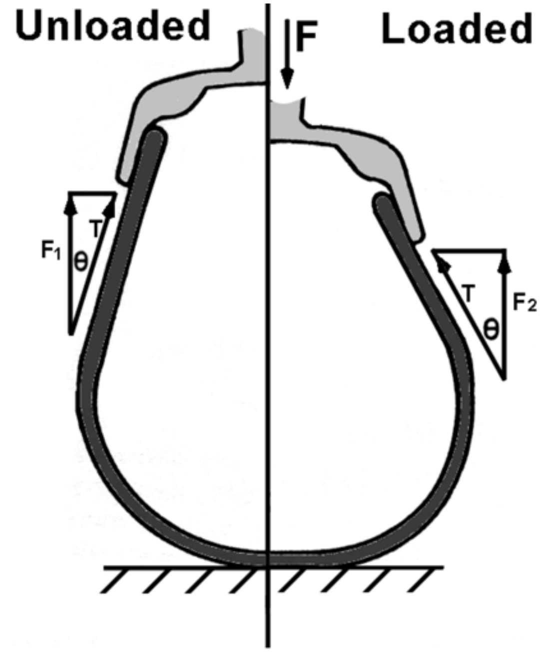



Fig. 2.2 **The in-line components ($F_1$ & $F_2$) of the side-wall tension are reduced by factors equal to the cosines of the angles of the side-wall. This reduction is greater with the loaded tyre resulting in a greater compressive force on the lower part of the rim.**

The above describes how the inflation pressure and tyre width increase produce forces to oppose the vertical wheel loading, but does not completely explain the detail of the mechanism by which these forces are transferred to the rim. The bead of a fitted tyre is an interference fit over the bead seat of the wheel rim, which puts this area into compression, the in-line component of the side-wall tension due to the inflation pressure reduces this compression somewhat. This component is shown as $\mathbf{F_1}$ on the unloaded half of fig.2.2 $\mathbf{F_1 = T.cos(\Theta_1)}$. The greater angle, $\Theta_2$, of the side-wall when loaded means that the in-line component of the tension is reduced, thereby also restoring some of the rim to tyre bead compression. This only happens in the lower part of the tyre circumference, where the widening takes place. So there is a nett increase in the compressive force on the lower rim acting upward, this supports the bike weight. The nett force is the difference between the unloaded and loaded in-line forces:

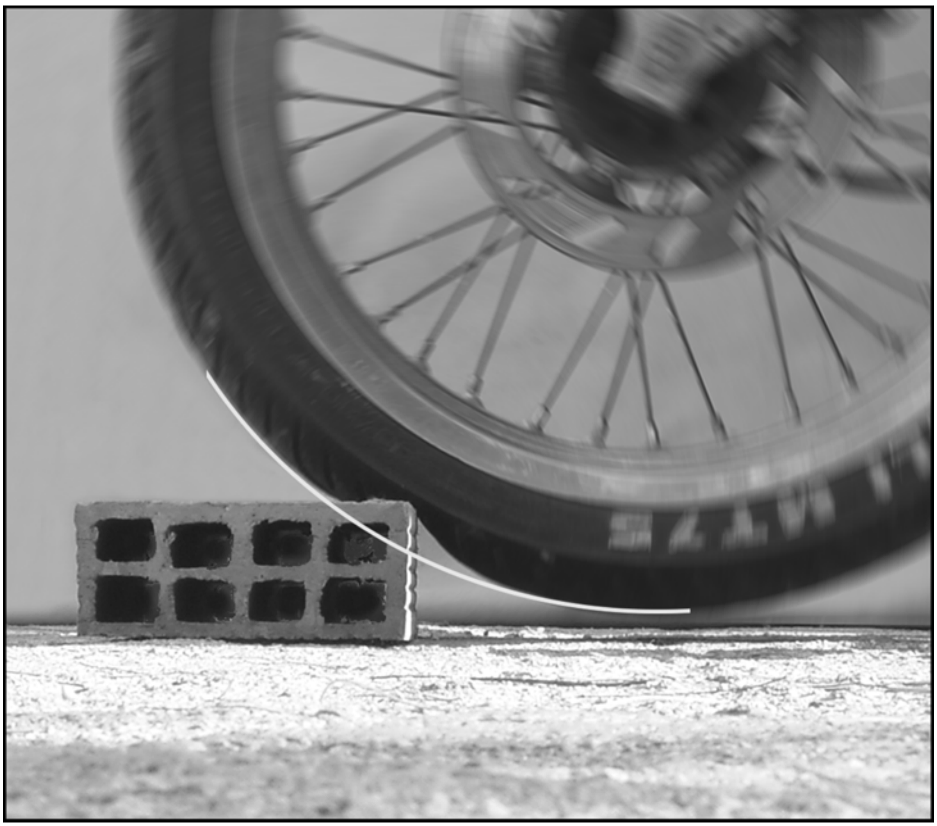
$$\mathbf{F = T.(cos(\Theta_1) - cos(\Theta_2))}$$

This is the principal but not the only mechanism which passes force from the wheel to the ground, the above ignores the effects of the flexure stiffness of the carcass itself, in addition to supporting the tension forces as outlined, the side-walls also have some bending resistance which can resist small wheel loads without any internal air pressure.

## Suspension action

In performing this function the pneumatic tyre is the first object that feels any road shocks and so acts as the most important element in the machine's suspension system. To the extent that, whilst uncomfortable, it would be quite feasible to ride a bike around the roads, at reasonable speeds with no other form of bump absorption. In fact rear suspension was not at all common until the 1940s or 50s.

Whereas, regardless of the sophistication of the conventional suspension system, it would be quite impractical to use wheels without pneumatic tyres, or some other form of tyre that allowed considerable bump deflection. The loads fed into the wheels without such tyres would be enormous at all but slow speeds, and continual wheel failure would be the norm.



**This photo shows a front wheel just after hitting a brick. The superimposed white line shows, what would have been, the un-deformed shape of the tyre, demonstrating the bump absorbing properties of pneumatic tyres. The rim forces would have been destructive without this cushioning.**

A few figures will illustrate the point:-- Assume that a bike, with a normal size front wheel, hits a 25 mm, sharp edged bump at 190 km/h. This not a large bump.

With no tyre the wheel would then be subject to an average vertical acceleration of approximately 1000 G. (the peak value would be higher than this). This means than if the wheel and brake assembly had a mass of 25 kg. then the average point load on the rim would be 245 kN. (about 25 tons). What wheel could stand that? If the wheel was shod with a normal tyre, then this would have at ground level, an initial spring rate, to a sharp edge, of approx. 17-35 N/mm., but rising as the tyre wraps around the step.

The maximum force then transmitted to the wheel for a 25 mm. step would be about 400 - 800 N. i.e. less than three hundredths of the previous figure, and this load would be more evenly spread around the rim. Without the tyre the shock loads passed back to the sprung part of the bike would be much higher too. The vertical wheel velocity would be very much greater, and so the bump damping forces, which depend on wheel velocity, would be tremendous. These high forces would be transmitted directly back to bike and rider.

The following five charts show some results of a computer simulation of vertical accelerations and displacements on a typical road motorcycle, and illustrate the tyre's significance to comfort and road holding. The simulated bike is traveling at 100 km/h. and the wheels hit a 25 mm. or 0.025 metre high step. Note that the time scales vary from graph to graph, to highlight the most significant features. To keep the graphs as uncluttered as possible it is assumed that both wheels hit the step at the same time.

Three cases are considered:

- With typical vertical tyre stiffness and typical suspension springing and damping.
- With identical tyre properties but with a suspension spring rate of 100 X that of the previous.
- With tyre stiffness 100 X the above and with normal suspension springing.

So basically we are considering a typical case, another case with almost no suspension springing and the final case is with an almost rigid tyre.

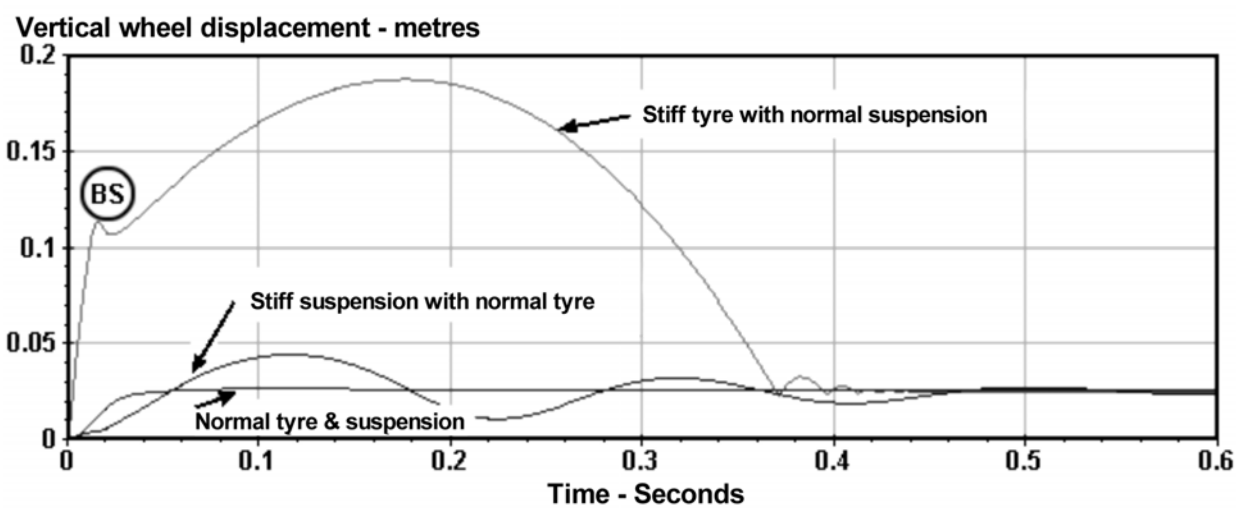



Fig 2.3 **This shows the vertical displacement of the front wheel. There is little difference between the maximum displacements for the two cases with a normal tyre, for a small step the front tyre absorbs most of the shock. However, in the case of a very stiff tyre, the wheel movement is increased by a factor of about 8 times. The tyre leaves the ground in this case and the landing bounces can be seen around 0.4 seconds. The sudden change of slope at the point marked (BS) is when the forks bottom out against the bump stops.**

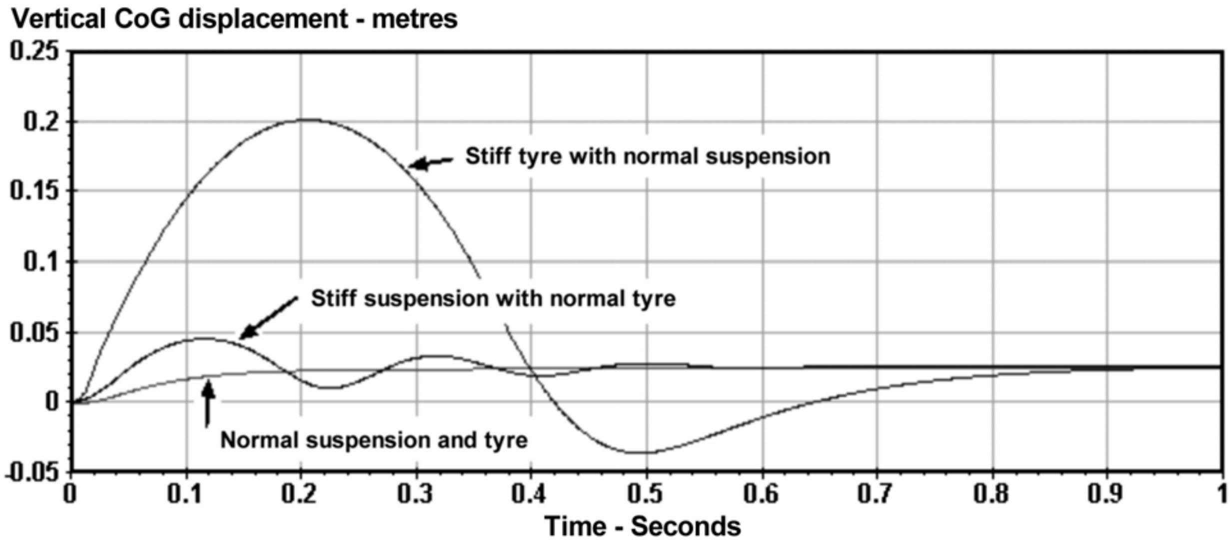



Fig 2.4 **These curves show the vertical movement of the CoG of the bike and rider. As in Fig 2.3 it is clear that the stiff tyre causes much higher bike movements, to the obvious detriment of comfort.**

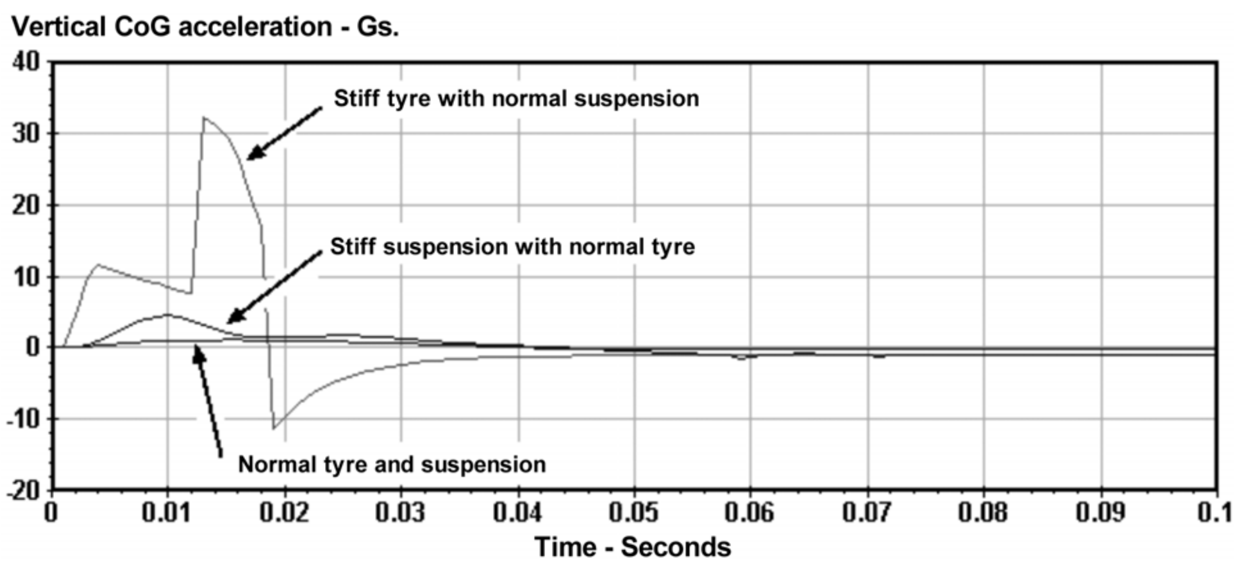



Fig 2.5 **Demonstrating the different accelerations transmitted to the bike and rider, these curves show the vertical accelerations at the CoG. The sudden step in the stiff tyre case at 0.012 seconds is due to the forks bottoming against the bump stops. (In this case, doubling the spring rate (curves not shown), prevented the bottoming out and the maximum acceleration was reduced to 12 Gs. from 32.)**

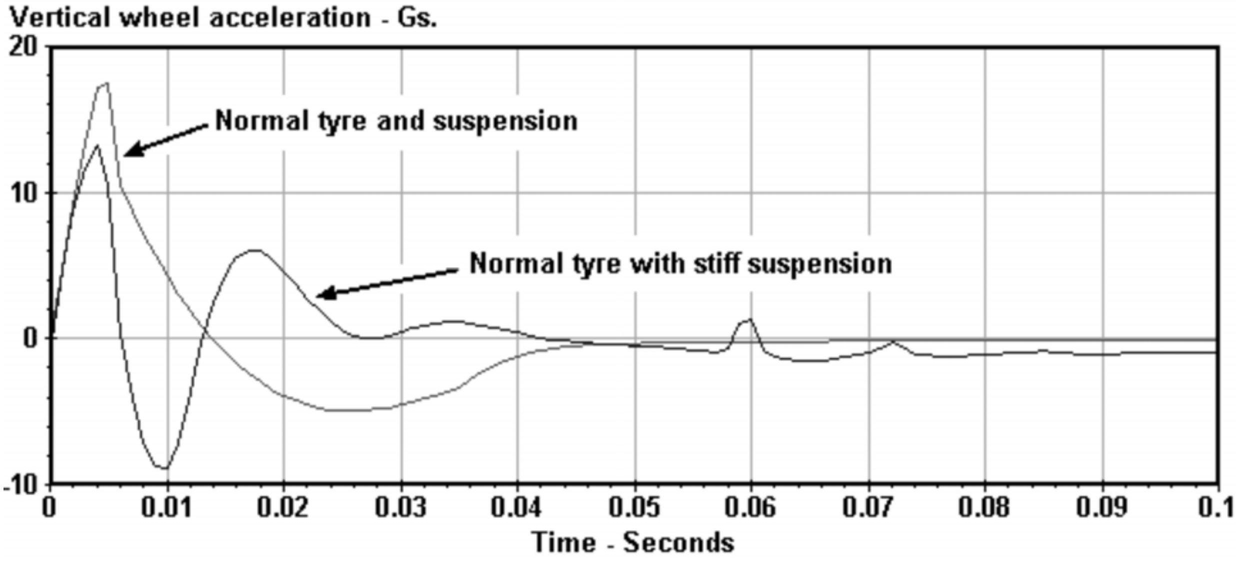



Fig 2.6 **Front wheel vertical acceleration for the two cases with a normal tyre. The early part is similar for the two cases, the suspension has little effect here, it is tyre deflection that is the most important for this height of step.**

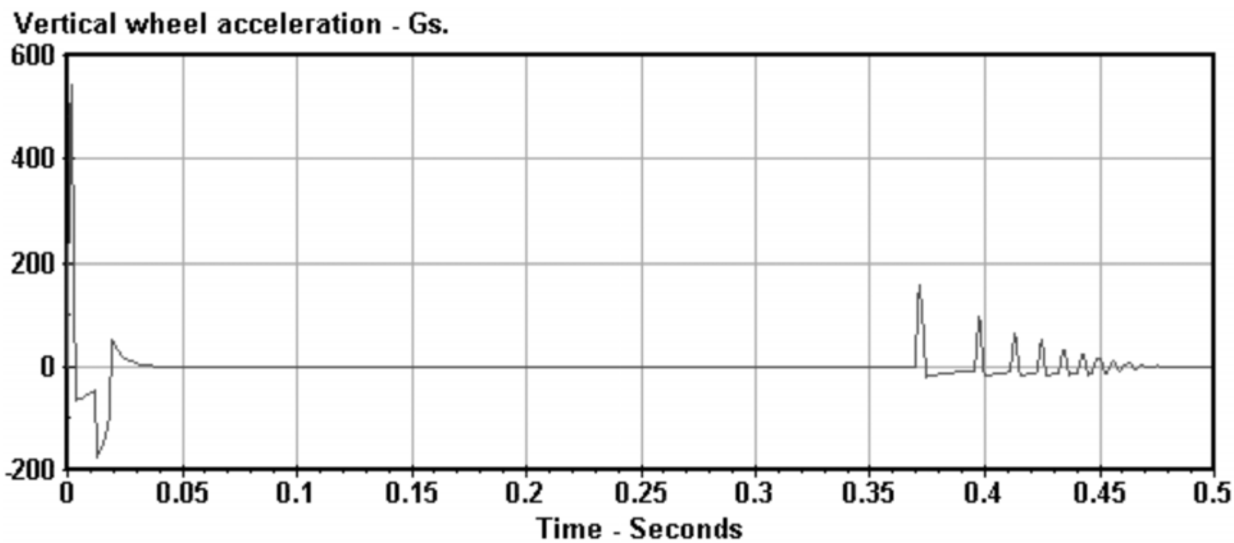



Fig 2.7 **As in Fig 2.6 this curve is of the wheel acceleration, but for the stiff tyre case, the values of the normal case are overwhelmed by the stiff tyre case, with a peak value of close to 600 G compared with less than 20 G. above. This high acceleration would cause very high structural loading.. Again note the effects of the landing bounces at 0.4 seconds.**

As the tyre is so good at removing most of the road shocks, right at the point of application, perhaps it would be worth while to consider designing it to absorb even more and eliminate the need for other suspension. Unfortunately we would run into other problems. We have all seen large construction machinery bouncing down the road on their balloon tyres, sometimes this gets so violent that the wheels actually leave the ground. A pneumatic tyre acts just like a spring, and the rubber acts as a damper when it flexes, but when the tyre is made bigger the springing effect overwhelms the damping and we then get the uncontrolled bouncing. So there are practical restraints to the amount of cushioning that can be built into a tyre for any given application.

Fig 2.8. **Not even the shock absorbing properties of the tyre are sufficient to prevent rim damage in all cases.**

**Normally such severe damage is restricted to crash situations.**

(Motorcycle Consumer News)

## Tyre stiffness or spring rate

The springing characteristics mentioned above are largely affected by the tyre inflation pressure, but there are other influences also. Carcass material and construction and the properties and tread pattern of the outer layer of rubber all have an effect on both the springing properties and the area in contact with the ground (contact patch). Under and over inflation both allow the tyre to assume non-optimum cross-sectional shapes, additionally the inflation pressure exerts an influence over the lateral flexibility of a tyre and this is a property of the utmost importance to motorcycle stability. Manufacturers' recommendations should always be adhered to.

It is often thought that the spring rate of a tyre is mainly due to compression of the air in a similar way to the action of a pneumatic suspension unit (described in chapter 6). This is not so, as a tyre is compressed there is only a minimal change in internal volume of the tyre and so it follows that there is very little pressure change, nowhere near enough to explain measured stiffness. It is the shape variation with load, as shown in figs. 2.1 and 2.2 that produce the force variation seen as the spring rate.

Fig. 2.9 shows the influence of tyre pressure on the vertical stiffness of an inflated tyre, when loaded on a flat surface. These curves are from actual measured data. Note that the spring rate is close to linear over the full range of loading and varies from 14 kgf/mm. at 1.9 bar pressure to 19 kgf/mm. at 2.9 bar.

The effective spring rate when the tyre is loaded against a sharp edge, such as a brick, is considerably lower than this, and is more non-linear due to the changing shape of the contact area as the tyre "wraps" around the object.

This spring rate acts in series with the suspension springs and is an important part of the overall suspension system. An interesting property of rubber is that when compressed and released it doesn't usually return exactly to its original position, this is known as hysteresis. This effect is shown only for the 1.9 bar. case, the curve drawn during the loading phase is not followed during the unloading phase. The area between these two curves represents a loss of energy which results in tyre heating and also acts as a form of suspension damping. In this particular case the energy lost over one loading and unloading cycle is approximately 10% of the total stored energy in the compressed tyre, and is a significant parameter controlling tyre bounce. These damping characteristics vary with loading frequency.

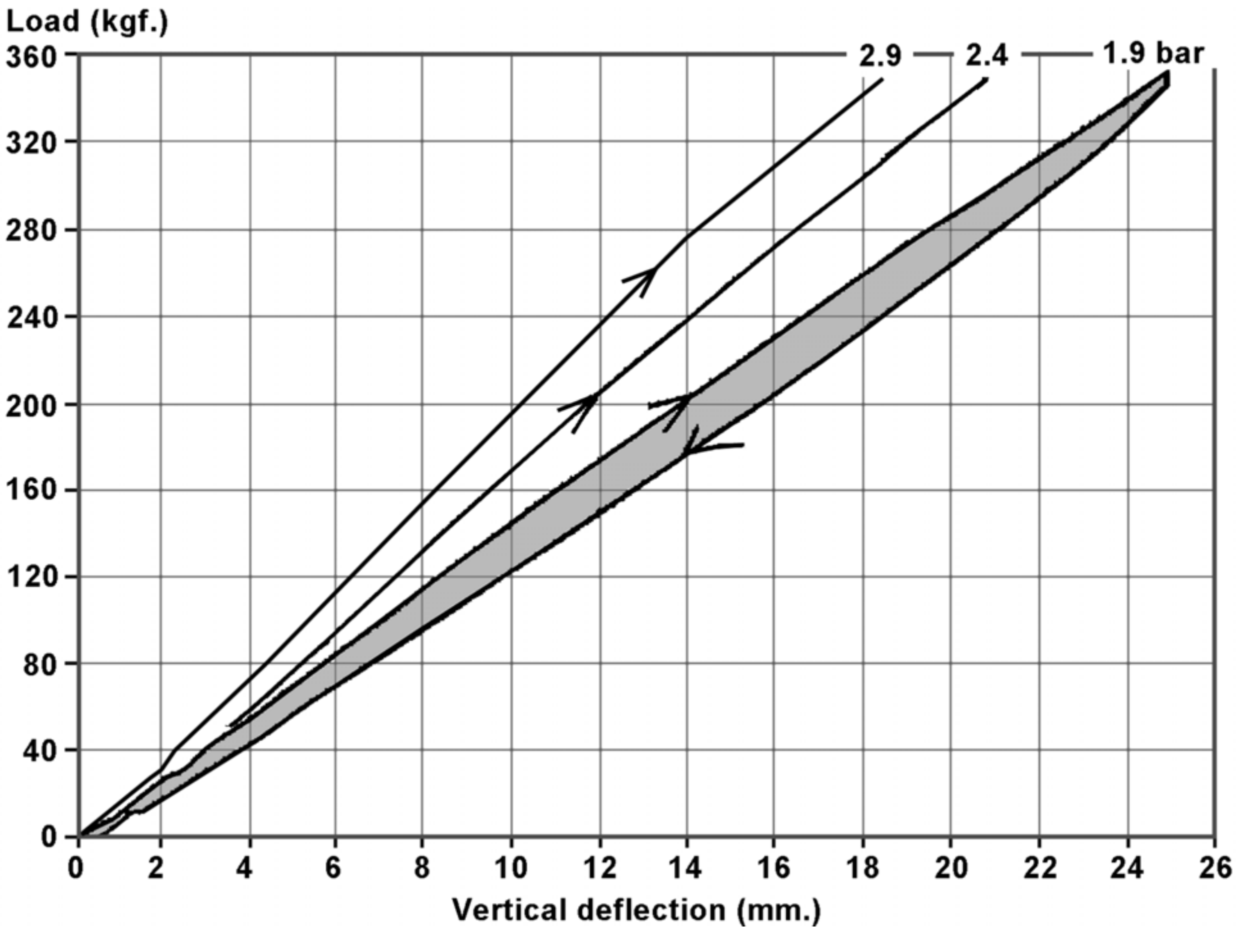



Fig. 2.9. **Vertical stiffness of a standard road tyre against a flat surface at different inflation pressures. This data is from an Avon Azaro Sport II 170/60 ZR17. The upward arrows indicate the compression of the tyre and the 2nd line with the downward arrow (shown only at 1.9 bar for clarity) shows the behaviour of the tyre when the load is released. The shaded area between the two lines represents a loss of energy called hysteresis. This acts as a source of suspension damping and also heats the tyre.** (From data supplied by Avon Tyres.)

Fig. 2.10 shows the lateral stiffness of the same tyre measured at two different pressures. In both cases the tyre was loaded vertically with its maximum rated capacity of 355 kgf. The lateral spring rate is less than half that of the vertical rate at 7.7 and 7.3 kgf/mm. at 2.9 and 2.5 bar respectively. It is interesting to note that at the higher pressure the tyre saturates or loses adhesion at the lower figure of 460 kgf. compared to 490 kgf. at the lower pressure.

Saturation is indicated when the curve more or less becomes horizontal, this is when the tyre cannot support any more lateral force and it displaces or slides sideways, with an approximately constant force. The contact patch area and pressure produced at the lower air pressure has allowed more static grip. However, these tests were done with the artificial case of an upright and non-rotating wheel on a serrated steel surface and hence it would be risky to extrapolate this grip characteristic to a moving machine on a normal road. Although not shown, the lateral deformation would also be subject to some hysteresis and this damping and the lateral flexibility exert an important influence over the weave stability of the motorcycle.

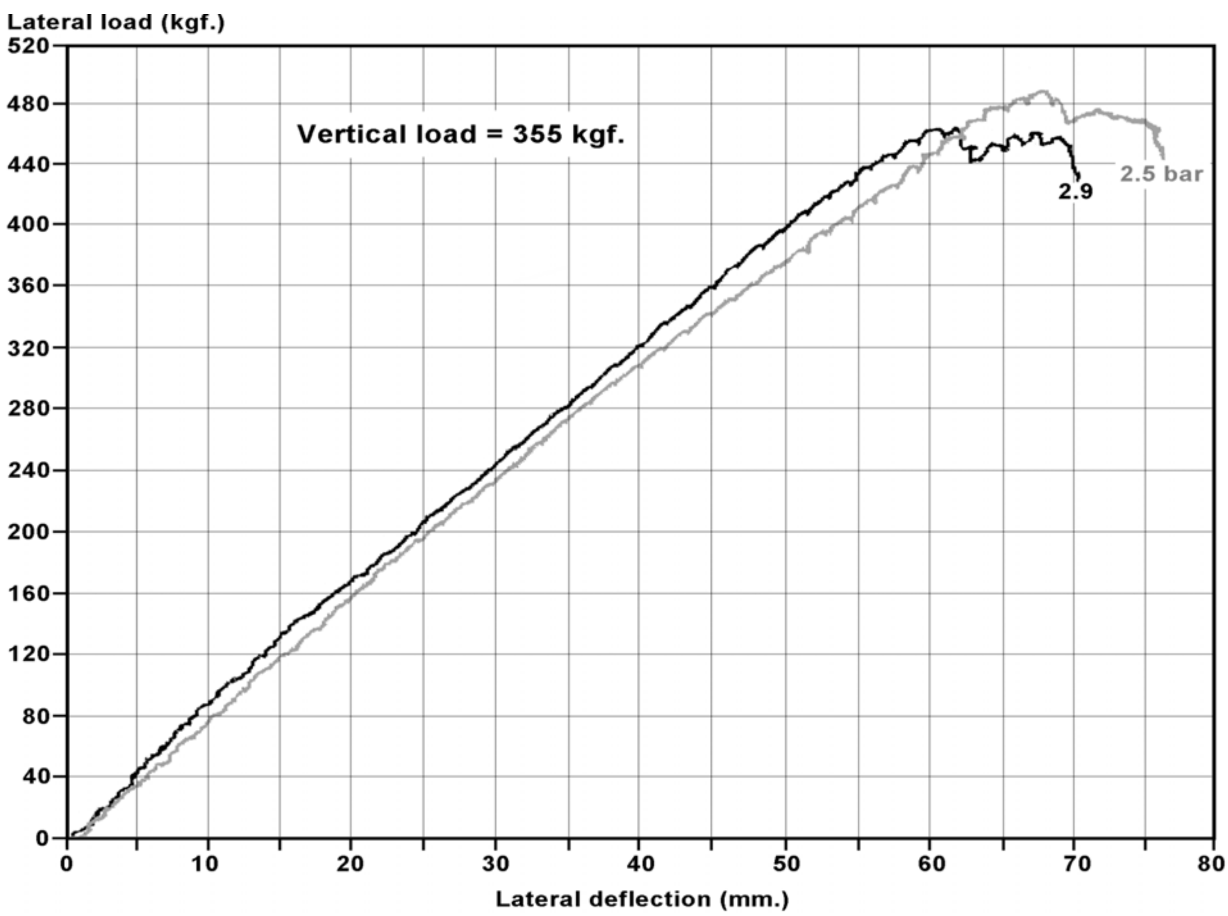



Fig. 2.10. **Lateral stiffness of the same tyre shown in fig. 2.9. The vertical load was constant at 355 kgf. and the wheel was kept vertical. As expected the tyre is somewhat stiffer with the higher inflation pressure but loses grip or saturates at the lower lateral load of 460 kgf. compared to 490 kgf. at the lower pressure.** (From data supplied by Avon Tyres.)

## Contact area

The tyre must ultimately give its support to the bike through a small area of rubber in contact with the ground, and so “contact patch area = vertical force ÷ average contact patch surface pressure”. This applies under ALL steady conditions.

The contact patch surface pressure is NOT however, the same as the inflation pressure, as is sometimes claimed. They are related but there are at least four factors which modify the relationship. Carcass stiffness, carcass shape, surface rubber depth and softness, and road surface compliance. If we have an extremely high carcass stiffness then inflation pressure will have a reduced influence.

Let’s look at this in a little more detail and see why:

If a tyre was made just like an inner tube, that is from quite thin rubber and with little stiffness unless inflated, then the internal air pressure would be the only means to support the bike’s weight. In this case the contact patch pressure would be equal to that of the internal air pressure. For an air pressure of 2 bar and a vertical load of 1.0 kN. the contact area would be 5003 sq.mm. If we now increased the air pressure to say 3 bar the area would fall to 3335 sq.mm.

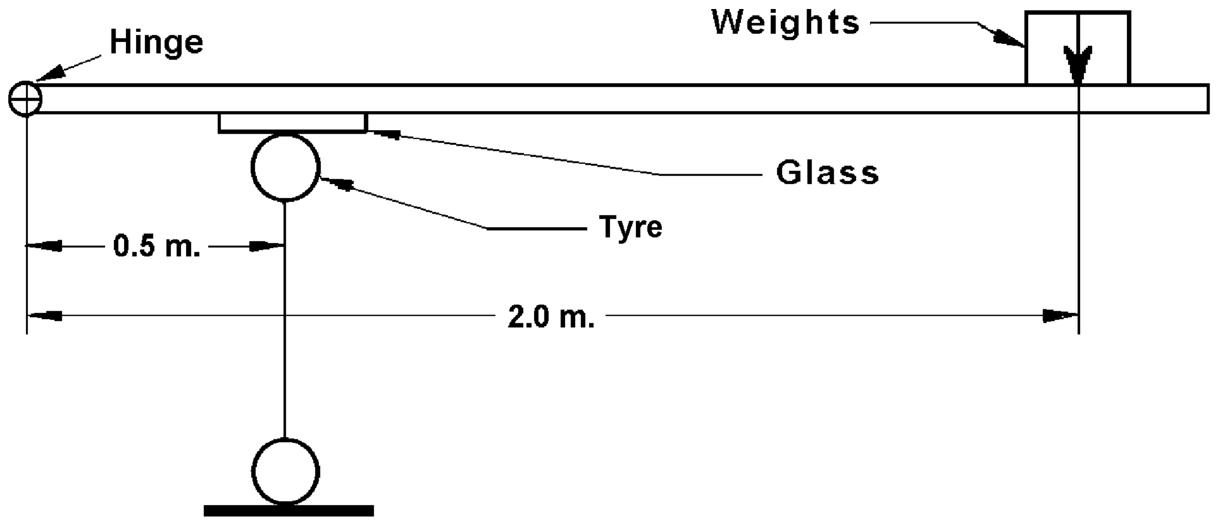
Let’s now imagine that we substitute a rigid steel tubular hoop for our rim and tyre, the area in contact with hard ground will be quite small. If we now inflate the hoop with some air pressure, it doesn’t take much imagination to see that, unlike the inner tube, this internal pressure will have a negligible effect on the external area of contact. Obviously, a tyre is not exactly like the steel hoop, nor the inner tube, but this does show that the carcass rigidity can reduce the contact surface area as calculated purely from inflation pressure alone.

On the other hand, let’s now imagine that we cover the previous steel hoop with a layer of thick soft rubber. Now, the actual contact area will be considerably increased and the average contact patch pressure will be reduced. Substitute this mental picture back to a real tyre and we see that the tread layer of rubber will give us a greater contact area and lower contact pressure than that of the inflation pressure alone. It is this compliance of the surface rubber that gives us more contact with the road when we increase tyre width and diameter, but this must be balanced against the opposing effects of the carcass stiffness. Radial and bias or cross ply tyres exhibit quite different characteristics in this regard.

The properties of the road surface are also important, a soft surface, mud and sand for example, will give support over a wider area of the tyre and so reduce the contact pressure. On a hot day with softened tarmac, even a normal road will deflect significantly enough to affect the contact patch.

To get a feeling for the degree of departure of the contact patch pressure from the inflation pressure, consider a completely flat tyre, in this case the rubber area will probably be no more than 3 or 4 times, at most, the area when inflated correctly. Based on the notion that rubber pressure = inflation pressure, we would expect the rubber area to be much higher, infinite in fact. Another extreme case to consider: imagine a knobbly tyre with very few knobs such that only one knob supports the bike. In this example the rubber pressure is simply the weight divided by the area of the one knob, this is regardless of the inflation pressure. These are extreme examples of course but should still demonstrate the lie in the proposition that rubber contact pressure = inflation pressure.

The following describes some simple measurements made by the author to determine the actual relationship between load, inflation pressure and contact area for one particular tyre. Various weights were placed on the end of a beam, which loaded the tyre via a thick plate of glass. The beam was arranged to apply the load to the tyre with a 4:1 leverage. So a 25 kgf. weight would load the tyre with 100 kgf. By tracing over the glass onto transparent paper the contact area was determined.




**Measurement setup. Various weights were placed on the end of a beam, which loaded the tyre via a thick plate of glass. The beam was arranged to apply the load to the tyre with a 4:1 leverage. So a 25 kgf. weight would load the tyre with 100 kgf. By tracing over the glass the contact area was determined.**

2 sets of tests were done. For the first, the tyre inflation pressure was kept constant at 2.4 bar and the tyre load was varied between 178 and 1210 N. (allowing for the weight of the glass and wooden beams). Secondly, a constant load of 1210 N. was maintained whilst varying the inflation pressure between 2.4 to 1 bar.

Even with a generous allowance for experimental error the effects are clear. The graphs in fig. 2.12 show that the results appeared to fit reasonably well to a smooth line, there wasn't much scatter. Point (1) on the curve with constant inflation pressure, shows how the actual contact patch pressure is lower (just over half) than the inflation pressure, or in other words the contact area is greater. This is due to the rubber surface compliance, thus this is more important at low vertical loads, whereas carcass stiffness became more important as the load rose as shown by points (3) to (6) where the actual contact pressure is higher than the air pressure, i.e. reduced area of contact.

The second graph shows how the carcass stiffness helps to support the machine as the air pressure is reduced, the contact patch pressure being considerably higher than the inflation pressure. It looks as though the two lines would cross at an air pressure of about 3.5 bar. (although this was not tested by measurement), at which point the surface rubber compression will assume the greatest importance. This is as per the steel hoop analogy above.

We can easily see the two separate effects of surface compliance and carcass stiffness and how the relative importance of these varies with load and/or inflation pressure.

These tests were only done with one particular tyre, other types will show different detail results but the overall effects should follow a similar pattern.

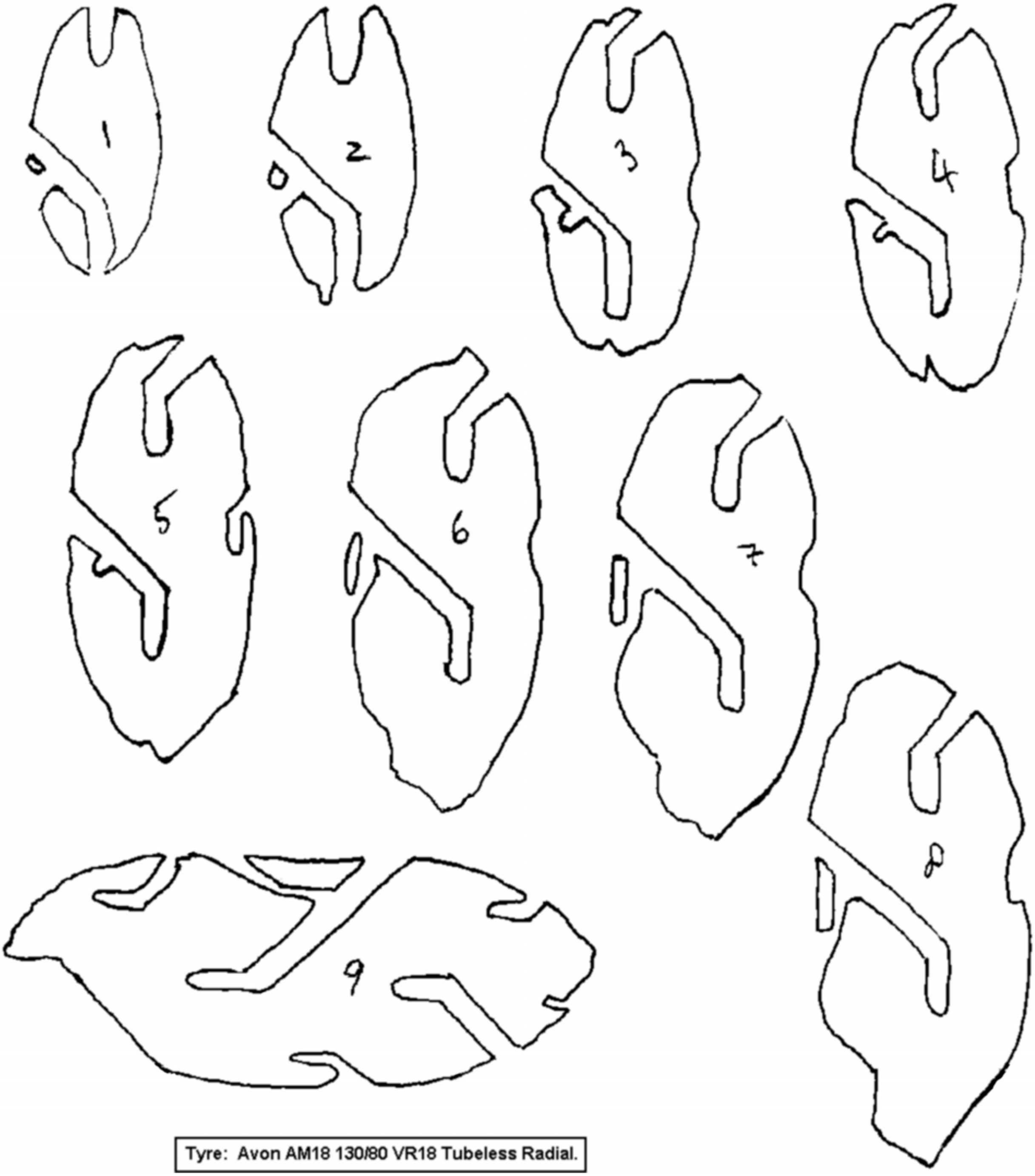
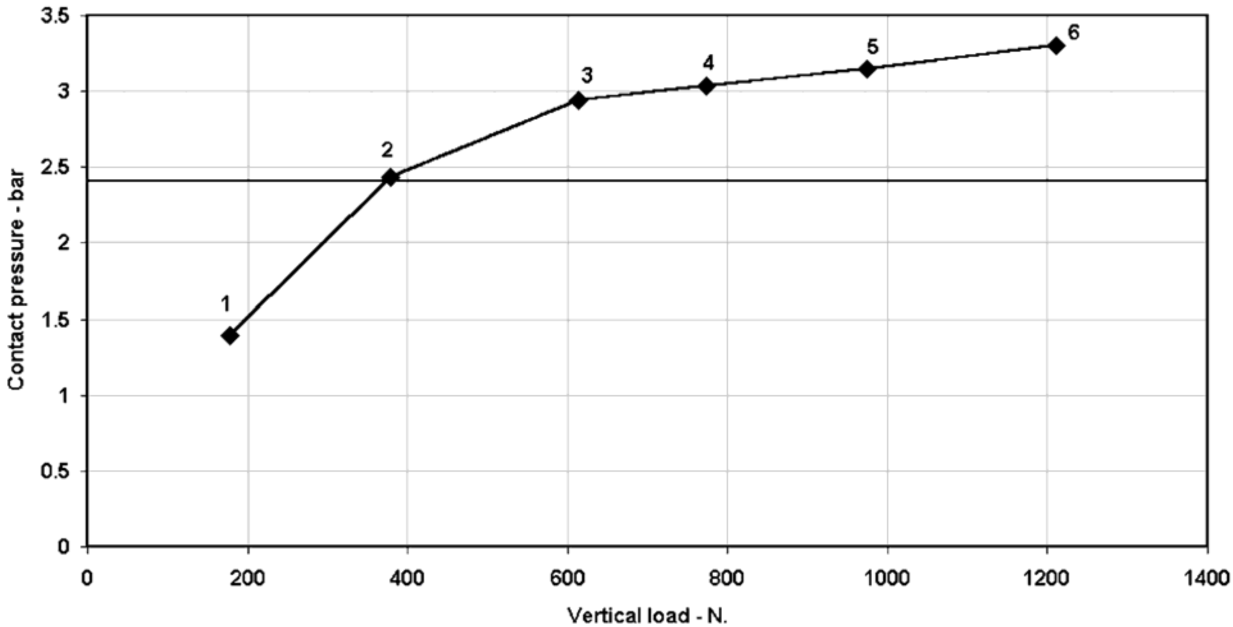



Fig. 2.11. **Tracings of tyre footprint for different loads and pressures. The numbers relate to the data points in fig. 2.12.**



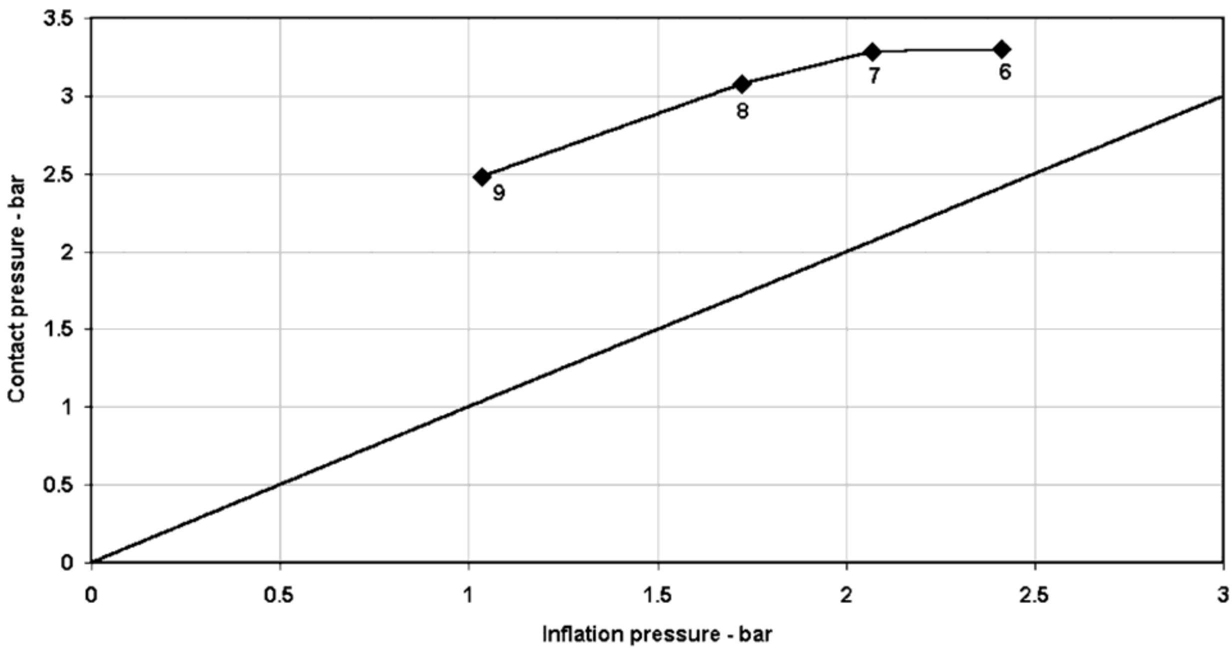




Fig. 2.12 **The top plot shows the measured contact patch pressure at various wheel loads for a constant inflation pressure of 2.4 bar. The lower curves show the contact pressure at various inflation pressures for a fixed load of 1210 N. The numbers at the data points correspond with the contact area tracings in the previous sketch, fig. 2.11. The plain line on each plot shows the case of the contact patch pressure being equal to the inflation pressure.**

## Area when cornering

Does cornering affect tyre contact area?

Let's assume a horizontal surface and lateral acceleration of 1G. Under these conditions the bike/rider CoG will be on a line at 45° to the horizontal and passing through the contact patch. There will a resultant force acting along this line through the contact patch of 1.4 times the supported weight.

This force is the resultant of the supported weight and the cornering force, which have the same magnitude, in this example of a 45° lean. The force normal to the surface is simply that due to the supported weight and does NOT vary with cornering force. The cornering force is reacted by the horizontal frictional force generated by the tyre/road surface and this frictional force is "allowed" by virtue of the normal force.

Therefore, to a first approximation cornering force will NOT affect the tyre contact area, and in fact this case could be approximated to, if we were just considering the inner tube without a real world tyre. However in reality, the lateral force will cause some additional tyre distortion to take place at the road/tyre interface and depending on the tyre characteristics, mentioned above, the contact area may well change.

Another aspect to this is of course the tyre cross-sectional profile. The old Dunlop triangular racing tyre, for example, was designed to put more rubber on the road when leant over, so even without tyre distortion the contact patch area increased, simply by virtue of the tyre shape and lean angle.

## Friction (grip)

Since they constitute our only contact with the ground, the tyres are crucial in providing the grip to transmit driving, braking and cornering forces. The amount of grip depends on the weight supported by each tyre, increasing the weight increases the grip. The ratio between the maximum possible grip and the vertical load is called the coefficient of friction. However, this coefficient is not constant but usually decreases with vertical load (i.e. increased contact-patch pressure). Reliable data is hard to come by but the reduction is of the order of 10% for a doubling of load. A further complication is that this relationship is not linear. This has far-reaching implications and is one reason for the general increase in tyre section on racing and sports machines, because, for a given wheel load, the bigger the section the lower the contact-patch pressure and so the greater the coefficient of friction, hence the grip on the road. Heavy braking (when as much as 100 per cent of the total weight may be supported on the front tyre) provides an interesting example of the effects of the relationship between load and friction coefficient.

The forward weight transfer increases the pressure on the front contact patch and reduces that on the rear, so reducing the coefficient at the front and increasing it at the rear. Therefore, the tyre with the reduced coefficient of friction is carrying most of the weight and vice versa. Hence, the total frictional force available for stopping is less than it would be on a machine with a smaller weight transfer. In other words, for maximum braking we need a long wheelbase and a low mass centre. This assumes that the rider is able to balance the two brakes to exploit available grip at each end to the maximum. In practice few riders can do this well and many racing and sports riders tend to favour only the front brake, in which case the greater weight transfer from a high CoG is beneficial. On the other hand many unskilled road riders use only the rear brake giving away most of the braking potential. This is elaborated upon in chapter 12, fig. 12.6.

Another reason for the trend towards larger tyre sections is, of course, the relentless growth in the weight and power of our machines, which would otherwise cause excessive tread wear and high temperatures.

To balance the grip forces with the individual loads, different tyre sections are used on the front and rear wheels. Since a motorcycle with the rider on board usually has a rearward weight bias, a larger tyre section is used at the rear. If we over-tyre at the rear (or under-tyre at the front) then the coefficient of friction at the front will generally be less than that at the rear. Alternatively, if we over-tyre at the front (or under-tyre at the rear) then the effect will be opposite. In either case, the end with the smaller coefficient of friction will lose adhesion first, at a lower cornering speed than if that tyre section was increased to balance. Compromise is unavoidable here because a change in throttle opening or in wheel loading (through carrying a passenger or heavy luggage) can drastically alter the front/rear requirement.

The profiles and lateral flexing characteristics also need careful balancing front to rear. Indeed, both the tyre and bike manufacturers undertake considerable testing to determine the best blend. A combination of tyres that gives excellent results on one model may prove lethal on another. Relative to their weight and power, motorcycles have smaller tyre sections than cars. This derives chiefly from our need to bank while cornering, in which case wide tyres may impair handling and stability, even though grip may be improved. Figure 2.13 shows how the contact patch moves away from the centre plane of the wheel or the steering axis as the machine is banked. A greater angle of lean is necessary to balance centrifugal force, and this may require a slightly higher centre of gravity to restore cornering clearance.

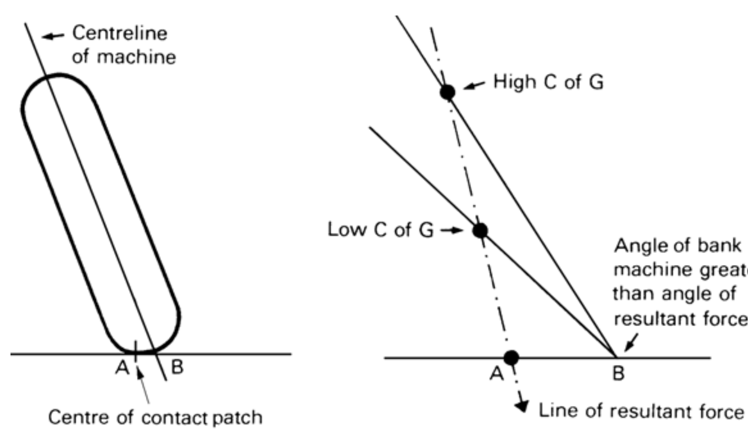



Fig. 2.13 **This shows the different angles of lean required with low and high centres of gravity, due to the width of the tyre. This effect opposes that shown Fig. 3.29.**

This adverse effect of large tyres is not confined to leaning. Even in straight-ahead riding, they can introduce other problems.

Because the forces fed into the front tyre are offset from the steering axis, steering forces are introduced which have to be counteracted by trail effects and the rider, usually requiring more physical effort. As figure 2.14 shows, a road disturbance such as a stone can cause a couple tending to steer the wheel to that side. A wide tyre is found by more stones and produces a higher couple.

Large tyres increase the unsprung mass too, to the detriment of roadholding. Larger tyres also increase precessional forces.

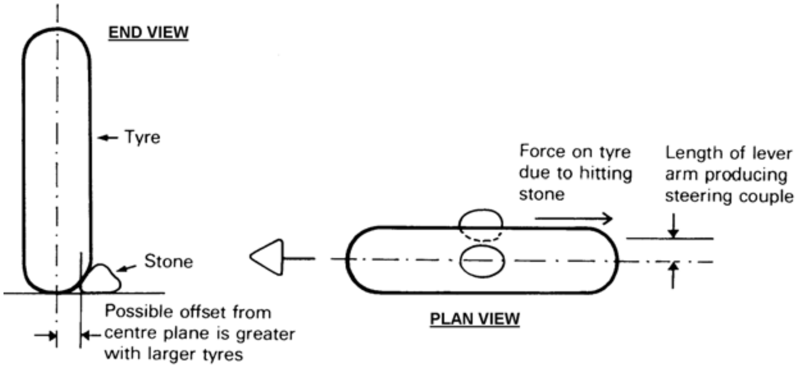



Fig. 2.14 **Steering torque produced by road disturbance.**

## Braking & driving

As the brake is applied a torque is transmitted through the wheel to the contact patch, where it manifests itself as a linear horizontal force at the road surface. The road pushes backward on the tyre and with equal intensity the tyre pushes forward on the road. Under power the situation is similar except that the directions of the forces are reversed.

## Cornering

It is when we wish to study the action of cornering forces and how these are generated that things become a little cloudy. As we negotiate a curve, the tyre must produce a centripetal forces to counter the bike's desire to travel on in a straight line.*** The road must react with the tyre to produce a force toward the centre of the curve.

*(*** This desire is due to the law of **"Conservation of Momentum"** which makes it appear that there is a force trying to push the bike outward, this imaginary force is commonly known as **centrifugal** force. This is explained in more detail in the appendix. **Centripetal** means inward toward the centre and **centrifugal** means away from the centre.)*

The detail mechanisms of how this actually happens is less obvious and more involved than may appear at first sight. Those who drive as well as ride may have wondered why it is necessary to turn the front wheels more on a car, than on a bike, to take the same corner at the same speed.

The essential difference is that a bike must lean inwards when cornering to maintain balance, whereas a car remains substantially upright.

## Mechanisms of grip

The detail mechanism of grip (what I call the micro explanation) involves a study of the detail deformation of the tyre around the contact area. However, this is very complicated and not yet fully understood, but we can get a good idea of the tyre's influence on handling and roadholding by using empirical methods derived from global tyre tests. I call this the macro explanation, and the following is based on that.

At all but the slowest of speeds the principal mechanism of grip between a tyre and a normal paved road involves some form of slippage between the tyre and the road. Moto-X machines driving through a corner in a deep rut naturally derive at least part of their cornering force in a different way. This idea of the need for slip may come as a surprise to many people as it is commonly thought that slip only takes place when riding on the limit and “drifting” the machine. The truth is, that at even moderate braking, accelerating or cornering loads the tyre is experiencing some slip over the surface. For braking and accelerating, tests show that we get the maximum amount of grip when the tyre is slipping by about 10% or more. That means that under hard acceleration the surface velocity of the tyre will be over 10% higher than the road speed of the bike. Conversely, under braking the wheel speed will be some 10% less than the road speed. If we plot a graph of the degree of grip against some slip parameter we can see that the relationship becomes highly non-linear as the tyre gets close to the limit, but the overall shape of the curve will be similar for both traction and cornering.

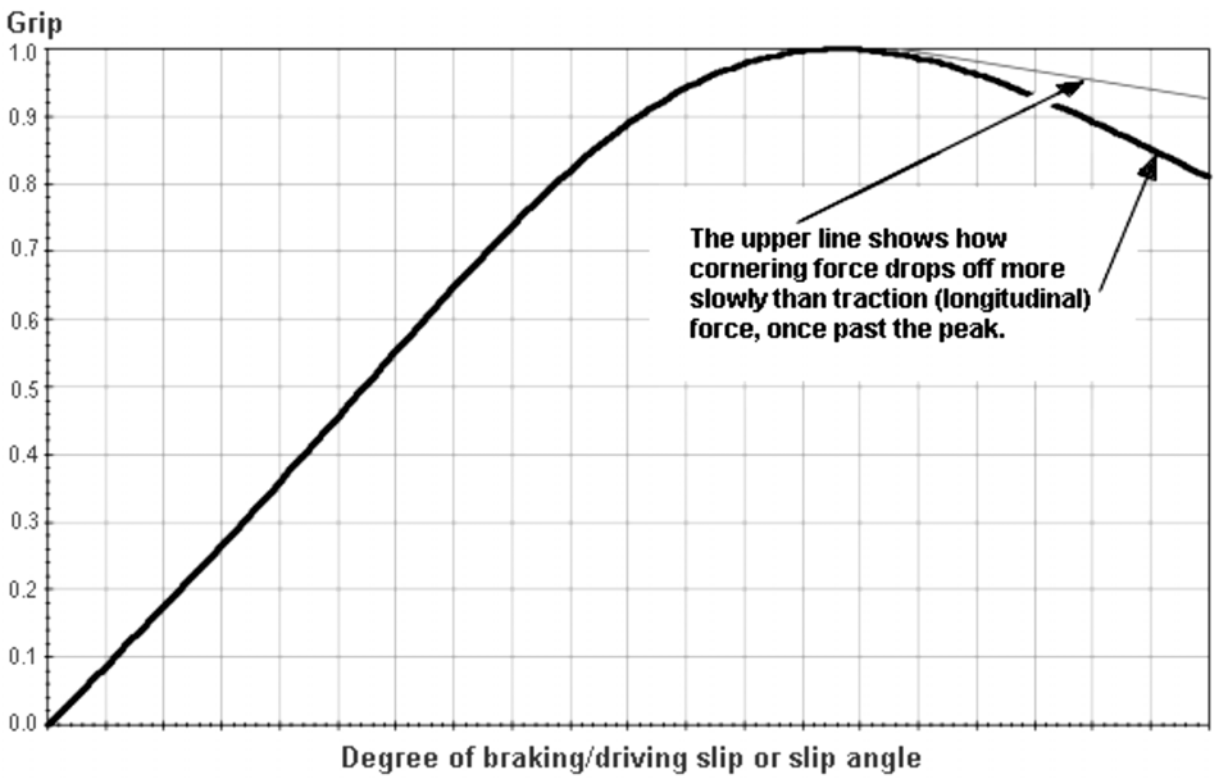



Fig. 2.15 **This curve is the general shape to be expected when considering the frictional force of a tyre compared to some slip parameter. For braking and driving this slip parameter is expressed as a percentage of the non-slipping wheel speed, and the peak in the graph occurs around 10 - 20%. For the steering case the slip is expressed in terms of a slip angle. With a motorcycle we also have to consider the effects of lean generated force.**

The initial slope of the linear part of the above graph is known by the term “stiffness”. When considering driving or braking we refer to traction or longitudinal stiffness. If we’re looking at the steering situation then we call it the cornering or steering stiffness. Although these values are not a true stiffness in a structural sense, it is the common way of expressing this tyre characteristic because the units are the same as structural stiffness. That is, a force related to some displacement.

## Slip angle

Consider a wheel, held upright as on a car, following a curved path as in fig. 2.16. If this wheel is aligned with the direction of the curve at any particular point on it (i.e. pointing in the direction of travel) then the wheel will tend to go straight on along a tangent to the curve and will generate no cornering force.

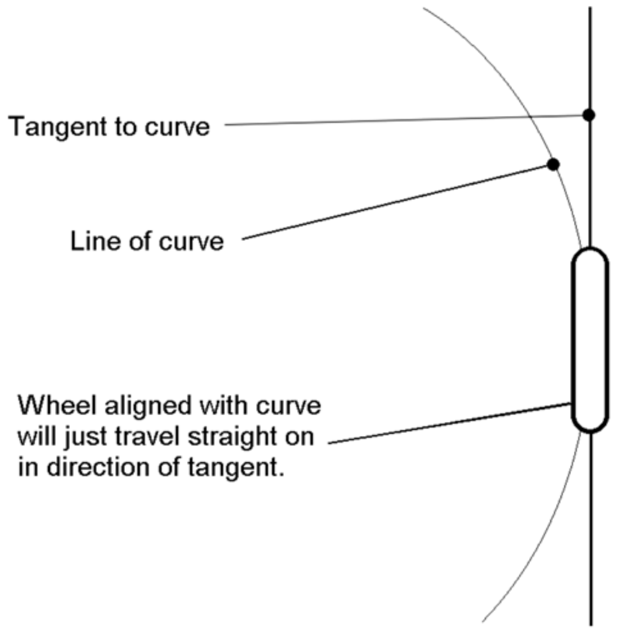



Fig. 2.16 **With an upright wheel if the plane of the tyre is aligned with the direction of travel it will continue on in the original direction. This is not surprising as we all know that we must steer or lean the wheel to make a vehicle turn.**

Clearly then, something else must be done in order to persuade the tyre/road junction to generate the required cornering or centripetal force, to direct the tyre toward the centre of the curve. On a car or other self balancing vehicle this is done by turning the wheel in more than the line of the curve. The angle between the plane of the tyre and the direction of the tangent to the curve is known as the 'slip angle'.

Reference to fig.2.17 will help explain how this works.

As the wheel is now no longer travelling exactly in the direction in which it is pointing, we can resolve its velocity tangent to the curve (this is the actual road velocity) into components aligned with the wheel and at right angles to it. This means that the peripheral tyre speed will be slightly less than the road speed around the turn but there is now a sideways velocity to the tyre, i.e. it is sliding sideways. This lateral movement produces a force at right angles to the wheel direction. The magnitude of this force depends on the amount of slip angle, increasing up to about 15° and then falling off, that's when the driver has lost it. There is no longer sufficient cornering force generated to support the required force necessary to continue around the corner at the desired speed.

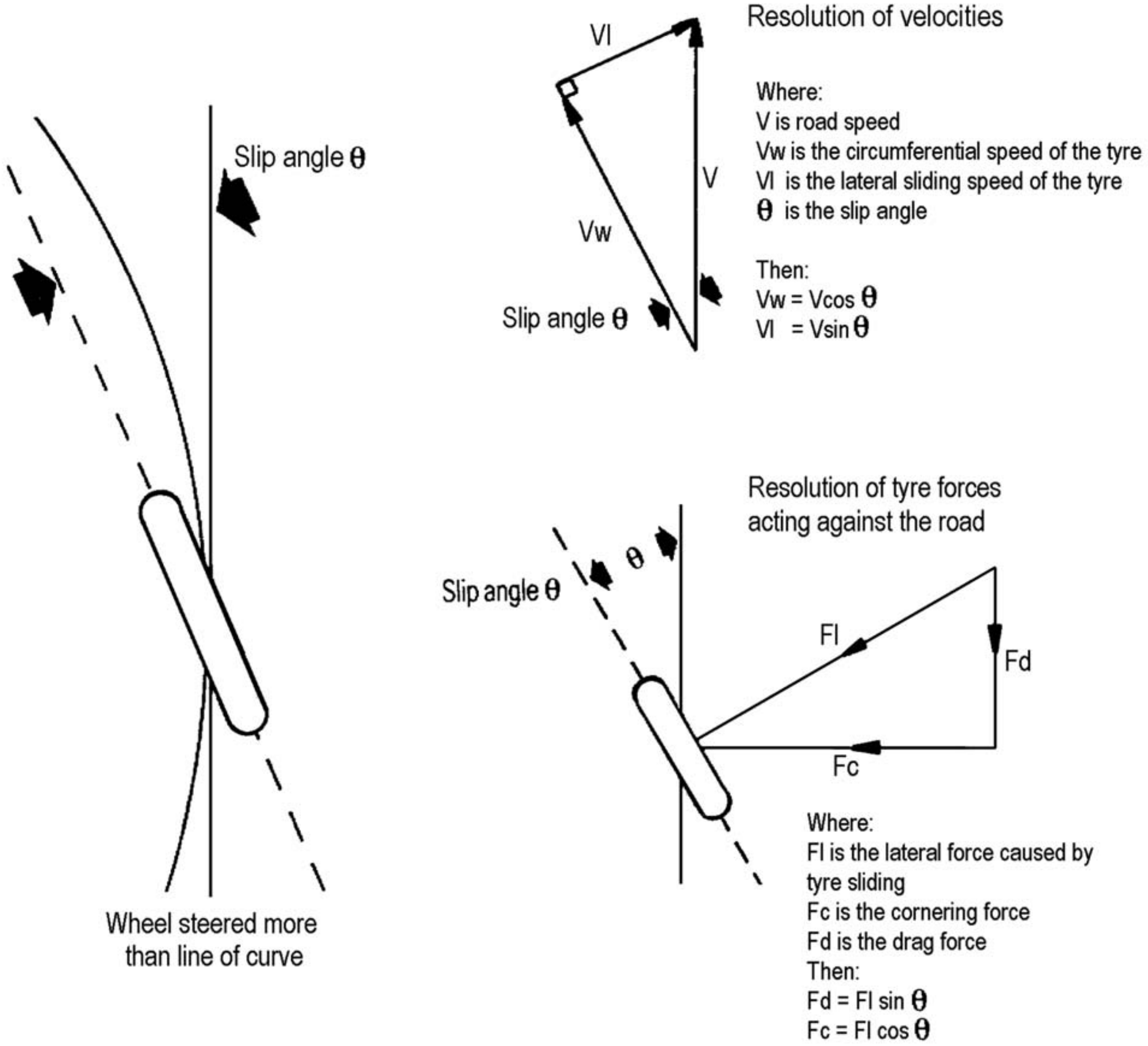



Fig. 2.17 **This shows how a slip angle results in a lateral sliding velocity, which causes a force lateral to the wheel's direction. This force can be resolved in two components. One, normal to the direction of travel which balances the cornering force and a second which is aligned with the direction of travel. This force acts against the forward motion of the vehicle and hence is a drag force. With a car this drag force, for a given corner, sets a limit to the maximum possible speed achievable from a given engine power, even if the grip of the tyre is not exceeded.**

This wheel lateral force can be resolved into a component at right angles to the direction of travel, cornering force or centripetal force, and into one aligned with the direction of travel, a drag force. It is this drag force which causes a car to slow when driven hard around a bend under constant power. So we can now see what generates and controls the forces that cause a car to corner, and why the wheels must turn more than the amount just necessary to align with the direction of the curve. We must have some lateral slippage in order to generate the required lateral force.

## Camber force (thrust)

The previous section explains how steering a wheel generates the force necessary to force a vehicle to turn around a bend. However, bicycles and motorcycles must lean when taking a corner and this leaning also creates a lateral cornering force. In fact at all but the slowest of speeds and cornering accelerations this force will likely be the major contributor to the total cornering force, and the steering effects will just make up for the difference between the required cornering force and that provided by the lean. Hence, the degree of steering necessary on a motorcycle is much less than that required by a car. The lateral tyre force due to the tyre camber angle is known as camber thrust or camber force. Let's look at Fig. 2.18 to see how this force is created.

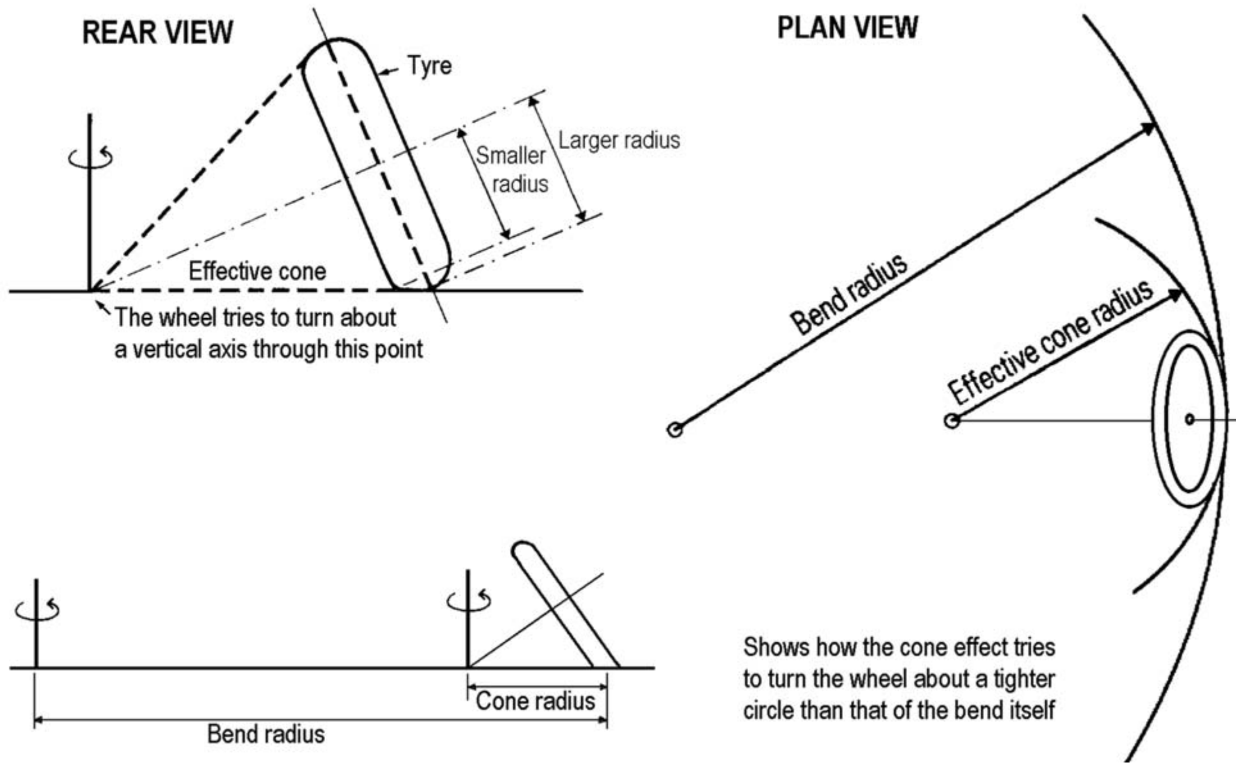



Fig. 2.18 **The top left sketch shows how the contact patch of the tyre flattens at an angle and effectively becomes a slice of a cone which tends to turn around the geometric apex of the cone. The other two diagrams show how this cone tries to turn a tighter circle than the actual bend radius.**

As the inside edge of the tyre is forced to adopt a smaller radius than the outer edge, then for a given wheel rotational speed, the inner edge would prefer to travel at a slower road speed, this happens if the wheel is allowed to turn about a vertical axis through the apex of the cone. Just as a solid cone on a table would, if given a push. If the bike was leaning over at 45° then for a normal size tyre the horizontal radius to the cone axis would be approximately 450 mm, an impossibly tight turn. However, we've seen before that **Conservation of Momentum** will want to make the bike go straight which tends to work against this desire to turn about the effective cone centre, these conflicting effects will form a balance where the actual corner radius described is considerably greater than the cone radius.

Another way to look at this is to imagine the bike trying to turn on the cone radius, with the above value of 450 mm., that would mean that the lateral acceleration required at 100 km/h would be 175 g, or in other words the tyre coefficient of friction would have to be 175. This is impossible and so we can consider that the bike slips outward until it reaches a radius at which the lateral acceleration balances the tyre force. At a lean angle of 45° or 1 g lateral acceleration that radius would be 79 m. at 100 km/h.

This then is the main mechanism for generating the cornering force on a bike, and is often referred to as "CAMBER THRUST".

If we were to plot a graph of camber force against camber angle then we would get a very similar shape to that shown in fig. 2.15. This is not surprising because, although not immediately apparent, the lateral force is accompanied by a similar side slipping mechanism, let's have another look at what's happening as in fig. 2.19.

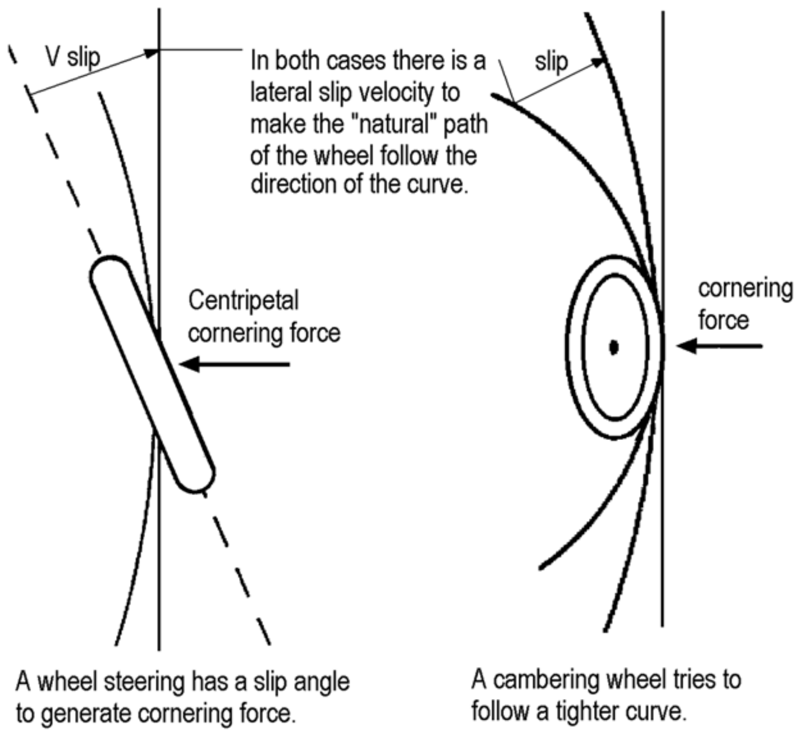



Fig. 2.19 **This sketch shows the similarity in the way that the two different methods produce cornering force.**

**In both cases the tyres would steer more into the turn without the influence of the forward momentum which forces the tyres to slip sideways across the road surface.**

**The centripetal force acts at right angles to the instantaneous direction of travel of the motorcycle. This is explained more in Chapter 4.**

To reiterate, as the wheel leans over, cambers, it can be considered to behave as part of a cone which tries to turn a very tight corner, but the centripetal force required is impossibly high, and forces a sideways slip which causes this tight curve to straighten out and follow the turn radius that the rider desires. Thus cornering force can be generated without the slip angle so necessary with a car.

However, because the camber angle of the bike is determined by the need to balance the machine for a given speed around a given corner (see chapter 4), it is unlikely that in all cases will the camber thrust be of exactly the correct amount.

Tyre size, construction and compound will affect this force for a particular angle of lean. Therefore, it is necessary to have an additional method to correct the cornering force to that which we need to negotiate the turn in question. This is simply done by introducing a small slip angle by means of the handlebars. If the camber thrust generated is insufficient to match the cornering force needed, then we just turn the bars a bit more into the corner, or in other words introduce a positive slip angle. At the rear the slip angle is provided by the bike adopting the attitude needed by the wheel. The whole bike is steered.

Under those circumstances where excess camber force is available (high camber stiffness) it becomes necessary to apply negative slip angles, i.e. to steer out of the turn.

## Combined effects

At low cornering requirements the lateral forces generated by lean (camber thrust) and by steering (slip angle) are additive to a first approximation. So for example if we are leaning at say 6° to balance a gentle turn ( 0.1 g ) and the camber thrust is X, and we apply a slip angle of 0.1°, which gives us a steering force of Y. Then the total cornering force will be close to X+Y.

However, at higher cornering speeds the situation changes, and the combined force becomes a much more complicated non-linear relationship. Imagine that a racer is cornering close to the limit at 50° of lean. This means that there are no more reserves of grip to be called upon, the rider is up to the maximum coefficient of friction. Thus it is easy to see that applying more steering angle to increase the slip angle cannot produce more grip. In fact it is more likely to take the tyres past the hump in the friction graph, fig. 2.15, and reduce the amount of total grip with predictable results.

The camber and slip angles combine in a way that produces a lateral force for each combination, but always this combined force can be no more than the maximum permitted by the frictional properties of the tyre.

Fig. 2.20 shows an example of how the total required cornering force is made up from a combination of lean and steer. (This is not data from a real tyre, but is descriptive of tyres in general.) In order to maintain a balanced cornering situation the tyres must produce exactly the required cornering force. A curve showing this requirement for different lean angles is super-imposed on the others. These curves assume a maximum coefficient of friction for this tyre to be 1.0. Which limits the lean angle to 45°. Real tyres may have a lower or higher value for the friction coefficient, depending on the type and condition of the road surface as well.

If we follow this curve from zero lean (fig. 2.20), up to about 15° we can see that the required force is almost exactly the same as the force provided by camber thrust alone without the need for any steering. From 20° and up there is an increasing shortfall in the camber thrust, and this must be made up by steering force. At 40° of lean we need a little more than 2° of slip angle, but then the non-linearity of the curves becomes apparent. At 45° lean we are on the limit of adhesion and need about 6°, it is interesting to note that the curves for both 5 and 6° of slip angle produce about the same cornering force. This means that the steering at this point is fairly vague or indefinite. Also note that the total cornering force at 45° lean would be reduced if we increased slip much more than 6° of slip angle. This behaviour affects handling feel and just goes to show how difficult it is to ride right on the limit of adhesion of the tyres, and why there are so few people in the world that get really close to it, and retain control.

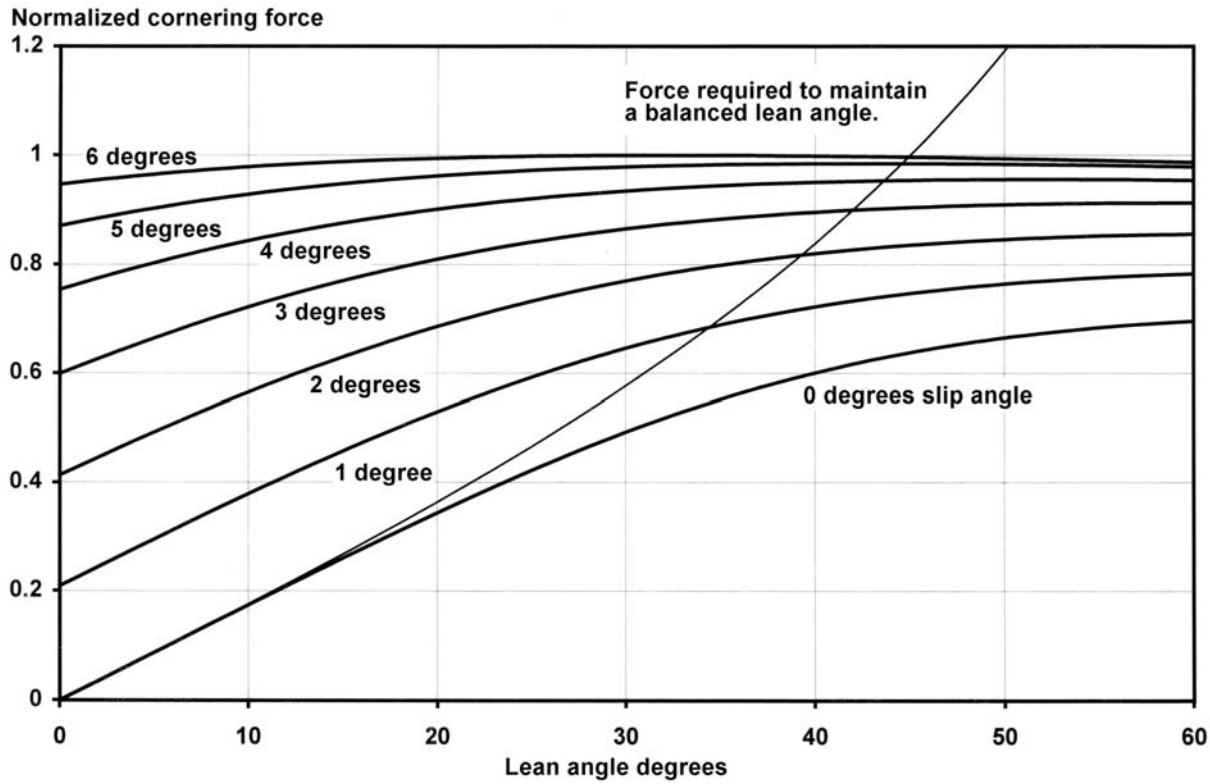



Fig. 2.20 **This plot is broadly representative of the lateral tyre force produced by a combination of leaning and steering. Super-imposed is a graph showing the tyre force required at any lean angle to maintain a balanced turn.**

Fig. 2.21 shows a set of similar graphs except that the tyre camber stiffness is 20% higher than that shown in fig. 2.20. For a tyre with these characteristics we see that up to a lean angle of about 28° the camber thrust is in excess of that needed, and so for these lean angles we would have to apply a negative slip angle, steer out of the corner in other words. However, at higher cornering speeds which need a larger lean angle we can see that we must add some slip angle to make up for a deficit in required lateral force, that is we must steer into the corner. At 45° lean the slip angle would still need to be about 6°. These two examples of the behaviour of tyres with different properties, clearly show why the handling characteristics and feel of a bike can change so much when we change to different tyres.

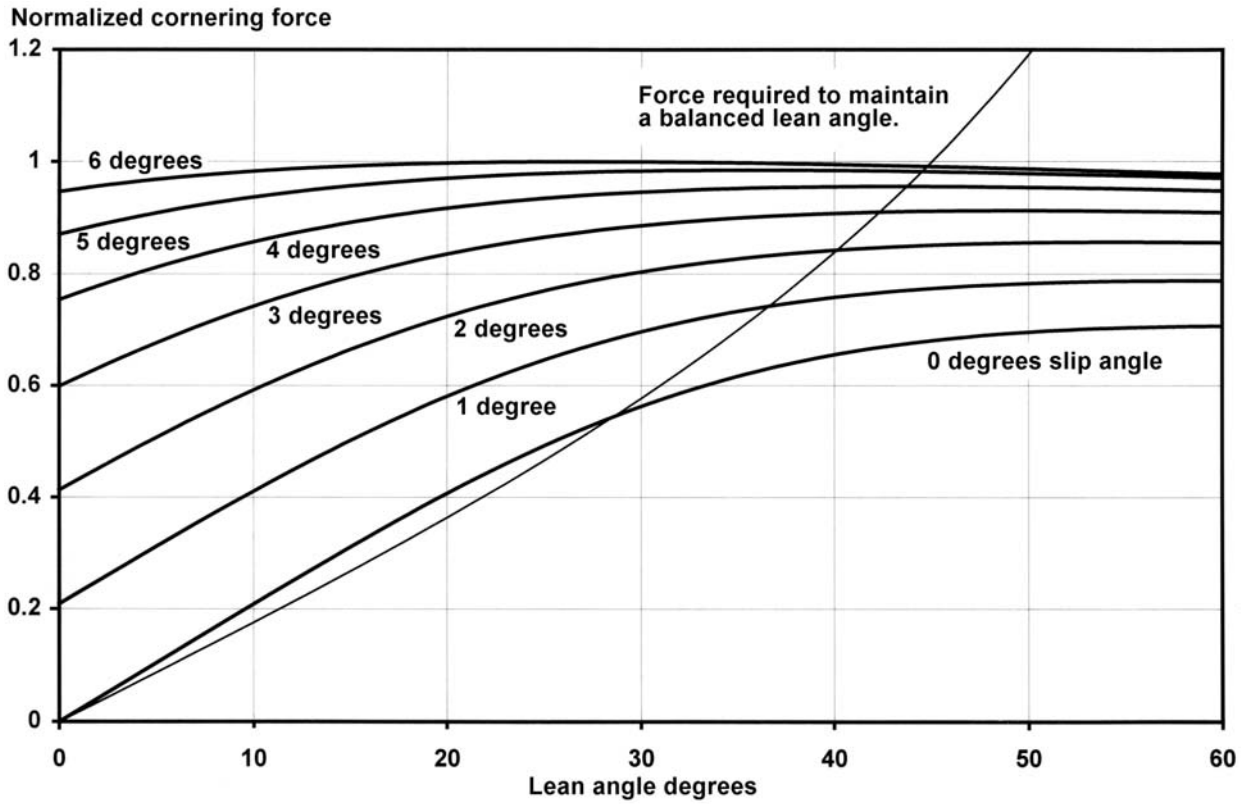



Fig. 2.21 **Tyre characteristics similar to those shown in Fig. 2.20, except that the camber stiffness is increased.**

## Longitudinal forces

As if things weren't complicated enough with trying to figure out the combined effects of camber and steer, we also need to remember that driving or braking forces add yet another dimension to be considered. If we use up some of the maximum potential tyre grip with driving force then the total available for cornering from both camber and steering will be reduced and so the above graphs need to be redrawn for each different power or braking requirement. Unfortunately such data is not readily available.

A much used graphical aid to understanding how cornering and traction (braking and driving) forces combine is the so called "friction circle", which Fig. 2.22 shows pictorially.

This assumes that the maximum tyre friction force possible in any direction is a constant, that is to say, if the tyre in its loaded state can support 1 kN. of cornering force, without any traction force, then it can also support 1 kN. of traction force in the absence of cornering force.

Any combination of traction and cornering force that give rise to a resultant force of no more than 1 kN. is possible also. This means that for the case where a tyre is capable of handling either a 1 G. turn or 1 G.

braking separately, it is also capable of supporting a 0.7 g turn and 0.7 g braking at the same time. In practice the friction circle is not truly circular, depending on the tyre characteristics it might be slightly elliptical. This effect is likely to be small.

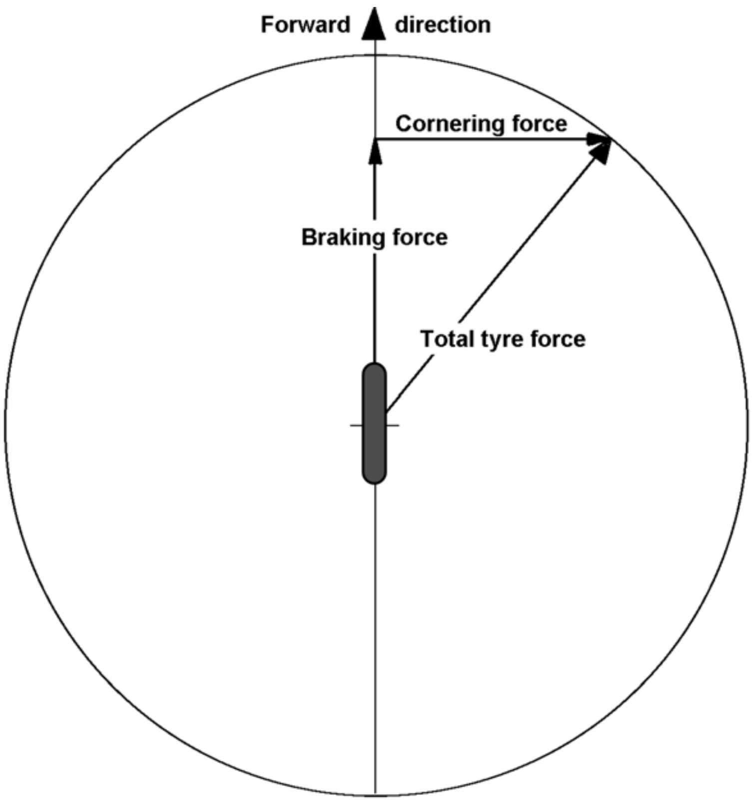



Fig 2.22 **Tyre friction circle. This is a conceptual plan view around a tyre, showing the maximum friction in any direction. Any combination of lateral and longitudinal forces is possible as long as the total tyre force does not exceed that defined by the circle.**

The “steering angle” of the handlebars depends on the required slip angle, but the “steering torque” to be applied by the rider to achieve this position depends on many other factors. Wheel diameter, tyre width, rake and trail are but some of the influencing parameters. On some machines it may even be essential to apply a negative torque to maintain a positive slip angle at the required level, the geometry may try to force the bike on to a tighter line if left to its own devices. A bike set up like this, tends to flop over easily, and may give the feeling that the rider needs to hold it up. Whereas another type of machine may need a positive torque to maintain the desired steering angle and this bike will feel as if it needs to be held down. A few numbers may help to show how different speeds and bend radii affect the lean angle and hence the effective cone radius, and camber thrust.

When cornering at an angle of 45° and 120 km/h. the turn radius will be 115 metres. and at half that speed, 60 km/h., the radius will be a quarter of that or 29 m. --- but as shown earlier the effective cone radius will be only about 450 mm (0.45 m). for both cases at 45°.

We can see that there is a very large difference between the bend radius and the effective cone radius, the tight cone radius generates a lot of camber thrust to support the large cornering forces present at a 45° lean angle. Now, if we slow down to 11 km/h. and tackle the same corners, we would need to lean over to only 2.5° and the cone radius would be up to 8 m., for the 29 m. radius bend, 0.57° camber angle and a cone radius of 30 m. would be appropriate for the 115 m. bend. For these cases the cone radii are much larger and are up to approximately 1/3 of the bend radii. Consequently the camber thrust is much reduced, in line with the smaller requirement of these gentle cornering speeds. For an average size motorcycle tyre at about 6 km/h. the bend radius and the effective cone radius will be equal to one another, in a balanced turn.

## Under- and over-steer

These terms have long been used to describe some handling behaviours of cars and are well defined when used in this context. It is not always realized but there are really two quite different regimes of under-/over-steer. There is the case at relatively low cornering speeds, say below cornering accelerations of around 0.3 G. In this range the tyres are operating in the linear part of their characteristics and the under-/over-steer is mainly a function of the relative steering stiffnesses of the two pairs of tyres, on motorcycles the relative camber stiffnesses are very important also because these define the degree of steer needed. The second case occurs at high cornering speeds and describes the behaviour when close to the limit, this is principally a function of the frictional properties of the respective tyres. In the linear range, when a car is cornering under such conditions that it needs to have a greater slip angle at the front than the rear then the vehicle is said to be under-steering. Conversely if the rear slip angle needs to be greater then over-steering is occurring.

If we relate this to tyre characteristics as in fig. 2.20 then we can say that an under-steering vehicle has a lower normalized cornering stiffness for the front tyres than does the rear. Fig. 2.23 shows an example case of different characteristics between front and rear tyres.

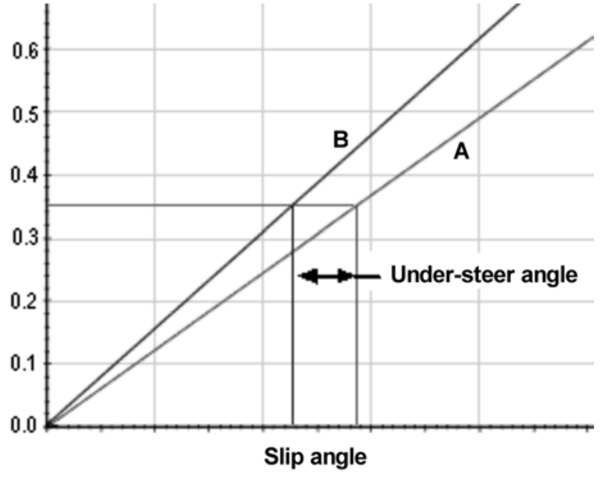



Fig.2.23 **Diagrammatic representation of under-/over-steering car tyre characteristics in the linear range.**

**If line 'B' is the front tyre then it needs a lower slip angle to provide its share of the total cornering force and so the car is said to be over-steering.**

**However, if line 'A' represented the characteristic of the front and 'B' the rear, then the vehicle would be said to be under-steering. The under-steer angle as shown is the extra slip angle that the front must adopt relative to the rear. It is effectively the steering angle of the front wheel relative to the main chassis.**

Actually the strict definitions of under-/over-steering are a little more complex but this visualization will suffice for our present purposes. The accurate definition of this characteristic effectively relates the under-steer angle to the lateral acceleration. *(In terms of the SAE definition of under-steer we can say that the gradient of the under-steer angle is greater than the gradient of the Ackermann (kinematic) steering angle.)*

Although these terms have started to be used, in recent years, in connection with motorcycles they are often misused and misunderstood. As we have seen, a motorcycle derives much of its cornering force from camber thrust and it is only the deficit or surplus that is provided by steering (slip angle). We have no real choice over the lean angle of a cornering bike. A given corner taken at a given speed defines an angle needed to retain balance, the rider can only exert a limited influence over the camber angle of the wheels by leaning his body more or less than the machine itself. Hence the relative camber stiffness (sometimes confusingly called roll stiffness) of the front and rear tyres has a great influence over the required slip angles, front to back.

It was shown in figs. 2.20 and 2.21 that the required slip angles could be positive (steering into the turn) or negative (steering out of the turn) depending on the camber stiffness of a tyre and the necessary lean angle. Imagine that fig. 2.20 represents a rear tyre and fig. 2.21 the front. We can see that at a lean angle of 15° the rear tyre needs no slip angle whereas the front requires a small negative slip of about –0.2°. At 30 degree lean, camber thrust alone is insufficient to provide all the needed cornering force at the rear and so a small positive slip angle of about 0.4° is necessary to make up the shortfall. At the front, the camber force almost balances that required and only about 0.1° of slip is needed. At 45° both front and rear need a slip angle of about 6° to make up for the fact that camber force is only capable of providing less than 70 % of the cornering force needed at this “on the limit” cornering speed.

It is clear that the steering requirements of cars and motorcycles are quite different and as a consequence we need to be careful when using terms and concepts that were developed for four wheeled vehicles. For example, when an under-steering car is cornering and the driver wants to turn sharper he can just simply apply a greater steering angle to increase the understeer angle as in fig. 2.23. On a motorcycle we have the added complication of counter-steer and if we just simply steered more into the corner, the bike would attempt to sit up and steer on a greater radius, not a smaller one, even though in the steady state the bike would need the greater steer angle to maintain the sharper turn.

In addition there are quite different ways in which the rider or driver perceives “under-/over-steering” between the two vehicle types.

Generally speaking, to apply a slip angle to a car front tyre the driver needs to apply a steering displacement in the same direction. However, as mentioned earlier, the rider applied steering torque on a motorcycle may be in the opposite direction to the actual steering angle of the front wheel. The reasons for this will be elaborated upon in a later chapter. The rider is more likely to perceive under-/over-steering based on his input steering torque than by detection of the very small steering angles needed. The formal definitions of these steering terms are however, based on slip angle rather than on steering torque, as a result it seems that we need to question the relevance of such parameters to motorcycle use. Perhaps we need to develop a completely new set of parameters to describe motorcycle handling, there is a dearth of such values compared to those available for the study and description of car dynamics, which has long been subject to much more rigorous analysis.

## Drifting

Those of you that have followed racing techniques over the years will be aware of a change in riding styles amongst the really fast riders, who now sometimes drift their machines to quite a significant extent. Fig. 2.24 shows an extreme case of drifting, as would be experienced in speedway riding. This might at first sight just seem like an exaggerated case of “over-steering”, but in fact there are quite significant additional effects.

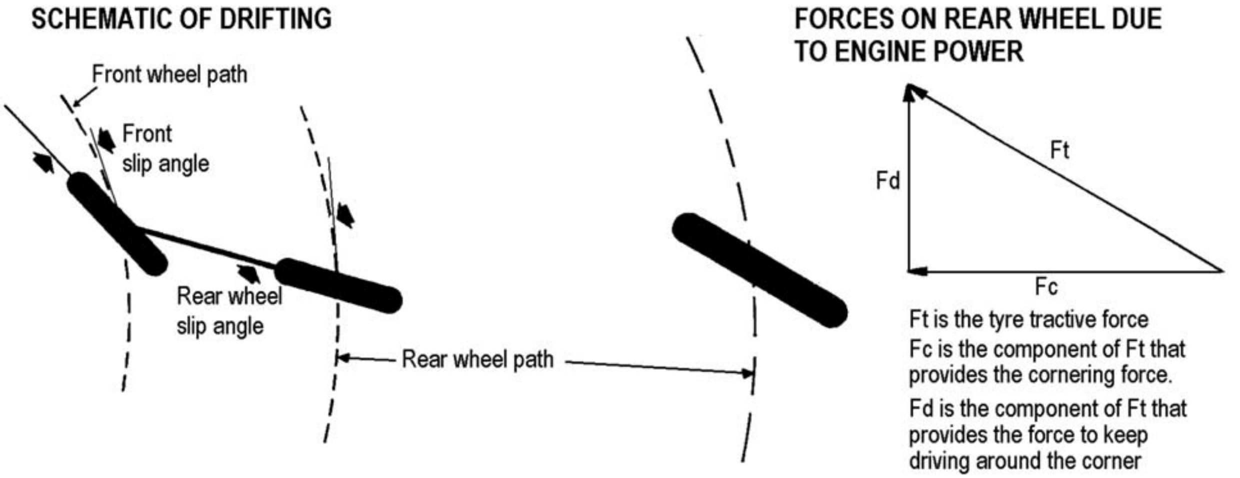



Fig. 2.24 **Showing the attitude of the bike and wheels during high levels of drifting, as for example in speedway.**

The cornering force is provided from two sources:

Camber thrust and slip angle as described above,

The component of the engine supplied driving force, that acts toward the centre of the corner. Fc in the following diagram.

This driving force itself, acts in line with the rear wheel, but as this is at a significant angle to the direction of travel it can be resolved into two components. One of which acts in the direction of travel and maintains the speed around the curve (Fd below), the other acts at right angles to this, and pushes the bike toward the centre of the curve (Fc below), i.e. provides some additional cornering force.

Because the front wheel is not driven it must produce its share of the cornering force by more normal means. But, as this wheel is more upright than in the non sliding case, camber thrust is reduced and so more of the cornering force must come from a slip angle, and the wheel will be turned more into the corner, than if the rear wheel was sliding less.

As a large part of the total cornering effort is derived from the engine power, it comes as no surprise that throttle position has a major influence over the cornering line. It has been demonstrated countless times in speedway that mid-corner engine failure or inexperienced shutting of the throttle, results in immediate intimate inspection of the perimeter fencing. This sudden loss of engine power results in an equally sudden loss of cornering power, and the bike succumbs to the effects of a lack of centripetal force.

The required lean angle varies as the angle of sliding changes (rear wheel slip angle), the reason for this can be seen in fig. 2.25 the over-balancing force acting on the bike is the component of the cornering force that acts at right angles to the machine. This force is less than the total cornering force which would act as the over-balancing force in the non drifting case.

So the greater the sliding angle, the smaller the over-balancing tendency that needs compensating by leaning in. Thus, a smaller angle of lean is needed, but the effect of this is often over stated, as a few more figures will demonstrate.

Imagine a road racer cornering at 1G. lateral acceleration, and let’s assume that the rear wheel is 300 mm. out of line with the front, and that the wheelbase is 1,450 mm, then the sliding angle is 12° . Without sliding at all the lean angle of the combined bike and rider would be 45° , but when sliding at 12° the required camber angle reduces to 44.4° , hardly a big difference! To return to speedway cornering styles, where a more typical slide angle may be, say 50° , then again for the 1G. case the lean angle reduces to 33°. So we can see that to reduce the lean angle by a significant amount requires a high degree of drifting, more in fact than is usual in road racing.

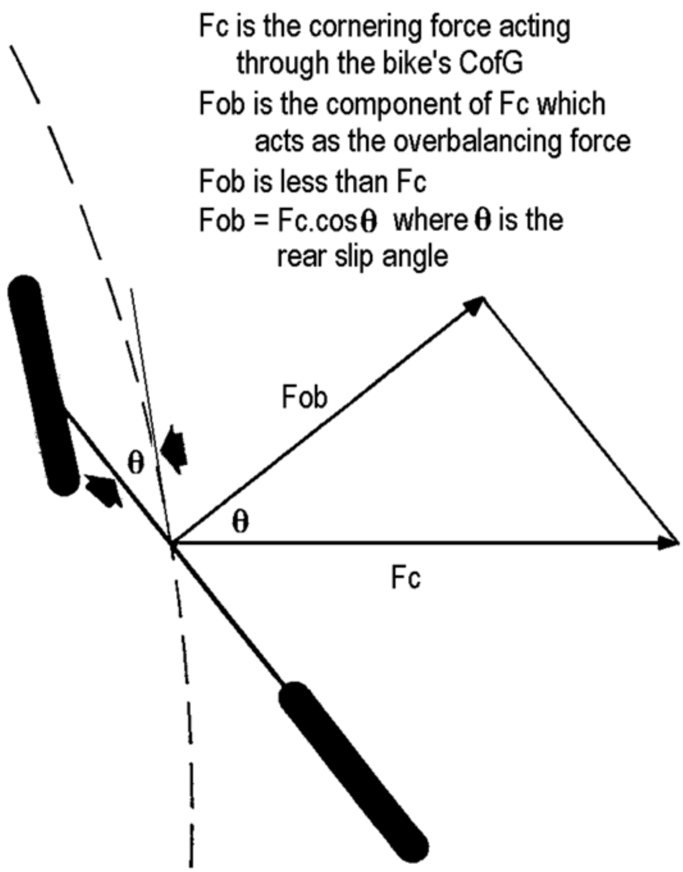



Fig. 2.25 **In this example of drifting we can see that the force $F_{ob}$ tending to make the bike lean outward is reduced. Thus necessitating a lower lean angle than normal to establish balance.**

So where is the benefit in drifting? --- Well, as any drag racer knows maximum traction is achieved when the tyre is spinning slightly. In fact the tyre companies tell us that approximately 10 - 20% slip is about optimum on tarmac, as shown earlier. So if sufficient power is available to cause this degree of slip the total friction force on the tyre can be divided between the normal lateral cornering force and that due to driving force, some of which gets resolved towards the bend centre, thus part of the total cornering force is under control of the throttle. Referring to the friction circle in fig. 2.22 this means that the resultant force moves rearward around the circle. In racing, feel and control over what the tyre is doing can be enhanced by skilled use of this technique. When cornering in the classic style it is hard to feel the limit of adhesion and there are minimal control options left to deal with the situation when that limit is exceeded. On the other hand when the bike has been setup with significant drift the rider has more control over the total tyre force through the throttle. Therefore when he feels that he is about to lose it, he can just close the throttle a little to restore adhesion somewhat. Feel is enhanced because the yaw attitude is very much under the control of the rider.

Racers often drift their machines for additional reasons. With a lot of power available the classic large radius line through a corner is not always the best and it is often beneficial to turn as sharply as possible at the bend entrance, and leave the straightest possible exit path to get the maximum acceleration out of the bend. Drifting the bike into the bend can sometimes help get the machine better lined up for this quick exit. Control over the yaw attitude is also useful to setup the exit angle from a corner, for example if the bike is steering outward too much, it can be made to turn inward by throttle application.


**"Ecca" Erik Andersson appears to have forgotten that he is on a road circuit and seems to be practicing speedway at Alastaro circuit. Such extreme attitudes in road racing are not normally considered a stable situation. However, in this case Ecca retained control.** (Esso Gunnerarsson)

The extreme drifting in speedway is necessary for slightly different reasons. The loose surface is incapable of supporting high cornering forces through the normal mechanisms of camber thrust and slip angle, but considerable force can be generated by a spinning rear wheel. Some of this is just due to the usual frictional processes just described, but an added effect comes from the rooster tail of shale thrown up behind. The tyre has to put effort into imparting momentum to these rocky particles and so in accordance with the previously mentioned laws of Newton, these particles must push back on the tyre. This is similar to the principle of a rocket, where the rapid ejection of combustion material pushes it along.

To take advantage of this rocket powered effect in a bend, requires the bike to be pointing toward the turn centre so that a significant component of this force is used as cornering effort.

## Construction

Even long after cars had almost exclusively changed over to radial tyres, those fitted to motorcycles retained the cross-ply form of construction and many tyre engineers claimed that radial tyres would never be suitable for motorcycles. It is true that cars and motorcycles have very different tyre requirements but recent history has shown that radials have benefits to offer motorcycles as well.

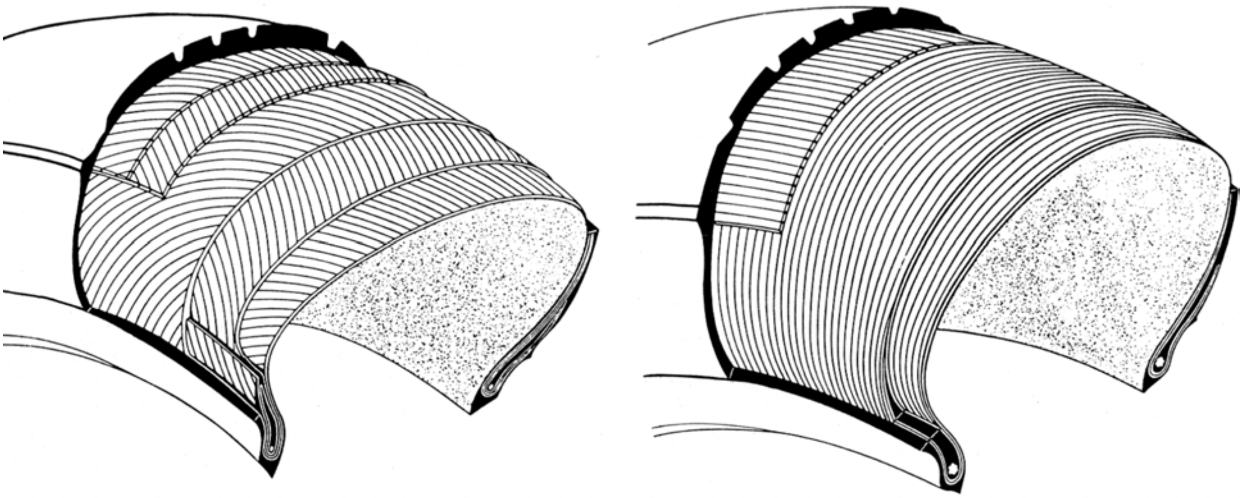


Fig. 2.26 **Showing the principal constructional aspects of cross-ply (left) and radial-ply (right) tyres. The cross-ply is made up of successive layers or plies with the cord angled across the tyre at approximately 45 deg. each layer applied at opposite angles to the previous. Hence the name of cross-ply. A true radial tyre has its cord running from one bead radially over the tyre to the opposite bead, a circumferential belt is added to reduce tyre fling at speed, generally stabilize the construction and enhance load capacity.**

The figure shows the basic differences in construction between the two types. The cross-ply tyre is usually made up from several plies of cord laid over each other with a bias angle of around 40 to 50°, and so cross each other at an angle of between 80 to 100°. Due to the layered and angled construction, the ply immediately under the tread of a cross-ply tyre does not react entirely symmetrical to the applied loads. This lack of complete symmetry can cause the tyre to run to one side even when rolling on a perfectly level and flat surface. This effect is known as ***Ply Steer.*** The cross construction of the carcass effectively forms parallelograms of cord imbedded in the rubber, this is not a strong form of structure and under driving/braking torque, vertical load and centrifugal effects these parallelograms distort, which allows squirming of the contact patch and tyre fling or diameter growth at high speed, reducing stability and heating the tyre. Such heating ultimately manifests itself as increased rolling resistance.

Prior to the cautious introduction of radial tyres in 1984, the bias belted tyre was developed. This followed the normal cross-ply construction but with the addition of circumferential belt plies. These belts weren't truly circumferential and the cord was set at a slight angle, often around 25°. Nevertheless, these angled cords effectively triangulated the main plies and so stabilized the overall structure with the results of less tyre fling, better stability, longer life and they were capable of better handling the ever increasing power outputs of the contemporary engines.

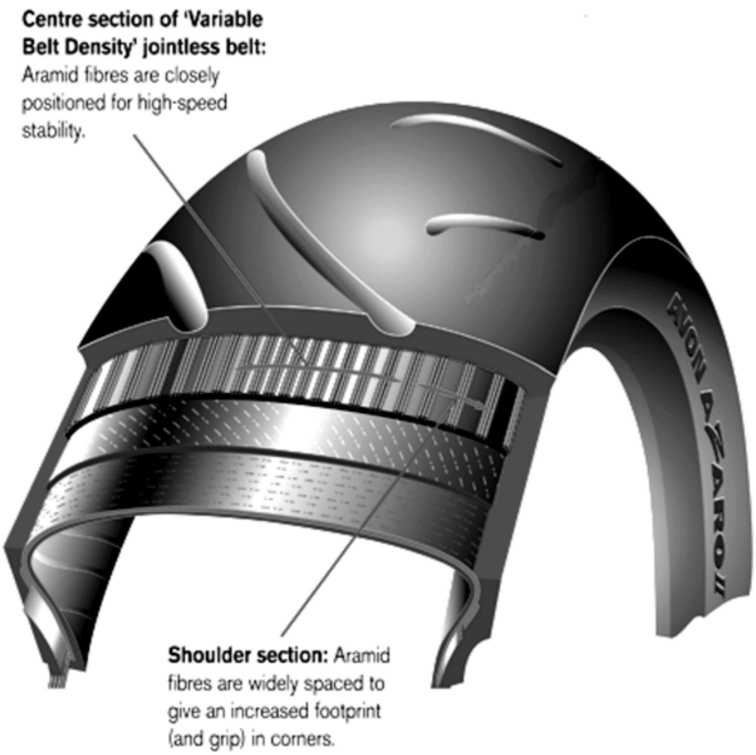



Fig. 2.27 **A wound on belt with wider spacing toward the outside of the tyre. This technique controls the stiffness of the tyre in order to provide the best compromise between thread stability at high speed and a large contact patch when lent over. The example shown here is an Avon Azaro Sport II.** (Avon)

When the first radials started to appear most manufacturers opted for a compromise approach and did not produce "true" radials with a 90 degree bias on the main cords and used a cross-ply layout with steeper bias angles, together with the angled belt cords of the bias belted tyre. Some tyres of that period that were called radials were obviously done so by the marketing departments as many couldn't justify the description on technical grounds. Many radials still use a biased belt but high performance tyres are moving to a 0 degree wind on belt, particularly for rears, which further restricts tyre growth at speed, enhancing tyre stability. An interesting development along these lines is the *Variable Belt Density* tyre. In this case the belt is wound closer in the centre of the tyre and wider apart at the edges as shown in fig. 2.27.

Radial tyres generally have less plies (1 or 2) than cross-ply tyres (3 or 4) and so have more flexible sidewalls. In order to restore overall tyre stiffness the sidewall height had to be reduced which is why they typically have much lower profiles, 50% to 70% being common, whereas cross-plies use around 90% to 100%. A low profile tyre has a smaller diameter than an otherwise equal size cross-ply and this is the principal reason that the trend to smaller wheel sizes reversed and 17” wheels have all but completely superceded the once popular 16”.

A consequence of using low profile tyres is that the tread shape will change more when fitted to various rim widths and so it is much more important to only fit radials to the reduced width range recommended by the tyre manufacturer.

## Materials

This is an extremely complex subject and is continually changing as new materials come along, we can only briefly review the subject here. Carcass cord fiber is generally nylon or rayon with belt cords being of aramid (commonly known as Kevlar) or steel.

The tread rubber compounds are a complex blend of chemicals, several types of rubber such as styrene butadiene, butadiene and natural rubber, carbon black and various oils. Silica based compounds are becoming more common now due to increased wet grip and cooler running. Adding more oil softens the compounds and increases grip. Increasing the natural and butadiene rubber content can increase the life but decreases grip. In other words like with many design problems it is a compromise between many conflicting requirements and it is an art to select the best compromise for any particular task.

As a way to reduce the compromise in compound selection there have been developments with multi or dual compound treads. Harder rubber is put in the centre section of the tread to combat wear, whilst softer rubber is put around the edges to increase cornering grip.

## Summary

Without something akin to the pneumatic tyre it is inconceivable that current motorcycles could have evolved to their present level of performance. For comfort and roadholding the tyre is more important than the suspension system.

A car derives its cornering force by means of a slip angle, which is to say that the driver turns the wheels in more than the line of the bend, as if he wanted to take a tighter curve. A bike on the other-hand achieves the same end through an effect known as camber thrust, which is a consequence of leaning into a corner. The tyre then acts like part of a cone and tries to turn a tighter circle than the actual bend. Centrifugal force straightens this circle closer to the desired path, and relatively small slip angles are then used to correct any shortfall or excess cornering force. There is some similarity between the mechanisms of steering both bikes and cars. i.e. it is necessary in both cases to set up a tendency for each vehicle to try and turn around a tighter bend than the radius desired. It is just the method of generating this tendency that differs.

Drifting is sometimes a useful way of using excess engine power to provide some cornering force, there is some reduction in lean angle but on tarmac this is not usually significant. The increased racing use of this technique has been made possible and sometime desirable by the vast power outputs of modern engines. Feel and control can be enhanced by the skilled use of such techniques.

The selection of tyre constructional methods and materials is a complex balancing act between the different requirements of low cost, high grip, high mileage, stability, load carrying ability and speed rating. It is still as much an art as a science.

# 3 Geometric considerations

## Basic motorcycle geometry

The elements of this are shown in figure 3.1.

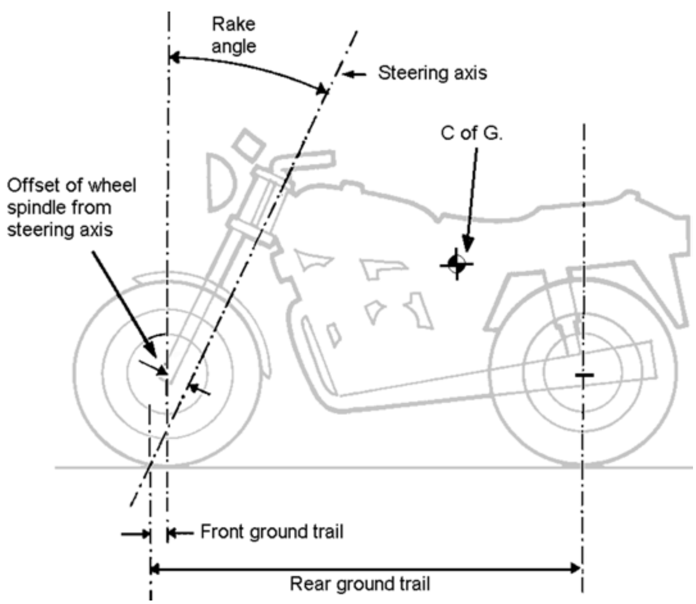



Fig 3.1 **The steering axis is the line about which the steering system turns. Rake or castor angle is the rearward inclination of the steering axis.**

**Ground trail is the amount by which the centre of the tyre contact patch *trails* behind the point where the steering axis meets the ground.**

**Back and front wheels each have their own value of trail. Offset of the wheel spindle from the steering axis is measured at right angles to that axis.**

**CofG is the centre of gravity and for most purposes, we are concerned with the combined CofG of rider and machine.**

## Trail

The primary function of trail is to build in a certain amount of steering stability, and it also is of great importance to the lean-in phase when cornering. We can see that both front and rear tyres contact the ground behind the point where the steering axis meets it, this gives rise to a castor (self-centring) effect on both wheels. The linear measurement of this castor along the ground (steering axis to centre of contact patch) is usually called the trail.

However, it would be more logical to use the distance between the ground contact patch and the steering axis as measured at right angles to that axis. This is the distance that creates a torque about the steering axis from any forces at the tyre, to distinguish between these two trail definitions, when necessary, I suggest that we call them *ground trail* and *real trail*. *Real trail* is approximately 90% of the *ground trail* for bikes with a typical rake angle, and is equal to the *ground trail* for a zero rake angle. Compare figs. 3.1 and 3.2 to see the difference. The importance of this distinction will become clear later in this chapter. *(The SAE refer to the real trail as "Mechanical Trail" to distinguish it from ground trail. The name used is less important than realizing the difference between the two)*

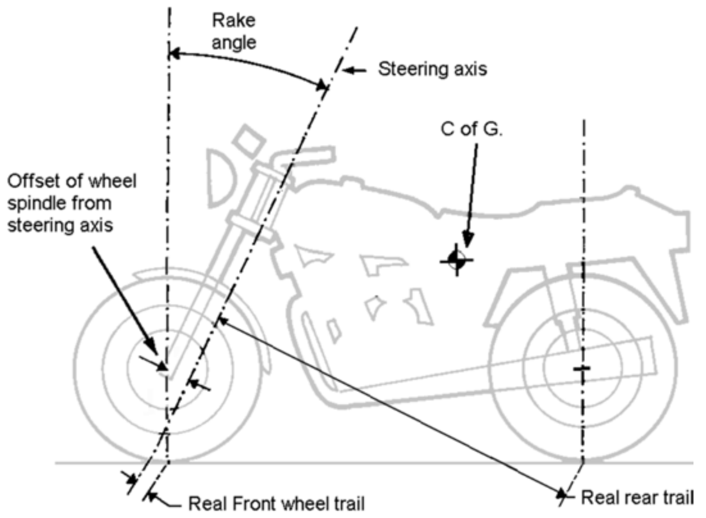



Fig. 3.2 **Real or mechanical trail is measured at right angles to the steering axis. Real trail is reduced from the ground trail by the cosine of the rake angle. For typical rake angles this is approx. 90%**

**Compare this with the ground trail as shown in fig. 3.1.**

How the trail causes a self-centring effect can be understood from figure 3.3 which is a plan view of a wheel displaced from the straight-ahead position.

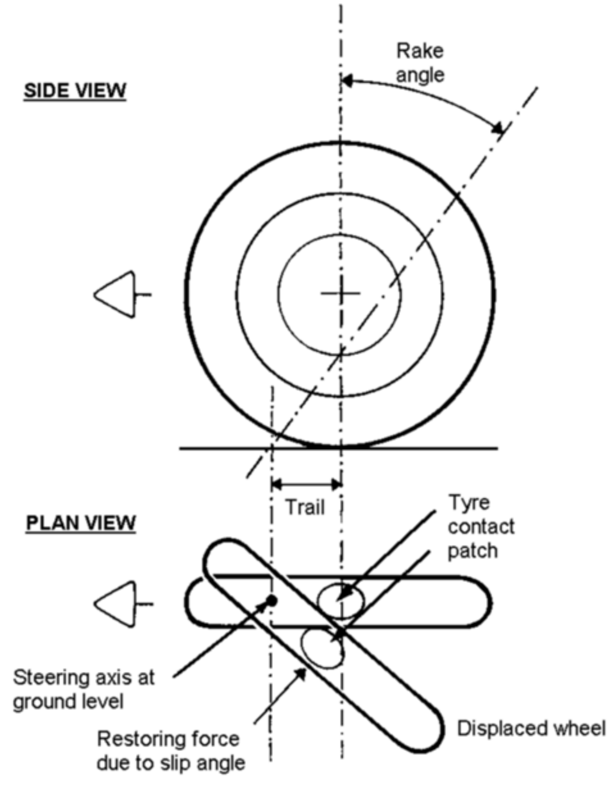



Fig 3.3 **Positive trail and the side force due to slip angle combine to produce a steering torque tending to restore deflected steering to the straight ahead position. This gives a certain degree of straight line stability.**

Because the wheel is at an angle to the direction of travel, slip angle is the technical term (see the chapter on tyres), a force at right-angles to the tyre is generated. Since the contact patch is behind the steering axis (positive trail) then this force acts on a lever arm to provide a correcting torque to the angled wheel. This lever arm is equal to the *real trail* and not the *ground trail,* which why the *real trail* is the more logical parameter to use. Despite the logic it is the *ground trail* which is most commonly specified. That is to say, if the steering is deflected by some cause (e.g. uneven road surface) then positive trail automatically counteracts the deflection and so gives a measure of directional stability. This also interacts with the stabilizing effects of the gyroscopic reactions, (see chapter on balance). If the tyre contact patch was in front of the steering axis (negative trail) then the torque generated would reinforce the original disturbance and so make the machine directionally unstable.

One may be forgiven for thinking that, because the positive trail of the rear wheel is much greater than that of the front (typically 50-100 mm. front, 1300-1500 mm. rear), the rear wheel would be the more important in this respect. The reverse is the case and there are several reasons for this. See figure 3.4. Imagine that the contact patch of each wheel is in turn displaced sideways by the same amount, say 12 mm. The front wheel will be turned approximately 7–10 degrees about the steering axis; this gives rise to a slip angle of the same amount and generates a sideways force that has only the relatively small steering inertia of the front wheel and fork to accelerate back to the straight-ahead position. However, the slip angle of the displaced rear wheel will be much less (approximately 1/2 degree) and so will the restoring force. Not only do we have a smaller force but it also has to act on the inertia of a major proportion of the machine and rider, hence the response is much weaker than in the case of the front wheel.

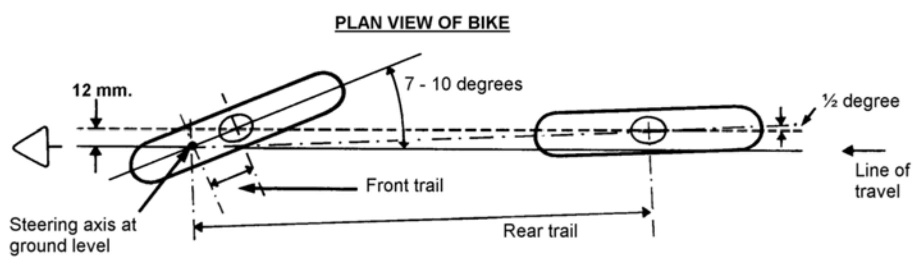



Fig. 3.4 **Despite the much larger trail of the rear wheel, its slip angle is much smaller than that of the front wheel for a given lateral displacement. Hence both the steering effect of the displacement and the restoring effect of trail are much less significant.**

From this, we can see that increasing the trail as a means of increasing the restoring tendency on the wheels is subject to the law of diminishing returns. It must also be emphasized that the disturbance to a machine's direction of travel, due to sideways displacement of the tyre contact patch, is less from the rear wheel than the front because of the much smaller angle to the direction of travel that the displacement causes. To summarize we can say that, while the large trail of the rear wheel has a relatively small restoring effect, the effect of rear-wheel displacement on directional stability is also small and so compensates.

Gyroscopic reactions, have an important influence on directional stability also and the amount of trail can have a large effect on these interactions. This is explained in other chapters.

It is important to realize that trail is not a fixed value for a particular motorcycle, there are several factors which cause the effective trail to vary during normal riding. Some of these are lean angle, steering angle and even the profile radius of the tyre. Fig. 3.5 is a contour plot of trail variation plotted against both lean and steering angles for one particular machine configuration.

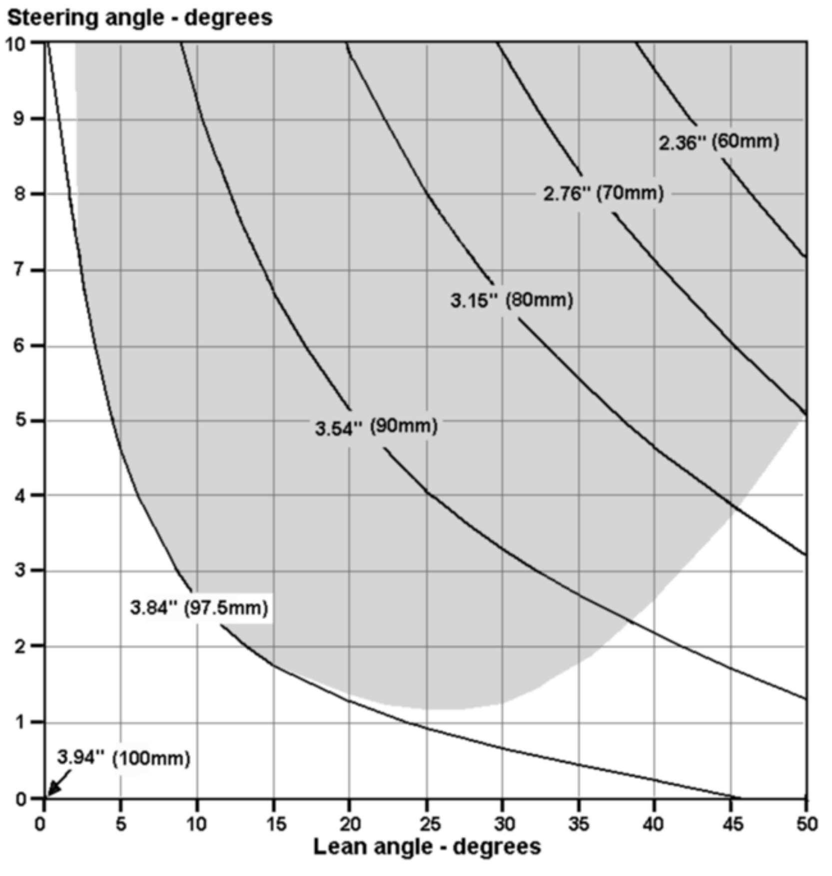



Fig. 3.5 **Shows values of effective real front trail for different combinations of lean and steering angles. Note that not all combinations of these angles are likely to occur in practice, for example when cornering with a lean angle of 45 degrees, it would be most unlikely that we would want to apply a 10 degree steering angle. The shaded area is a rough guide to the area unlikely to be used.**

**Considering only the most likely values, there can be up to about 25% reduction in trail during riding.**

**The bike had the following basic parameters.**

**Wheelbase: 1.4 m or 55”**

**Offset: 80 mm or 3.15”**

**Rake angle: 30 degrees**

**Tyre radius: 360 mm or 14”**

**Tyre profile: 55 mm or 2.17”**

**Initial real trail: 100 mm or 3.94”**

**Initial ground trail: 115 mm or 4.55”**

Basic data supplied by Roberto Lot of Padova University

Although the primary purpose of front-wheel trail is to provide a degree of directional stability, there are various side effects too. Let us consider two of them.

## Steering effect

If we lean a stationary machine to one side and then turn the handlebar the steering head rises or falls, depending on the position of the steering. The weight of the machine acting at the tyre contact patch causes a torque about the steering axis which tends to turn the steering to the position where the steering head is lowest (i.e. the position of minimum potential energy).

For a given amount of trail, this steering angle is affected by rake and wheel diameter, as discussed later. Fig. 5.17 in the chapter on aerodynamics gives another insight on this subject. When in motion, the effective weight of bike and rider supported by the steering head is reacted to the ground through the tyre contact patch. As shown in the chapter on cornering, the weight and cornering forces largely balance each other out and the nett steering torque from this aspect is reduced. Wide tyres and rider lean-in ensure that there is some residual torque and hence the amount of front-wheel trail (amongst other parameters) affects the amount of steering torque the rider must apply (hence the feel of the steering) to maintain the correct steering angle consistent with the radius of the turn and the machine’s speed.

### Straight-line feel

As we all know, even when we are riding straight ahead the steering feels lighter on wet or slippery roads than on dry. This is because our seemingly straight line is actually a series of balance-correcting curves, with the handlebar turning minutely from side to side all the time. As we have seen earlier, a small steering displacement causes a tyre slip angle, which produces a restoring torque. For a given slip angle, this torque depends on tyre properties, surface adhesion and trail. Thus, through the steering, we get a feedback on road conditions that gives us a feel (dependent on trail) for the amount of grip available.

## Rake or castor angle (steering axis inclination)

The basic reason for rake is less easy to explain than that for trail because it is just possible that we don't need it. Why then, do all current production machines have the steering axis raked back between about 23 and 30 degrees from the vertical – 27 degrees often used to be considered as a magic figure? There is no simple answer and several factors are probably relevant. There is no denying the greater convenience of construction (see figure 3.6).

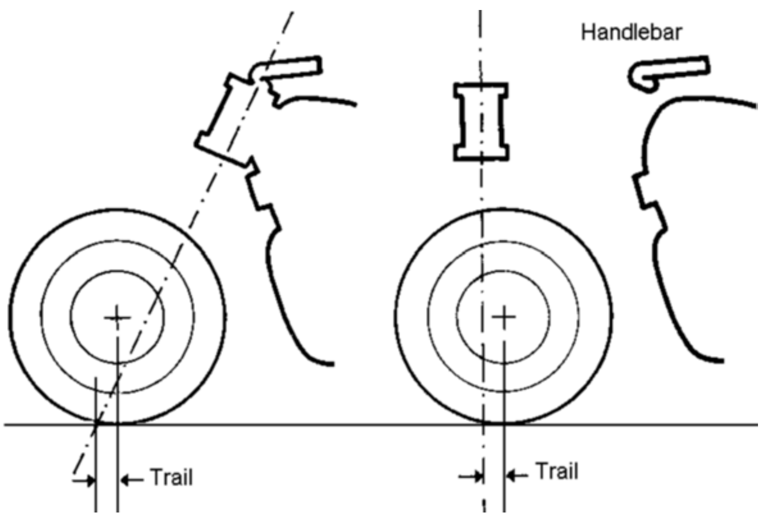



Fig. 3.6 **With a conventional high steering head, a normal rake angle (left) is convenient for frame construction and direct handlebar mounting. For the same trail, a vertical steering head (right) brings problems in both respects.**

(See the appendix on the author's experiments with rake angles.)

In most designs of hub-centre steering, simply for space reasons, the wheel spindle is not offset from the steering axis, hence trail is wholly dependent on rake angle, which is typically between 10 and 15 degrees to give the required result. This is a much steeper angle than usual, yet hub-centre layouts are renowned for their stability and steering. This reputation may stem from reasons other than the steep rake angle, but it certainly seems that this departure from the norm is not harmful and may indeed be beneficial. There is an appendix to this book that details some of the author's experiments with steep rake angles. Basically, these experiments showed that there were distinct benefits from the use of small rake angles, which were not generally offset by compensating disadvantages. It is interesting to note that since the first edition of this book in 1984 detailing those experiments, there has been a gradual trend toward the use of steeper rake angles, particularly for racing and sports machines. Considered as unstable values a couple of decades ago these machines now commonly use castor angles of 20 to 23 degrees. However, some of the reason for change probably comes back to ease of construction again.

Over the past couple of decades there has been a trend toward more forward weight distributions and the rider has been moved forward. These changes are made easier with a more forward head stock and hence steeper castor angle. The reduced rake will also help compensate for the higher steering forces required by the increased load on the front wheel.

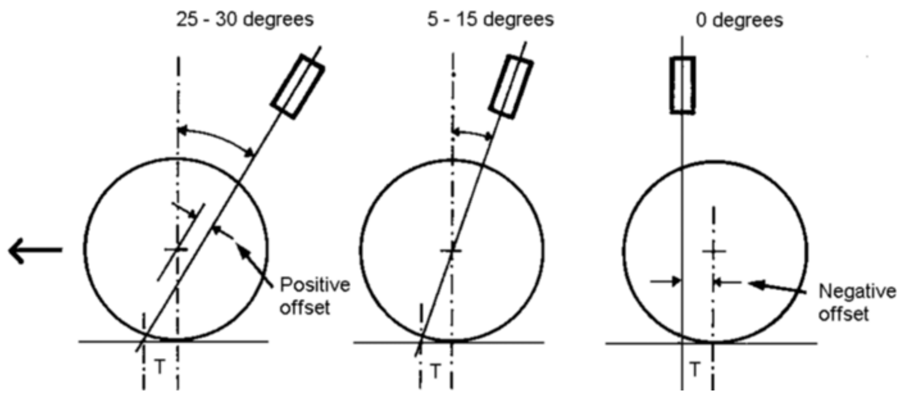
Let us now examine the main effects of rake angle. Figure 3.7 shows three different rake angles, all giving the same amount of ground trail. The real trail will be reduced in the first two cases. The reduction will be by about 10% for the normal range of rake angle, and around 3% at 15 degrees of rake.

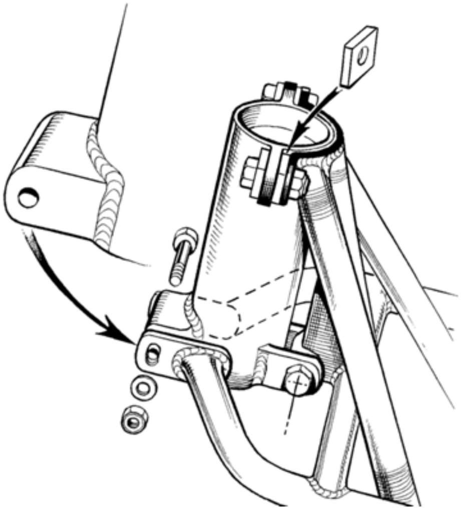
Fig. 3.7 **Three possible rake angles giving the same ground trail.**

*Left:* **Conventional system.**

*Middle:* **Rake angle for zero spindle offset (as in many types of hub-centre steering).**

*Right:* **Zero rake angle (vertical steering axis), note the negative offset.**



**Yamaha used this method of rake angle adjustment for some of their GP racers in the 1960s.**

**Today's racers generally use a larger diameter fixed head tube with eccentric inserts to support the steering bearings. the rake angle is adjusted by changing the offset of the eccentric inserts.**

**In common with any system that changes rake at the head tube, there are many knock-on effects, other parameters are changed also, viz:**

- **Trail**
- **Wheel-base**
- **Weight distribution.**

## 1) Reduction of castor effect

For a given amount of ground trail, the self-aligning torque on the front wheel and fork depends on the length of the lever arm from the centre of the tyre contact patch to the steering axis, measured at right angles to that axis, real trail in other words. As can clearly be seen in figure 3.8, this lever arm is shortened as the rake is increased. In practice, this means that to maintain the same real trail we need more ground trail as the rake angle is increased.

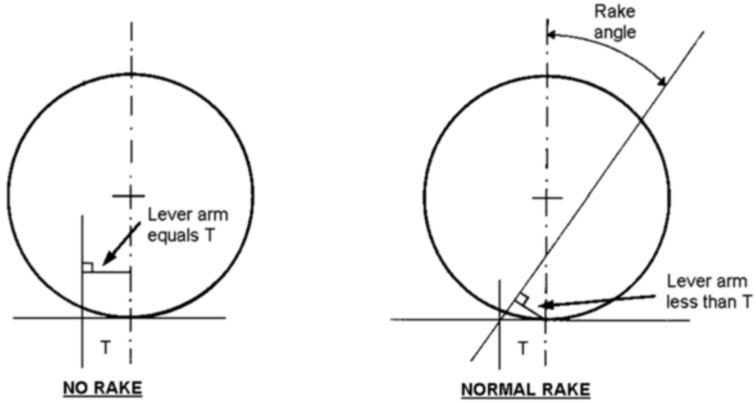



Fig. 3.8 **Positive rake reduces the self centring torque of a given ground trail dimension. Real trail = Ground trail multiplied by the cosine of the rake angle.**

This reduction in real trail is emphasized even more as we apply some steering angle, fig. 3.9 shows the reduction in ground trail at different castor angles for various degrees of handle-bar steering angle up to an extreme value of 80 degrees.

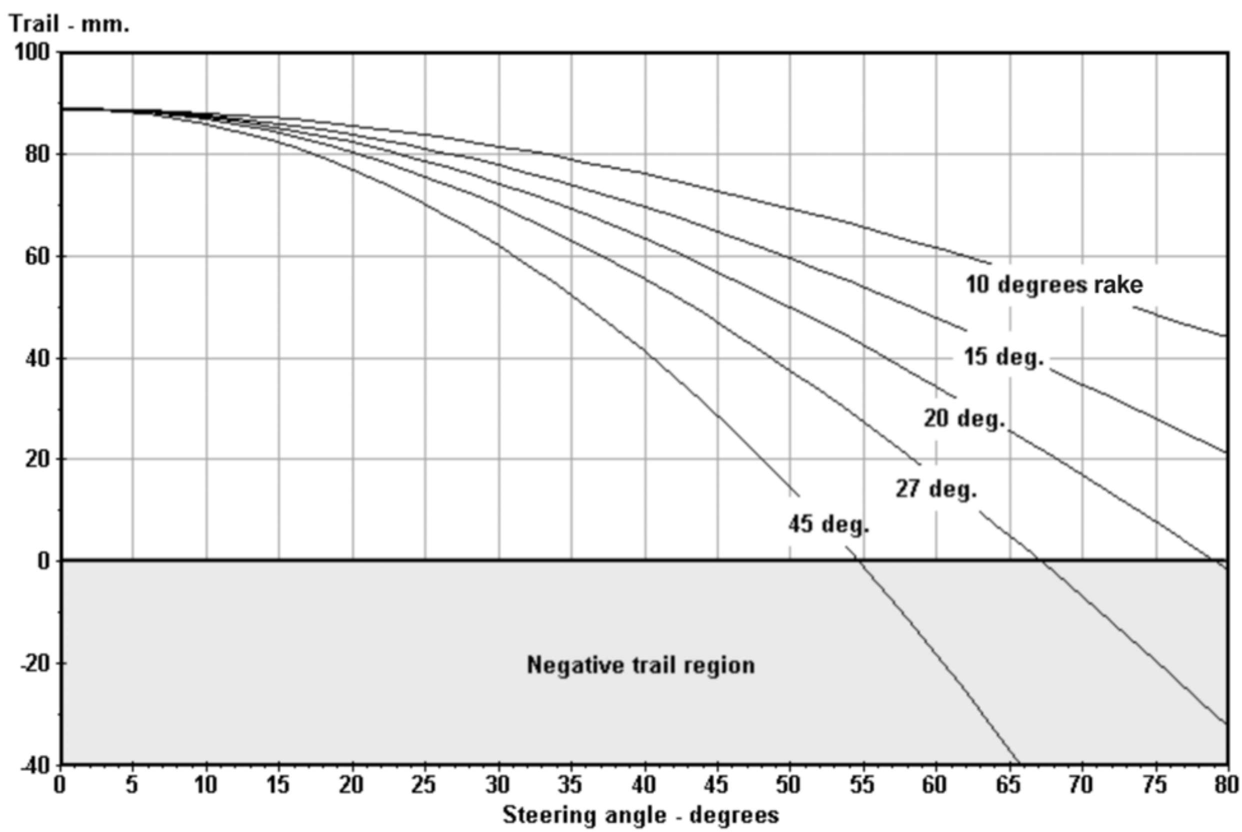



Fig. 3.9 **Reduction in ground trail for several values of castor angle plotted against steering angle. See fig 3.10.**

Road bikes will seldom have more than about 45 degrees of steering lock, but these curves show that at a typical castor angle of 27 degrees the ground trail will have reduced from 89 mm. to 46mm. at a steering angle of 45 degrees. Fig. 3.10 shows the same data plotted for the real trail. Compare the difference between the two sets of curves at low steering angles. Both of these sets of graphs are plotted for a bike held vertically and with a 305 mm. radius front tyre and 89 mm. of ground trail.

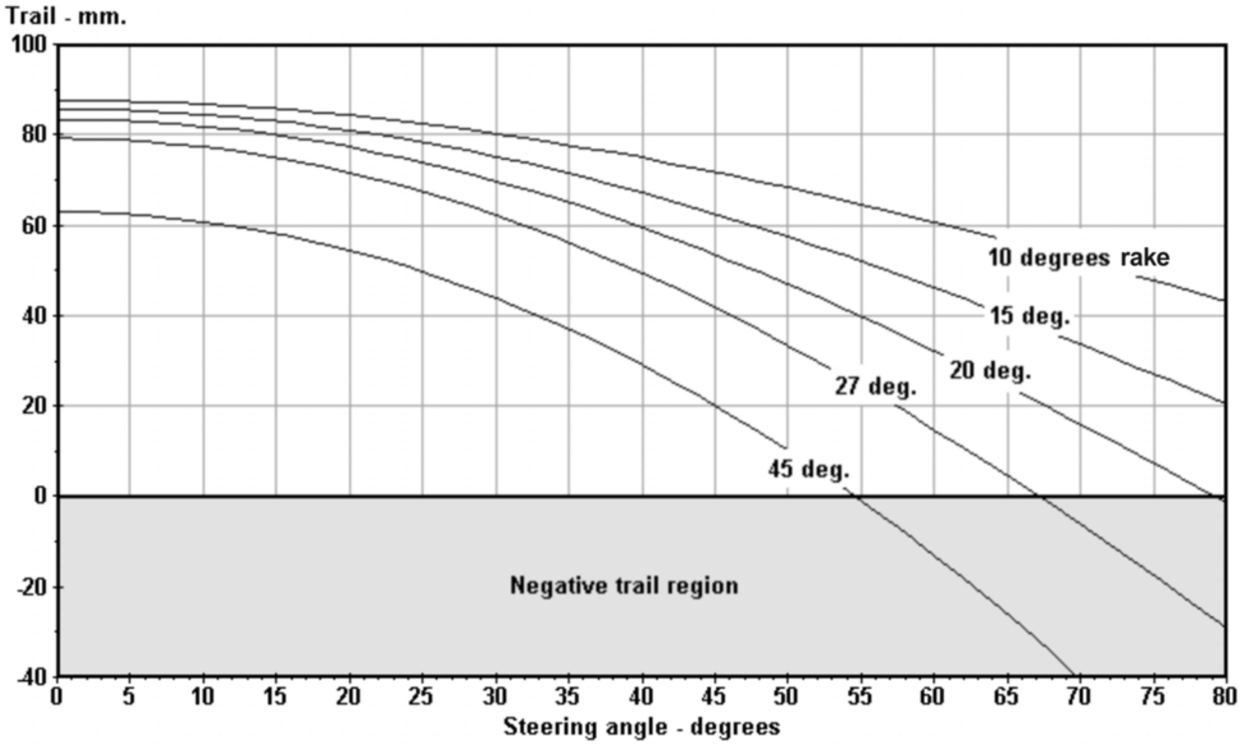



Fig. 3.10 **Similar graphs to those in fig. 3.9 except that the curves show the effects on real trail.**

## 2) Negative castor

At large steering angles, the rake can even cause the trail effect to become negative, though sufficiently large angles are possible only at very low speeds. Reference to figure 3.11 clearly shows what happens physically, and the curves in figs. 3.9 and 3.10 demonstrate the range of values over which it occurs. Even though very large steering angles are needed to produce negative trail there is still some reduction in trail at small steering angles and this may necessitate using a larger initial trail than would otherwise be so. A convincing demonstration of this effect can be obtained by wheeling a pushbike and turning the handlebar far enough for negative castor to take over, whereupon the steering will try to turn even further. This is one reason why trials bikes generally need steep steering axis, since their tricky manoeuvres at low speeds often involve extreme steering angles. At these steeper castor angles there is less reduction in effective trail.

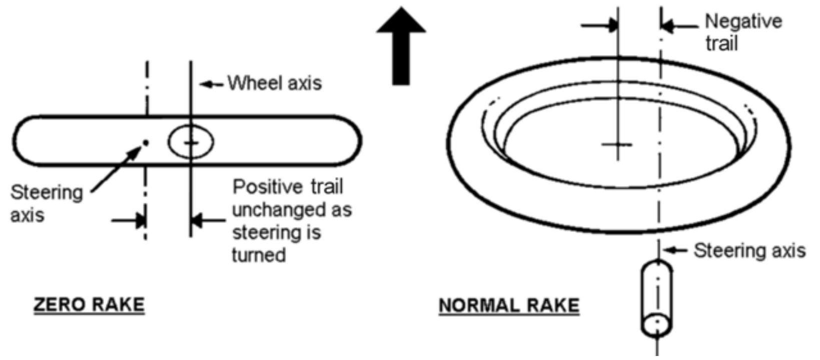



Fig. 3.11 **Plan view with the steering turned 90 degrees to the left. This clearly shows the retention of full positive trail with zero rake angle and the negative trail effect that can occur with a normal rake angle.**

## 3) Steering-head drop

With a normal motorcycle (i.e. with positive trail and positive rake angle) held vertically, the steering head would drop as we turn the handlebar to either side. (With negative trail, which is not normal, it would rise.) The greater the rake angle, the greater the drop. This can be best appreciated by visualizing an extreme rake angle, as in figure 3.12.

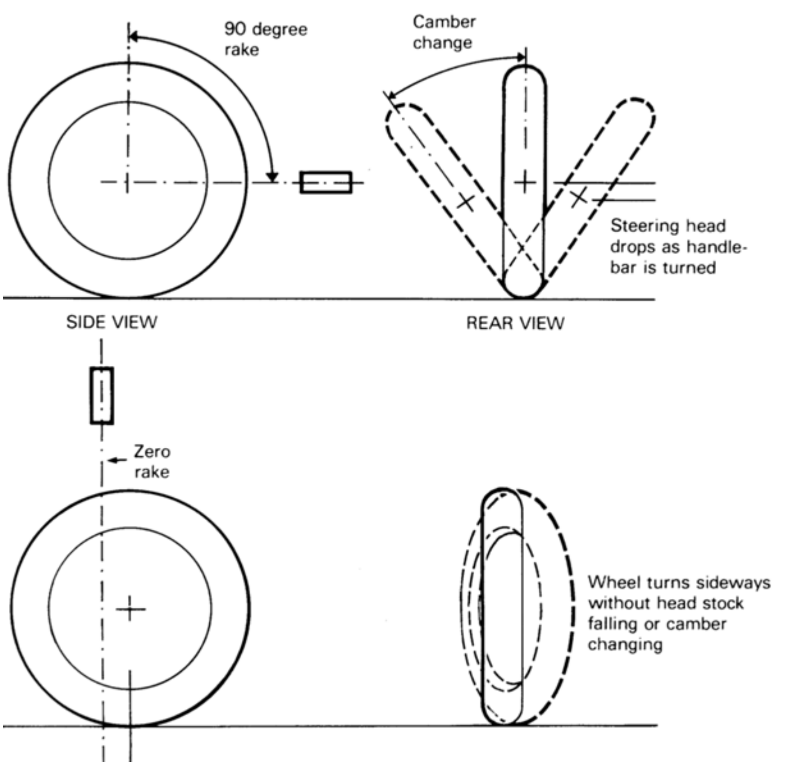



Fig. 3.12 **With a 90 degree rake angle (top), steering-head drop and camber change on turning the handlebar can be easily seen.**

**With zero rake (bottom) these effects are absent.**

This drop tends to work against the self-centering effect of castor because, to return deflected steering to the straight-ahead position, we must lift the considerable weight supported by the steering head.

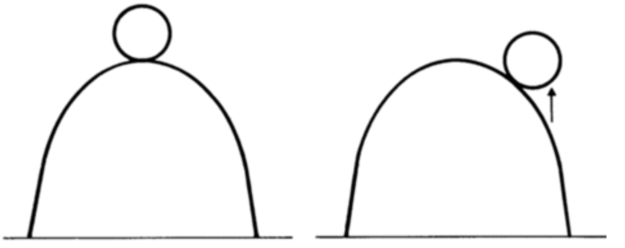


Fig. 3.13 **Analogous to steering head drop is this case of a ball on a mound. Virtually no force is necessary to keep it in place at the top. If allowed to fall, we must lift its weight to restore balance. This condition is know as unstable equilibrium, because any departure from its balanced state causes forces to further de-stabilize the system.**

While this effect is detrimental to balance (hence another reason why trials bikes have steep head angles), and to directional stability while travelling in a straight line, it helps to steer the wheel into a corner when banked over. However, it is important not to over-dramatize such effects. The following figs. 3.14 and 3.15. put the actual values into perspective. The first of which shows the front end drop and the second shows the steering torque needed to lift the bike back up. At 27 degrees rake and 45 degrees of steer the front end drop is only about 8.2 mm. and the restoring torque needed is below 6 Nm. for each 50 kgf. of front end weight. At less extreme handlebar angles the effect is much less, up to 10 degrees of steer the drop is no more than 0.5 mm. with a 27 degree rake angle.

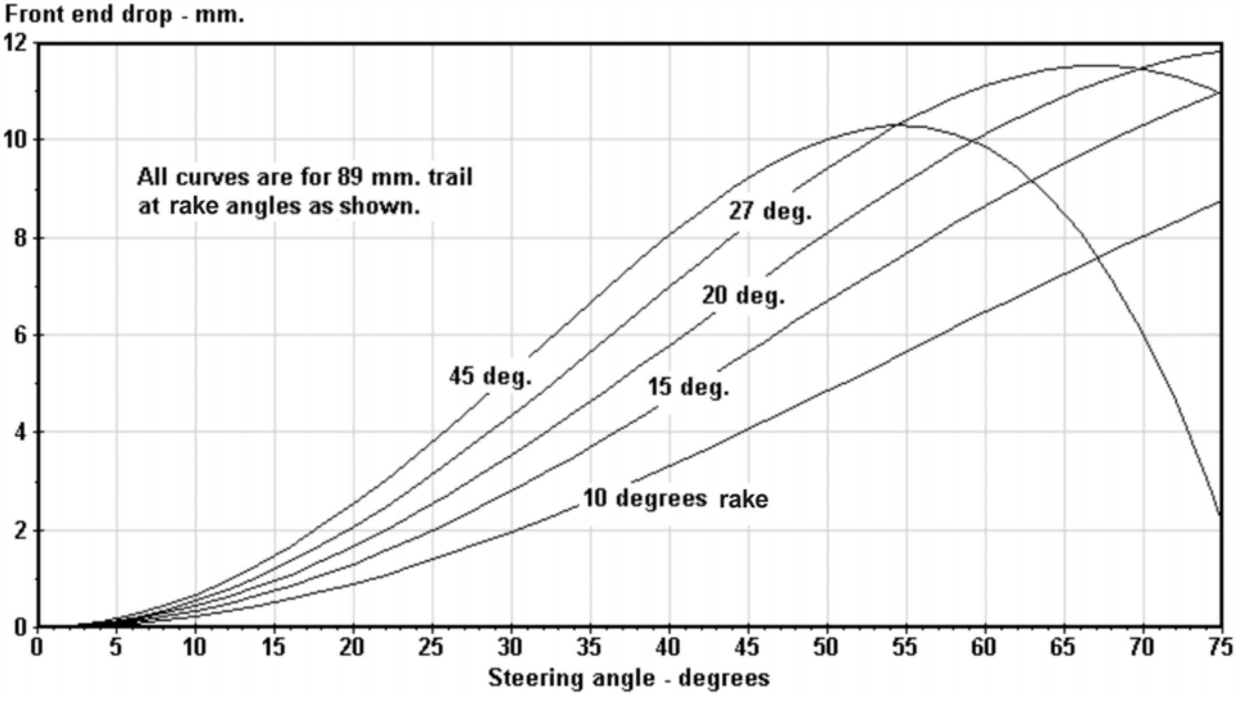



Fig. 3.14 **Front end drop for a vertical bike at various rake angles over a range of steering angles.**

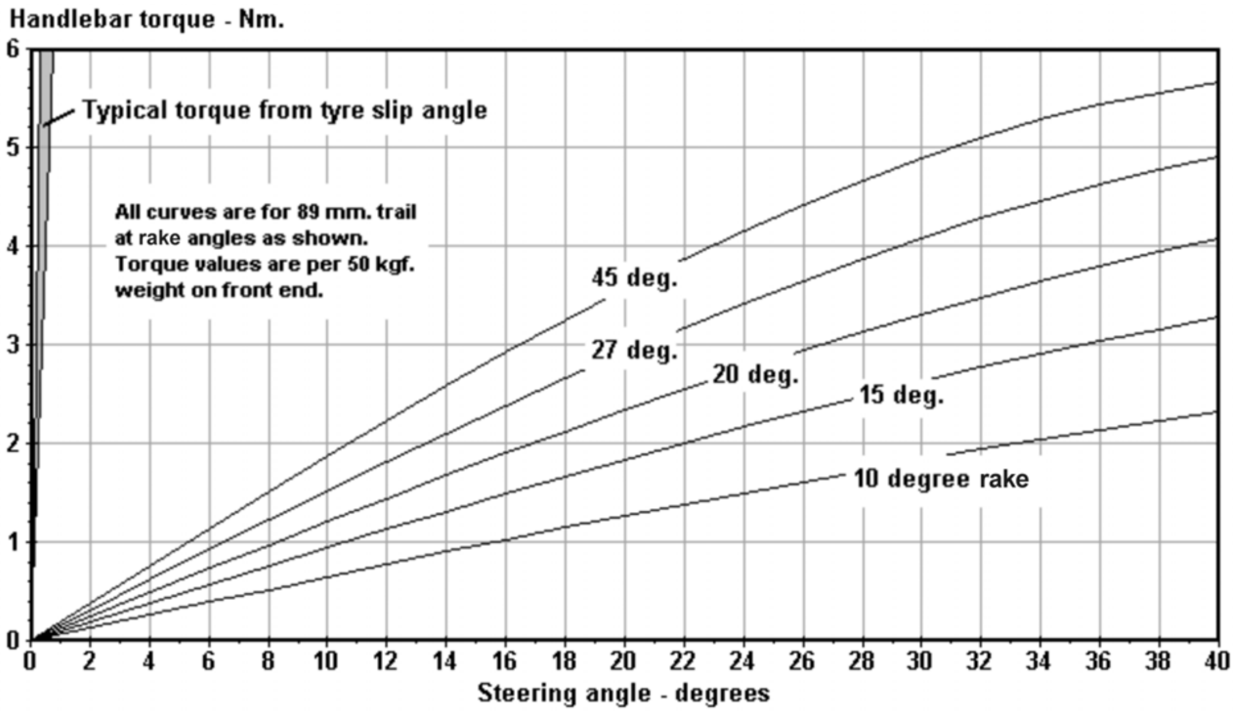



Fig. 3.15 **Average additional torque needed to restore the steering to the straight ahead position due to front end drop. Note that this torque is approximately only 1% of that due to the slip angle at low steering angles.**

## 4) Camber change

Figure 3.12 shows what happens with a 90 degree rake angle – camber change. Although the effect is less pronounced with conventional rake angles, it is still there – and it means that, when a motorcycle is cornering with the steering pointing into the corner, the front wheel leans over a little more than the rear.

## 5) Wheel-spindle offset

Reference to figure 3.7 shows the difference in offset required with different rake angles to achieve a given amount of trail, zero rake angle requires the greatest offset. However, since a normal rake angle tends to reduce the effectiveness of the ground trail dimension, a zero rake angle would require less trail, hence less offset. In general, all other factors being equal, it is an advantage to have the minimum offset, since this usually gives minimum steering inertia. In this respect, hub-centre steering seems to have much in its favour.

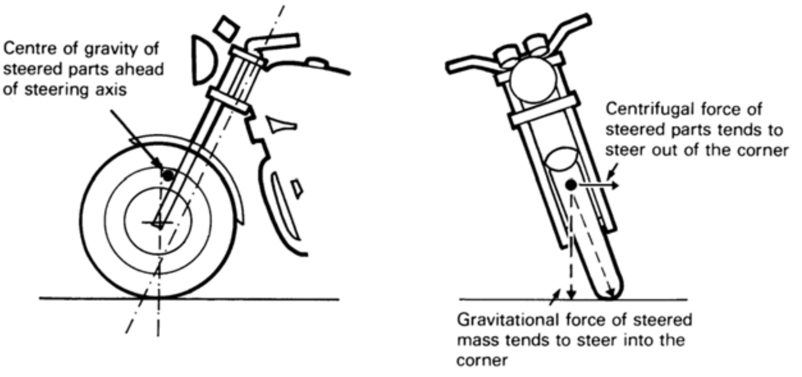



Fig. 3.16 **The forward offset of the CoG of the steered parts causes turning moments due to gravitational and centrifugal forces which balance one another provided that the rider's CoG is in the machine's centre plane. Leaning in farther than the machine reduces the gravitational moment and increases the centrifugal moment, hence tending to steer out of the bend.**

It is sometimes stated that, because the offset places the centre of gravity of the wheel and fork ahead of the steering axis, this produces a torque tending to steer the wheel into the curve while the machine is banked over. This is true only when the bike is stationary. Fig. 3.16 shows what happens during cornering, centrifugal force acts through the offset to steer the wheel *out* of the turn, but this effect is almost completely balanced by that of the gravitational force tending to steer *into* the turn. Actually with zero width tyres and the rider in line with the bike these two effects balance exactly. Hence, the offset has little nett effect on the self-steering characteristics of the machine, except when using very wide tyres.

## 6) Gyroscopic effects

When explaining the automatic balancing effects of gyroscopic precession in the next chapter, we will consider the subject as if the steering axis is vertical (i.e. zero rake). In the case of a typical rake angle the situation is modified, the components of the precessional forces acting as described are reduced and components are introduced that act in a contrary way, so reducing the effect of the gyroscopic forces.

## 7) Steering angle

A rake angle reduces the effective steering angle between tyre and ground compared with the angle through which the handlebar is turned. This can easily be seen with our usual trick of visualizing an extreme rake of 90 degrees. See figure 3.17. In this case, no true steering angle is developed but a camber angle is produced equal to the handlebar angle. At more normal rake angles, a small camber change is produced (as in 4 above), and for a 27 degree rake, this reduces the effective steering angle to approximately 90 per cent of the handlebar angle.

Fig. 3.18 is a contour plot showing the effective steering angles over a range of lean angle for both 0 and 27 degrees of rake angle. Note that the effective steering angle is actually increased with increasing lean angle

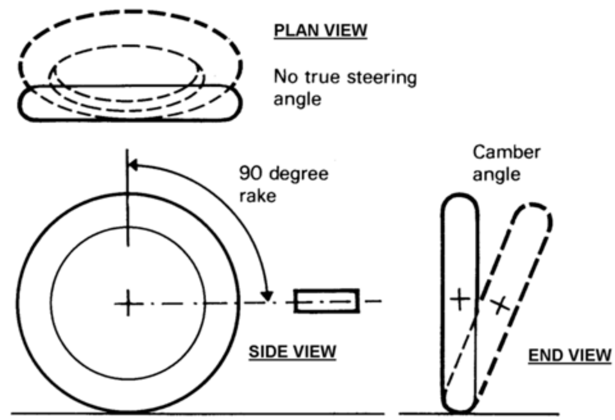



Fig. 3.17 **This extreme example shows how steering rake reduces the effective steering angle at ground level.**

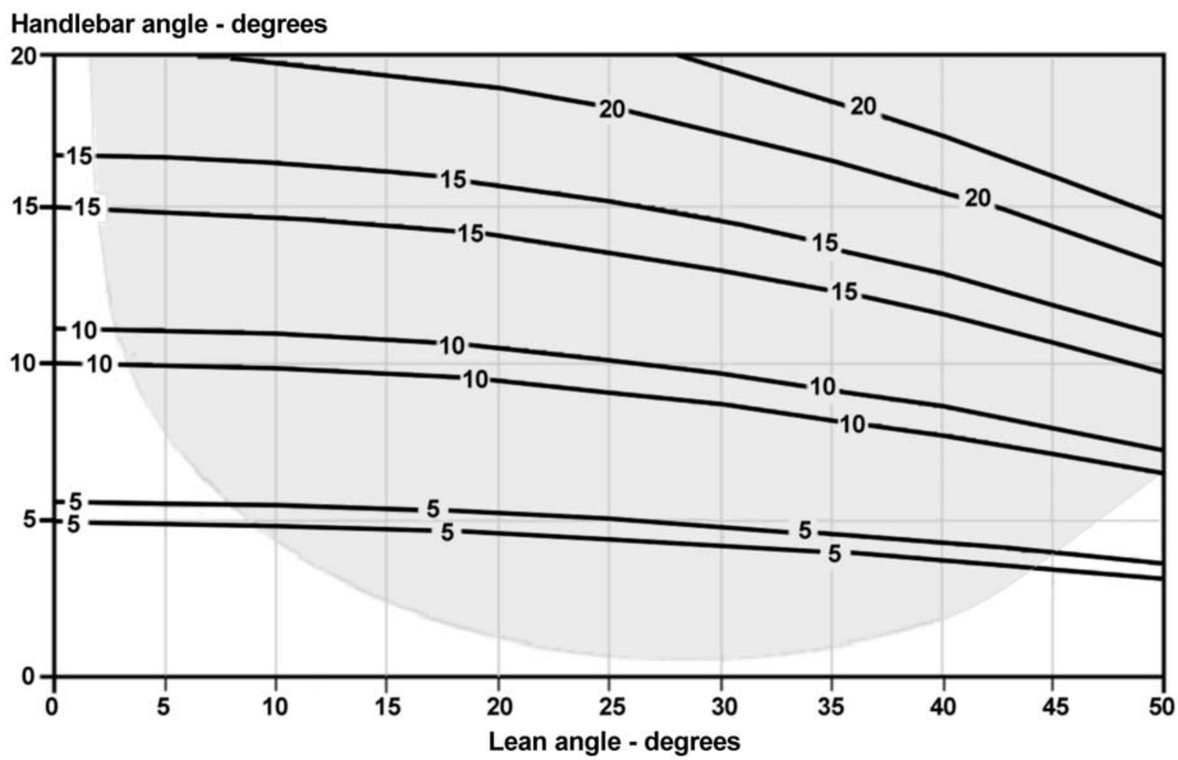



Fig 3.18 **True ground level steering angle plotted against lean angle for a range of handlebar angles. The upper curve of each pair is for a rake angle of 27 degrees whereas the lower line is for zero castor angle. Note for example that for zero lean , with 27 degrees of rake, the handlebar angle must be increased to achieve the required true steering angle. To have a ground level steering of 10 degrees needs approx. 11 degrees of handlebar angle. On the other hand, increasing lean angle tends to increase the effective steering angle, although this is relatively unimportant within the normal range of use. The shaded area is a rough guide to combinations of lean and steer which we are unlikely to use.**

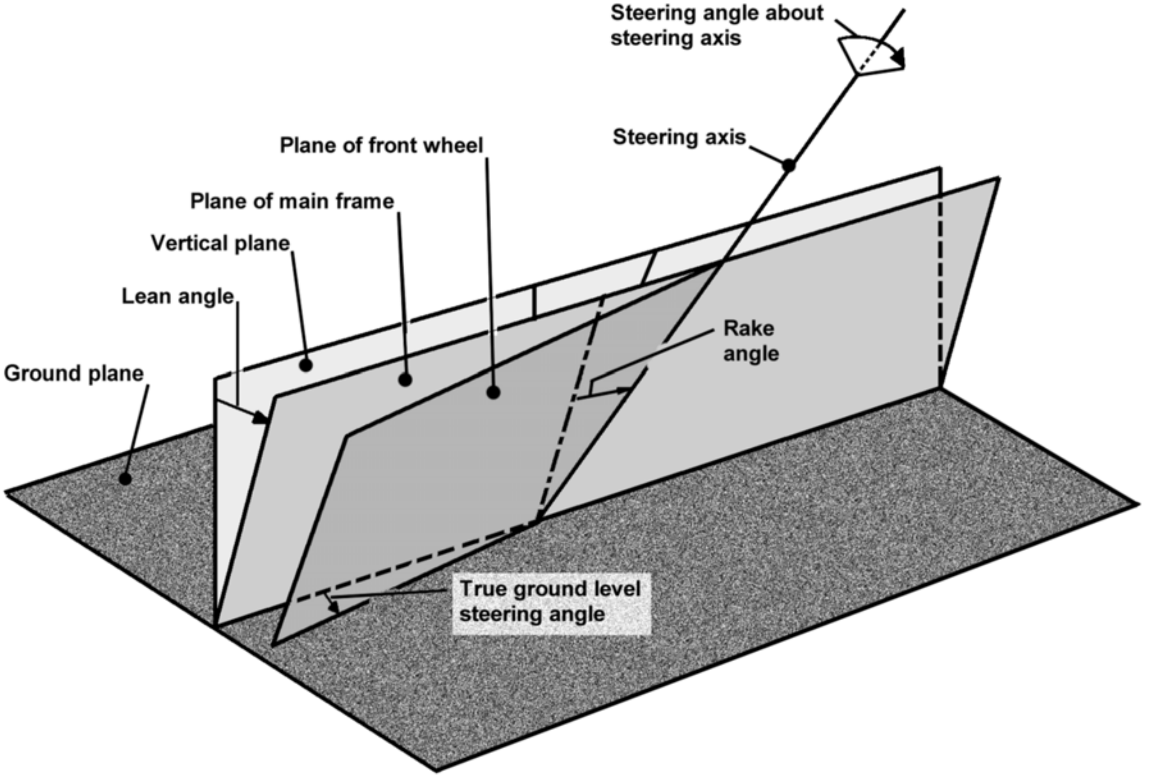



Fig. 3.19 **This representation of a bike, as two planes which intersect along the steering axis, can help visualize the various geometric aspects caused by both leaning and steering. One of the planes is that of the main part of the frame which includes the rear wheel, the other is the centre plane of the front wheel. The front plane can rotate about the steering axis contained on the rear plane and the rear plane can lean relative to the vertical.**

## Summary of rake effects

Except for the case of minimum offset in 5 (above) , it seems that a non-zero rake angle has no particular advantages. If this is so, why do conventional motorcycles handle and steer as well as they do? The answer lies in the very small steering angles involved in normal riding. The detrimental effects of rake become more pronounced at greater steering angles. The weight-assisted self-steering effect may or may not be beneficial – it is possible to have too much of a good thing. When cornering at any particular bank angle and speed we need a self-steering effect to give us just the right steering angle, too much and the rider must apply a reverse effort to the handlebar, too little and he needs to steer into the corner. The steering angle required for a given angle of bank depends on many parameters, and so it is not possible to build in a self-steering effect that is perfect for all conditions – which is just another example of the unavoidable need for compromise. A change of tyre type would be enough to alter things.

## Wheelbase

The distance between the wheel centres has several effects but, in general, the longer the wheelbase the greater the directional stability and the greater the effort needed to negotiate bends. There are three main reasons for this.

### 1) Required steering angle

Figure 3.20 shows how, for a given bend, a long-wheelbase machine needs the front wheel to be turned farther into the bend. Consequently more effort is required for cornering; also, a given deflection of the front wheel (say, from bumps) will have less effect on its directional stability.

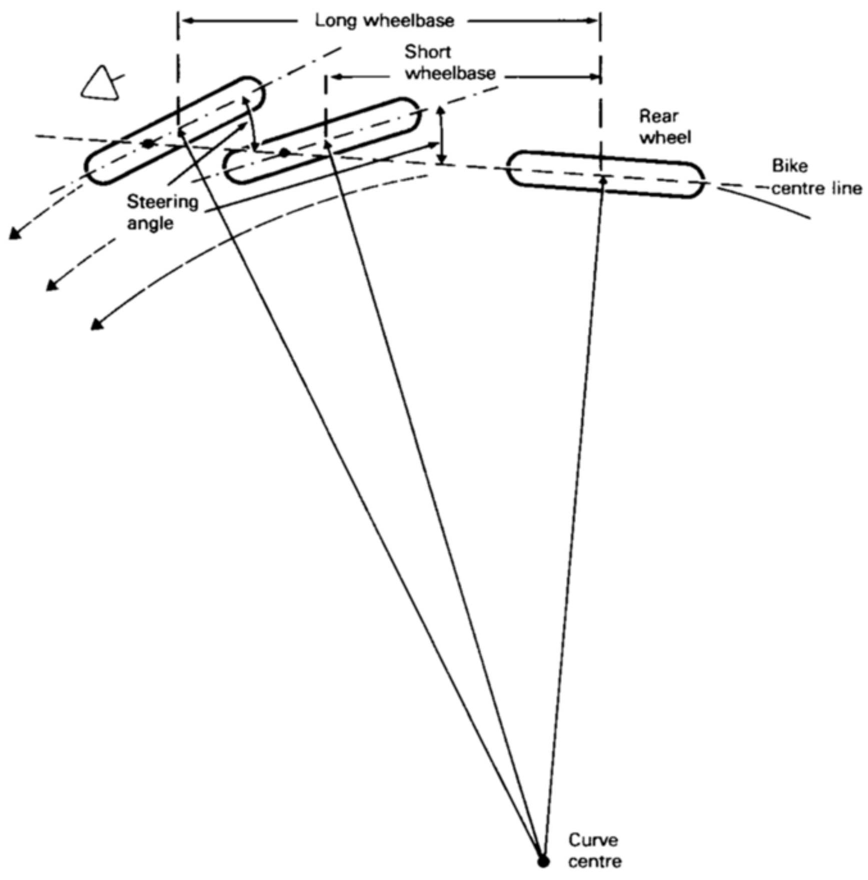



Fig. 3.20 **For a given radius bend, a longer wheelbase calls for a larger steering angle. This diagram shows this for Ackermann or kinematic based steering, but similar considerations apply to more realistic cases with real slip angles.**

A practical consideration for trials bikes is that, for a given degree of steering lock, the minimum turning circle is smaller with a shorter wheelbase. For this reason, trials machines have wheelbases as short as 1240-1270 mm.

### 2) Rear-wheel angle

It is also clear from figure 3.21 that, for a given sideways deflection, the angle of the rear wheel to the direction of travel is smaller with a longer wheelbase, thus improving directional stability.

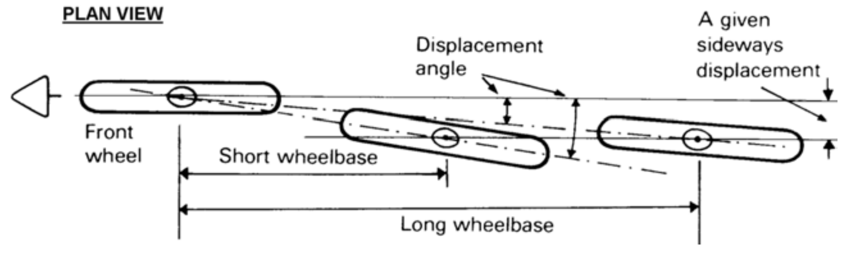



Fig. 3.21 **A long wheelbase enhances directional stability by reducing the displacement angle of the rear wheel.**

### 3) Inertia effects

The wheelbase has an effect on load transfer under braking and acceleration: for a given centre-of-gravity height, the longer the wheelbase the smaller the load transfer. In addition, the moments of inertia in the pitch and yaw planes are increased, which makes a long-wheelbase machine more sluggish and stable.

## Summary of wheelbase effects

In common with most design features, wheelbase is a compromise and varies with the intended use of the machine. Trials machines need good maneuverability, mainly at low speed, and so have short wheelbases. Large touring machines need good directional stability and slow pitch movements for relaxed riding but not the razor-sharp handling of a racer; hence touring machines have longer wheelbases (about 1470-1520 mm.), though too long a wheelbase impairs maneuverability in traffic. Some feet-first machines, such as the out-of-production Quasar, have wheelbases of 1960 mm. or so. A racer must compromise between the requirements of stability at very high speeds, quick handling and load transfer. Actual figures tend to be between 1270 mm. in the smaller classes and 1400 mm. for the larger, faster machines.

# Wheel diameter

With both road bikes and the racers that many emulate, wheel diameter has varied considerably with time. During the 1950s and at least some of the 60s most of the larger machines were equipped with 19 inch (483 mm.) rims. During the course of the 60s there was a definite trend toward the use of the 18 inch (457 mm.) wheel. This became the most common fitment until around the end of the 70s. when the 16 inch became slowly accepted. For the past decade or so there has been a halt to this continual reduction in diameter and 17 inch (432 mm.) has become the standard, recently however, there have been experiments with 16.5 inch (419 mm.) wheels on the larger racing machines. These 16.5 inch wheels have been used with taller tyres so that the overall diameter is similar to the normal 17 inch. As yet, there does not seem to be a definite consensus as to which is best, some riders reporting juddering with the new size, likely the result of the increased tyre profile height.

Tyre and wheel size is yet another area for the inevitable need for compromise, as there are advantages and disadvantages on both hands, as follows:

- For a given tyre section, a small wheel reduces both the unsprung mass (to the benefit of roadholding) and the steering inertia. This is welcome in all cases.
- Wheel size also effects gyroscopic forces. For a given tyre and rim section, these forces are proportional to the road speed and the square of the wheel diameter. Thus, bigger wheels will start to give their balancing effort at lower speeds. It was the continual widening of tyres for improved cornering ability that was probably the main driving force to reduce diameters, in order to avoid excessive gyroscopic reactions, which tend to slow the steering responses.
- As figure 3.22 shows, a smaller wheel drops farther into holes, similarly, it feels raised bumps more sharply. Fig. 3.23 demonstrates how a smaller wheel must mount a step in a shorter time than a larger wheel, this increased vertical velocity places more demands on the suspension system and transfers more shock back to the sprung mass of the bike. So touring machines need larger wheels for comfort and roadholding on rough roads, while trials and motocross machines have 21 inch (533 mm.) front wheels, the better to ride the bumps. True, these bikes have 18 or 19 inch rear wheels but the large tyre section there brings the overall diameter up close to that of the front.

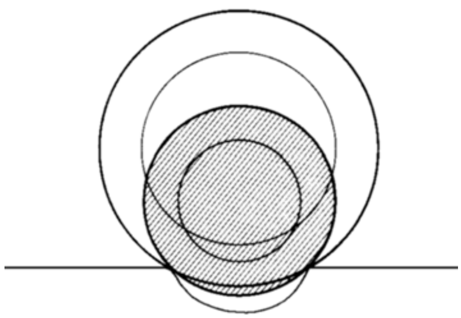


Fig. 3.22 **A smaller wheel drops farther into holes than a larger wheel.**

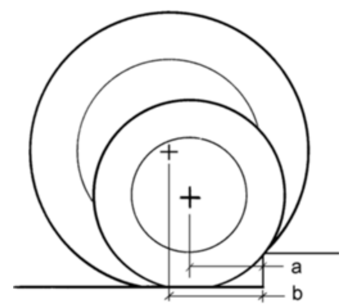



Fig. 3.23 **The smaller wheel travels a smaller distance "a" to mount the step than the larger wheel which travels "b".**

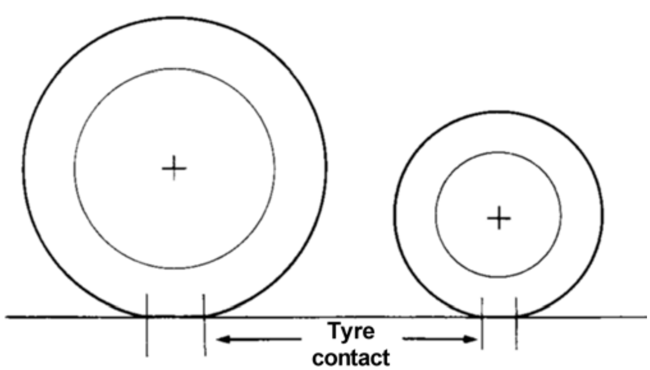



Fig. 3.24 **The larger diameter tyre generally gives rise to a longer contact patch. The exact shape and size of the contact area will also depend on width, inflation pressure and other tyre properties.**

- For a given tyre section, the area of rubber on the ground is generally greater with larger wheels. With smaller wheels, we could restore the area by widening the tyre (as is done in racing) but this can bring other problems, as discussed elsewhere.
- The self-steering effect of trail and rake mentioned earlier is emphasized by the use of smaller wheels.
- For purely structural reasons, smaller wheels are stiffer.

# Other considerations

## Frame stiffness

In building a chassis suitable for our purpose, we have more problems than those involved in simply achieving the best compromise between the various geometric parameters discussed so far. For if the chassis is not rigid enough to maintain our chosen geometry in use, then all our calculations will be set at nought. There are many sources of flexure in a motorcycle and all must be dealt with to achieve good handling. It is especially important to maintain the alignment between the centre planes of the wheels and the steering axis, otherwise directional stability will suffer and the tendency for wobble type instabilities may be increased. At the front end, which is the more important, this alignment is governed mainly by the lateral stiffness of the fork and wheel. Hub-centre steering scores heavily here because, in most layouts, only the wheel constitutes a possible source of flexure away from the steering axis.

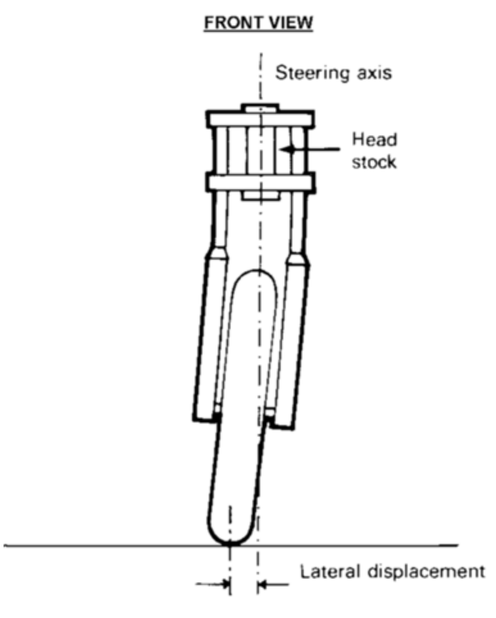



Fig. 3.25 **Lateral displacement due to fork and wheel flexure.**

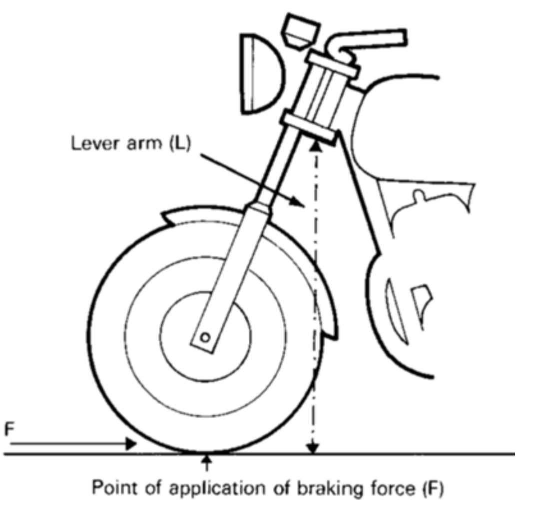



Fig. 3.26 **The long lever arm causes high bending moments in the fork legs and steering head, which can give rise to juddering and wheel hop.**

Telescopic forks supported by a conventional high steering head also lack stiffness in a fore-and-aft plane, but this is of less concern except perhaps under braking, when the weaker designs may give rise to shuddering and wheel hop. This is yet another problem reduced by hub-centre steering.

Keeping the rear wheel aligned with the steering axis involves not only the lateral stiffness of the wheel but also the torsional and lateral stiffness of the main frame and pivoted rear suspension.

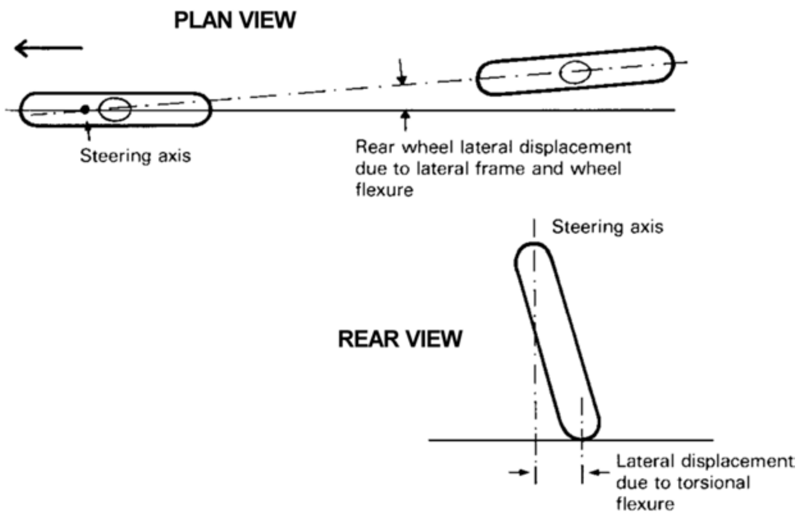



Fig. 3.27 **Lateral displacement of the rear wheel due to lateral and torsional frame flexing.**

In normal road riding, torsional stiffness between the handlebar and the front-wheel spindle is not often a big problem. However, it is very important in trials and motocross, where high steering torques are often necessary to get out of ruts.

A practical point that is sometimes over-looked in connection with handling is the lateral stiffness of the sub-frame supporting the seat. The rider receives much of his feedback on the machine's behaviour through the proverbial seat of his pants, if the seat itself is flapping about independently of the chassis he will get the wrong message. Many cases of suspected bad handling have been rectified by stiffening the seat support. Handling of the featherbed Manx Norton became noticeably more taut when the rear subframe was welded, instead of bolted, to the main frame; this modification also cured a spate of fractures of suspension-strut piston rods, probably caused by lateral flexure. Done properly there is no reason why the subframe can't be bolted on and many modern frames have reverted to this type of seat frame, and some racing machinery even just use a composite seat assembly without the use of a tubular structure. From the point of view of crash repair it has much in its favour.

## Weight (mass) and its position

Generally speaking, the less mass a machine possesses the better. Under the influence of a given force, the smaller the mass the more quickly it will accelerate. Not only does this mean a brisker performance for a given engine power; it also means a more responsive handling performance for a given effort by the rider. Just as important as the amount of the mass of the machine is its distribution and the location of the mass centre, as the following considerations show.

**Balance.** Low weight and a low centre of gravity both facilitate good balance. Figure 3.28 shows that for a given degree of lean, the unbalancing couple is directly proportional to the weight and the height of the centre of gravity.

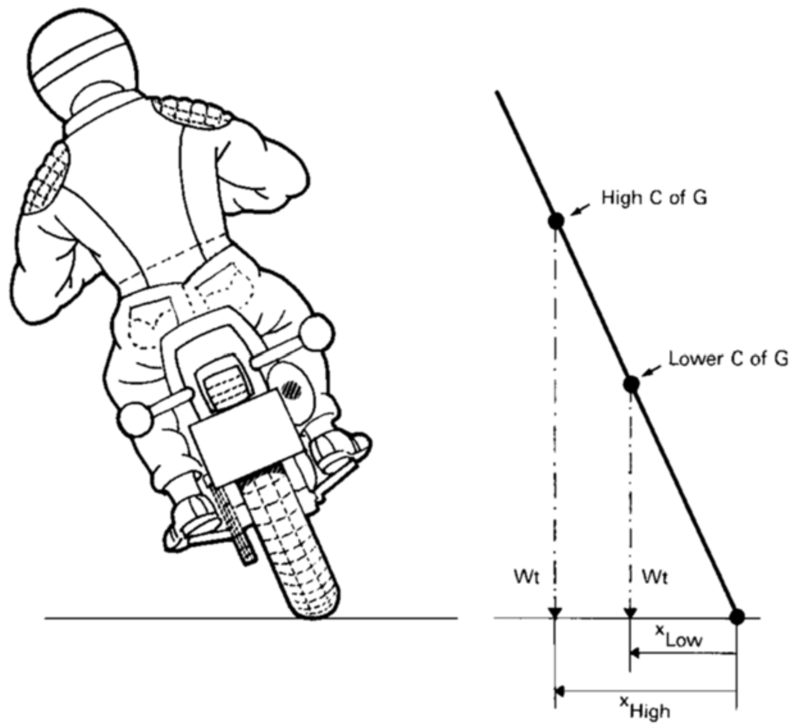



Fig. 3.28 **The unbalancing couple is equal to Wt.x i.e. the weight (Wt) multiplied by the lever arm (x). Since the lever arm is proportional to the centre of gravity height, a higher C of G gives a greater unbalancing effect.**

**Load transfer.** Under braking, vertical load is transferred from the rear wheel to the front, under acceleration, the transfer is in the opposite direction. Lengthening the wheelbase decreases the load transfer, as also does lowering the mass centre height and reducing the mass. Load transfer is not affected by the longitudinal location of the centre of gravity, though this controls the static weight supported by each wheel.

**Traction.** Since the driving force that the rear wheel can deliver to the ground is limited by the load carried by the wheel, a rearward weight distribution improves traction. However, we must balance this requirement against the need to keep the front wheel on the ground for steering. A forward mass bias also helps directional stability, as it does in a dart or an arrow.

**Angle of lean.** The angle of lean necessary to balance centrifugal force when cornering is slightly affected by the centre-of-gravity height. See Figures 3.29 and 3.30.

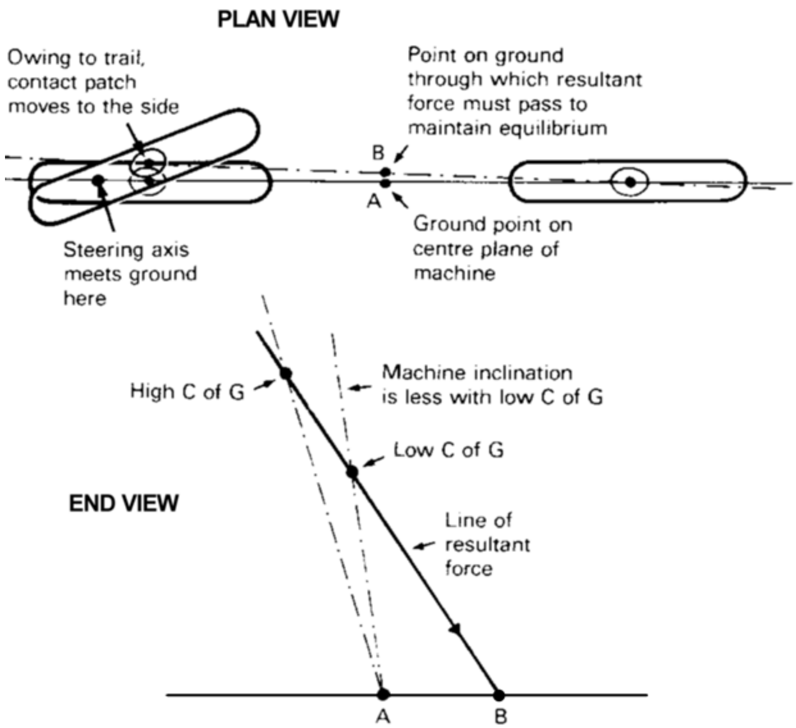



Fig. 3.29 **Steering into a corner moves the line through the contact patches outward and a lower centre of gravity then requires a smaller banking angle, the effect is small in practice and countered by an opposing effect due to the finite tyre width. Shown below.**

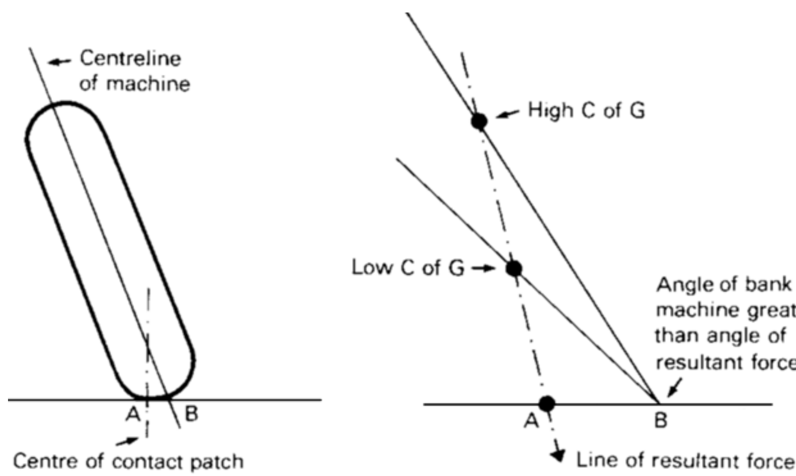



Fig. 3.30 **This shows the different angles of lean required with low and high centres of gravity, due to the width of the tyre. This effect opposes that shown above.**

## Angular motions

As far as linear motions are concerned, it is the *amount* of the machine's mass that is important. But when it comes to the angular motions of pitch (about a transverse axis), yaw (about a vertical axis) and roll (about a longitudinal axis), the *distribution* of the mass is all-important because that governs what are called the moments of inertia. These are a measure of the inertia effect about the particular axis and its value determines the ease with which we can apply angular acceleration to the machine about that axis.

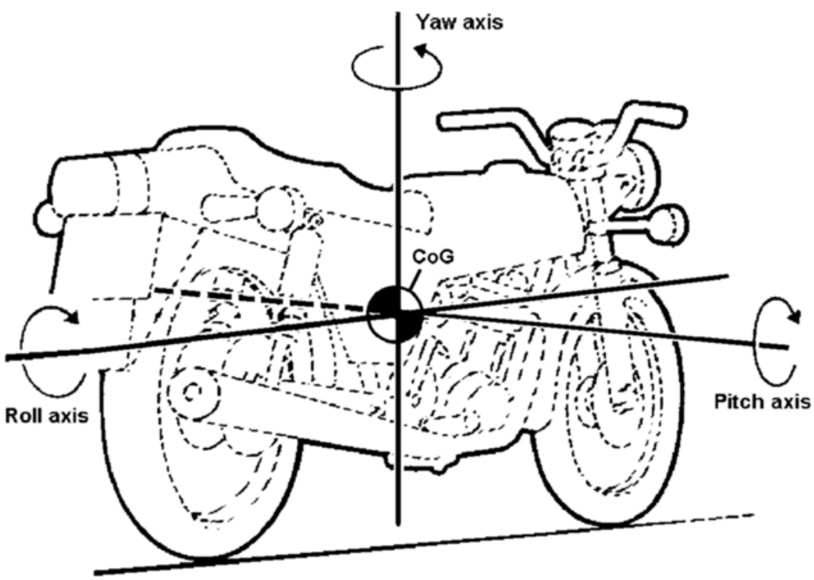



Fig 3.31 **The three principal axis about which a machine can rotate. This just shows the direction of these axis, drawn through the CoG. Their actual location depends on many dynamic factors as shown in other chapters.**

**Pitch.** Pitch inertia controls how rapidly the bike pitches forward or backward under the various inputs due to braking, accelerating and road bumps. Except in trials and motocross, there is no great need for a fast pitch response and so a large amount of pitch inertia is not generally harmful. Indeed, it may contribute to comfort when braking or when traversing bumpy surfaces, as shown in the chapter on suspension. It is not easy to geometrically define the axis about which a machine pitches because it varies with the configuration of the bike. For example, if a machine is sprung at the front but not at the rear, it will pitch about the rear-wheel axle, while a machine with the opposite arrangement (i.e. sprung rear, rigid front) would pitch about the front-wheel centre. In the case of a conventional machine, sprung at both ends, the pitch axis depends on suspension geometry and spring rates. Of the three angular motions, pitch is unique in that it comprises mainly of movement of the sprung part of the machine with respect to the wheels, whereas roll and yaw are global movements of the bike including wheels.

**Yaw.** Any vehicle will be subject to some degree of yaw acceleration during corner entry. In this case there are conflicting requirements for both a high yaw moment of inertia and a low one. For example, a high value enhances directional stability while a low value facilitates rapid changes of direction and minimizes the effects of a slide. Within practical limits, it is found better to aim for a low yaw moment of inertia, which involves concentrating the mass of the bike as close as possible to the longitudinal centre. Naturally, this tends to produce a low pitch moment of inertia also.

**Roll.** The roll moment of inertia is the sum of all the individual components of the total mass multiplied by the square of their distance from a fore and aft axis drawn through the CoG. A low roll moment of inertia is desirable for a rapid and effortless change in banking angle.

The terms roll axis and roll centre have for long been used in the car world and are geometrically defined by the lateral suspension layout of the car, at any particular body lean angle. Fig. 3.32 shows the derivation of the roll centre as applied to a car. The roll axis is simply the axis joining the front and rear roll centres, which are normally at different heights. It can help visualize the roll motion of the body with respect to its wheels and CoG. In this age of computer design the use the roll centre concept is largely seen as outdated in the four wheeled world.

It is ironic then and unfortunate also, that these terms have recently become somewhat fashionable in the motorcycle world without any corresponding relevance. Motorcycles, being single track vehicles don't have any similar side to side suspension mechanisms and so the relevance of this term to motorcycles is hard to justify and consequently is most often completely misunderstood.

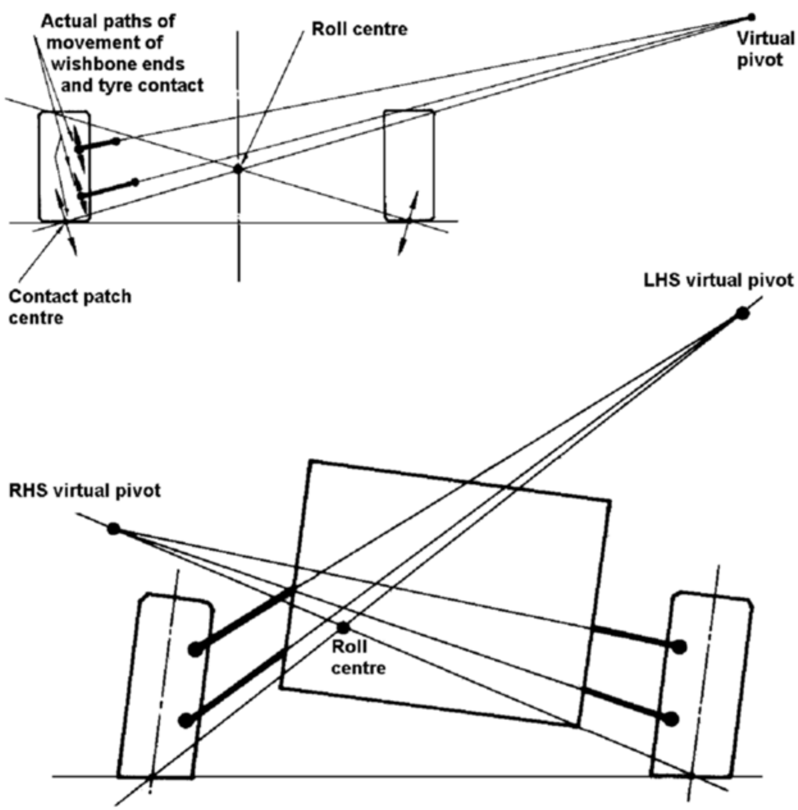



Fig 3.32 **The upper sketch shows how, with a double wishbone system, a virtual pivot point is defined.**

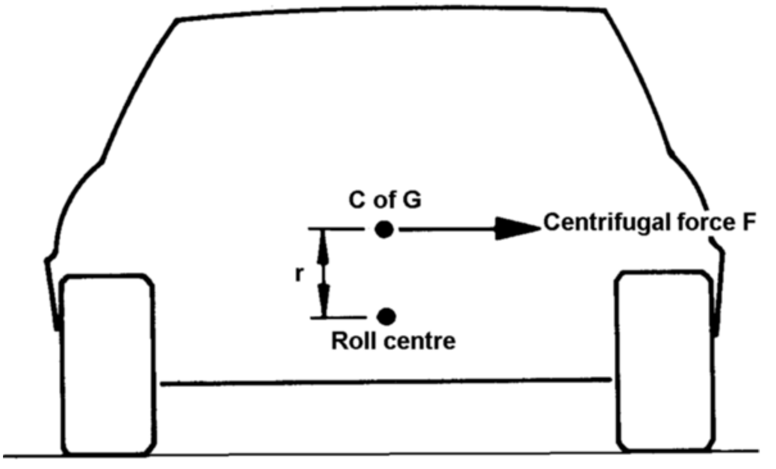
**The instantaneous movement of the tyre contact patch is at right angle to its radius from the virtual pivot. Where these right hand side and left hand side radii meet is defined as the roll centre. This is the only point which satisfies the motions of left and right hand sides, at any roll position. (see appendix 6)**

**The lower sketch shows how the roll centre moves as the car rolls and the wishbone angles differ from side to side.**

Fig. 3.33 shows how a centrifugal cornering force can be considered as acting through the centre of gravity which is normally above the roll centre. The distance between the CoG. and the roll axis produces a torque (usually called the roll couple) which causes the car body to lean.

Fig 3.33 **During cornering the centrifugal force produces a torque about the roll axis equal to F.r, this is called the roll couple or roll moment.**

**If r = 0 then no roll occurs but this causes other undesirable effects such as suspension jacking up.**




It is easy to see that there is no corresponding analogous concept between cars and bikes in this regard. Unlike a car the roll moment generated by steady cornering force on a motorcycle is exactly balanced by the opposite roll moment created by gravity acting on the leant over machine. It is only during the transient lean-in phase that any unbalanced roll torques exist and these are not caused by the same mechanisms as those shown above for a car. This will be explained elsewhere. If we were to apply the car criterion for determining the roll axis to a motorcycle then it would have to be near ground level, through the tyre contact patches, to produce the appropriate roll couple.

It might seem better to develop a motorcycle specific roll axis term, but in reality we can define a roll axis to be anywhere we want it to be depending on our frame of reference. We can be viewing from outer space, from the earth's surface fixed to one point, from the earth's surface following the bike, or from different positions on the bike. Each different view point will produce a different apparent roll axis.

Imagine watching the rear of the bike from a point fixed on the ground, and let the machine be half way through the lean-in phase. At this point the bike will be moving laterally toward the centre of the turn but will also have a rotational motion as shown in fig. 3.34. These two motions add together to produce a combined motion at the CoG. as shown in fig. 3.35. Clearly then, the centre of rotation of this larger arc is some way below ground level. As the relative lateral and rotational components of the motion vary throughout the cornering process then a roll axis as defined in this way will vary in position also. This is one valid way of looking at roll motions but it is hardly of any use to us.

Returning to fig. 3.34, let’s consider things a little differently. Viewed from the ground all points on the machine will have some combination of rotary and lateral motion except for the centre of the tyre profile. This has only lateral motion and all the other points will be rotating about it. Therefore if we need a conceptual roll axis then this would seem to be the most logical one to use.

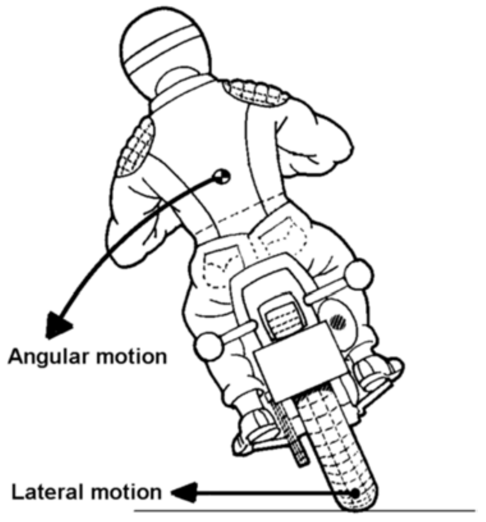



Fig 3.34 **Shows both the angular and lateral motions**

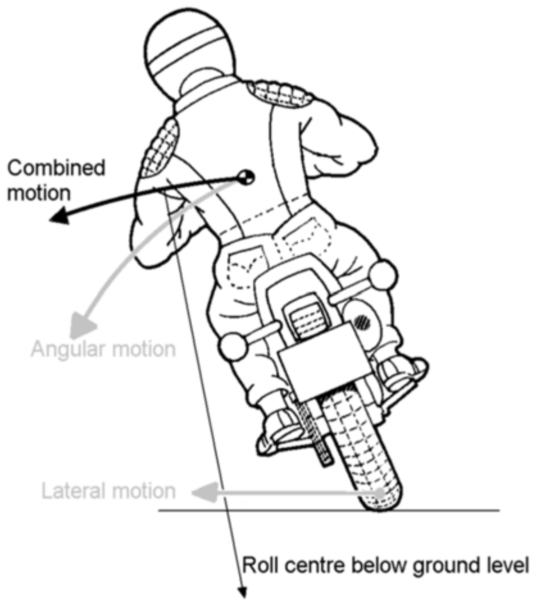



Fig 3.35 **Adding the two motions produces a combined motion with a larger radius as shown.**

It is sometimes suggested that the bike rolls about its CoG., but it is not difficult to see that if this were so then the tyre contact patch would have to rise up off the road surface as the bike rolled. Even though for estimating roll performance we normally calculate the roll moment of inertia about an axis through the CoG., as we shall see when we discuss the dynamics of cornering, this is not to say that this has to be considered as the roll axis. In practice there seems be no useful reason to develop a roll axis concept for motorcycles that is analogous to that used for cars.

# 4 Balance and steering

## Balance

As a single-track vehicle, a motorcycle lacks inherent static balance, i.e. it falls over if left to its own devices when stationary. Once moving above a certain speed however, even the most uncoordinated riders find that the machine seems to support itself.

Therefore, we can see that there are two aspects of the balance process, the low speed case and that in the higher speed ranges. There have always been those, like expert trials riders, who can balance indefinitely on a stationary bike, but this is exceptional and most of us need a minimum of forward motion before this is possible. However, at these low speeds it is necessary to move the handlebars from side to side to stay upright, and as all trials riders know, it is easier if we stand on the footrests instead of sitting down.

Let's examine why. Fig. 4.1 shows the rear and top views of a bike and rider. Now, if the combined centre of gravity (CoG) of bike and rider, is vertically above the line joining the front and rear tyre contact patches, then balance is achieved, but this is an unstable situation, any small disturbance such as a light breeze will be enough to start a topple over, i.e. the CoG moves sideways.

This can be prevented by either of two methods or a combination of both, one is to move the tyre contact patch line back underneath the new position of the CoG. We can steer the bike to place the position of the tyre line wherever we need it, and this is one reason why it is easier to balance when moving.

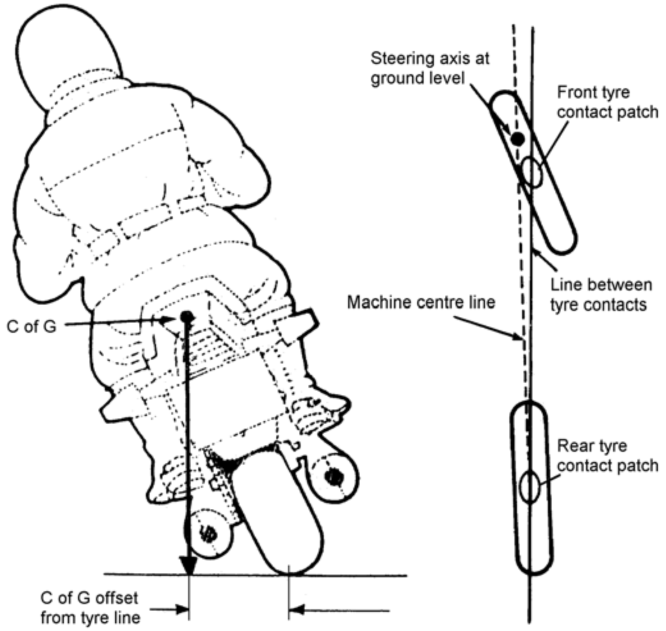



Fig 4.1. **The rear view shows that if the CoG is not vertically above the line joining the tyre contact patches, then an over-balancing moment is generated. However, the plan view shows how the sideways relationship of the machine centre-line and the tyre line can be changed by steering from side to side. This is of great importance to very low speed balance.**

The other way to maintain low speed balance is by moving the combined CoG of both the rider and machine back over the line joining the tyre contact patches. If the bike is stationary this can only be done to a limited extent by moving the handlebars, because of the front wheel trail the centre-line of the motorcycle will move inboard of the contact patch as we turn the bars. This is also what trials riders are doing when moving their bodies from side to side whilst standing up. The high CoG of the rider has more effect on the toppling over moment and also gives more control over the position of the bike's CoG. Thus to a great extent the process of low speed balance is dependent on the individual skill of the rider. In addition, some bike parameters can also affect the ease of remaining upright, the main ones being:

- A low CoG height helps.
- A large trail changes the position of the tyre line more for a given handlebar movement.
- A small rake angle reduces the fall of the steering head when the bars are turned away from the straight-ahead position, assisting with the balance process.

The balance mechanism at higher speeds is more complex, but at least is largely automatic and largely independent of rider ability. To understand the action fully it is helpful to look at a few properties of gyroscopes, which is another way of describing spinning motorcycle wheels and the rotating parts, mainly crankshaft and clutch, of the engine and gearbox. *(The reader is advised to refer to Appendix 4 where this subject is explained more fully, but which is briefly revisited here for completeness.)*

*A spinning wheel is often said to have a stable axis of rotation, due to the conservation of angular momentum, i.e. a tendency to maintain its plane of rotation. In other words, we have to apply a torque to change that momentum. If the torque is applied about an axis at right angles to the spin axis then the momentum change is manifest as an angular velocity about a third axis at right angles to both of the previous, this is known as gyroscopic precession. Applied to a bike, this means that when leaning into a corner there will be a roll velocity which must be accompanied by a corresponding steering torque. Such effects are proportional to the rotational velocity of the wheel and so become more important at higher road speeds. Nowhere are these laws of nature better demonstrated than with a spinning top, which can balance on a point whilst spinning fast but topples over when rotating slowly or when stationary.*

*A few experiments with a bicycle wheel will not only show the strength of the effect but will also show some of the subtleties that contribute to both balance and steering.*

*Firstly, hold the wheel upright, as in fig. 4.2, get someone to spin it briskly so that the top of the wheel is moving away from you, as if it were the front wheel of a machine you are riding. If you then try to tilt the spindle to the LEFT (equivalent to banking your machine) you will find that the wheel tries to turn instantly and strongly to the LEFT, as if steered by an invisible hand. In other words, your attempt to tilt the wheel about its fore-and-aft axis by applying a torque has produced a angular velocity about its vertical axis. Now start again but this time turn the wheel to the LEFT about a vertical axis, just as sharply and strongly it will try to bank to the RIGHT. Try both these manoeuvres again, but do it at different wheel speeds and tilting torques, you will see that the precessional reactions depend strongly on these factors. Note particularly, the directions in which these reactions operate, as this is important for the automatic retention of balance.*

Let us now see how these and other forces help keep the machine balanced and on a relatively straight path without assistance from the rider. Suppose the bike, whilst travelling along at a normal speed, starts to fall to the left under the action of some extraneous influence. As we have just seen, gyroscopic precession of the front wheel tends to steer it to the left. Also, as shown in chapter 3, trail will also cause a left steering effect when the bike leans left. This sets the machine on a curved path (to the left), so creating a centrifugal force (to the right), which counters the lean and tends to restore the machine to the vertical, the gyroscopic reactions are thus reversed tending to restore the steering to the straight ahead position. In practice, that which we regard as riding in a straight line, is really a series of balance correcting curves, if we could look at the actual paths taken by the centre-lines of the wheels, we should see that the front wheel path continually crosses that of the rear.

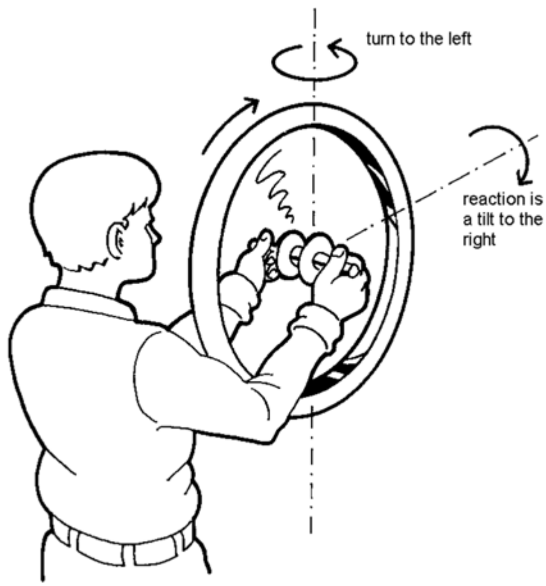



Fig. 4.2. **Gyroscopic precession. When a bicycle wheel, spinning as shown, is turned to the left it tilts strongly to the right. However, when it is tilted to the left it turns to the left.**

In the explanation above, we only considered the effects on the front wheel, gyroscopic and steering forces are at work on the rear also, but it is much harder to steer the rear wheel independently, as the whole bike must yaw, rather than just the wheel and forks, as on the front. Hence, only a small contribution is made to the auto-balance mechanism by the rear. We have now considered balance in a straight line, but as we lean when cornering, there must be other factors at work to maintain equilibrium under these conditions.

## Steering

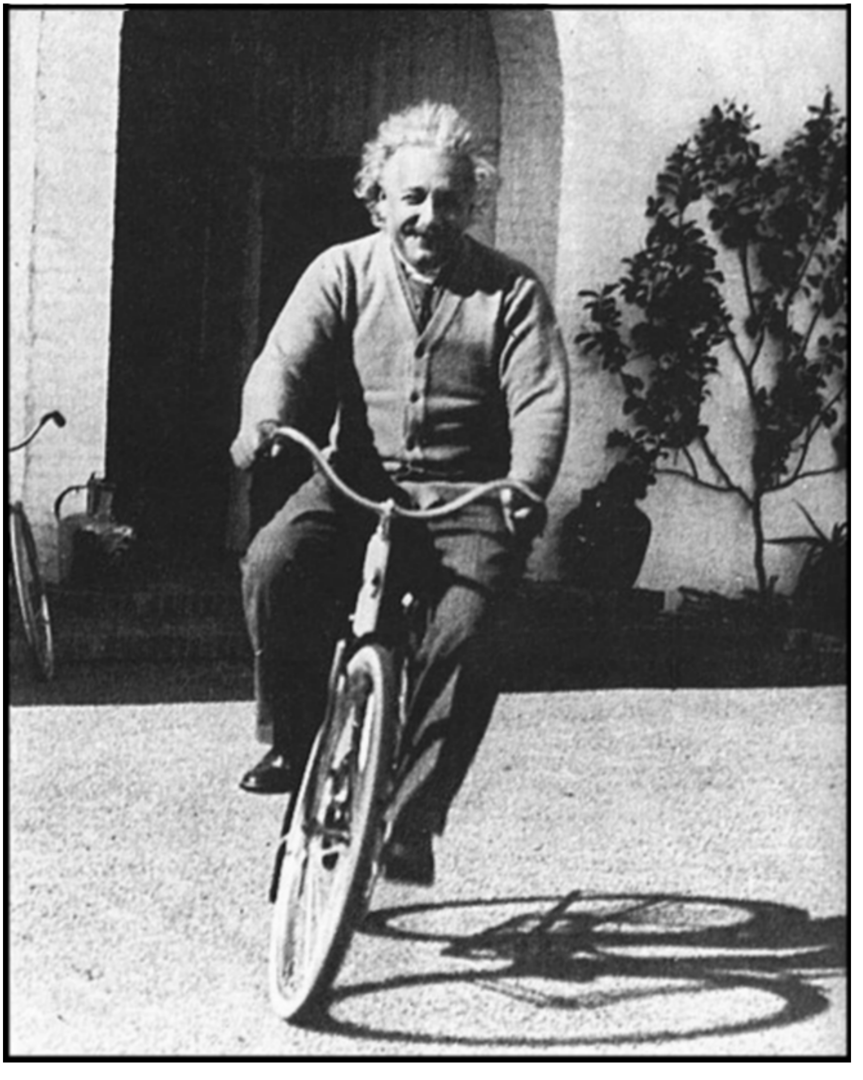
So much for balance. Our next problem is steering and, for the purpose of analysis, we can divide this into two phases:

- initiating the turn.
- maintaining the turn.

## Maintaining a turn

Since the second phase is easier to analyse, let us consider it first. It is not feasible to steer a solo motorcycle through a corner in a substantially upright position, as it is with a sidecar outfit or car, because the centripetal force generated by the tyres would cause it to fall outward. Hence, we must bank the machine inward so that this tendency is counteracted by the machine's weight tending to make it fall inward. Equilibrium is achieved when the angle of lean is such as to balance the two moments – one due to centrifugal force acting outward and the other to gravitational force acting inward. Fig. 4.3.

The actual angle, depends on the cornering or lateral acceleration, which in turn depends on the radius of the turn and the speed of the machine, fig. 4.4 shows the relationship between speed, lean angle and corner radius. The lean angle is that at which the resultant of the two forces passes through a line joining the front and rear tyre contact patches, fig. 4.3 shows the basics necessary for cornering balance.



**We can only speculate as to whether Albert Einstein saw this bicycle as merely convenient transport, or was it the object of an investigation into the laws governing the behaviour of single track vehicles?**

**In any event it is unlikely that he saw in it any need for the application of his theories of relativity. He would have been quite happy to have stayed with Newtonian physics in this application.**

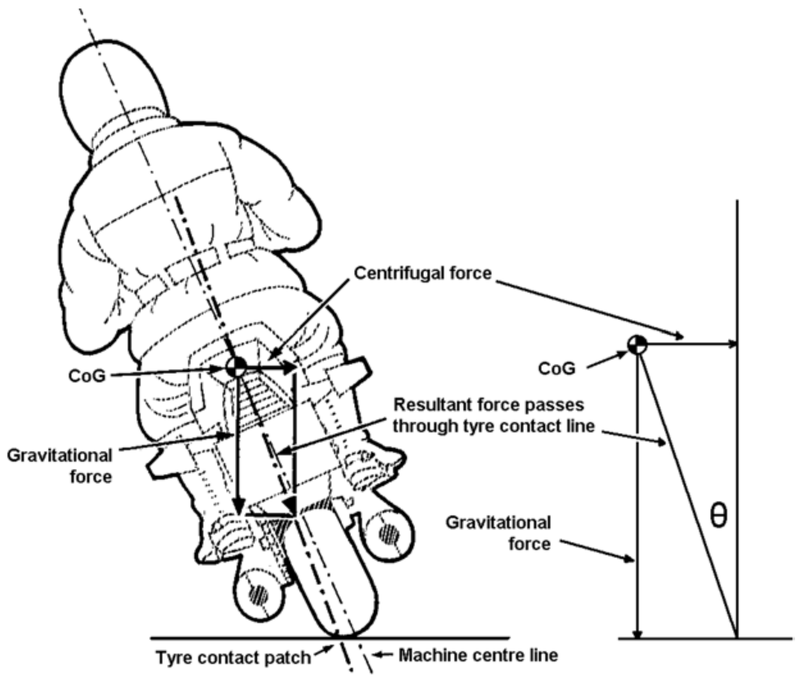



Fig. 4.3 **Equilibrium in steady cornering is achieved when the resultant of centrifugal force and gravitational force (both acting through the mass centre) passes through the line joining the contact patches of the front and rear tyres.**

**If $\theta$ is the lean angle measured through the tyre contact patch and $A_l$ is the lateral cornering acceleration expressed in Gs, then**

**$\theta = \arctan A_l$.**

**When $A_l$ is 1G. then $\theta = 45°$ and**

**when $A_l$ is 0.5G. then $\theta = 27°$.**

***A conceptually better (though little used) way of looking at this is to consider the centripetal and gravity forces acting at the tyre surface. The lean angle remains the same in any case.***

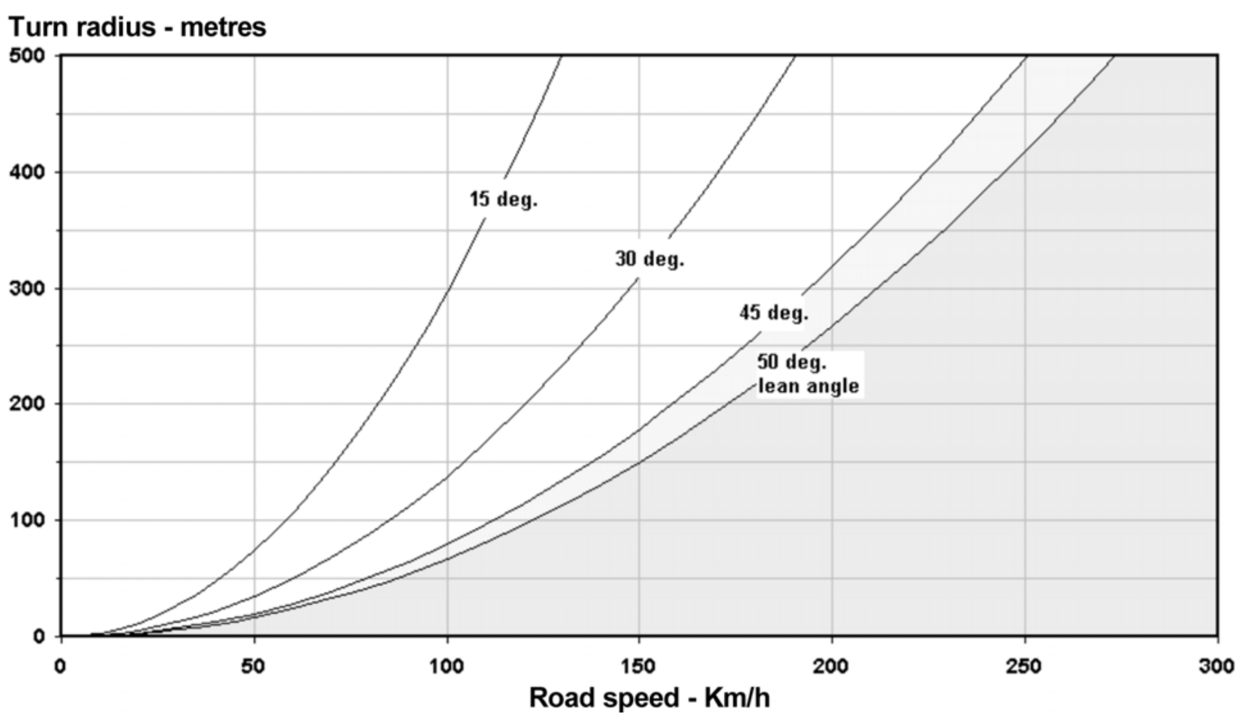



Fig. 4.4 **This graph shows the turn radius that corresponds to various road speeds for different lean angles.**

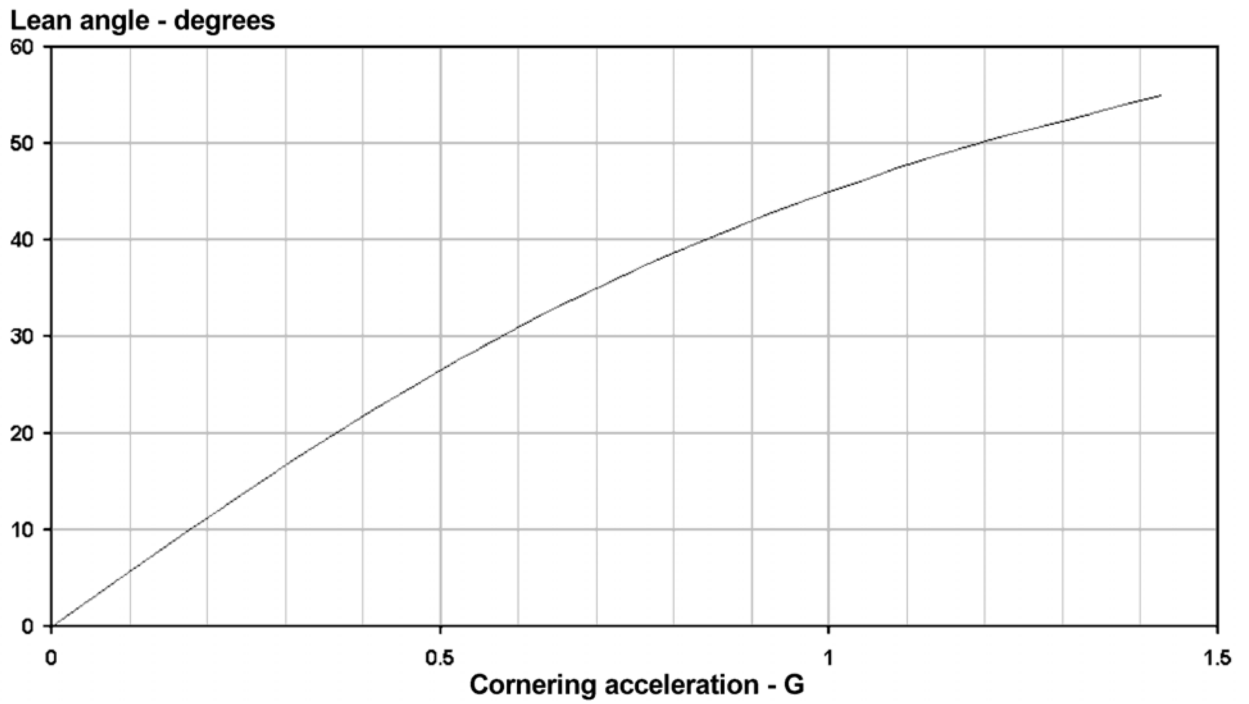



Fig. 4.5 **Relationship between lateral cornering acceleration in Gs. and the required lean angle to maintain balance. The lean angle is measured through the tyre contact patch and the CoG of the combined bike and rider.**

There is a secondary effect that influences lean angle. Gyroscopic precession from the rotation of the wheels and engine parts about the centre of the corner needs a balancing roll torque tending to lift the machine, this must be balanced by an increased lean angle of approximately 1 degree at 1G. cornering acceleration. A detailed explanation is given in the appendices.

## Initiating a turn

### The basics

How do we actually initiate the turn, we have already seen that a moving motorcycle has an automatic tendency to stay upright? Do we lean first or steer first? If we were to turn the handlebar in the direction we want to go, both centripetal tyre force and gyroscopic precession of the front wheel would cause the bike to topple outward. Therefore, if we momentarily turn the bar in the opposite direction, then the centripetal tyre force and, to a much lesser extent, the gyroscopic reactions will cause the machine to bank to the correct side. There are four main effects to be observed.

- A steering action to the *right*, will cause the machine to start turning *right,* and just as in a car, centripetal tyre force will cause a lean to the *left*. This is the major banking influence.
- This steering action as we have seen, will also produce a small precessional tendency to lean the machine to the *left.* This is a small effect when the wheels are on the ground, as shown later.
- Gravity will then initially augment the banking effect, but this will become less important as the tyre cornering force builds up and balances the gravitational moment completely, when the bike reaches the steady state lean angle.

- The velocity of banking or roll rate will give rise to gyroscopic torques which oppose the rider's counter-steering input helping to steer the front wheel into the curve. This gyroscopic torque is in opposition to the rider applied steering torque, and in fact balances most of his input and hence works against rapid steering. However, without this "negative feedback" the bike would be rather unstable and very hard to control, as we shall see.

These forces will also act on the rear wheel which, because it is rigidly attached to the bulk of the machine, will tend to make the machine yaw into the curve. However, this reinforcing effect is secondary to that of the front wheel. Steering rake and front-wheel trail, also help steer the machine into the curve as the lean angle builds up. When we have established our correct lean angle, the processes for maintaining balance, as described above, will come into effect and help keep the bike on our chosen path.

We have seen, then, that a turn can be initiated by steering momentarily in the "wrong" direction. Termed "counter-steering", for most riders this action is accomplished subconsciously. In racing, riders often make use of deliberate counter-steering to achieve the high roll rates necessary under those extreme conditions. Briefly, it is the combination of gyroscopic moments and centripetal force that requires this counter-steering action, we don't have a choice in the matter. There are those that would have us believe that counter-steering wasn't known about until the 1970s. or '80s.. This is nonsense, it is well documented that around the start of the 20$^{th}$ century the Wright Bros. were well aware that this was the mechanism for turning a bicycle. In the early 1950s., whilst chief engineer at the Royal Enfield motorcycle factory, Wilson-Jones did a series of tests with real motorcycles to investigate this further. The results of these and some of his other experiments into steering geometry were published in engineering journals.

However, counter-steering doesn't explain how we can corner "hands-off". Although, whilst it is possible to do this, it is accomplished only with a lot more difficulty. So let us consider what happens if we try to lean without being able to steer. As there is nothing solid for us to push against, the only way we can apply bank is to push against the machine with the inertia of our own body. To lean the bike to the left, we must therefore initially move our body weight to the right. The left leaning bike will now generate camber forces from the tyres tending to lean both rider and machine over to the right, the roll rate will again cause a gyroscopic steering torque which helps ensure correct balance. The initial bike lean to the left might well be considered as a 'counter-lean', analogous to the 'counter-steer' of hands-on turning. Anyone that has tried changing direction 'no-hands' will know that we have far less control over the machine with just body movement available. The mechanisms involved with counter-steering produce much greater response and more finesse of control.

So, we now have two possible methods of initiating a turn and it is interesting to note that in both of them (banking and counter-steering) our physical effort is in the opposite sense to that which might be thought natural. When learning we adapt quickly and the required action becomes automatic. It is these reverse actions that require us to learn to ride in the first place. The required responses are clearly counter intuitive. When learning most of us initially wobble about out of control until our sub-conscious latches on to the fact that counter-steering and counter-leaning is the way to do it. Once the brain has switched into reverse gear, it becomes instinctive and is usually with us for life, and we can return to riding after a long layoff with no need to re-learn the art of balancing or steering.

In practice, we sub-consciously combine both methods, with some steering and some body motion. The relative proportions by which we combine the two methods depend partly on riding style but also on speed and machine characteristics. For example, a heavy machine with light wheels at low speeds demands a different technique from that appropriate to a light machine with heavy wheels at high speeds and hence a different feel. However, humans adapt quickly and the correct technique soon becomes second nature.

## More detail

Having considered the basis of initiating a turn in fairly general terms, let's now look at this very important aspect in more detail.

Consider a racing bike approaching a corner and the rider needs to heel over as fast as possible. He strongly applies counter-steer and the machine starts to lean over rapidly, but getting a quick roll acceleration also means we need a quick roll deceleration. We start off upright with no roll velocity and we end up at 45-50 degrees lean, again with no roll velocity. In the process the roll velocity must have increased up to a maximum value somewhere around about half of the final roll angle, and then decelerated back down to zero roll velocity at the final lean angle. So the whole lean-in process is not just as simple as a bit of counter-steering followed by straightening out at the end. Basically, we use counter-steer to lean the bike in, at about half way through the roll we have to remove it and possibly give it some "pro-steering" to cause the roll deceleration.

For a long time those that read various motorcycle magazines may have been somewhat confused by the rather conflicting "explanations" often given for this process. There would appear to be two conflicting theories and the adherents of one seem to deny completely any possibility of validity in the other. We might term these two theories

- **Gyroscopic or precessional theory**. Where it is taken as read, that at least the majority of the lean-in torque comes from gyroscopic reactions.
- **Steering out from under theory.** Which basically assumes that as the front tyre steers out from under the CoG., gravity will then continue the lean as the steering straightens up.

I imagine that most open-minded people interested in this subject would be inclined to the view that there are probably some truths and untruths in both points of view with reality lying somewhere in a combination of both. We shall see that this is indeed the actual situation, but we shall also see that the physical mechanisms from either theory alone is capable of explaining the motorcycle lean-in. However, neither theory alone properly explains all the observed phenomenon. Both theories however, require that we use "counter-steering" i.e. the initial rider's input is counter to that necessary for a very slow speed turn.

The whole process of establishing a stable cornering attitude is extremely complex and to understand it properly needs a mathematical explanation outside of the scope of this book, but the following is a detailed description of the process using graphical rather than mathematical results from computer dynamic simulations. In order to fully understand what's happening some of the simulations represent impossible situations, but are never-the-less useful. For example, in the first simulation we consider the case in which the tyres produce no lateral force, thus leaving us with only gyroscopic reactions to lean the machine. Another simulation is done with no gyroscopic effects, this approach allows us to clearly see the individual forces and is useful to test the two theories above. The simulations allow us to investigate the effects of parameter combinations that just aren't possible with a real machine, and this can provide valuable insights to the detail behaviour. All the simulations are for a bike travelling at 100 km/h, and the rider is aiming for a final lean angle of 44 degrees. The bike data is not for any specific bike but, except where noted, are typical of an average large capacity machine. Before tackling the following, the reader is advised to read Appendix 4 describing the mechanisms of gyroscopic effects. The following text is quite detailed and possibly tedious to read, but is included for those that want to better understand the detailed mechanisms of the lean-in process.

## Gyroscopic effects only

In this simulation the tyre parameters are set to produce no lateral tyre forces. This is a situation that is difficult to achieve in practice, the closest we might manage would be to ride on an ice-rink with slick tyres. However, this simulation is very useful to test the "gyroscopic only" proposition above.

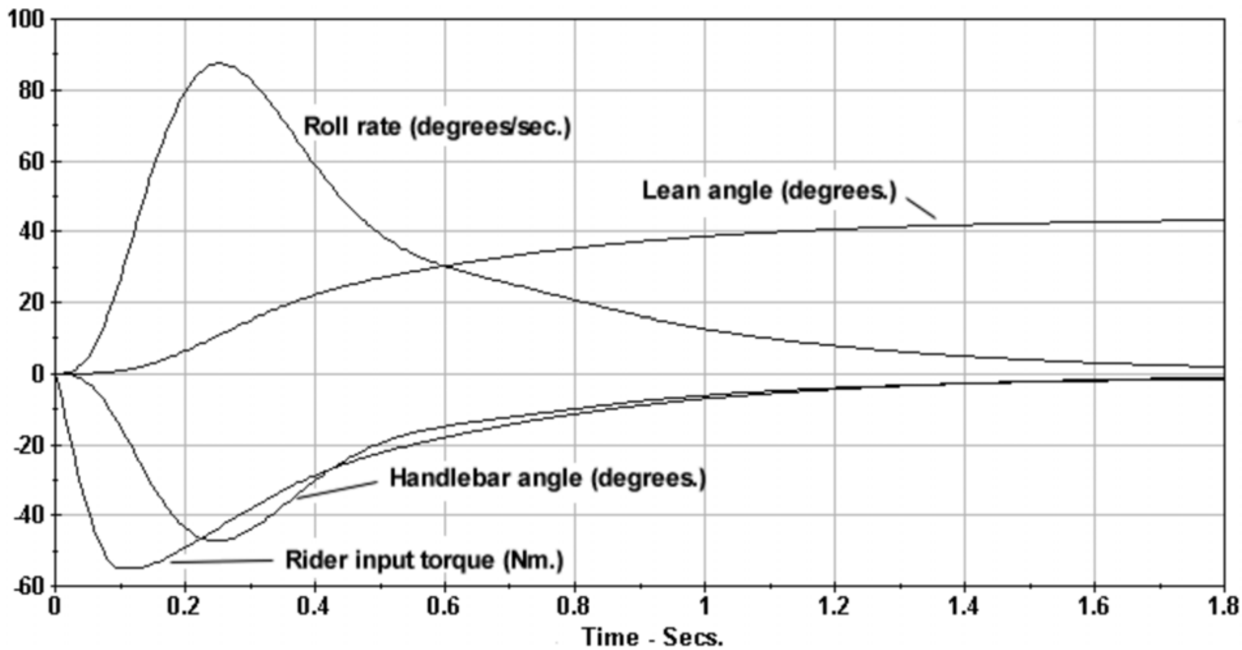
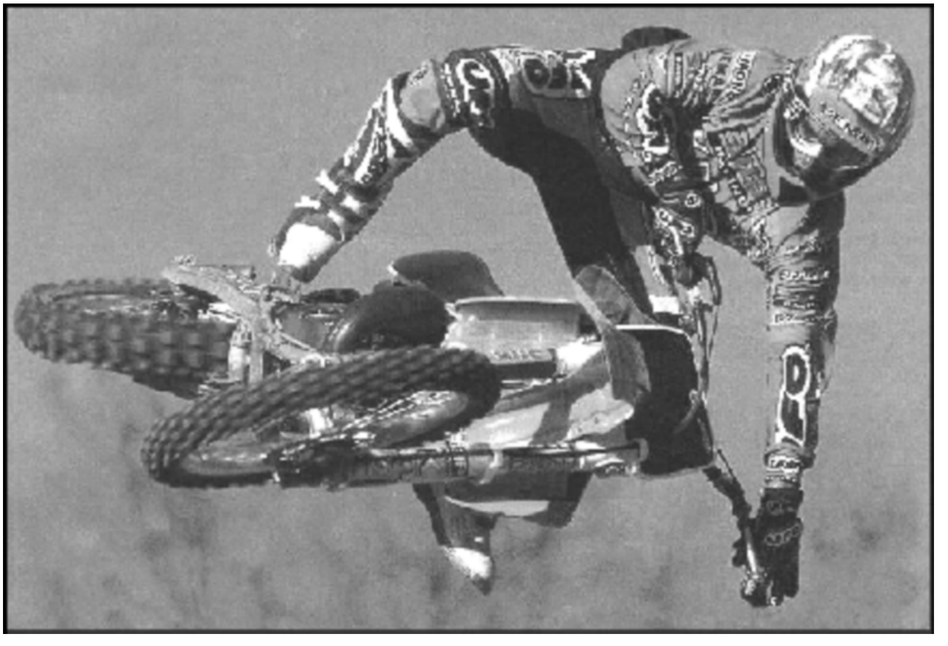



Fig. 4.6 **Lean-in performance with no lateral tyre forces. Note how the leaning velocity or roll rate builds up to a maximum and then decreases to zero as the final lean angle is reached. Note also that the steering or handlebar angle does not exactly follow the shape of the input torque.**

Studying the first set of graphs in fig. 4.6 we see that a negative rider applied steering torque has resulted in a positive lean angle. This agrees with the observed phenomenon that countersteering is used to lean a bike. We can also see that a lean angle close to our desired 44 degrees has been built up in a little over 1 second, which is quite a good lean-in performance. Following a decrease in rider applied torque the lean angle levels off at our target value. So at first sight it looks as if gyroscopic reactions alone, initiated by a negative input torque, can explain the motorcycle lean-in process. But let's look a bit closer at some of the other calculated values. The steering angle initially goes negative to a value of nearly 50 degrees. That is way beyond the steering lock limit built into most bikes, and is also clearly way beyond what we experience in practice. We all know that for normal cornering at 100 km/h. the handlebar movement is barely perceptible.

Not drawn on the graphs but the computer calculations showed only a very minimal sideways movement of the bike and so we're not going to get very far around the turn despite a reasonable lean angle. Obviously there is something very wrong or deficient with the idea that gyroscopic effects alone are all that's necessary to start a bike turning. Or is there…..?



**A perfect demonstration of the power of gyroscopic reactions, in the absence of any tyre forces. Note that the rider has applied a very large steering angle toward the right hand side but the lean of the bike has been to the left. This is known as counter-steering.**

(Oleguer Serra)

Clearly what we've just described is the case of a moto-X rider in mid-air using the precessional effects of the still spinning front wheel to lay his bike over by turning the handlebars. Our simulation fits this situation well. The steering is turned by a relatively large amount and there is little sideways movement of the bike as a whole, but the machine can be leant over quite quickly to any desired angle. Just like our calculations this is done with no forces acting through the tyres.

There are several different torques that act on the steering assembly, which we must sum to get the overall residual torque. Let's now consider another set of graphs showing some of these separate steering torques, fig. 4.7. The main ones being :

- Rider input torque.
- Gravitational torque. As shown in chapter 3, the lean of the bike causes a component of the weight on the front wheel to act through the trail to produce a steering torque.
- Gyroscopic torque. This is due to the roll velocity.
- Damping torque. This comes from various sources, not just from the addition of a steering damper.
- Tyre torque. The lateral tyre force acting with the lever-arm of the real trail produces a torque about the steering axis.

The calculations show that tyre feedback and steering torque from the weight of the machine acting about the steering axis are both zero. This is hardly surprising as we eliminated tyre forces from the system and our air-bound motoX'er only experiences gravity in so far as its effect on bringing him back to earth. Of the remaining graphs the *Residual torque* is the sum of all the other contributing torques, some cancelling others out. This total torque is what's left to accelerate the steered inertia about the steering axis, we'll see later when considering other cases that it is normally lower in value than in this case. This is quite simply explained because in this case there have been very large steering movements and so the torque necessary to accelerate up to these large steering angles needs to be correspondingly larger also. Note too the magnitude of the damping torque, we'll also see that this is much higher than normal, the reason also being linked to the large and hence rapid steering motions.

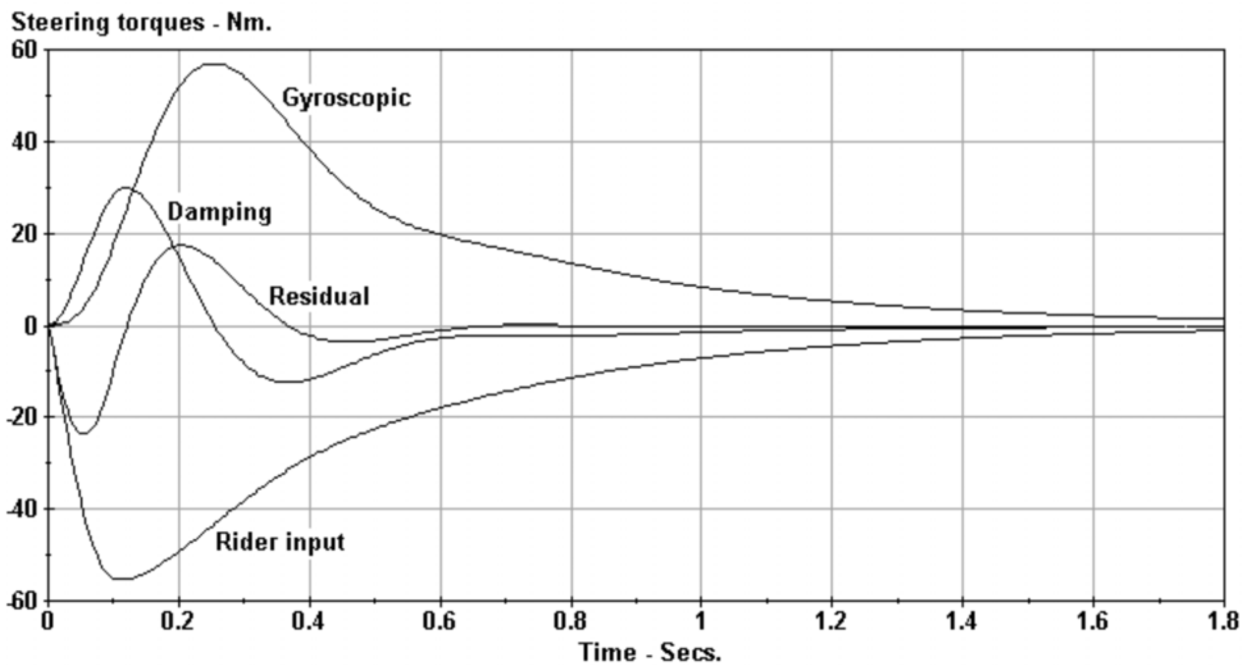



Fig. 4.7 **Torques about the steering axis. For simplicity the damping is all assumed to be viscous and thus proportional to the velocity of steering motions. Note how the residual or total torque follows the input torque in the initial stage before the opposite damping and gyroscopic torques have built up.**

The rider applied torque and precessional torque curves follow a roughly similar but opposite shape, and comparison with the previous set of graphs (fig. 4.6.) shows that the precessional torque curve mimics the shape of the roll-rate curve. We need to refer back to the discussion of gyroscopic reactions to see why this all fits together as it does. Remember that we saw that a torque about the steering axis must be balanced by a roll precessional velocity and also that a roll torque is synonymous with a steering precession velocity.

There are a lot of simultaneous interactions occurring in parallel and so it is not easy to view the situation strictly as a cause and effect serial process. However, as long as we don't try to take it too far, it is often useful as an aid to understanding to try and follow a sequence though.

Initially, the rider applied steering torque builds up to a maximum in the first 100 msecs. During the first 25 msecs. or so, the total or residual steering torque is almost equal to the applied torque because neither of the opposing gyroscopic nor steering damping torques have built up. This residual torque starts to accelerate the steered assembly. The associated rotational velocity of the front wheel about the steering axis will have to be balanced by a gyroscopic torque about the roll axis. This torque will start to rotate the bike in roll as shown by the *Lean angle* and *Roll rate* curves in fig. 4.6. The angular velocity in roll (the roll rate) which is now building up will also have to be met by a corresponding torque about the steering axis. Which is why the *Gyroscopic torque* curve in fig. 4.7 is the same shape as the *Roll rate* curve in fig. 4.6.

Both the, now increasing, precessional and steering damper torques will balance out some of the applied steering torque until at about 120 msecs. the residual torque passes through zero and builds up as a positive torque. At this time (when the total steering torque is zero) the steering rate will be at its maximum and so too the steering damping force. The now positive residual steering torque will act to slow down the negative steering rate and its corresponding roll torque. At around 250 msecs. the steering angle and also the roll rate (and precessional steering torque) will have reached their maximum values, the steering rate will have reduced to zero also.

From about 150 msecs. the rider decides that with the increasing roll rate it is time to ease off and starts to gradually decrease his applied torque. From this point on, the rider continues to reduce his input and the roll rate, steering angle and precessional torques all gradually return to zero. With little or no roll torque the lean angle levels out at its final value. Recall that when discussing the theory of precession we saw that the final angular displacement about the roll axis is determined by the area under the steering torque curve. Therefore, there can be many different rider inputs that end up with the same lean angle, a small torque held for a longer period can produce the same final lean angle as a high input torque for a shorter time. However, the time that it takes to reach a steady lean angle will depend on the rider's initial input.

### Summary

The applied rider steering torque causes a steering velocity which demands a matching gyroscopic roll torque. This roll torque causes a roll angular velocity which in turn must have a corresponding steering torque to balance. This gyroscopically generated feedback torque operates in opposition to the original rider applied torque so forming a self stabilizing system.

The large steering angle and the lack of significant lateral movement indicate that gyroscopic effects alone do not explain the observed behaviour of a motorcycle on the road. However, the simulation does (at least qualitatively) fit the case of an airborne machine.

## Gyroscopic with tyre camber force only.

There is another possibility for investigating lean-in behaviour without applying the "steering out from under" forces. If we set the tyre steer stiffness to zero but use a normal value for the camber force coefficient then we can generate lateral tyre forces as we lean over but actual steering of the handlebars will produce no additional tyre force. Although this is not a situation that we can easily achieve in practice it is probably a better way of looking at the results of removing the tyre steering force. The data settings for this simulation are the same as the previous case except that the tyres can generate camber force, front and rear. Fig. 4.8 shows the same parameters as those in fig. 4.6 except that the lateral movement of the bike into the corner is now shown.

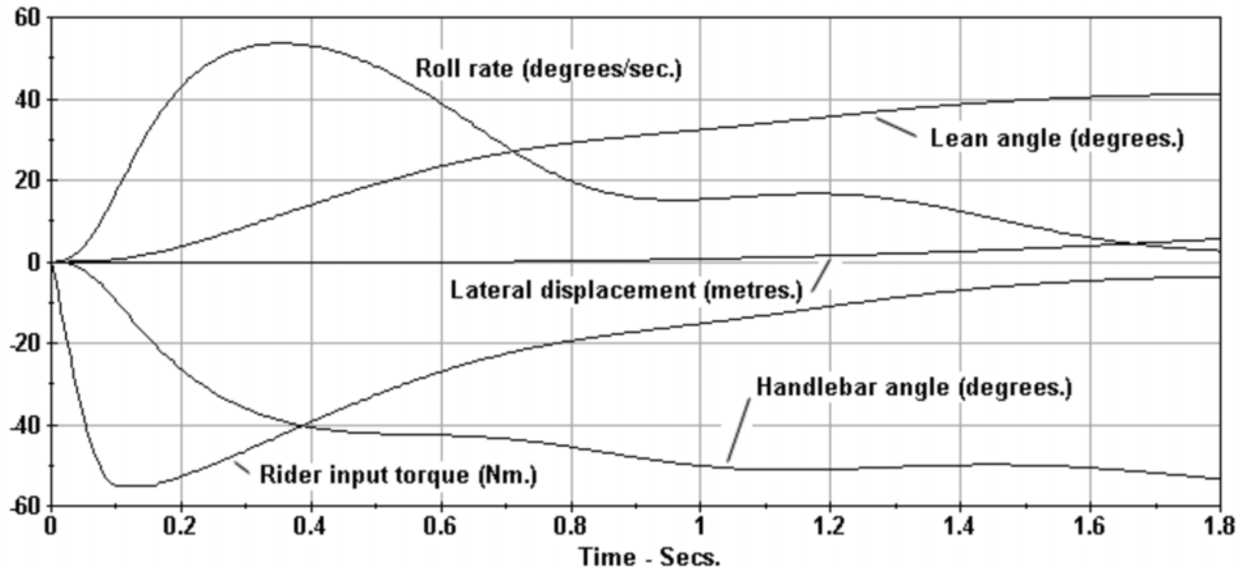



Fig. 4.8 **Similar to fig. 4.6 except that the lateral displacement is added, note how this sideways movement starts to build up. The maximum steering angle applied is similar to the previous case, although the detail shape of the curve is different. Experience tells us that such steering angles are not realistic.**

Comparing those two sets of graphs we can see that there are differences. For our present purpose the important one to note is that the lateral displacement is building up. After 1 second this has reached about 0.8 metres and at 1.8 secs it is 7.5 metres. In other words, now that we can generate lateral tyre forces as the bike leans, it also starts to move into the corner.

We can also see that the other problem with the gyro-only case, that of the large maximum steering angle, has been little changed by the addition of this lateral tyre force. With this large steering angle also goes a large steering angular velocity and so too a gyroscopic roll torque.

## Summary

Whilst the first case of the gyro-only model does in fact describe the real situation of an air-borne motorcycle, this second case has no real world equivalent. In practice we can’t remove the tyre steering forces without also removing the camber forces. However, we saw that the addition of a lateral force capability removed one of the problems with the gyro-only model, the bike will now actually turn the corner on the road. We are still left to explain the large steering angles involved which are at least 10 times more than we would expect from simple observation. These large steering velocities also give rise to roll torques and it is mainly this gyroscopic roll torque which causes the lean. Fig. 4.9 shows the total roll torques for both of these cases, in the first case this torque is due entirely to the gyroscopic reaction. In the second case the roll torque is mainly due to this same effect but there is an additional effect (not shown for clarity) from the difference in roll torques between the lateral tyre force and the balancing gravitational moment.

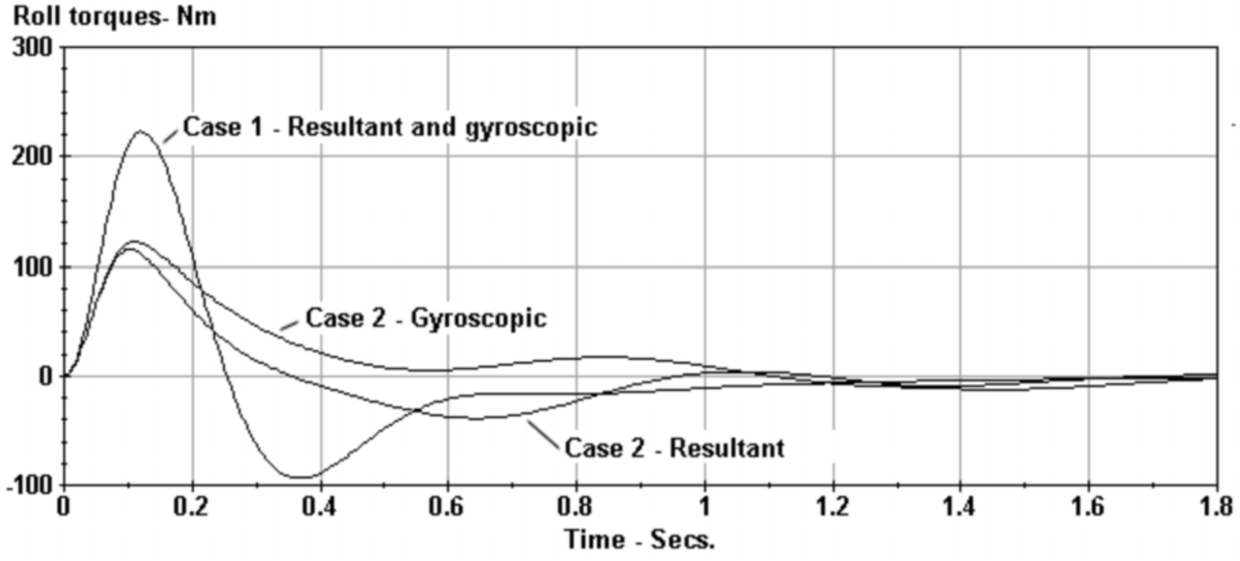



Fig. 4.9 **Resultant roll torques for both of the two cases mentioned. In case 1 all the roll torque is due to gyroscopic effects, but in case 2, although the gyroscopic torque has the major influence the tyre and gravitational torques begin to have an effect also. Compare this with fig. 4.11.**

## Gyroscopic with tyre camber and steer forces.

For this simulation reasonable values were input for the properties of the tyres, that is, both steering and camber produce lateral forces. This should provide us with a realistic simulation that shows all the main aspects of the lean-in process. To be of value it must overcome the deficiencies of the two previous cases and it must also produce results in accord with our experience and observations.

To reiterate, the results must show:

- That initial counter-steering causes the bike to lean-in and that rider control can determine the final lean angle and path taken by the bike.
- The steering angle during the countersteering phase and the subsequent steady state phase is quite small, only a few degrees at most.
- “Hands off” riding is possible, the machine is auto-stable..

The input parameters are as before except that both tyres have normal properties.

Whilst most of the curves in fig. 4.10 are similar to those in the previous cases (figs. 4.6 and 4.8), the important difference is in the graph of the front slip angle. Note that the scale factor on this line has changed to x10. That means when the graph value is 10 then the real angle is actually 1 degree. We can see that in this example the value of the initial counter-steer slip angle is about 1.8 degrees which compares with a value near 50 degrees in the other cases. Clearly, despite the similarity between the other curves the introduction of lateral steer forces has had a major impact on the value of this parameter, and has now brought it into line with values that we’d expect from experience.

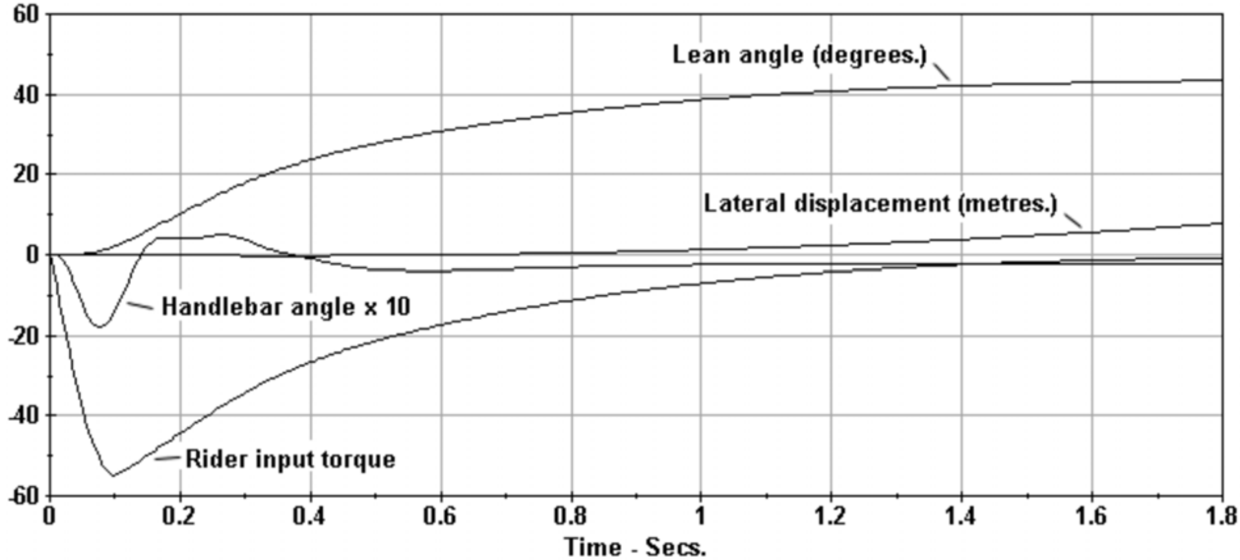
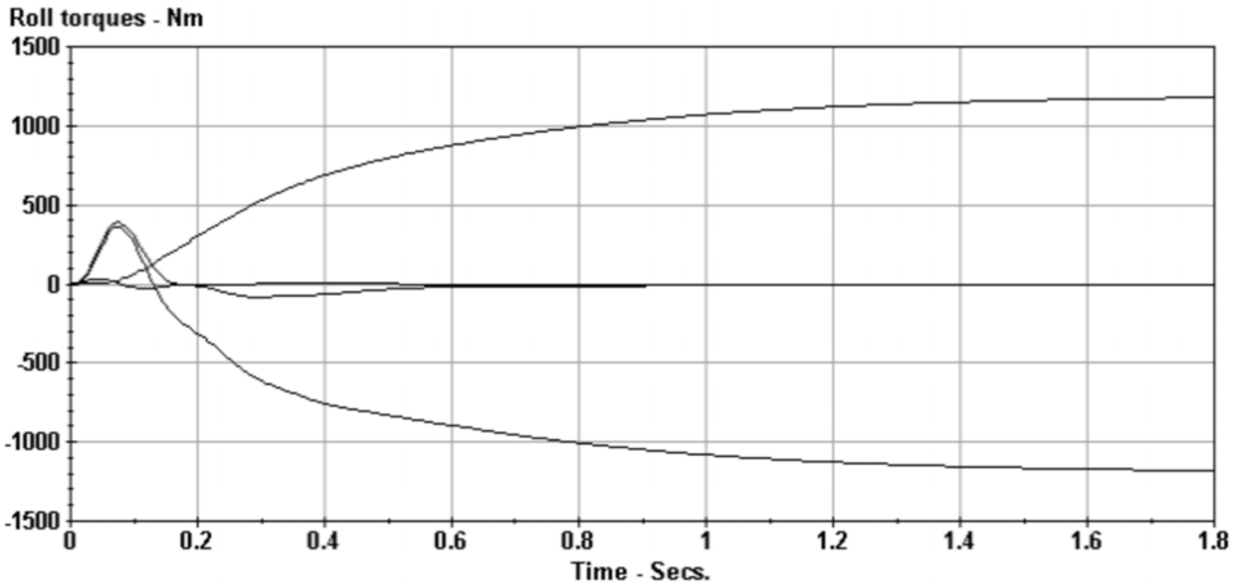



Fig. 4.10 **These results have been calculated using realistic tyre properties and allowing for gyroscopic forces. This simulation now appears to represent the observed real world behaviour reasonably well. Note the scaling on the handlebar angle.**

As this current simulation seems to describe the lean-in process reasonable well, lets look in more detail at some other calculated parameters and see if we can work out exactly what's going on. Let's start by looking at the *Roll* torques because after all it is the residual roll torque that causes the lean motion, fig. 4.11.

We can see that the precessional roll torque in this case has a relatively minor effect on the overall roll torque, in complete contrast to the previous two cases. Actually the peak value of the gyroscopic torque in this initial period is 32 Nm. whilst the peak tyre induced torque is 360 Nm. during the initial counter-steering period. Following the curves along from the start we see that up to about 100 milli-secs. the overall roll torque is determined mainly by the tyre force, after that time the increasing lean angle means that the gravity torque also builds up and adds to the total. After approximately 500 msecs. the tyre and gravity torques almost balance and the residual torque remains close to zero.



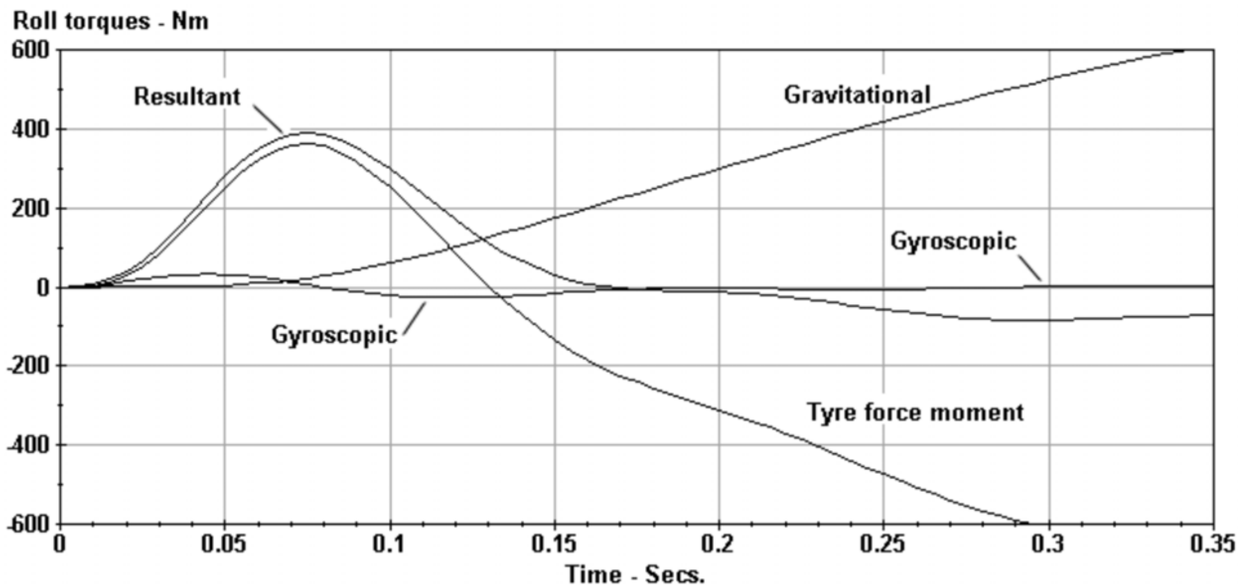




Fig. 4.11 **These two sets of roll torque curves are the same data, the time scale has just been expanded in the second to show more detail during the important initial period. Note the almost insignificant contribution of the gyroscopic moment to the overall roll torque, in complete contrast to the two previous cases, in fig. 4.9.**

It is interesting to observe the detail shape of the resultant torque curve. During the first 150 msecs. or so there is a positive hump, this is mainly caused by the initial counter-steering effects of the front tyre.

This torque causes a roll acceleration and the bike starts to lean. Prompted by the rider's reduced input torque the overall roll torque starts a longer negative phase. Thus the accompanying negative roll acceleration will continue until the roll rate is reduced to zero and the bike adopts its final steady state lean angle. The fine control of all this being due to the rider's input. What is important is that there is a positive torque to begin the lean followed by a negative torque to slow the lean to its final position.

As the ***roll*** torque is little influenced by gyroscopic effects it starts to look as if precession plays little or no part in the lean-in process when we simulate it with a reasonable tyre model. However, the precessional ***steering*** torque has the single largest value of all contributions to the total steering torque, as shown in fig. 4.12.

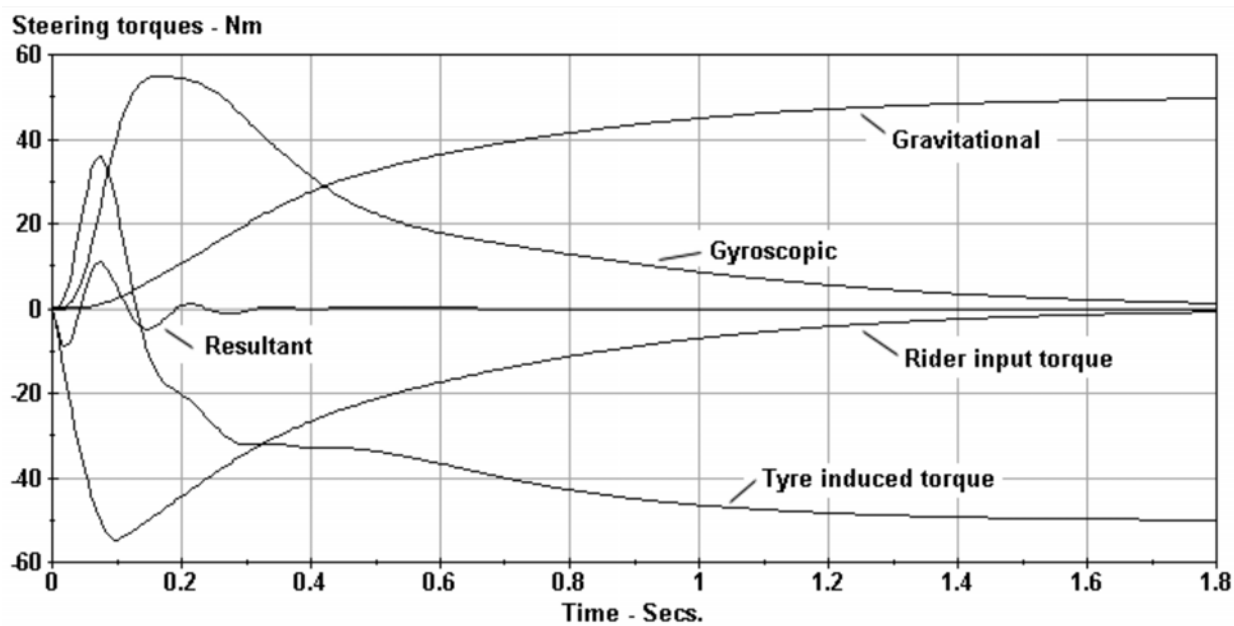



Fig. 4.12 **The main four steering torques are shown here. We can see that for the first 20 or 30 msecs the resultant torque is due mainly to the rider input, but as the countersteering tyre force builds up the resultant begins to reduce. As the roll velocity begins to increase a matching gyroscopic torque is produced. After the initial torque swings have occurred a pattern emerges as the machine steadies. The gravitational and tyre induced torques tend to follow similar but opposite shapes, thus cancelling each other, but also the gyroscopic torque and rider input tend to mirror one another. In other words the rider input mainly goes to balance the gyroscopic steering torque. This explains why steering requires less effort with lighter wheels.**

These apparently conflicting observations are easily explained by considering the angular velocities about the steering and roll axis. Firstly, we know that there is very little steering movement associated with normal turning, and hence there is little steering angular velocity. This in turn means that there is minimal gyroscopic torque developed about the roll axis. On the other hand there is considerable roll angular velocity developed during the lean-in and with this must go a corresponding high steering torque.

Initially (during the first 20 – 30 msecs.) the rider applied negative (counter-steering) steering torque mainly goes toward accelerating the steered assembly, also in a negative direction. This steering movement builds up a negative slip angle which produces a negative lateral force on the front tyre. This tyre force acts through the tyre contact patch at a distance behind the steering axis equal to the value of trail. Thus producing a torque about the steering axis. This lateral tyre force also acts on the lever arm of the vertical CoG height to produce a positive roll torque, which in turn leads to a positive and increasing roll velocity. This roll velocity then must have a matching gyroscopic steer torque. Now this gyroscopic torque and the previous tyre feedback torque both act in opposition to the rider applied torque. As a result the residual steer torque is reduced to zero within 50 msecs. and peaks with a positive value at 75 msecs. These changes to the residual torque causes a reduction in the steer and front slip angles which returns to zero at about 150 msecs. The negative tyre steer force and the *tyre feedback* steering torque follow suit. The now increasing lean angle starts to produce significant camber force to add to the now positive total tyre force.

Except during the initial period, the applied rider torque mainly balances the gyroscopic torque. In fact it is necessary for the rider to continue to apply such relatively high torques to counter the effects of roll precession. This is why less rider effort is necessary with lighter wheels, in racing this is important to achieve fast lean-in to corners. After a while the rider decides that the roll angle and roll velocity have built up to such levels that it is time to ease off on his input torque. This sets off a chain of events, similar to but opposite in nature to those set off by the original countersteering, which slow the bike's roll down to its final steady state position.

Simulations with zero rider input, equivalent to riding no-hands, showed that the bike was capable of balancing itself. Sharp pulse input steering torques where also simulated with no-hands, representing disturbances such as hitting an off-centre bump, the results showed that after an initial wobble the bike would continue to remain balanced.

### Summary:

With this simulation, using reasonable values for all the most important physical parameters we have been able to look at the detail of the motorcycle lean-in process. The calculated results seem qualitatively in accord with our practical experience and observations. The main shortcoming of this case is the assumption of zero tyre width, this will affect the steady state steering torque required to maintain a balanced turn. However, for current purposes this is not a problem, we know from experience that balance and turning can be done with both narrow (pedal bicycles) and wide tyres. So obviously tyre width is not of prime importance to this process.

We have seen from the previous two cases (with no steering tyre forces), that gyroscopic reactions can produce high roll torque. However, in this example the gyroscopic roll torque is of minor importance, because there is very little steering movement. This is not the same with the steering torque, here the gyroscopic effects have a major influence, and hence the moment of inertia of the front wheel has great control over the steering performance.

## Tyre forces only – no gyroscopic effects.

In the previous case we looked at a reasonably realistic situation which implicitly included the gyroscopic effects, whether we like them or not. Experience tells us that when we reduce the moment of inertia of the wheels, which reduces gyroscopic effects, then we can increase the rapidity of the turn-in. So would we be better off without the interference of these reactions? In this final test case of this series we'll consider the behaviour with zero gyroscopic effect.

We saw above that precessional effects had little influence on the overall roll torque but were of major importance to the total steering torque, acting in opposition to the rider's applied torque. We might easily conclude that lean-in performance would improve without the gyroscopic effects, so letting the full effect of the rider's torque be applied to the job in hand. This is difficult to test in practice and would require counter rotating wheels or flywheels to be mounted concentrically with the road wheels. These would have to be configured to have the same angular momentum as the main wheels. This has been done with the front wheel of a pedal bicycle (Jones) and was shown to be ridable but with much greater difficulty than normal. Anyway this is something that we can do quite easily with computer simulation, we just simply set the moment of inertia of the wheels to zero. With no moment of inertia the wheels can have no angular momentum and hence no gyroscopic effects.

With the remaining parameters as before, the virtual bike and rider would crash within a very short distance. As we saw in the previous case the rider applied torque largely balances the gyroscopic torque and so it seemed reasonable that if the rider could adapt his input control responses to something appropriate to this machine setup then it might become ridable. A real human rider has a huge range of learning possibilities and responses that enable him to control a wide range of devices after a relatively short period of practice. Indeed by adjusting the control responses of the model of the virtual rider it was possible to make the machine marginally ridable. However, it is easy to adjust the responses of the virtual rider but that is no guarantee that a real rider could easily duplicate those responses and it was apparent that the bike was very critical to control and would require lightening fast reflexes. It was not possible for the bike to remain balanced without rider input, in other words it couldn't be ridden no hands. This simulation adequately demonstrated the importance of gyroscopic reactions to the overall balance and steering of a motorcycle.

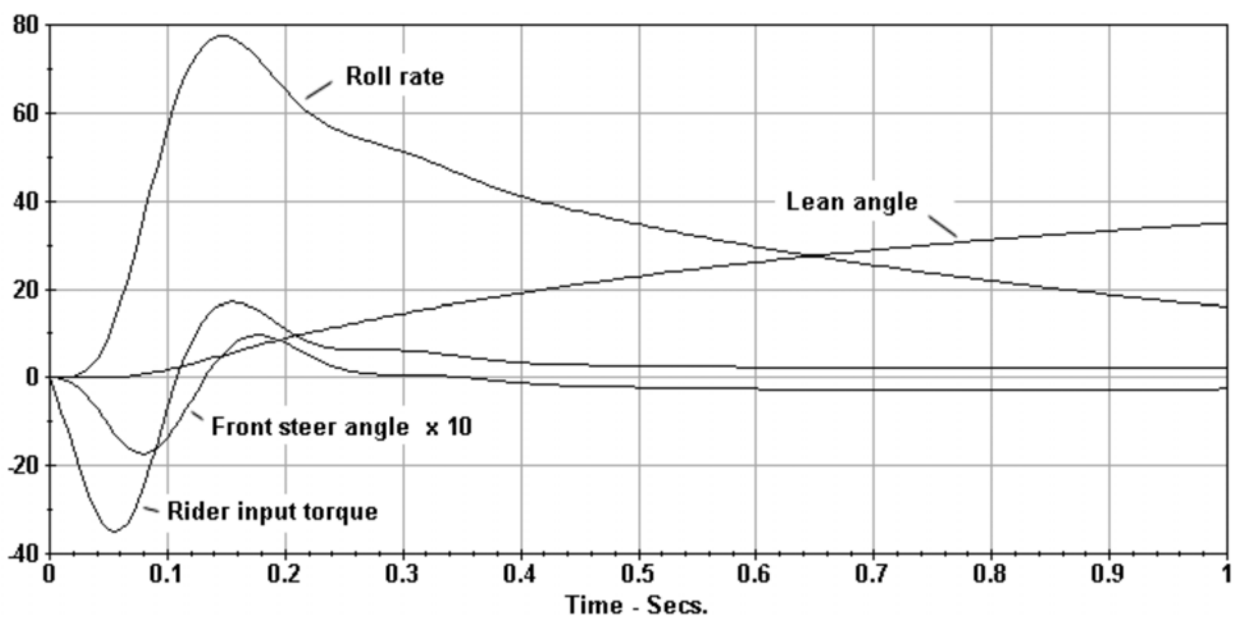



Fig. 4.13 **Behaviour with no gyroscopic reactions. Compare with fig. 4.10. The main difference is in the shape of the curve of the rider input torque. In the previous case this input has a higher magnitude and remains negative for much longer. Without gyroscopic torque to oppose the rider's input, the rider himself must modify his response so that the resultant steering torque remains within acceptable limits.**

Fig. 4.13 differs in important aspects from the applied torque curves in fig. 4.10 with the gyroscopic reactions accounted for. Firstly, the counter-steering torque is applied for a much shorter period, which is then followed by a longer period of "pro-steering" torque which then gradually returns to zero. Despite the average steer torque being much lower during this shorter counter-steering torque period, about 110 msecs. the build-up of lean angle seems to be very similar. The roll rate must be slowed down as the machine approaches the required lean angle, in the previous cases the roll rate produced a gyroscopic negative feedback steering torque which would make the appropriate adjustments to the steering. In the current simulation the rider must make these adjustments himself, hence the need for the "pro-steering" or "counter-counter-steering" torque. The rider would therefore be expected to apply rather different control responses to those that he's currently used to.

Extending these considerations, it becomes obvious that the rider must be very attentive because any small disturbances have to be corrected by his own actions. This would be more difficult in practice as human reaction times are of the order of 0.25 secs. and we have seen that a lot can happen in that time. So the rider would have great difficulty responding fast enough to unexpected disturbances.

### Summary

Whilst we could construct a motorcycle that had counter rotating wheels to cancel the precessional effects, this case really describes an unrealistic situation. Nevertheless, there are useful lessons to be learnt from it. We saw that even though a bike without the interaction of gyroscopic reactions appears to have no auto-stability, it could be made stable by appropriate input response. A simple model of our simulated rider could be adjusted to give such input. Whether such responses are possible to be learned and applied by real riders is a matter outside the scope of this book. It seems fairly safe to assume that a vehicle configured in this way would be very tricky to control at a minimum.

Analogous situations exist in other fields, for example some military aircraft are made somewhat unstable to achieve performance goals, then safe control is beyond the ability of the pilot, and computers are used for rapid control in what's become known as *"fly-by-wire"*.

## Body lean only – no steering.

Whilst we normally control a motorcycle by steering, or rather counter-steering, this is not the only way possible. We can also shift our body weight around to achieve a limited degree of directional control. At first sight this might appear to be easily explained, but surprisingly when we get down to the fine details, the interactions between tyre forces, precessional effects, steering torques etc. are no simpler than those that we've previously considered.

Note that the physical bike parameters are as before in the previous simulations but there is no input steering torque from the rider, this being replaced with a body movement that causes the combined bike and rider CoG to move sideways by 50 mm. (relative to the bike centre-plane) within a time of 0.15 secs. This is probably unreasonably fast and is chosen only to emphasize the reactions, as an aid for us to follow the motion. Firstly let's look at the graph of roll torques.

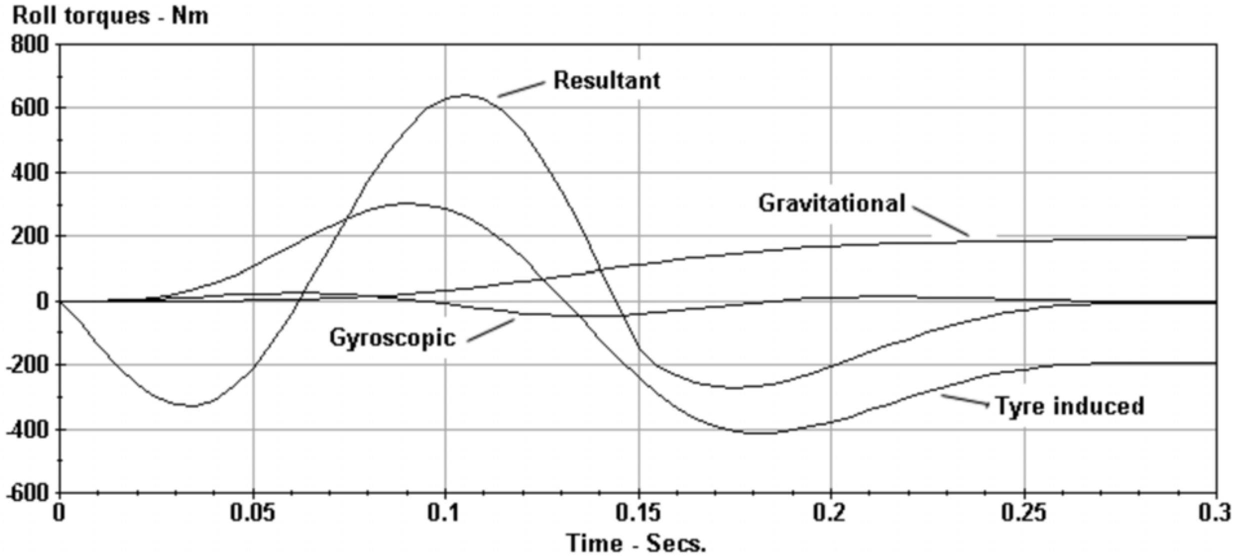



Fig. 4.14 **Roll torques due to lateral rider weight shift without any input steering torque, (hands-off case). Note that the maximum roll torque from gravity is much lower that in previous which is simply due to the smaller lean angle developed without rider steering control.**

Note that the time scale in figs. 4.14, 4.15 only shows the first 300 msecs. which is when the main action takes place.

The only way that the rider can move himself to a position offset from the central plane of the bike is by applying a negative force to the bike itself, this force in turn represents a negative roll torque on the bike. This torque is not plotted on the above graph page directly, but it is plotted as part of the total roll torque. The total roll torque will cause the machine to begin a negative roll motion, and we can see the roll rate start to build up on the following graph in fig. 4.15.

Switching to look at the graphs of steering torques, lower set in fig. 4.15, we can see the roll rate reflected in the precessional steering torque, as this is the first of the steering torques to build up, the total steering torque initially follows it causing a negative steering slip angle to build up as we can see in fig. 4.15. This slip angle leads to a negative lateral tyre force build up, which causes both positive roll and steering torques to occur. This tyre steering torque quickly offsets the original negative gyroscopic steering torque causing the net steering torque to turn positive which in turn starts to slow and reverse the negative steering angle. Around this time the lean angle will have built up enough to create increasingly significant gravitational steer and roll torques.

This complex interaction of many parameters continues in like fashion for a short time and settles down to a steady turn. This whole process is much slower than in the cases with steering input, this is of course completely in accord with practical experience.

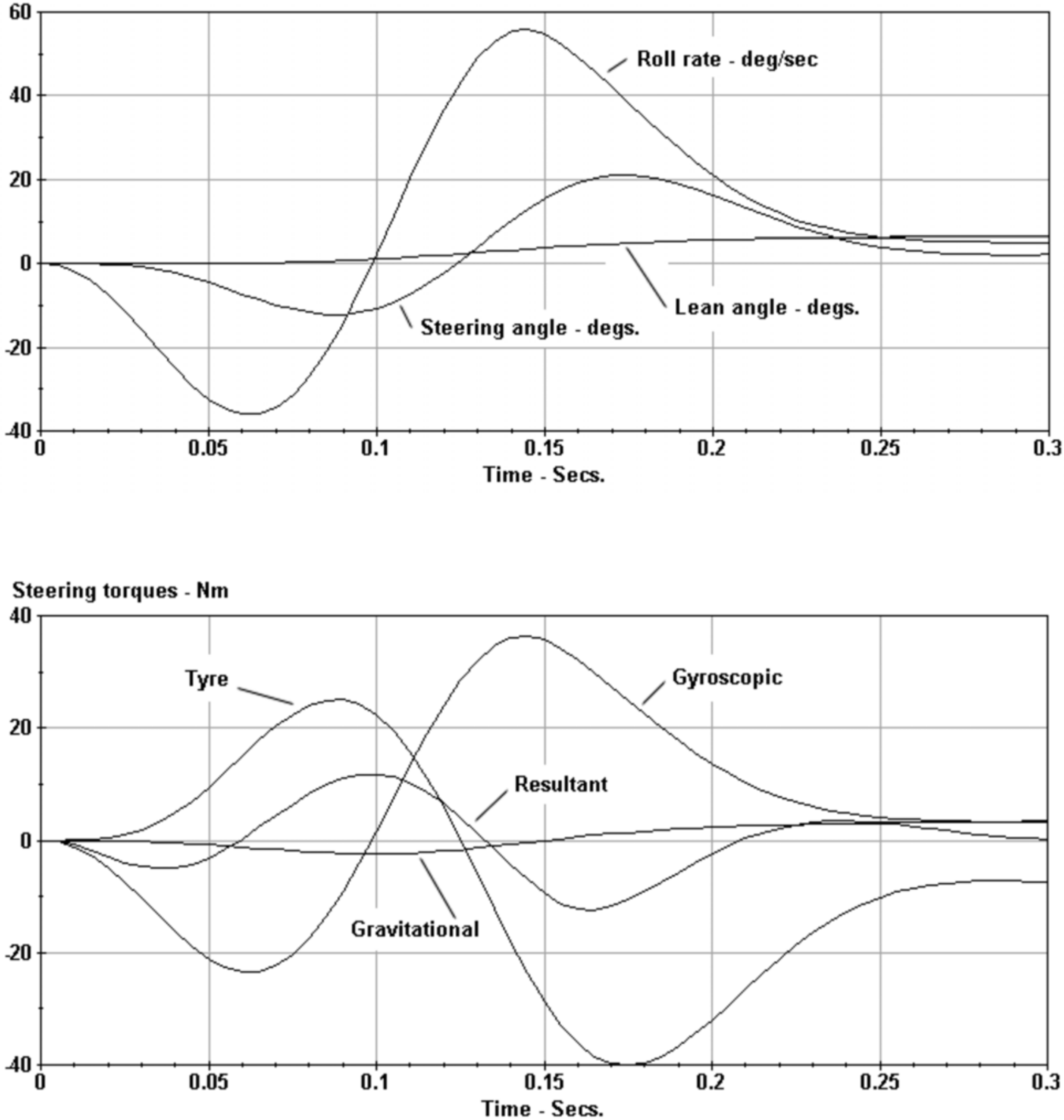


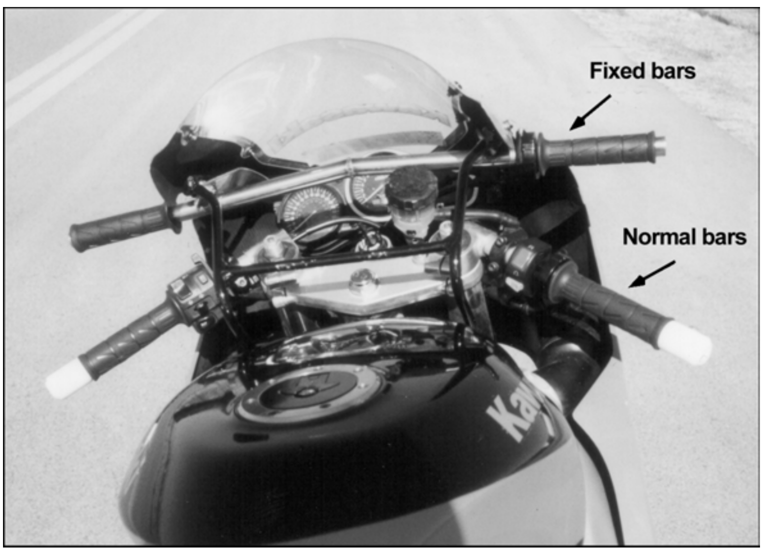

Fig. 4.15 **Cornering 'hands-off'. Note the initial negative roll velocity of the bike as the rider pushes against the machine to force his body inward. The final lean angle is much smaller than in previous cases, we know from experience that we have very limited control when riding 'no-hands'.**

### Summary

This simulation just confirms what we know from everyday riding, whilst no-hands control is possible it is far too sluggish to be of any practical use in the absence of normal handlebar steering. It is interesting to note that in this case too it is counter-steering/counter-lean actions that start the process. True the rider must lean into the bend but that initially causes a “counter-lean” of the bike itself, gyroscopic and tyre forces get to work and lead to a “counter-steer” motion of the front wheel.

At first sight it looks like a racer’s tendency to lean in at the start of a turn could add to and assist the standard handlebar control, as well as just keeping the bike more upright.

*N.B. The timing and other numerical values mentioned in the above examples are valid purely for the test cases described. Whilst the general trends will be similar the actual values may well vary considerably with other basic input parameters.*




**An interesting experiment by Keith Code, well known owner of a Californian riding school.**

**A second handle-bar was added which did not steer, it was fixed to the frame of the bike. The idea was to compare cornering with body movements only to that of normal steering. Many people, students, racers and press testers have tried this machine and it is hardly a surprise that none were able to corner by body movements only. Rider comments do seem to indicate that it was harder to ride than doing it “no-hands”. This is probably a psychological effect, the lack of a normal response from the handle-bar conflicting with the rider’s sub-conscious expectations. With no hand contact our response expectations would be quite different.**

## Conclusions

We’ve used a simplified mathematical model to simulate the important aspects of the lean-in process. We looked at cases that separated out the action of steering tyre forces from the gyroscopic reactions in order to assess the effect of each. Whilst lean-in can be explained with either of these processes alone, using similar maximum values of rider applied steering torque, not all of the accompanying observed behaviour could be satisfied except by the model that used both effects.

During the first 10 msecs. or so, the gyroscopic roll reactions to the steering velocity built up quicker than the tyre counter-steering force. Had we been allowing for tyre relaxation this difference would have been greater. However, despite this head start on the part of the gyroscopic reactions they are of minor importance to the overall roll torque, which for around the first 100 msecs. is due mainly to counter-steering tyre forces. As the counter steering phase comes to an end, a lean angle will have built up sufficiently to make the gravitational roll torque significant. This gravitational torque is mainly balanced by the opposing tyre induced torque (as the lean angle builds up), except during the counter-steering phase when the gravitational torque is small anyway. Even when the tyre torque and gravitational torque balance, the roll motion continues due to the roll momentum which has been built up. Approaching the final lean angle the rider changes his control input and this begins a process opposite to that of the counter-steering period in order to slow the roll to its final steady state position.

It is interesting to compare the shapes of the steer/slip angle curve with the applied torque curve. Until the steady state is reached the applied torque always remains negative, however, the steering movement shows a more complex shape. Initially it is negative and this, *(through the tyre steering stiffness)* gives the initial negative tyre lateral force that starts the roll process. During the final part of the manoeuvre, prompted by the reduction in applied torque the slip angle shows a positive bump in its curve, which in turn through the tyre steering stiffness will slow the roll. So if we talk about applied torque then counter-steering lasts for the whole duration of the transient phase, whereas, talking about steering movements the counter-steering only lasts for a brief initial period.


**This picture of Chris Walker half way through a rapid transition from full left lean to full right lean shows the counter-steering still being applied to force the bike to change direction as rapidly as possible. Just as we need counter-steering to start entry into a corner, we also have to use it to begin the rise up out of a corner. However, counter-steering in that case means initially steering deeper into the corner.**

**Note that the front wheel has risen slightly off the ground, some of the reason may be the application of power but also the bike is subject to centrifugal force due to its high roll rate at this point in the direction change, which tends to lift the bike. This is most noticeable halfway through an 'S' bend.**

(Motociclismo)

The roll torque is little affected by precession as we have seen but this is far from the case with the steering torque. The roll velocity must be accompanied by a corresponding steering torque. This gyroscopically induced steering torque can easily be a major component of the total steering torque during lean-in. We have seen that these precessional effects are of great importance to the maintenance of auto-stability. Looking at the graphs of steering torques from case 3, we can see that the overall torque available for angular acceleration of the steering is much smaller than any of the individual components, it is the difference between other much larger valued parameters, as such, a small percentage change in any of the contributing torques can have a disproportionably large effect on the overall torque, it is through the negative feedback nature of the gyroscopic torque that this balance is kept under control. Remember how difficult it was to get stable control in the case with no gyroscopic reactions.

Modern racing techniques demand rapid lean-in to corners, so therefore we need to optimise the use of the rider's input as much as possible. From fig. 4.12 we can see that during the initial counter-steering phase the rider's torque is opposed firstly by the tyre force induced steering torque, followed shortly afterward by gyroscopic steering torque. Thus to make better use of the rider's effort we need to minimize the tyre feedback torque and the gyroscopic torque. The tyre feedback is largely dependent on trail and this is one of the prime reasons for trail reduction to speed turn-in, although there are competing influences on the selection of trail. Gyroscopic reaction is dependent on the polar moment of inertia of the wheel and so a reduction in general in wheel weight will also result in improved lean-in.

For any given bike then. we see that to a great extent it is the rider's strength that determines roll performance. However, there are other fundamental considerations that relate to ultimate lean-in performance, that will not be so easy to solve as sending the riders to a gym.

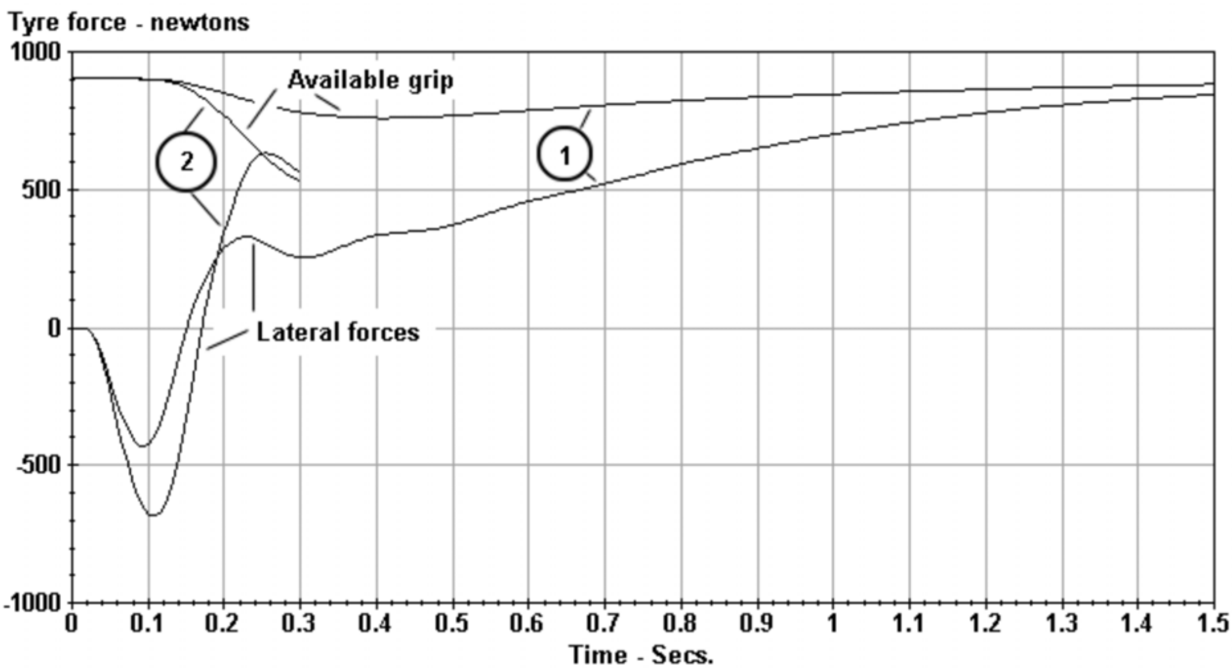



Fig. 4.16 **Two examples of the front tyre forces on entry to a corner. The curves (1) are for a relatively normal lean-in and (2) are for a rapid turn in with the trail reduced from 90 mm. to 60 mm. and the wheel moment of inertia reduced from 0.41 to 0.25 Kg.m$^2$ . In both examples we can see how the available grip is reduced due to centrifugal force, but in case (1) this is relatively minor and the lateral tyre force required is well below the available grip until the turn is nearly fully established. On the other hand, the rapid roll rate in case (2) both reduces the available grip and also requires more grip during this period. The result being that at about 0.25 secs. the required grip exceeds that available and the front slides out. This is the final limitation on the maximum turn-in performance possible.**

To start to turn quickly (as opposed to lean quickly) we must build up the inward lateral force on the tyres as soon as possible, without that lateral force we just go straight on regardless of the speed of leaning. The maximum lateral force is determined by the tyre properties and the vertical tyre load. However, the vertical load can be reduced very significantly by the unloading effect of centrifugal force due to a high roll rate. So the final limit to initial cornering performance is the balance between rapid lean-in and build up of tyre force, as shown in fig. 4.16.

Depending on various machine parameters, it is possible that you can get around the corner slightly quicker by leaning in slightly slower. This is basically a problem with the front tyre, the build up of tyre force is much quicker at the front than the rear. Therefore we have some unused reserves of grip at the rear during the transient phase, if we could put this to good use then we would have the potential for greatly improved turn-in speed. Perhaps one way to exploit this would be to introduce some form of “Two Wheeled Steering”, but that is another story. See further comments in chapter 18.

# 5 Aerodynamics

The size and shape of a motorcycle, together with the rider and any aerodynamic aids such as a fairing, windscreen or legshields, affects its drag and lift, and hence its power requirement, the more so as speed increases. Many complete books have been written on the subject of vehicle aerodynamics and so we can only scratch the surface of the subject in one chapter. Most texts and articles on aerodynamics usually concern themselves with extolling the virtues of low drag and the accompanying effects on performance and fuel economy. However, the creation of slippery motorcycles is probably the easy bit.

In common with all medium to high speed vehicles the effects of wind pressure on stability and controllability must be considered, but not unexpectedly this becomes very complex on a bike compared to self balancing vehicles such as cars. Trail, gyroscopic reaction, yaw and roll coupling and their interaction with the steering, are the cause of many stability and control problems not present on cars, lorries, etc..

Fairing design for roadsters and road racers usually leaves much to be desired, for most designers concentrate their attention on the front end and neglect the rear. In racing, this is a result of the tight, post 1957 framework of FIM rules to which fairings must conform. In touring, it is due to designers either copying racing designs or thinking only of styling or weather protection for the rider.

Here we'll briefly consider some elementary factors regarding drag and then look at some of the aerodynamic considerations unique to single track vehicles.

## Drag

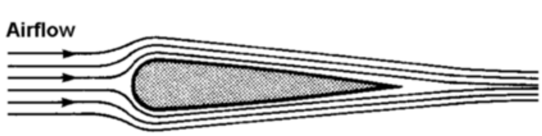
Firstly, let's consider just what causes air drag -- which is after all the largest thief of our engine power output at all but the lowest of speeds. Drag is a force trying to prevent rapid movement of a bike though the air, this force is generated through a difference in pressure between the front and rear of the machine. This pressure difference acts on the frontal area of the bike to give the drag force, and hence the larger the frontal area the greater is the drag. It is not always appreciated but it is the viscosity of air that is responsible for drag. Without viscosity even the most un-streamlined shape would create zero drag at high speed.

According to the well established Bernoulli theorem, the total pressure in a streamline of gas is made up of two parts, one is the static pressure and the other is the dynamic pressure due to its longitudinal velocity. The total pressure (the sum of those two) remains constant, so if the gas is speeded up, its dynamic pressure rises and the static pressure goes down. As the air meets a moving object such as a motorcycle, it will be speeding up and slowing down as it flows over the various shapes, and so the pressure felt on the surfaces of the bike and rider will vary from place to place. If we sum all the individual components of this air pressure over the whole machine then the resulting force will comprise the drag and lift forces. If there is a side component to the air stream there will also be a side force.

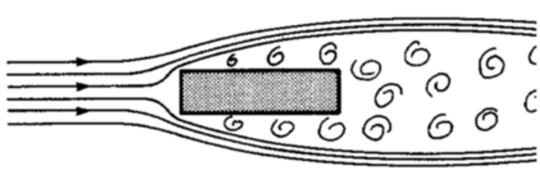
However, using the classic laws of physics with what's called an "ideal gas" would result in a calculation that gave us no drag at all. In other words the drag force from gas pressure acting on front facing surfaces would be exactly balanced by a propulsion force due to the gas pressure acting on rearward facing surfaces. This apparent lack of drag is sometimes known as the d'Alembert paradox. Experience tells us that we have drag and so we need to look for other causes.

Near to the surface of the bike and rider the air does not behave like an ideal gas, viscosity causes internal friction. Right at the surface the air speed will be that of the moving machine and the internal gas friction will tend to drag adjacent layers of air along with it. As we get farther from the surface the air will be largely unaffected, so there is a relatively thin layer of air near the surface which has a strong velocity gradient. That is, at the surface the velocity of the air particles is equal to that of the vehicle and outside of this layer their velocity will be that of the surrounding air. This layer is known as the boundary layer and the behaviour of this layer is so important to the aerodynamic properties of a vehicle.

When the local airflow over part of a body is slow (either because the vehicle is travelling slowly or because of its shape), the velocity gradient is small and adjacent layers within the boundary layer slip over each other in an orderly fashion and the flow is known as laminar. However, at higher local velocities the velocity gradient is likewise high and the friction between layers causes them to trip over one another (like ocean waves breaking on the shoreline), and create eddies. This type of flow is known as turbulent. Except at very low speeds we will have a mixture of turbulent and laminar flow over a moving vehicle. Generally speaking, the frontal aspect of a vehicle is such that the air has little choice but to follow the shape, but as the air passes the widest cross section the boundary layer becomes thicker and when the shape of the body departs too much from the ideal, the air can no longer follow the shape and the boundary layer becomes detached from the body. The object moving through the air then leaves a turbulent wake behind. The size and shape of the object and the speed are the most significant factors affecting the size of this wake. The pressure within the wake tends to be constant and less than normal atmospheric pressure, and certainly less than the average pressure acting on the front of the object and hence there will be a nett drag force. Studies have shown that the drag force is approximately proportional to the size of the wake.

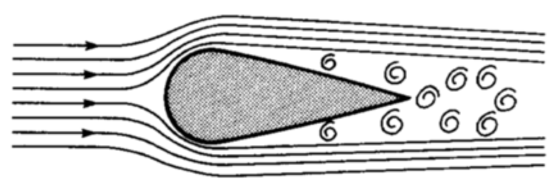



**Laminar flow around a good shape**

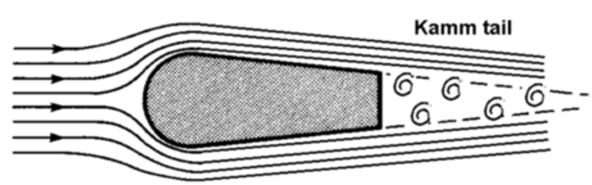


**Turbulent flow around a bad shape. Drag is proportional to the size of the wake.**

Fig 5.1 **With a shallow rear angle, only possible with a long thin object, it is easier for the air to follow the object's shape and keep the size of any wake down to a small value. A poor aerodynamic shape causes early separation and a large wake, and thus high drag.**



**Too steep an angle at the rear causes early separation and increased drag.**




**Shallower angle with the same length and sharp cut-off leaves a smaller wake and less drag.**

Fig 5.2 **A teardrop shape with too steep a rear angle also allows early separation with a large wake and drag.**

**However, the gentler angle and sharp cut-off of the Kamm tail keeps the same length but the separation point is later and surrounds a smaller cross section, and so produces a smaller wake and less drag.**

Thus viscosity causes a change in the pressure distribution around the vehicle and when we sum the components of pressure we will have a nett effect which gives drag and lift forces. This then, is briefly how viscosity is the cause of the major part of aerodynamic drag. There is another more direct effect of viscosity and that is sometimes called surface or viscous drag. To push a body through a viscous fluid we have to provide a shear force to cause the adjacent layers to slide or shear over one another. This force also acts to slow the body and so is an addition to our drag force previously described. On normal road vehicles this viscous drag is only a small proportion of the total drag, but is very important to highly streamlined solar and human powered machines.

The point where the boundary layer detaches from the body is known as the separation point and we generally want to design any bodywork so that this point is as close to the rear and at the smallest cross section possible to reduce the size of the wake. Air, like anything else, prefers not to change direction quickly and to prevent this separation we must avoid too sharp a rate of reduction in cross sectional area as we move toward the rear, and so we get the classic long thin teardrop shape for minimum drag.

However, the length to width ratios needed for minimum drag would render a normal motorcycle too long. To shorten the body by bringing the rear in at a steeper angle would only cause early separation with a large wake and increased drag. It is better to design a longer body and just cut it off at the tail to give the required length, the ideal point of cut off would be at the point of separation. Behind that point of separation the shape has little significance and so the absence of bodywork is unimportant. This is known as a Kamm tail after Dr. Kamm its inventor.

Even with a Kamm tail a conventional bike is difficult to streamline really efficiently because of the various cut-outs needed for practical reasons, such as getting on and off, and putting one's feet down when stopped etc. Improvements can certainly be made though, against most current emphasis, the place to start is at the rear of the machine, the flow around the high pressure areas at the front can reasonably well take care of themselves, but anything that smoothes the flow at the back and helps reduce the size of the wake would really be beneficial. In fact many pointed front shapes would produce less drag if turned around. Machines designed for very high-speed record attempts not only have the smallest possible frontal area but also streamlining extended well behind the rear wheel.

The actual drag force is proportional to the frontal area of the vehicle, so if we double the area we double the drag and power requirement. However, the effect of velocity on drag follows a square law, that is to say, if we double the airspeed we quadruple the drag. The effect of velocity on the power requirement is more dramatic, this is proportional to the third power of the air speed and so we need to increase power by eight times to double the speed.

In order to numerically compare the aerodynamic qualities of different shapes a dimensionless parameter known as Cd. is used. It is sometimes incorrectly stated that a flat plate being pushed though the air has a Cd. of 1.0. This refers only to a special theoretical case when all the air ahead of the plate is pushed away in front of it, and none is spilt around the sides. To compare the actual drag of various machines we need to multiply the Cd by the frontal area, this is known as the CdA and its units are those of the area measurement.

The question of size is a very interesting one, because it affects the validity of extrapolating the results from model tests to the real machine. Wind tunnels large enough to accept a full sized bike or car are very expensive to build and run, and so if scale model tests can be meaningful a great deal of money can be saved. However, the use of models is not always plain sailing. Firstly there are the fairly obvious problems such as model accuracy, e.g. a tolerance of 1 mm. on the real thing becomes a tolerance of 0.2 mm. on a fifth scale model.

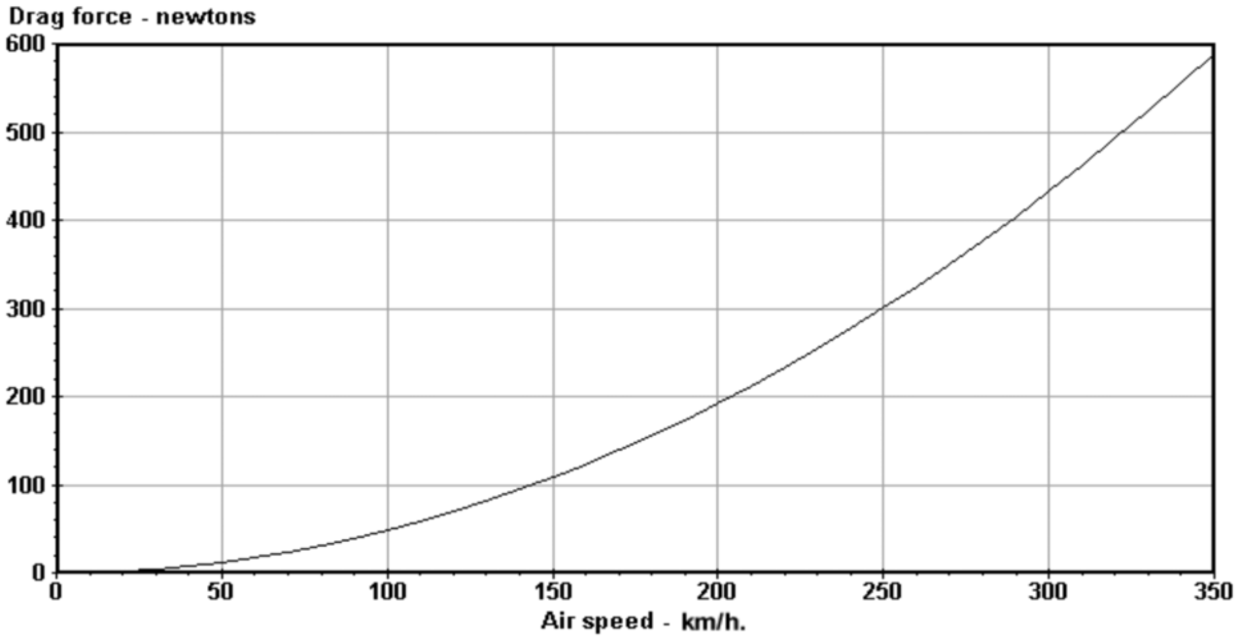
| Description of bike | Rider prone | Rider sitting |
|---|---|---|
| Yamaha Venture | | 0.75 |
| Honda V65 Magna | | 0.61 |
| Honda Blackbird | 0,44 / 0,49 | 0,72 / 0,81 |
| Honda VF1000F | 0.40 | 0.46 / 0.45 |
| Aprilia Mille | 0,52 | 0,61 |
| Ducati 916 | 0,49 / 0,57 / 0,53 | 0,61 / 0,69 / 0,61 |
| BMW R1100 RT | 0,53 | 0,97 |
| BMW K100RS | 0.40 | 0.43 |
| Yamaha R1 (1998) | 0,57 | 0,62 |
| Yamaha FJ1100 | 0.43 | 0.48 |
| Kawasaki GPZ900R | 0.36 | 0.43 |
| Suzuki GSX1100EF | 0.41 | 0.44 |
| Suzuki GSXR750 | 0.32 | |
| Suzuki Hayabusa | 0.31 | |
| Kawasaki ZX-12R | 0.34 | |
| Yamaha OW69 | 0.32 | |
| Honda 1996 RS125 | 0.20 | |
| Honda 1990 RS125 | 0.19 | |
| Honda RS500 | 0.24 | |
| Rifle faired Yamaha | 0.15 | |

**Whilst the value of the Cd is useful for comparing the relative drag qualities of different shapes, it is not so useful for comparing different bikes. It ignores the other main ingredient of drag - frontal area. This table shows the value of the Cd multiplied by the area, this parameter is known as the CdA and is the best way of relating the drag performance of diverse machinery. The lower this value, the lower will be the power needed to produce a given speed. This data has been compiled from various sources and so should only be considered as a guide because there is bound to be variation from measurements made at different sites. Rider size and clothing may be different for example.**

**Multiple values in the table indicate that data for these machines was available from multiple sources.**

**The last machine in the list was especially prepared for a fuel economy contest.**

Reynolds, an early investigator into fluid flow, discovered that the turbulence characteristics of similar shaped but different sized objects was dependent on their size and air speed. In fact, if you halve the size of your object you need to double the air speed to achieve similar air flow patterns. This effect was formalised into a mathematical expression known as the Reynolds number. In order to achieve similar flow patterns over different size models we need to try and keep this number constant, but in many cases this is very difficult. Assume that we wish to investigate the aerodynamics of a motorcycle at 150 km/h. by using a sixth scale model in a wind tunnel. This would then demand (for a similar Reynolds No.) that the tunnel air speed be approximately 900 km/h., which immediately introduces yet more problems. Immense power would be needed to create this flow rate, and the forces on the model would be extremely high, creating mounting difficulties. In fact, at the same Reynolds No. the drag force on the model would be equal to the drag force on the real thing. As if this was not bad enough, 900 km/h. is very close to sonic velocity, the speed of sound. As we get closer to this speed compressibility problems take over and disrupt the whole flow pattern. The end result of this is that small scale model tests have to be done at low Reynolds numbers and much skill and care is needed in their interpretation. In the future it is likely that computer based mathematical models will become good enough to substitute for a wind tunnel in many cases. This is development is known as CFD or Computational Fluid Dynamics.



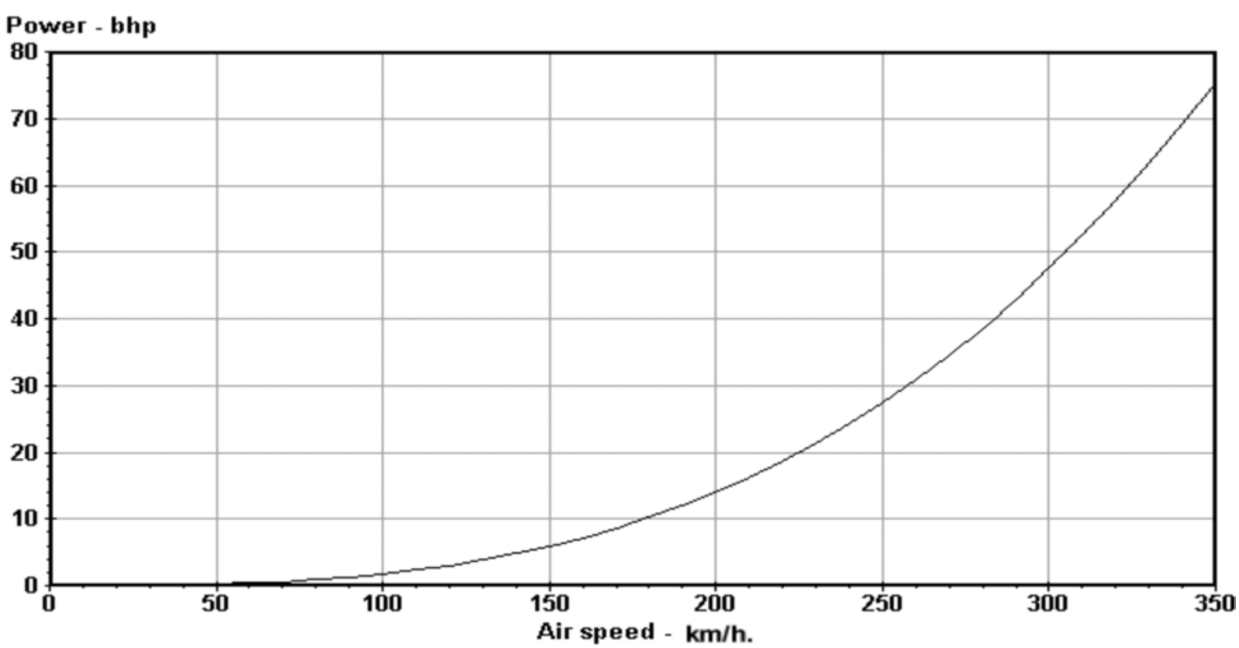
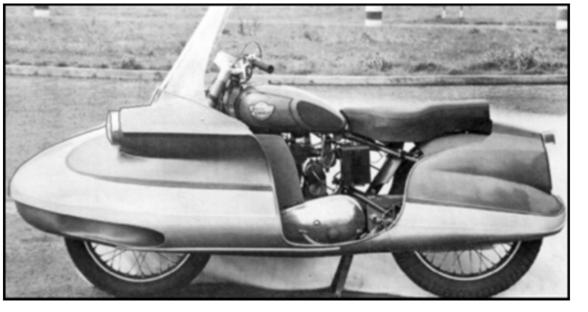




Fig 5.3 **Drag force against air speed. The first graph shows the drag on an object of 1000 sq.cm. frontal area with a CD = 1. The second shows the required power against air speed for an object of 1000 sq.cm. frontal area with a CD = 1.**

The previous graphs, fig. 5.3 show how the drag force and power requirement to overcome aerodynamic forces vary with air velocity up to 300 km/h. The air speed must include any head or tail wind effects. These curves are plotted for 1000 sq.cm. of frontal area with a Cd = 1.0. Therefore to get the drag or power requirement for a real motorcycle we must multiply any values from the graph by the relative frontal area and Cd of that particular vehicle.

The value of reduced drag on everyday motorcycles was clearly shown back in 1956 when "Motor Cycle" magazine commissioned Laurie Watts, the technical artist, to design a semi-enclosing streamlined body shell to be built around a then five year old 350 cc. Royal Enfield. This resulted in a 45 lbf. (20.4 kgf.) increase in weight, but, such was the value of lower drag that even acceleration in the 15 – 40 mph. (24 – 64 km/h.) range was significantly improved, as the table of performance in fig. 5.4 shows. These tests were carried out by the late Vic Willoughby at MIRA using electronic timing gear.



Left: **Vic Willoughby testing the Motor Cycle magazine sponsored Royal Enfield “Dreamliner”.**

**Even though its shape appears to owe more to styling than to optimum aerodynamics, its performance figures are impressive. It added an extra 45 lbf. (20.4 kgf.) to the weight but still achieved improved acceleration and reduced fuel consumption: 35% less at 60 mph. (97 km/h.).**

Above: **Side view showing front and rear fairings.**

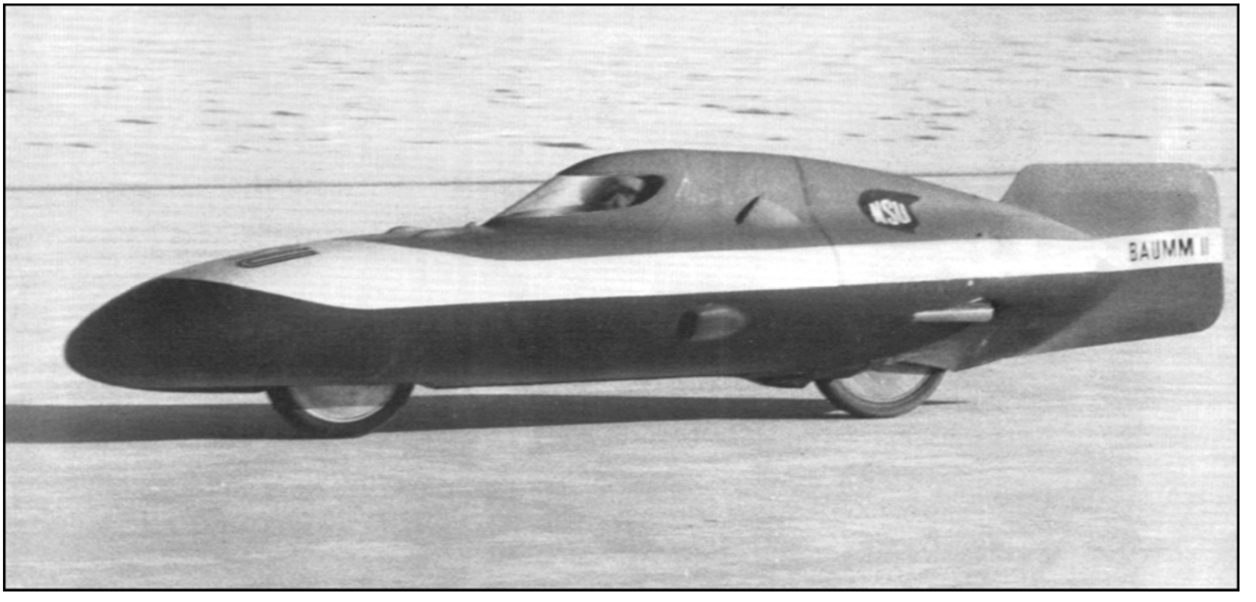
Trying to reconcile these results into a percentage reduction in drag has its problems, largely because all tests were done with unchanged gearing. Thus, the engine RPM. and hence the BHP. produced would have been different during the maximum speed runs, depending on whether the bodywork was fitted or not. The engine power being produced at top speed may have been more with the fairing fitted as the machine was normally over-geared. However, by making one or two assumptions, I have calculated that the drag with the body fitted was probably in the range of 75 -- 85% of the unfaired machine. So, if this order of performance improvement is possible with such a body, just imagine what could be achieved with a purpose built device, -- a well streamlined FF. for example, with a smaller frontal area as well as the possibility of a better shape.

## Table of performance

| Nature of test. | Standard machine | With bodywork |
|---|---|---|
| | | |
| **Fuel consumption** | | |
| 30 mph. (48.3 Km/hr.) | 102 mpg. (2.77 l/100 Km) | 126 mpg. ( 2.24 l/100Km) |
| 45 mph (72.4 Km/hr.) | 83 mpg. (3.42 l/100 Km) | 105 mpg. (2.69 l/100 Km) |
| 60 mph (96.6 Km/hr.) | 55 mpg. (5.14 l/100 Km) | 74 mpg. (3.81 l/100 Km) |
| 70 mph (112.7 Km/hr.) | Couldn’t maintain around the track. | 60 mpg. (4.71 l/100 Km) |
| | | |
| **Acceleration** | | |
| 15 – 40 mph second gear | 6.4 secs | 5.5 secs |
| 40 – 65 mph top gear | 15.7 secs | 13.6 secs |
| Standing ¼ mile (402 m.) | 20.3 secs | 19.6 secs |
| Speed at end of ¼ mile | 61 mph (98.2 Km/hr.) | 64 mph (103 Km/hr.) |
| | | |
| **Top speed** | | |
| Sitting up, with head wind | 59.5 mph ( 95.8 Km/hr.) | 66 mph (106.2 Km/hr.) |
| Sitting up, with tail wind | 70 mph (112.7 Km/hr.) | 78 mph (125.5 Km/hr.) |
| Sitting up mean | 65 mph (104.6 Km/hr.) | 72 mph (115.9 Km/hr.) |
| Laying down, with head wind | 64.5 mph (103.8 Km/hr.) | --- |
| Laying down, with tail wind | 78 mph (125.5 Km/hr.) | --- |
| Laying down mean | 71.3 mph (114.7 Km/hr.) | --- |
| | | |
| **Coasting** | | |
| From 60 mph (97 Km/hr) to 0 | 388 yards (355 metres) | 500 yards (457 metres) |

Fig 5.4 **Table of performance for Royal Enfield “Dreamliner”. A very low performance machine by today’s standards but still dramatically illustrates the performance benefits of lower drag. Despite the extra weight the acceleration figures also show an improvement.**

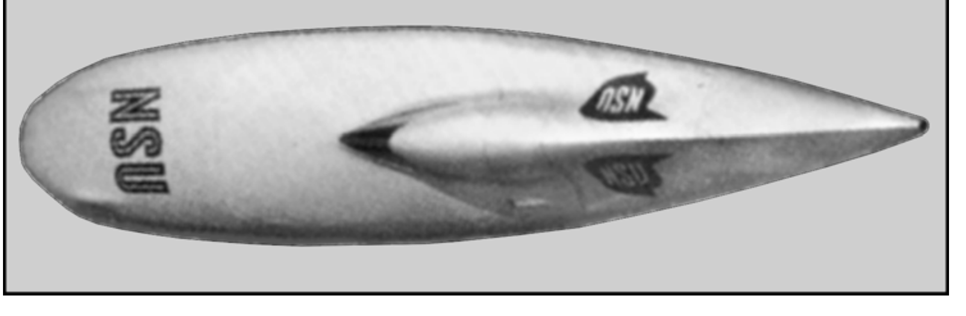
So what shape is best from the drag view point? Well, the classic tear drop shape as shown in fig. 5.1 would be hard to beat, and an approximation to this was used very successfully by NSU to take many world speed records in the smaller classes. There were plans to use shorter versions of these streamliners in GP racing, and later on road machines, but the FIM changed the rules in 1957 and outlawed all but the most rudimentary of fairings. A legacy we are still left with today.



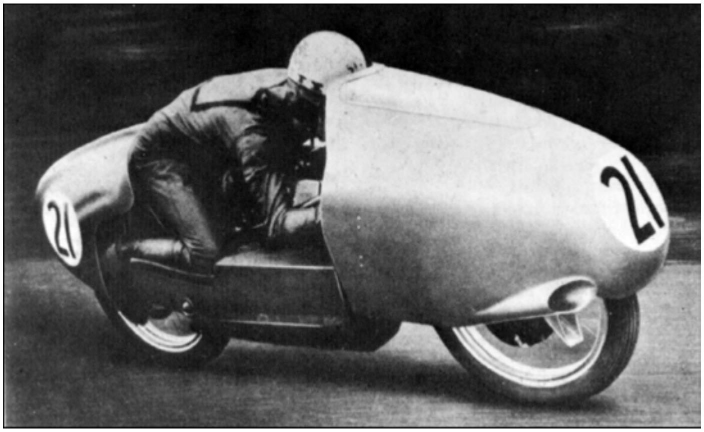

**H.P. (Happy) Mueller streaks across the Utah salt at 240 km/h in the 125 cc NSU "flying hammock" in 1956. Probably one of the most aerodynamic motorcycles built. The tail fin was only fitted for runs above about 240 km/h.** (MCW)

The classic tear drop has a circular frontal cross section but when placed near the ground as for a road vehicle, the ground proximity interferes with the symmetrical airflow pattern and so the ideal shape would need modifying to account for this. This interference with the airflow is also most likely to introduce a lift force.

**This top view of an NSU record breaker shows how close its designer came to a perfect tear-drop shape. Note the long length to width ratio, which allows for a shallow angle toward the rear.**



# Evolution of the racing fairing.

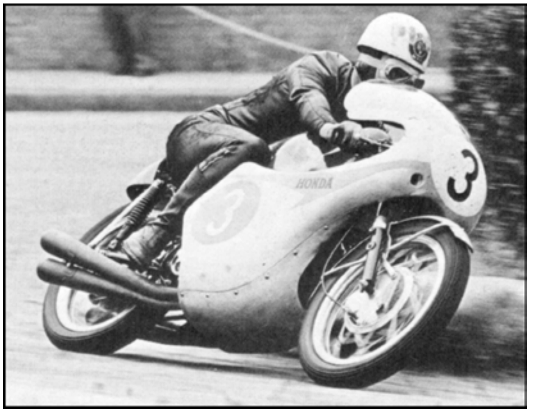



**Although it never set a fashion for racing, this 1953 kneeler Norton was used to increase the one hour record to 134 miles (216 km.) at the Montlhery circuit with Ray Amm aboard.**


**It is not hard to see how the term "dolphin fairing" originated. This 1954 NSU Rennmax is a direct ancestor of today's streamlining. There can be little doubt that the nose gave rise to a lot of lift.**


**Whilst full frontal fairings were popular in all capacity classes in the 1950s. Tail fairings were rarer, generally being fitted to the 125 cc. machines, which needed all the help they could get. Slipstreaming by following competitors was rendered less effective because of the reduced wake. This is multi-world-champion Carlo Ubbiali winning the 1956 Belgian GP on a 125 cc MV Agusta.**


**With their own full size wind tunnel the Carcano designed Moto Guzzis of the 1950s. probably had the most efficient "dustbin" fairings of the period. Shown here is a 1957 500 cc. single cylinder at the TT with Keith Campbell aboard. Its 240 km/h. top speed from just 47 bhp. being testament to the quality of the aerodynamics.**





**After the 1957 FIM ban on certain types of fairing, the "dolphin" shape quickly became the norm by the end of the 50s and early 60s. This shot of the great Bob McIntyre on a 250 Honda in the 1961 TT shows the basic form that has continued until the present time.**

**The aerodynamics of this 1999 125 cc. Aprilia are only different in detail to that of the older Honda on the left. The belly pan is narrower and fills almost completely the gap between the two wheels. Note also the front mudguard which is faired in to blend the air over the front forks. At the rear the seat back is larger and higher to help smooth the flow as it leaves Jeronimo Vidal's back.**

## Internal air flow

The needs of the engine also complicate the issue and present requirements that inevitably increase drag. Any bike needs a supply of cool fresh air to breathe efficiently, and to provide engine cooling. Many attempts have been made to use the high pressure at the front of the machine to slightly pressurize the air fed to the induction system, to achieve a supercharging or ram effect at high speeds. In the past, carburetion problems made this difficult to exploit properly, but with modern electronic fuel injection systems it has become a more practical proposition. An alternative technique is to totally enclose the carburettors inside a large air box fed with cool air from ram tubes in the front of the fairing. At normal road going speeds the potential power benefit of ram charging is quite minor; very high speeds are necessary for it to become significant. At 160 km/h. the maximum possible benefit is a bit over 1% and we have to be travelling at around 320 km/h. to get a 5% increase in power. This might add about 5 km/h. to the top speed at 320 km/h. but only about 0.5 km/h. at 160 km/h. However, these values are assuming that we can make use of the maximum potential, in practice various intake losses prevent us from achieving those figures. Generally the main benefit of ducting air from the front is to ensure a supply of cool air to the intake system.

Figure 5.5 shows actual airbox pressure measurements taken from a GP 250 machine. We can see how the airbox pressure rises in like fashion to the speed increase. The spikes in the pressure curve occur when the throttle is shut for a short time whilst changing gear. The engine is not sucking any air out of the box at that time and the airbox pressure rises towards the full ram pressure available.

Whether air or liquid cooled we have to provide a flow of cooling air. Air cooled engines give us no choice over where this air must be directed, but with liquid cooling the radiators can be mounted in a variety positions to suit space and/or aerodynamic considerations. The usual location is the obvious frontal position, but this is largely obstructed by the front forks. Alternatively radiators have been side mounted (NR500 Honda) and there have been experiments with mounting under the seat area (Britten).

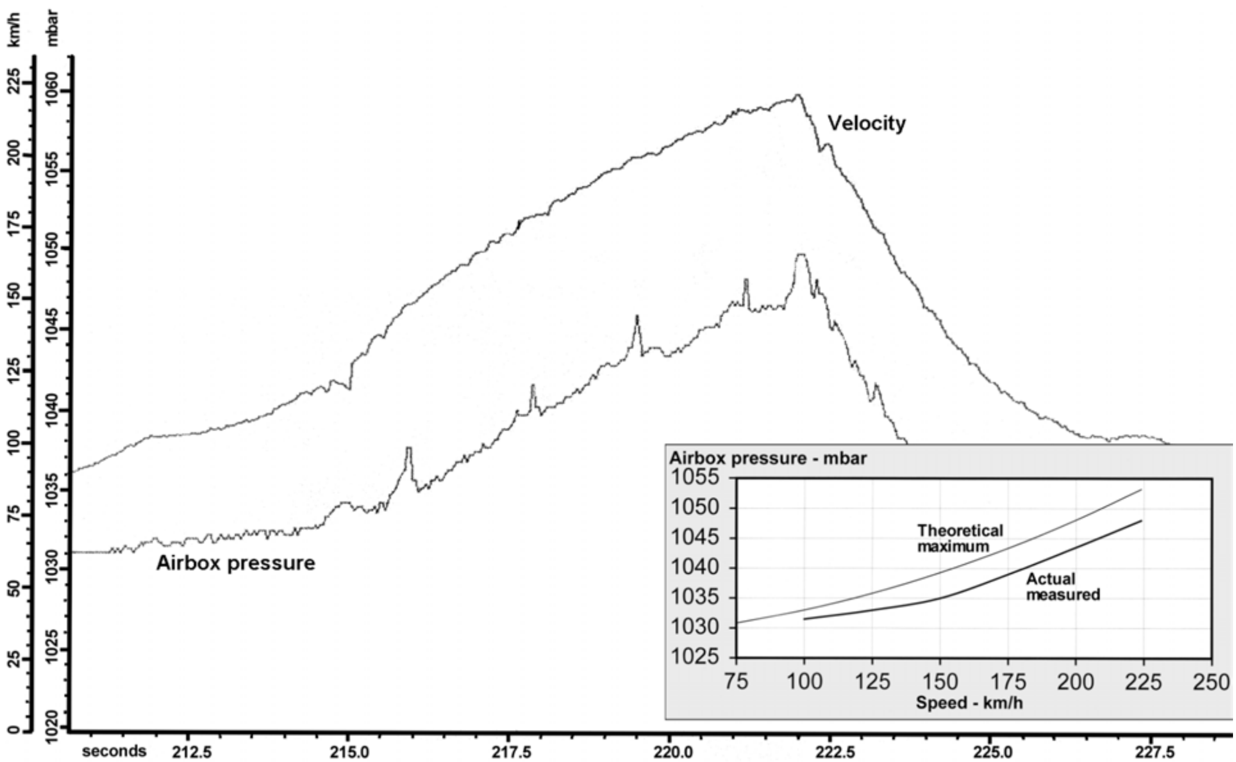



Fig. 5.5 **Traces from the data acquisition system of a 250 GP racer. The horizontal axis shows elapsed time in seconds, the upper curve shows velocity in km/h and the lower curve shows the airbox pressure in milli-bars. The inset shows the same data re-plotted against speed. Note how the real data is less than the theoretical maximum.**

(From data supplied by 2D)

## Lift

In the same way that we can group together the various masses of a motorcycle into one equivalent point and call it the Mass Centre or Centre of Gravity (CoG), we do the same with the aerodynamic forces. Viewed from the front and side there will be one point through which the total force can be considered to act. This is called the Centre of Pressure (CP), fig. 5.6. This resultant force can be resolved into vertical and horizontal components. If the bike is not laterally symmetrical or if there is some side wind then there will be a lateral component also. The fore and aft part of the horizontal component represents the drag force, whereas the vertical component will be the lift or down-force, depending on its direction. In addition to the linear forces acting at the CP, the pressure distribution may be such as to create moments. For example, if we have lift toward the front and an equal downforce toward the rear, then the nett vertical force is zero but there is a moment tending to rotate the machine backwards. These are termed the aerodynamic moments. In addition to visualizing the aerodynamic forces and moments as acting at a point, it is often useful to consider the relationship of this point to the CoG location. These forces create moments about the CoG which can have a large impact on dynamic stability.

Aerodynamic lift as described above reduces the total load on the tyres, and so is potentially dangerous, because it reduces the maximum possible grip of the tyres on the road.

If the lift CP is in front of the CoG then this lift force will cause a greater reduction of load on the front compared to the back, and vice versa. It is not always realised, but even with a body shape that produces no nett lifting force (only drag) nor any direct aerodynamic pitching moment there is still a tendency for load to transfer off the front wheel and onto the rear due entirely to the effect of the drag force, as fig. 5.7 shows.

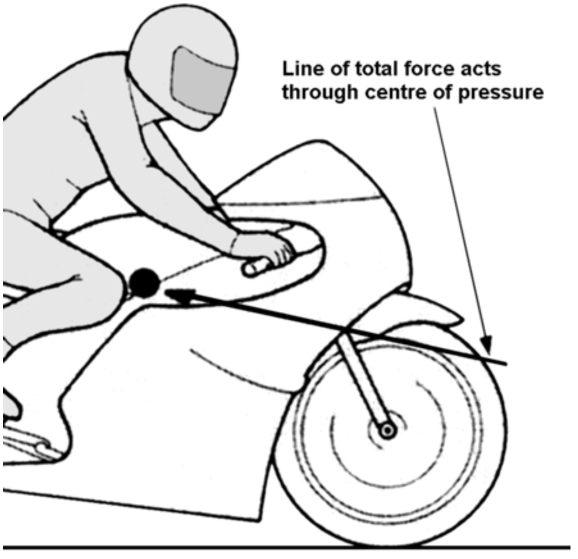



Fig 5.6 **In the case shown, the resultant aerodynamic force is acting upward. The horizontal component is the drag force and the vertical component represents a lift force.**

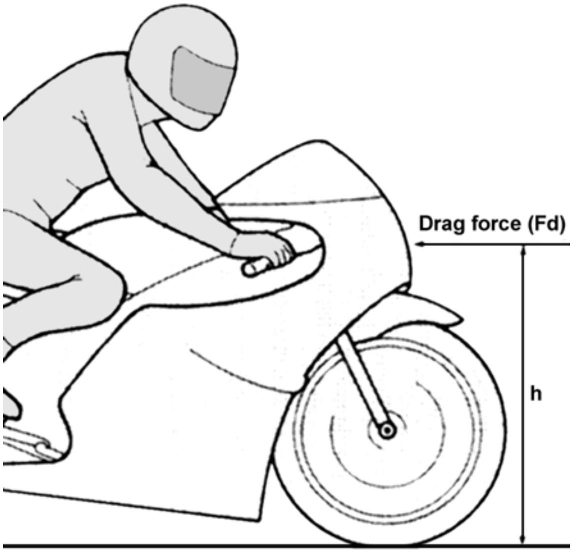



Fig 5.7 **The drag force Fd acts at height h to create a moment Fd.h tending to unload the front wheel and load the rear.**

The drag force, Fd, acts through the frontal CP, at a finite distance above the ground, h. This produces a moment trying to rotate the bike backward, thus lifting the front and loading the rear. It is easy to put some values to this to get an idea of the relative importance. Fig. 5.8 shows how serious can be the load transfer off the front wheel from about 130 km/h. and upward. These graphs are plotted for a fictitious motorcycle with a frontal area of 4000 sq.cm. with a CD = 0.7, a wheel base of 1500 mm. and a CP height of 750 mm. Two curves are drawn, one assuming that the loaded bike has a static load of 115 kgf. on the front wheel and the other with 90 kgf. At 200 km/h. the load on the front wheel is reduced by 270 N. and the load on the rear will have increased by 270 N., this is a significant amount and will have an effect on the handling and cornering abilities of any motorcycle. This example represents a fairly low drag racer and so gives a conservative view of the load transfer.

All else being equal, anything that reduces drag and/or the effective height at which it acts, will help reduce this load transfer. In addition to just reducing drag the ideal situation would be for the body shape to create a down force at the front and a lifting force at the rear to exactly balance the above effect.

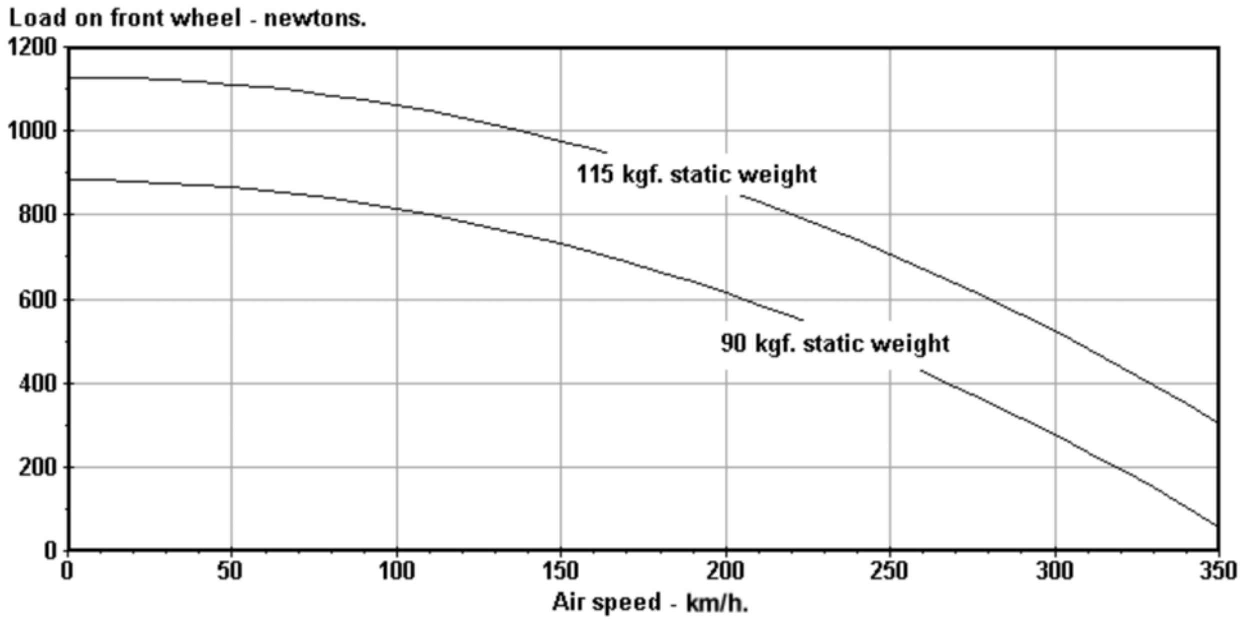
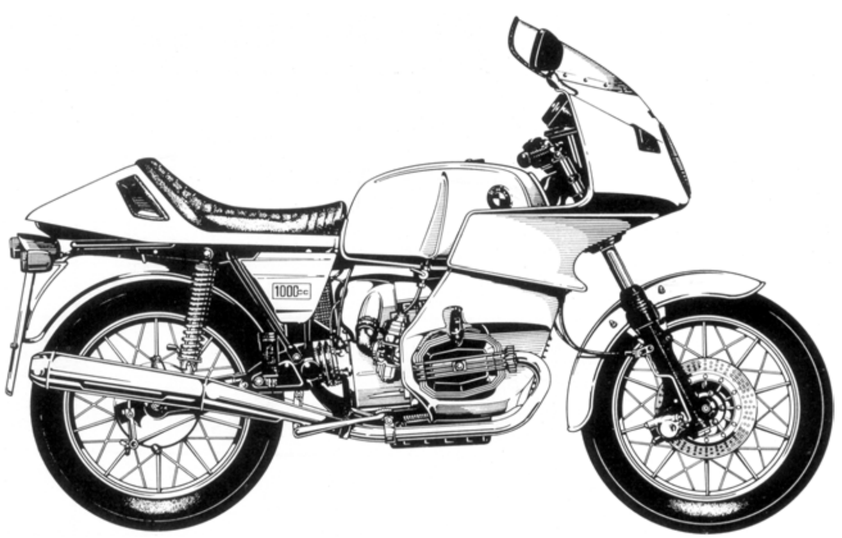



Fig 5.8 **Drag induced reduction of load on front wheel at different air speeds. Shown for 2 static front wheel weights, for an example motorcycle with a Cd of 0.7 and a frontal area of 4000 sq.cm.**

Originally led by BMW. with their R100 RS. fairing, and also on some racing Suzukis, this problem of cancelling the lift is now receiving more attention from the manufacturers.




**Not only were BMW one of the first to fit a full fairing as standard on the R100RS, but considerable effort was put into reducing front wheel lift. Note the sloping flatish surface above the hand covers as well as the lateral fin just behind the top of the front wheel.**

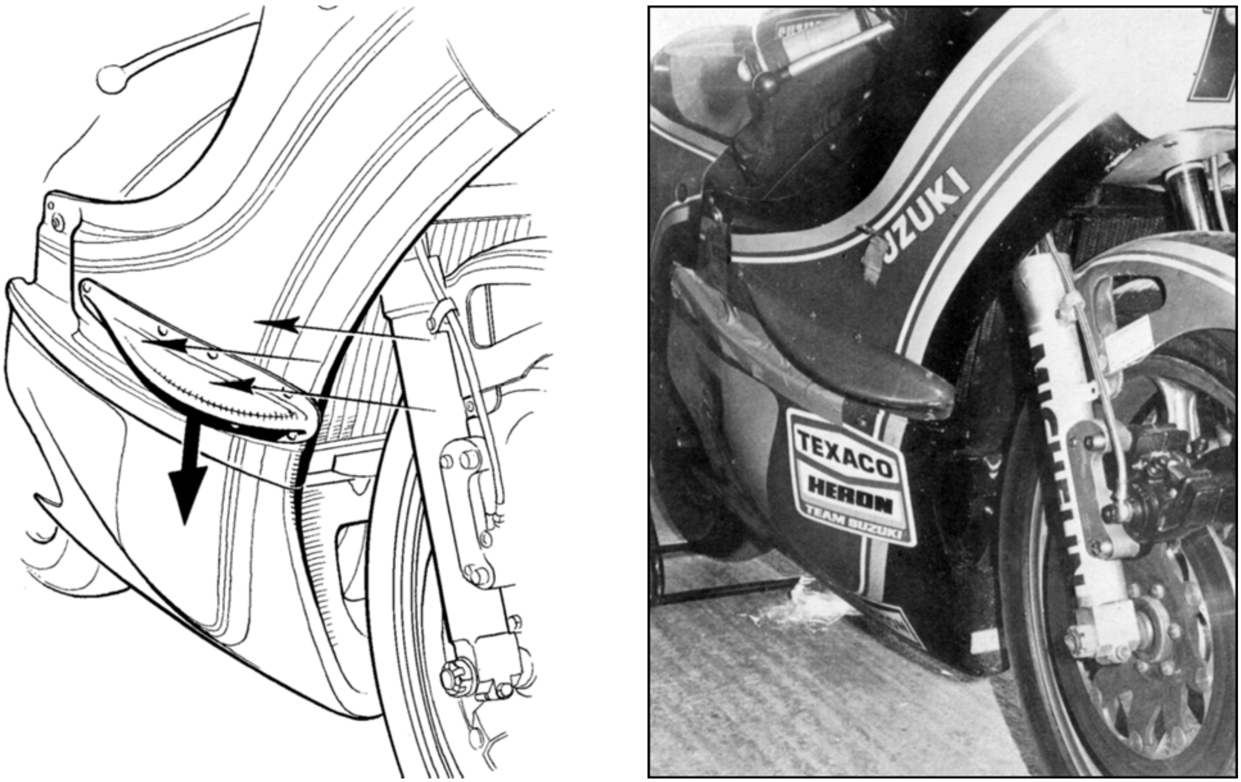



Fig. 5.9 **Small winglets on this GP Suzuki were designed to provide some measure of downforce near the front to counter the tendency to lift the front end at speed.** (MCW)

We must remember that this drag induced lift effect is not the only aerodynamic effect that is changing the load on the tyres. To this we must add any overall lift, which will reduce the load on both wheels and any aerodynamic pitching moments which might tend to lift one end and press down on the other.

Since racing cars have achieved dramatic increases in cornering speed through the extra tyre grip produced by aerodynamic downforce from front and rear wings and 'ground-effect' chassis, it might be thought that the same principle could be exploited on a solo motorcycle. Because our machine has to be banked for cornering, however, the situation is more complex than it is on a car. Indeed, a fixed wing or other means of generating aerodynamic downthrust might well reduce our cornering speed. Figure 5.10 representing a banked motorcycle, shows that the down-force provided by a fixed wing acts in line with the bike and has both vertical and horizontal components.

However, although the vertical component is available at the tyre contact patch to increase grip, this extra grip only counteracts the extra horizontal component, and so there can be no nett gain. As explained in the tyre chapter, available tyre grip depends also on the contact-patch pressure, with the result that the extra tyre loading from the downforce reduces the coefficient of friction. Thus, the grip available to generate the centripetal cornering force is also reduced, so that the maximum cornering speed is slower.

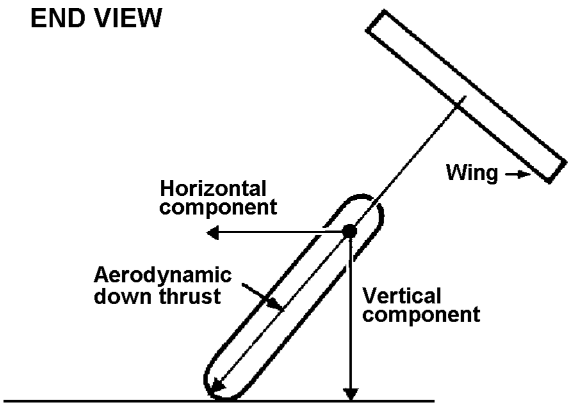



Fig 5.10 **A fixed wing across the bike just increases the horizontal and vertical components acting on the tyres in the same proportion and hence there is no nett gain in cornering performance. Probably the reverse.**

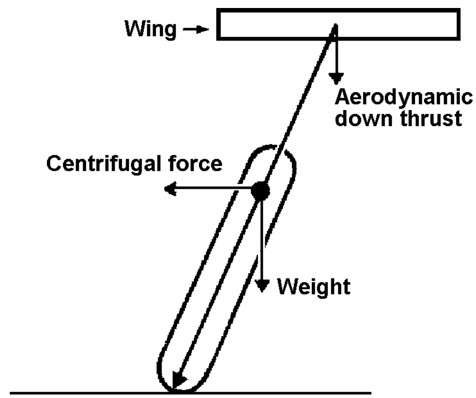



Fig 5.11 **In theory, a tilting wing that remained horizontal could enhance braking, traction and cornering power. As a side effect, a smaller angle of lean would be required for a given cornering speed.**

**Colin Lyster with his "wing" designed for a 450 Honda racer. The angle of attack was to be adjusted by fore and aft movement of the rider in the hope of better braking with the added downforce and extra drag.**

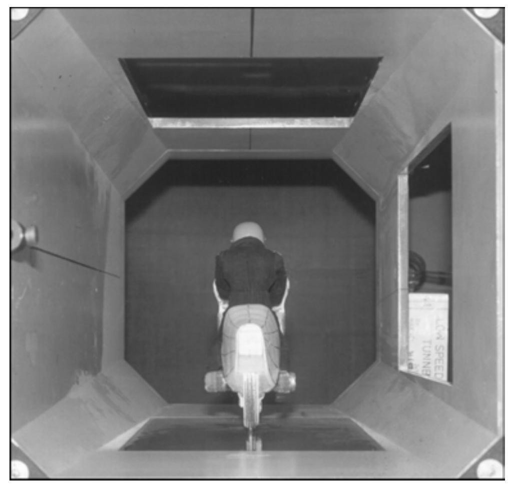
**However, the drag of this high mounted wing would create a backward moment which would reduce the vertical load on the front wheel.**

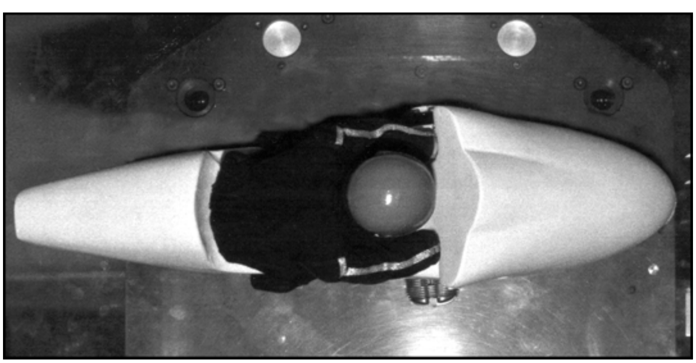
**Other disadvantages would include: greater roll moments due to side winds and even with a light-weight wing structure its high position would increase the roll moment of inertia quite considerably, any lateral flexibility in the mountings would most likely result in some form of weave instability.**

**The dynamics of motorcycles are very different from the dynamics of cars and not all "improvements" from the car world are applicable for motorcycle use.** (MCN)

Conversely, aerodynamic lift might actually increase cornering speed – though at the expense of high-speed braking and traction forces and of stability. Taken to a ridiculous extreme, a motorcycle would then become an aeroplane, which can corner at lateral accelerations limited only by the ability of the structure to withstand the forces generated. If we could generate aerodynamic down-thrust in a vertical plane only (say, by a tilting wing) then increases in braking, traction and cornering speed would follow (as in a car). An interesting side effect would be that a smaller angle of lean would then be required for a given cornering force.

As can be seen from fig. 5.11 the moment due to the downforce helps counteract the centrifugal moment. Hence the moment due to the machine's weight must be reduced – which means a smaller angle of lean. However, any benefits from such a device might well be outweighed by practical and stability problems. In a straight line, any aerodynamic features that create extra down force will load the tyres more and make available more traction. Provided that our brakes are good enough and that we have sufficient power, this greater grip will allow for better acceleration and braking.



**Whilst ultimate aerodynamic efficiency was not the prime goal of the author's QL, it was desired to improve upon a standard machine within the bounds of practicality. The photos above right and lower left show a 1/6 scale model in a wind-tunnel. Measurements from this model were compared with a 1/6 scale model of a standard R90s BMW. This comparison indicated that the QL had 65% of the drag of the standard bike. Whilst the low Reynold's numbers may have caused relative differences in the separation points it was clear that drag was significantly lower. Road tests later confirmed this as well as excellent weather protection and stability.**


**The mid 1980s Ducati Paso featured this almost totally enclosing bodywork, with internal ducting. As a road bike the rider would be expected to have a sight line above the windshield which, like the author's QL, was not transparent.**

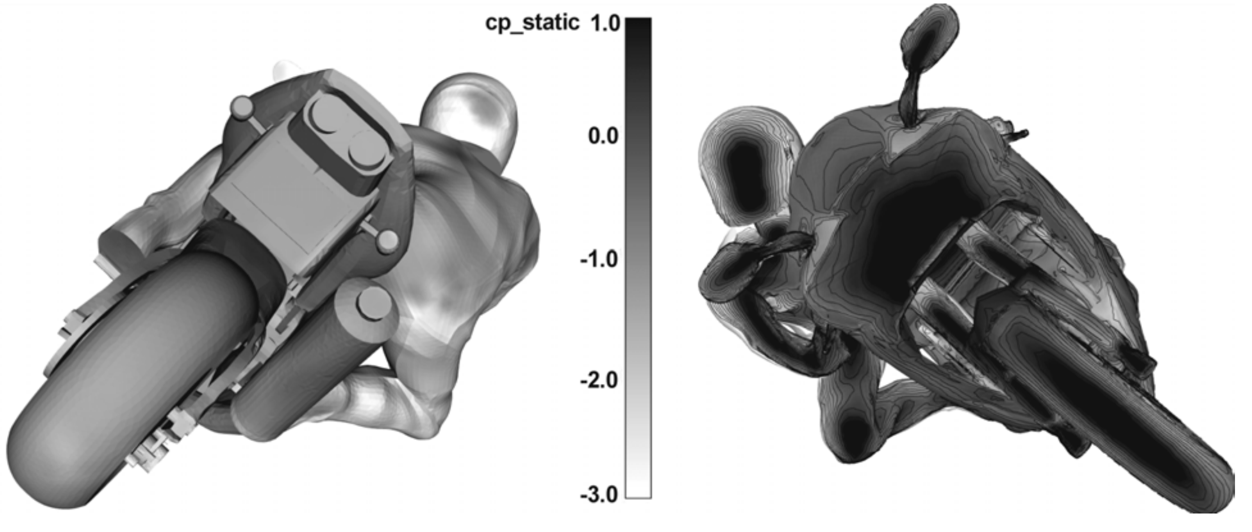
**The NSU record breaker being tested in the full size wind tunnel at the Stuttgart Technical College. The size of such installations is quite apparent from this photograph. Compare this to the much smaller tunnel used by the author to test the 1/6$^{th}$. scale model of the QL, pictured earlier.**

## Airflow evaluation

Traditionally the actual aerodynamic performance of vehicles has been evaluated in wind tunnels, but this is a very expensive undertaking. We have already seen some of the problems of using small scale models in smaller tunnels, namely model tolerances and achieving similar Reynolds numbers. There is no real substitute for full size tunnels and actual wind speeds, but such installations are large and expensive to build. Considerable power is also required to run the fans. Using such tunnels is very time consuming to refine the shape of a vehicle. Tests must be run, then the vehicle or a model of it needs to be modified where the tests indicate, then the test/modification cycle must be repeated, possibly several times. However, the advent of small powerful computers looks set to offer an alternative, CFD or Computational Fluid Dynamics. CFD is a method of dividing the air surrounding a vehicle into a vast number of small cells and then calculating the pressures, velocities and turbulence in each cell, but taking into account that these variables must balance from one cell to the next across the cell boundaries. The problem is that millions of cells are needed to model the flow around even fairly simple shapes. Computers are thus needed to:

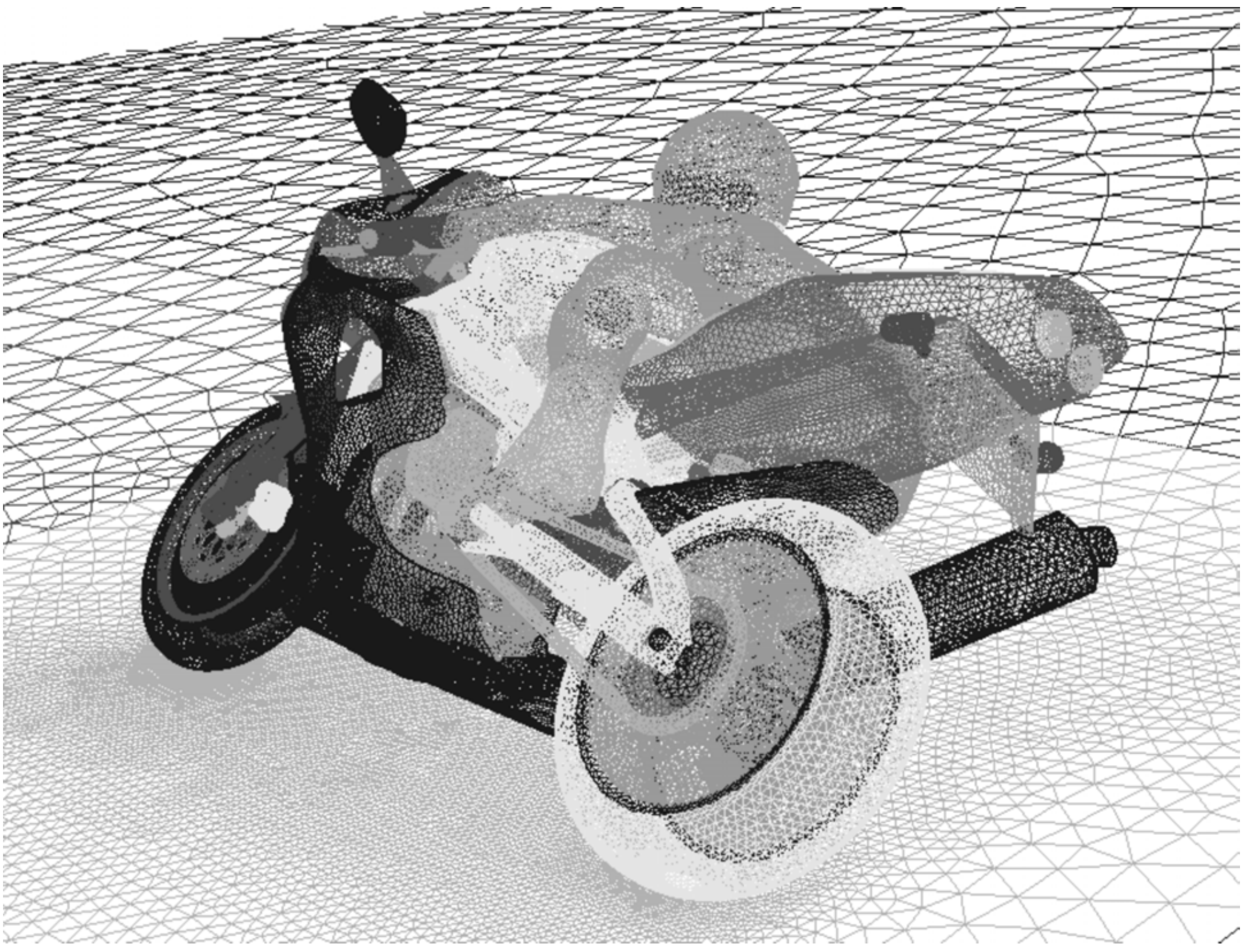
- Create and define the numerous cells from CAD drawings or digitised shapes from laser scanning.
- Assemble the properties of all the cells into a set of non-linear equations.
- Solve these millions of equations simultaneously.
- Modern computer graphics are then used to present the huge volume of numerical output data into a form that is easy for humans to digest.

Large cost savings and rapid development times are the promise of such techniques. There is no need to build physical models, these are just defined from the CAD drawing of the vehicle. To iterate through a wide range of alternative designs needs only a few surfaces on a drawing to be changed, and the computer run repeated.

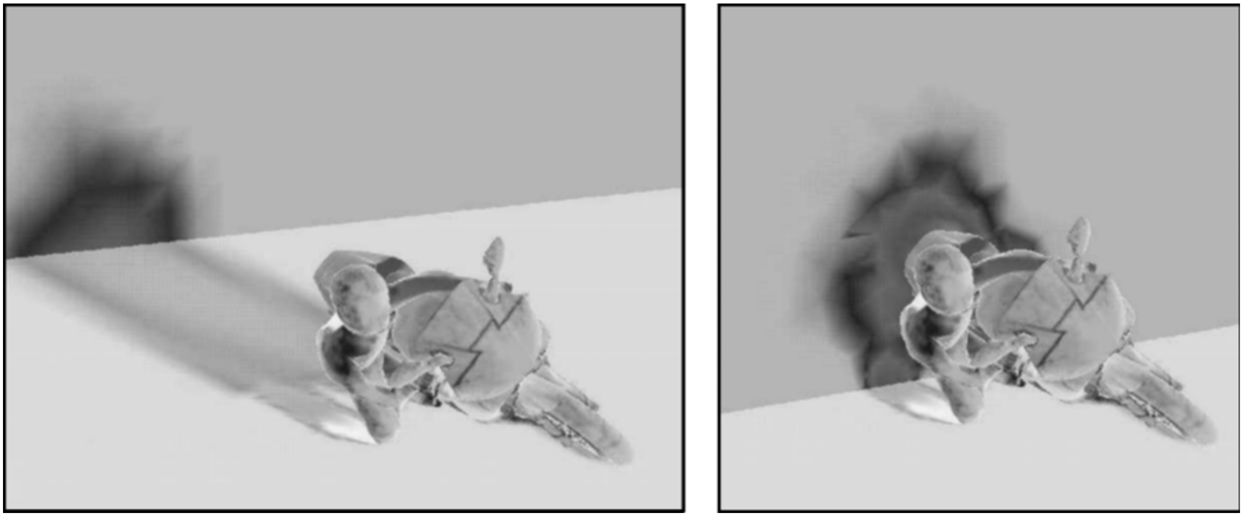



**CFD plots of the pressure distribution on bike and rider. The shading indicates the static pressure, as per the central scale. Such plots are normally in colour which highlights the differences much better than these grey scale images. The right hand sketch is also marked up with iso-bars, or lines of constant pressure.** (Advantage CFD, Reynard Motorsport Ltd)

At the moment computer technology is at a level which still demands that quite extensive computer facilities are necessary for large models. The current high-end stand alone desktop computer being suitable for smaller problems such as the air flow inside air-boxes. Future generations of desktop computers will have more power and undoubtedly that will make CFD techniques much more accessible as has been the case with FEA (Finite Element Analysis) for structural design. Nearly forty years ago the author worked with FEA, but at that time such techniques were solely available for specialists with access to large main-frame computers, and required that the user had a high level of skill in the techniques. Nowadays, such methods are available to any designer with a desktop computer and any mid to high level CAD programme. CFD is mathematically similar to structural FEA, in fact it is really the FEA of fluid flow, but the big differences are the vastly greater number of elements or cells needed to adequately model a fluid flow problem, and more importantly the greater complexity of the non-linear equations necessary.



**A 2D representation of the CFD grid of cells around the bike and rider. In reality the cells are 3D and fill the whole volume around the machine, literally millions of cells are necessary to model the flow successfully.** (Advantage CFD, Reynard Motorsport Ltd)

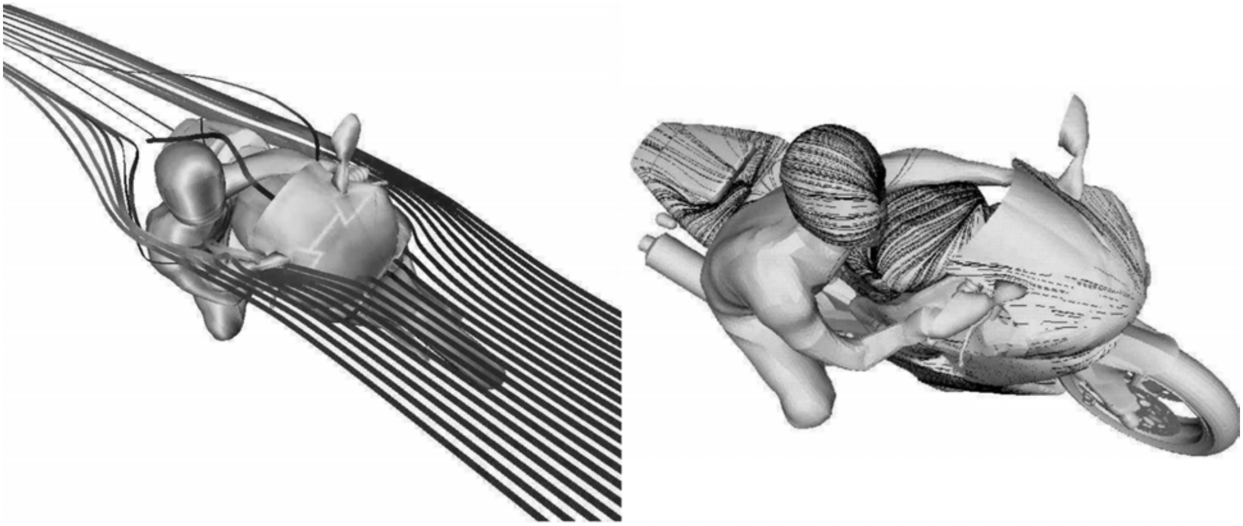


**These CFD output illustrations show the size of the disturbance at two imaginary vertical slices through the air. The first is some distance behind the bike and the other is immediately behind the seat. Simply put these show the size of the hole punched in the air by the bike and rider.** (Advantage CFD, Reynard Motorsport Ltd)

At the present time CFD is still a technique that mainly requires specialist facilities and staff, and is generally performed by dedicated consulting entities. One such is Advantage CFD, a division of Reynard Motorsport, makers of racing cars. As an example Advantage CFD recently applied these techniques to the study of a cornering motorcycle, to evaluate the feasibility of using the aerodynamic characteristics, of the flow between the ground and the inboard side of the fairing and rider, to improve cornering. We noted earlier that creating down-force in line with the bike would likely be counter-productive, but the thrust of this analysis was to create a sideways “lift” force which would add directly to the cornering force. The results of this work indicated that small modifications to the belly pan shape could in fact increase cornering speed by a small margin. Similar work in a wind tunnel would have been very hard, due to the difficulty of simulating the leant over bike travelling a curved path. The illustrations show just some of the flow and pressure visualizations output from the CFD analysis of this project.

Just as structural FEA has resulted in lighter and stronger engine and chassis parts over the past decade or so, then in the future it is certain that, as CFD becomes more accessible and accepted, aerodynamics will be improved also. However, except for simpler or smaller problems that will require the next or a future generation of computer technology to enable. Studying the air flow around a complete motorcycle and rider combination in detail requires a lot of computing, especially for the study of dynamic stability in gusting conditions.

For the present, CFD techniques are sufficiently advanced, when used by knowledgeable specialists, to be of practical benefit to designers of F1 cars, who commit millions of dollars to it each year. It might seem strange that motorcycle manufacturers and well funded race teams have so far ignored this technology, particularly considering the economics of modern top level racing. It is hard to see why, although history has shown that motorcycling in general is slow to adopt state of the art design and analysis tools. Undoubtedly CFD will gradually come into use, helping not only racers go faster but also improving weather protection and stability for tourers. These techniques can also be applied to air flow analysis through air-boxes, engines and exhaust systems.



**Two different types of CFD flow visualization. On the left we see overall streamlines and on the right we see what's happening right on the surface of some parts of the rider and bike. Visualizations like this can be produced for any viewing angle.** (Advantage CFD, Reynard Motorsport Ltd)

## Side wind stability (traditional view)

Most descriptions of motorcycle aerodynamics follow the stability criteria of non-leaning vehicles and low speed aircraft, however this approach neglects some very important yaw-roll-steering coupling effects inherent in single track road vehicles. Let's look at the traditional explanation first and then see how a real motorcycle differs:

"*Aerodynamic stability is often said to occur automatically if the centre of pressure is some appropriate distance behind the centre of gravity. If this is not the case, then the rider must apply a steering correction. It is this same stability criteria that requires aircraft to have vertical fins at the rear, and darts and arrows to have rear feathers. When exposed to side wind, if the CP is rearward, the back of the machine is blown away from the wind, thus causing the front to point into the wind and automatically correct for the tendency to be blown off course.*

*On the other hand, when the CP is towards the front then this end supposedly gets blown with the wind and the machine will steer with the breeze away from its original course, a basically unstable condition. In much the same way as a weather vane keeps its arrow pointing into wind, so the tail fin on aircraft gives it directional stability."*

In fact achieving a rearward CP is harder than talking about it. Conventional motorcycles tend to have most if not all of their bodywork in front of the rider, partially a result of the short-sighted FIM. decision in 1957 to severely restrict the design of racing streamlining, and so a forward CP is inevitable with a normal layout.

Ironically, even with a purpose designed streamlined body with plenty of side area to the rear, it is often even more difficult to achieve the desired end. To understand why, we need to look a little closer at the detail of the airflow around such a machine. Fig. 5.12 shows how, at speed, a side wind does not really act as a side wind at all but combines with the effective head wind, from the bike's motion, to create an airflow at an angle to the machine. e.g. If the bike is travelling at 120 km/h. with a side wind of 25 km/h. then the effective breeze will be at 123 km/h. acting at an angle of about 12°.

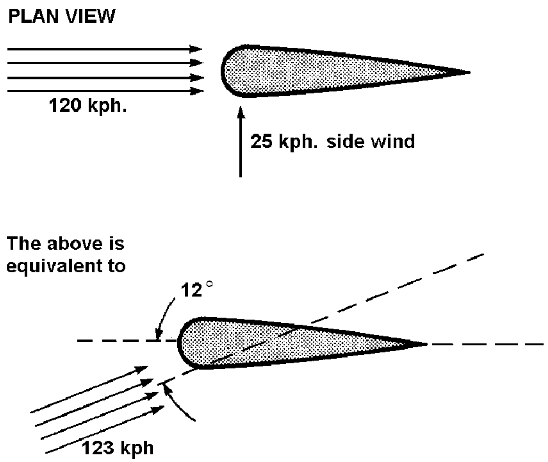



Fig 5.12 **Adding the effects of the forward speed of 120 km/h and the side wind of 25 km/h results in a total equivalent air flow of 123 km/h acting at an angle of about 12 degrees.**

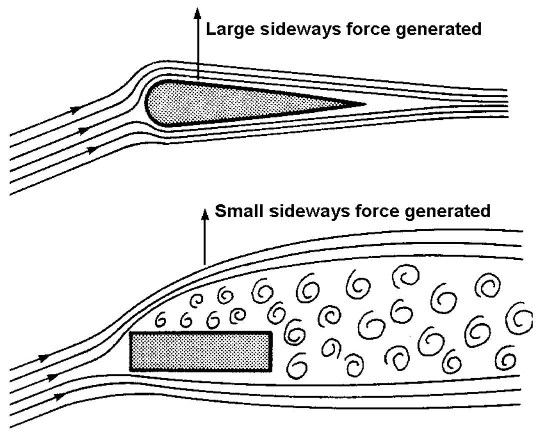



Fig 5.13 **Shows how a well streamlined shape such as an aircraft's wing produces a side or lift force, whereas the brick shaped object is less affected by the inclined air stream.**

A well streamlined section, such as an aircraft wing will produce very high forces at such an angle of incidence, on aircraft wings these forces act as lift, and their magnitude is amply demonstrated every time a 747 takes to the air. On a road vehicle these forces act as side forces on the machine, and we wish to reduce their magnitude so that the disturbing effect is minimised. A badly streamlined shape would make a very poor aircraft wing because the degree of lift would be low and the same consideration applies on the road, the magnitude of the sideways disturbing forces will generally be reduced with a high drag design. A real motorcycle fits somewhere between the near ideal teardrop shape (for drag reasons) and a brick with wheels. The air flow around the front area of the machine will be reasonably well ordered but at some point along its length separation will occur and the flow rearward of this point will be turbulent. When this occurs the side area exposed to this disturbed flow will not be effective in keeping the CP towards the back. As speed gets higher, the separation point tends to move frontward and so does the CP. Streamlined record breakers often have huge tail fins in an effort to redress the balance.

Let us continue to see how this is explained by the traditional view:

*"Suppose, for whatever reason, the machine adopts a yaw attitude at high speed, when aerodynamic forces are significant (i.e. it is pointing at a slight angle to the direction of travel, as shown in fig. 5.14). Initially, the machine's momentum tends to keep it moving in the original direction, regardless of its attitude, so that the wind force created by its motion acts on the whole of the forward-facing flank. The pressure on the area ahead of the mass centre gives rise to a moment that reinforces the disturbance, while the pressure on the area behind the mass centre tends to correct the yaw. Which of these opposing effects predominates depends on the relative positions of the mass centre and the centre of pressure."*

This rather traditional explanation is quite accurate for aeroplanes and most multi-track road vehicles, however, it needs to be modified to allow for the unique attributes of leaning single track machines.

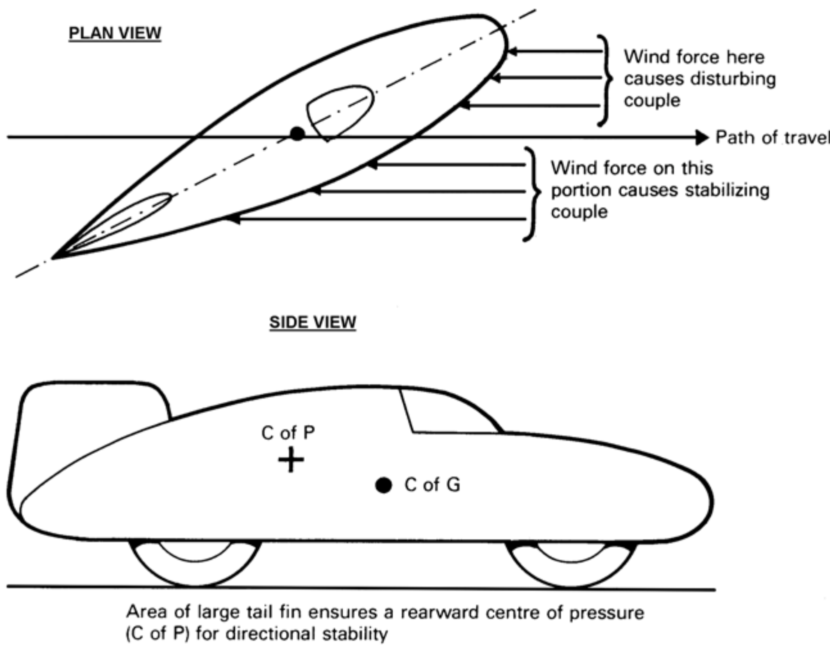



Fig 5.14 **This simplified diagram shows the basic idea behind the large tail fin such as sometimes used by record seeking machines, and how it moves the centre of pressure rearward to try and achieve automatic directional stability. The idea is that the rear stabilizing couple has to exceed the front disturbing couple. Fig. 5.13 gives a better idea of the actual airflow around a streamlined shape.**

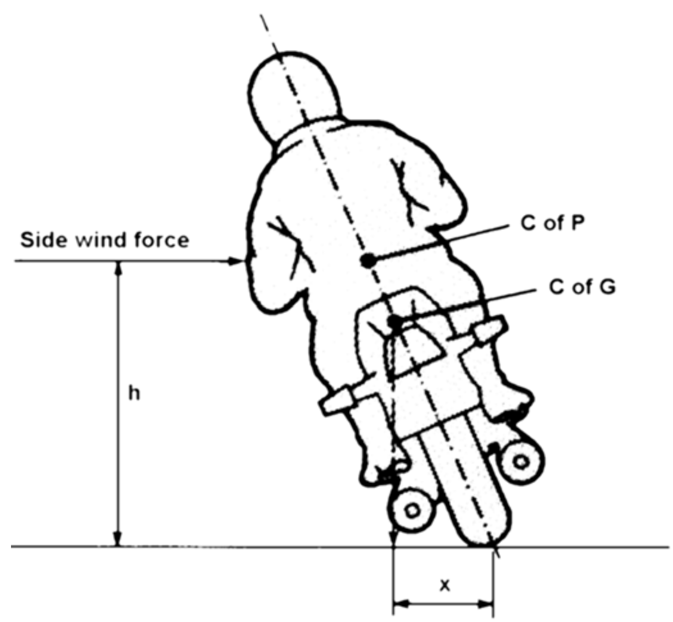



Fig 5.15 **The weight acting through the CoG at a distance x from the tyre contact produces a counter balancing torque to cancel that from the side force acting at height h. More weight and a high CoG reduce the lean angle needed to maintain balance.**

**The lean angle is defined by :**

**$\Theta$ = arctan( Fw.h / Wt.x ) where Fw is the wind force,**

**Wt is the weight and $\Theta$ is the lean angle.**

Let's now consider how some of the realities, of motorcycle steering, control the reactions to a side force, such as that produced by a side wind. (The author wishes to thank Douglas Milliken and Dr. Andreas Fuchs for providing the stimulus to look closer into these aspects.)

## Steady state directional stability.

Steady state means riding along with a uniform side wind, which thus creates a constant side force acting through the lateral centre of pressure. Clearly, this causes a roll couple, which must be balanced by leaning into the wind. To minimize the angle of lean required to balance a given side force, we need a low centre of pressure, a high centre of gravity and a large weight. Fig. 5.15.

To prevent the side wind from gradually blowing the bike off course, it must be steered into the wind to compensate. Fig. 5.16.

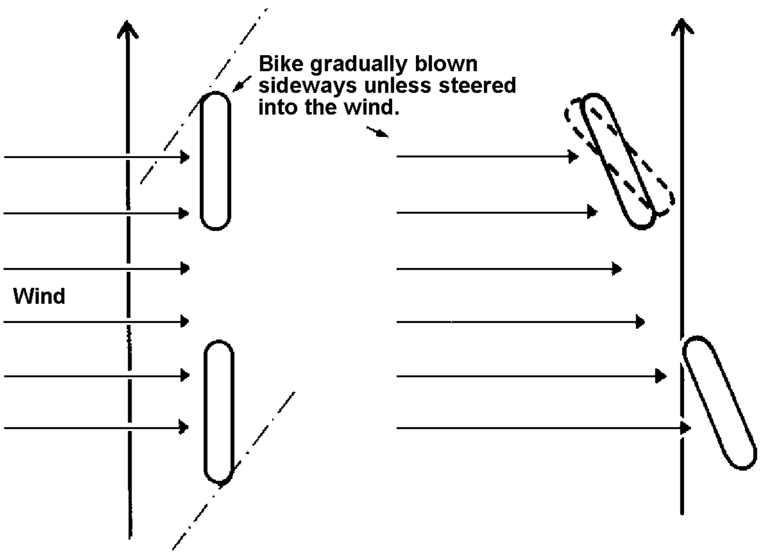



Fig 5.16 **Side wind effect in steady state. We need to steer into the wind to prevent being blown sideways. Due to the required lean angle these steering forces may come almost entirely from camber thrust. In all probability any yaw and steering angles will be small.**

Fig. 5.15 shows how we have to lean into the wind to maintain balance, but a few experiments with a pushbike, fig. 5.17 show some additional and very important steering effects.

The purpose of this experiment is to demonstrate that a side force, such as wind, can introduce steering tendencies and that their magnitude and direction depend on the point of application of the side force. This has important implications for the behaviour of a motorcycle subject to a side wind.

Our experiment shows that this ‘self steering’ is controlled more by the location of the side force than by the lean angle. Let’s look at this in a little more detail, figs 5.15 & 5.18 show how different CP heights change the balance of the forces acting normal to the bike centre plane. We can see that as the CP is lowered a smaller angle of lean is required and the normal component of the weight force is reduced, and the opposite occurs when the CP is raised. It is only when the CP is at the same height as the CoG that these normal forces balance each other.

**Fig. 5.17** *Attach a short bar (about 600mm. long) vertically and midway between the wheels as shown in the photo below, some plastic cable ties or thin wire are all that's needed to hold it in place. It is useful if this bar has a few holes in it.*

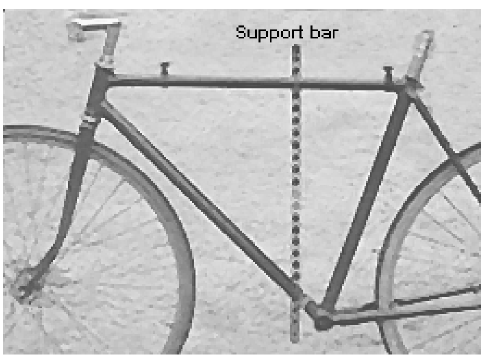
Now attach a length of thin rope or wire to the top of this bar, balance the bike upright and straighten the steering. Allow the bike to lean away from you and take the weight with the rope held horizontally.

Repeat this several times, but with the rope being attached to progressively lower positions on the vertical bar.

When the rope was tied near the top of the bar, you'll notice that the steering turns in toward the direction of lean. However, when the rope is at the lower end of the bar the steering is out away from the lean and as we might then expect there is an intermediate vertical support position that allows you to lean the bike without any tendency to steer one way or the other.

Now remount the bar as far forward as possible without interfering with the front wheel and repeat the experiment.

The main difference to note in this case is that the position for zero steer is higher up the bar than in the previous case. If you do this again with the bar mounted towards the rear you'll find that the rope location for zero steer is much lower.




**Left shows the bar mounted midway between the wheels, at right it is closer toward the front.**

**The other pics. show the three steering effects depending on the height of the horizontal force.**

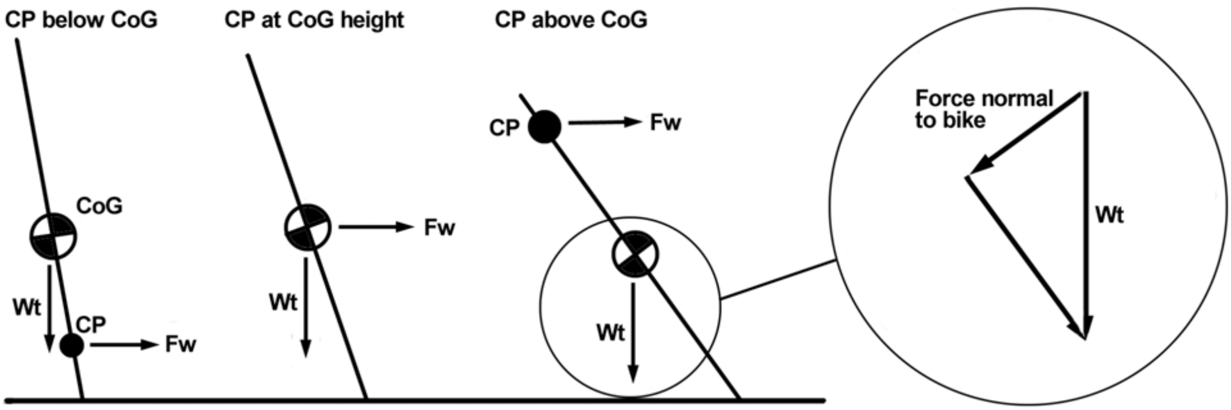




Fig. 5.18 **With a given side wind force (Fw) and weight (Wt) the angle of lean is less when the CP is below the CoG. As the lean angle increases the component of the weight acting at right angles to the bike also increases but the normal component of the wind force will decrease. When the CP and CoG are at the same height these normal forces balance.**

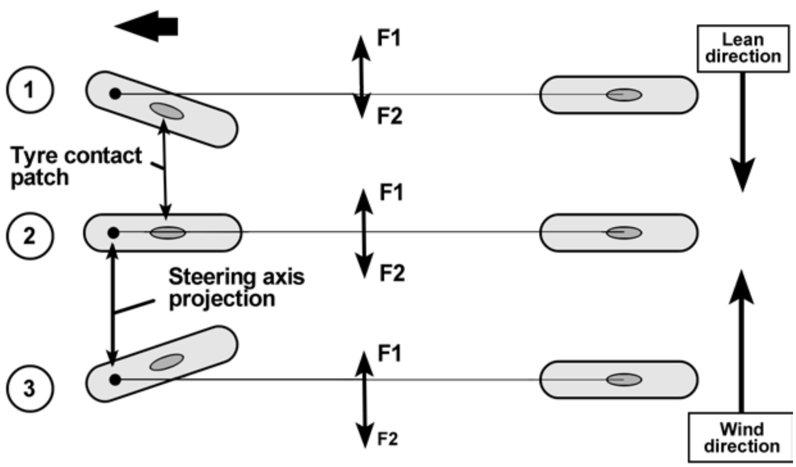



Fig. 5.19 **Looking down on the bike's centre plane we can see how unbalanced normal forces cause a steering action. F1 is the normal component of the wind force and F2 is the normal weight force.**

**The projection of the steering axis to the ground is forward of the front tyre contact patch (trail) and so any lateral imbalance in the forces will tend to twist the steering about the contact patch as shown. When F2 is greater than F1, i.e. the CP is higher than the CoG, as in (3) it will steer into the lean. With a low CP (1) the steering will be away from the lean. When the forces F1 & F2 are equal (2) there is no steering effect.**

Fig. 5.19 shows how any imbalance in these normal forces will cause a steering effect, this is completely in accord with the results of our experiment which showed that this effect varied with where we applied the horizontal force. Figs. 5.18 & 5.19 implicitly assume that the fore and aft location of the CP is the same as that of the CoG, but we also noticed with our tests that this longitudinal position has an effect on the CP height required to give zero steering action. Basically it is the yaw moments that we must balance not the lateral forces. Some simple calculations show that the point of application of a side force which gives no steering action must lie on a line drawn through the rear tyre contact patch and the CoG, as shown in fig. 5.20.

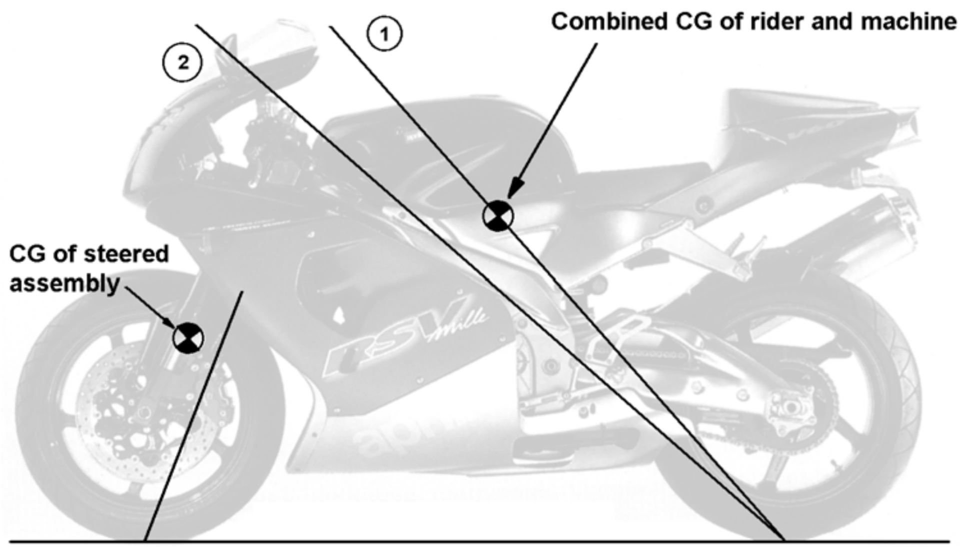



Fig 5.20 **Line (1) defines the range of positions of the lateral CP which give zero steering effect, as described above.**

**Line (2) takes into account an additional steering tendency caused by the offset of the steered weight from an axis through the front tyre contact patch.**

The previous explanation only accounted for one of two steering effects from lateral wind forces. The second arises from the steered assembly. The CoG of this assembly is usually offset forward from an axis through the tyre contact patch, we won't go into the details of this here except to say that the neutral steer line moves forward slightly as shown by the second line in fig. 5.20. Except where the front wheel is covered by a full fairing there will also be a separate CP of the steered components which will in most cases introduce further steering torques, these won't be considered in the following text to avoid unnecessary complication, but be aware of their existence.

So we can see that if the CP lies on this neutral steer or trim line the motorcycle can adopt a stable attitude without the application of any additional steering torque, it can be ridden *no-hands* in a steady side wind. It is automatically stable in other words. If the CP is elsewhere then the rider must apply a continuous steering torque to maintain the correct course. It is therefore obvious that the requirements for pure aerodynamic stability (rearward CP) are not quite the same as those to give overall bike stability. Whilst automatic neutral steering is assured with a CP anywhere along the trim line, all positions do not react the same. We saw that the lean angle is dependent on the relative heights of the CoG and CP and not on their longitudinal location. Therefore a high and forward CP may be stable but the lean angle will be much more than if the CP was lower and more rearward along the trim line.

As we saw earlier, the CP is invariably well toward the front, but fig. 5.19 shows that the trim line is likely to be excessively high at the front end. Stability would therefore be improved if we could raise the CP or lower that line, and to do that we must lower the CoG height and/or move the CoG forward. Lowering the CoG means that the lean angle must increase, which is undesirable, so we are left with moving the CoG forward. It is probably fair to say, that within the range of practical positions of the CP, steady side wind stability is enhanced by the most forward CoG position possible. The author's QL (shown earlier) had more bodywork high up toward the front than is usual and, although not tested, the CP was probably higher than normal, it was very easy to ride in windy conditions.

Under steady conditions, a lack of automatic stability is not too great a problem because the rider only has to apply a steady correcting steering torque to keep to his desired path. This raises the question of controllability as well as stability, the two may be quite different. Depending on whether the CP is behind or in front the trim line, the vehicle will naturally turn away from the wind or into the wind. Each requiring opposite control responses from the rider. If the bike steers away from the wind then the rider must steer back against the wind, and vice-versa. It would seem likely that only one of these required response actions would be intuitive. Fortunately perhaps, in practical cases the CP is forward of the trim line, and we've seen that this can cause the bike to steer back into the wind. A naturally more stabilizing reaction. As we shall see the issue of controllability is more important under dynamic conditions.

Stable or not, as suggested in fig. 5.16 the tyres have to produce sideways forces to oppose those of the wind, but as we've seen in the chapter on tyres, leaning produces camber thrust. i.e. a lateral force at the tyres which will act in the opposite direction to the wind force. Now if the camber thrust at each end of the machine, at the required angle of lean to maintain balance, is exactly equal to the wind force, at each end, then no additional steering is required to keep on track. In the chapter on cornering we saw that in most cases only a small steering angle, if any, was needed to make up for the shortfall or excess camber thrust. The same can apply to wind induced lean.

Assume that the camber stiffness of both tyres is the same, and that the total camber thrust, from back and front tyres, is equal to the wind force. Then if the CP is forward of the CoG, the component of the side force acting on the front tyre will exceed the front camber force and the front will need to be steered into the wind to create additional tyre force by means of a slip angle. In addition, the camber thrust at the rear will now exceed the rear part of the wind side force and so will cause the back end of the bike to yaw toward the wind. Surprisingly, this reaction is needed for the rear, as the bike yaws, a negative slip angle and steer force will build up to counter the excess camber thrust. So now we have a strange bike attitude, whereby the rear part of the bike is pointing with the wind and the front is steering into it. If the CP is behind the CoG the reverse attitude would have to be adopted. However, these yaw and steering angles are not likely to be large.

When the CP is on the trim line the front will automatically steer to the correct slip angle in a similar manner to that described above for the rear. The trim condition only balances the steering ***torques***, the steering ***angle*** is free to adjust according to the tyre forces, and as all applied forces must balance, the steering angle will automatically adjust to that which gives the right combination of camber thrust and steer force. The camber and steer stiffness of the tyres will therefore have a large effect on the yaw attitude and steering angle that the bike has to adopt in any given steady wind condition.

## Dynamic directional stability

The dynamic state refers to riding in gusty conditions, such as when passing gaps between hedges or buildings or when over-taking large trucks. Suddenly passing from still air into an area with a steady wind is also classed as a transient condition. When analysing the steady state case above we basically had only to establish a lean angle to counter the roll moment of the steady side force, and apply a steady steering torque to balance the lateral force according to tyre properties and the relationship of the lateral CP to the neutral trim line. In the dynamic situation there are many more factors to consider and the reactions are extremely complex and not yet fully understood, we can only briefly look at some of the main points here.

The response of a bike to gusting side winds will depend on several important features:

- Aerodynamic design, the magnitude and point of application of the varying lateral load.
- Frequency of the gusting.
- Relationship of the CoG and CP positions, this controls the basic aerodynamic stability as per the explanation of the traditional view, and also affects the transient roll and yaw moments.
- Relationship of the CP to the neutral trim line, this controls wind generated steering torques as per the explanation of static stability.
- Rider responses.
- Mass distribution and moments of inertia about the main axes.

There are other influences also, such as structural compliance of the machine and the tyre characteristics.

As pointed out in previous chapters, roll or leaning motions of a bike are intertwined with yaw and steering motions. Any quick roll motions produce steering effects (through trail and gyroscopic precession). Therefore, it is important to keep the transient banking change to a minimum, which needs a low centre of pressure and/or a high roll moment of inertia.

It is notoriously difficult to mentally visualize the behaviour of complex dynamic systems, without testing and/or mathematical analysis. A factor which currently limits the worth of such analysis is that in dynamic conditions the lateral wind force and CP location varies in very complex relationships. In the future as computer power takes a leap forward there is the possibility of combining CFD with dynamic simulations. The CFD would be used to calculate the aerodynamic forces acting on the machine in a particular attitude, this information would then need to be fed into the dynamic calculations, repeated at small time steps. Each time that there is a significant change in attitude the aerodynamic forces would have to be recalculated.

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To finish this topic I shall relate a couple of examples from my own experience. Neither case comes under the heading of a scientific test but are interesting and provide food for thought none the less.

I once converted an old Honda Gold Wing from telescopic forks to one of my double-link front ends, for its owner, Wayne Boys. After he had had it for a while we discussed the various effects that he noticed in normal riding, many observations were as I had expected but I was a bit surprised when he pointed out that it was far more stable in gusty side winds. At first I thought that this was probably just due to the greater lateral stiffness of the new suspension, which is generally more stable anyway. But a bit more thought as to the differences between the two set-ups, shed a bit of light on why they should behave so differently under these conditions. There seemed to be three main differences, in addition to the stiffness already mentioned:-

- 16" wheel instead of 19",
- Less trail,
- A 17 deg. rake angle which needed zero offset between the steering axis and the wheel centre-line to achieve the desired trail.

It can be argued that all three of these changes work toward improving the performance in gusts.

**Smaller wheel** The precessional forces will be reduced in line with the reduction in weight close to the wheel/tyre circumference. This will reduce the coupling between yaw and roll movements.

**Less trail** The component of the wind side force, acting at the front of the machine is passed to the road surface through the tyre via the steering axis, but the steering axis is in front of the tyre contact patch by the amount of the trail, and hence the sideways force on the bike will tend to steer the wheel. (See the bicycle experiment above.) This steering torque will be reduced with less trail.

**Zero offset between the steering axis and wheel axle.** With normal steering geometry with about 25 – 50 mm. of offset, the bulk of the wheel side area is forward of the steering axis, this combined with the effect of today's large tyres and discs means that a considerable steering force can be generated by the action of a side wind on the wheel. But with the zero offset geometry used on the double arrangement, this wind force is balanced about the steering axis and no turning effect is produced.

**This is the Honda Gold Wing mentioned in the above text. Note the small 16" wheel at the front. For a number of reasons this machine had improved stability in gusting wind conditions.** (photo: Kerry Dunlop)

An interesting feature that I noticed when riding my own QL., was the effect that tyre pressures had. The QL. has a streamlined body with a fairly large side area, and so the magnitude of the side wind forces may be expected to be high. Now, whilst the machine felt very stable and easy to ride in gusty conditions, it would squirm about alarming if the tyre pressures were allowed to drop. This happened once due to a slow leak at the valve on the front wheel. It appears that the reduced lateral stiffness of the tyres when under-inflated, allowed the machine to move about excessively under the action of the high lateral wind force. A similar machine without the bodywork, was generally less stable in wind but proved to be much less sensitive to tyre pressures in similar conditions.

## Summary

Aerodynamic design of motorcycles is more than just a matter of producing a low drag, low lift body with a CP. behind the CoG. Stability is harder to achieve with well streamlined low drag bodies, this is due both to the greater side area present with such fairings and to more efficient production of "sideways lift" due to the angle between the airflow direction and the direction of travel. So ideally we want a combination of sometimes conflicting requirements:--

- Minimal drag for performance and fuel economy.
- Low frontal CP. to reduce drag induced load transfer.
- Low and rearward lateral CP. to reduce the unbalancing moments, and give directional stability.
- A shape and value of side area that minimises the side force produced.
- A high CoG. combined with a large weight and high roll moment of inertia, to minimise the effect of whatever side forces are generated.

# 6 Suspension principles

The primary function of the suspension system is to insulate both the rider and the bulk of the machine from road shocks – the first for his comfort, the second for mechanical reliability and longevity. In doing this, it is vitally important to keep the wheels in the closest possible contact with the ground for maximum control and roadholding.

Motorcycles present a huge challenge to suspension designers because of the vast range of conflicting demands brought about by the vehicle layout and the need to lean for cornering. The low wheelbase to CoG height ratio gives rise to large longitudinal load transfer under braking and acceleration. Under braking the front suspension may be called on to support 100% of the machine's weight, add this to braking forces and the static suspension load will be nearly tripled. Whilst doing this it will also have to retain its ability to absorb road shocks. Acceleration on a modern superbike or racer is quite capable of causing the reverse problem of transferring all of the load onto the rear. Cornering can load both ends by an extra factor of about 50% with the cornering capabilities of modern tyres. Off road bikes are also called upon to withstand the additional loads of landing after quite high jumps, and this must be done within a movement range of around 300 mm. A high degree of sophistication is necessary in the damper design to provide the suspension performance that we expect and enjoy today. The wide range of conditions to be catered for means that there is no one optimum design. If there were then we would not hear so much about top racers struggling to get their settings correct for a given circuit. Even in racing, compromise is inevitable.

There are four main parameters that affect suspension performance:

- Springing.
- Damping.
- Sprung and unsprung masses.
- Tyre characteristics.

We will consider each in turn and then see how they combine into a complete system.

## Springs

For our purposes the most important characteristic of a spring is its "rate". This is a measure of its stiffness and is determined by measuring the extra force needed to compress (or extend in some cases) the spring by a given small amount. This can be expressed in N/mm (kN/m gives the same numerical values) and in the imperial system of measurement is usually expressed in terms lbf/inch. So a spring with a 10 N/mm. rate will need an additional force of 100 N to compress it by a further 10 mm. In some cases this rate does not vary throughout the useful range of movement of the spring, and is termed linear. On the other hand some types of spring exhibit a different rate in various parts of the range of movement, this is often known as a progressive rate spring and in motorcycle use this progression is usually positive, i.e. the rate increases with added load. It is very important to understand the distinction between rate and load. Load is the total force being supported by the spring, whereas rate is the ADDITIONAL force needed to compress the spring by an extra given amount. Springs can take many forms and be made from many different materials, but the practical range is more limited.

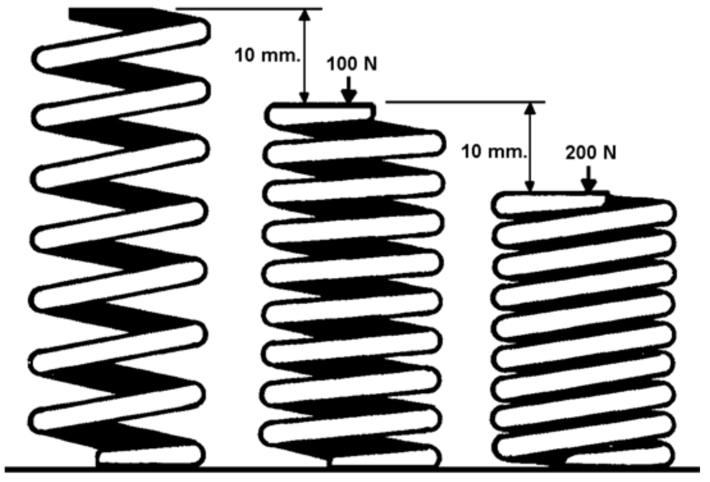
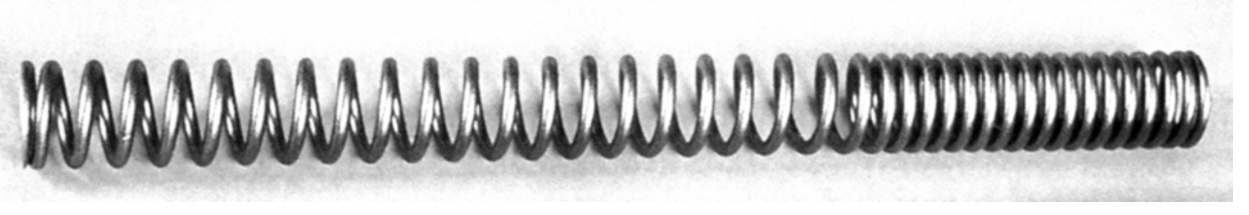



Fig 6.1 **Showing the meaning of spring rate. The spring shown has a constant rate of 10 N/mm.. Each additional 10 mm. of movement requires an additional 100 N of axial load. This doesn't apply when the spring becomes coil bound, then it effectively becomes solid.**

## Coil Springs

Coil springs in steel are the most common by a long way. They may be evenly wound (constant pitch) to give a linear rate, or they may be wound with a varying pitch to give a progressive rate. In this case as the spring is gradually compressed, the closer wound coils become "coil-bound" (i.e. touch each other and act as a solid mass) and so the rate rises.



**Typical dual rate front fork spring. There are two distinct pitches, as load is applied the closer wound coils on the right will become coil bound and act as a solid spacer, hence leaving less coils to the left to deform under load. That is, as the load increases the spring rate also increases**

At one time, due to manufacturing costs, it was quite common to have dual or triple rate springs made by stacking two or three springs together, rather than winding a spring with varying pitch along its length.

There are two ways that this can produce the desired result:

Use springs with the same rate but wound with different pitches, so that the one with the closest pitch becomes coil bound first.

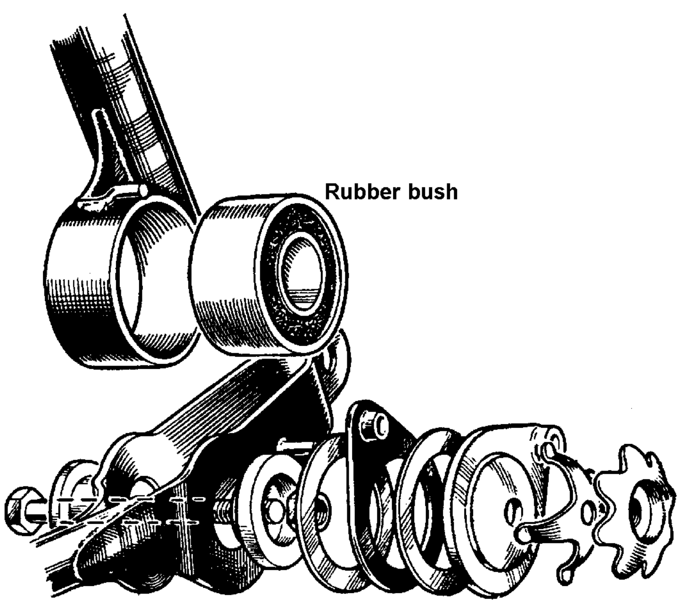
Use springs with different rates, then the softest will become coil bound before the others.

As an alterative to steel, titanium is a very attractive spring material. It is twice as flexible as steel and little more than half the weight, size for size. Theoretically, this should lead to a spring of one quarter the weight of a similarly stressed steel one. In practice, the need to have closed coils at the spring ends to seat properly, means that this saving is not fully available and an actual titanium spring will be closer to 1/2 of the weight of the steel one. There is really only one disadvantage with this material that prevents more wide spread use, and that is one of cost, confining its use mainly to exotic works racers.

Fig 6.2 **Dual rate spring by stacking two springs, the right hand side spring is wound with a closer pitch and so it will become coil bound before the other. It is usual to separate the springs with some form of spacer, often made in aluminium or a hard plastic.**

## Rubber

Rubber has many properties that make it an interesting spring material, and it can and has been used in a variety ways. It is not suitable as a direct replacement material to make a coil spring. It must be used in an appropriate design. Greeves used it in the form of large bonded bushes that served as both the pivot bearings and the springing medium in their leading link front forks. The rubber itself was loaded in shear, though the bush as a whole was loaded in torsion.




**Rubber springing. Exploded view of early Greeves front fork pivot showing bonded rubber bush. Although the bush was loaded in torsion the rubber itself was stressed in shear.**

**In this example damping was by means of a concentric and adjustable friction damper. Later versions used linear hydraulic dampers mounted inside of the fork stanchions.**

In some Hagon telescopic forks, designed for grass track racing, rubber bands provided the springing medium. This provided easy adjustment by adding or removing bands as required. Rubber has a natural progressive rate, and it also provides a small amount of inherent damping, though this generates heat which may become a problem with very hard use in rough terrain. It is a versatile material and its characteristics can be tailored to suit different requirements by varying the composition and/or the mechanical design. Although it would be less adaptable for individual suspension tuning, than coil springs which can easily be obtained in a wide range of rates and lengths, on a mass production basis, rubber springing lends itself to low unit costs. The ubiquitous Austin Mini was suspended thus, and adequately demonstrated the potential, this material merits consideration in any new design, and may be overdue for a revival on low cost machines with the current interest in progressive rate suspension.

## Gas springing

To complete our survey of spring materials we must consider air or gas, which automatically provides a progressive rate. This can easily be demonstrated with a bicycle tyre pump. First extend the pump then cover the outlet with your finger to seal it, and compress the pump. You will find the initial movement generates little resistance but as the movement increases the required force increases very rapidly. The load supported by a pneumatic unit depends on its internal pressure, which in turn depends on the initial static pressure and internal volume. This pressure is inversely proportional to the volume. i.e. If the internal volume is halved then the pressure doubles and the unit will be supporting twice the initial load. This relationship between pressure and volume is known as "BOYLE'S LAW".

The extent of the progression in rate is determined by the compression ratio of the unit (i.e. the ratio of the gas volume at the two extremes of travel). Fig 6.3 shows this variation between two units that start off by supporting the same load.

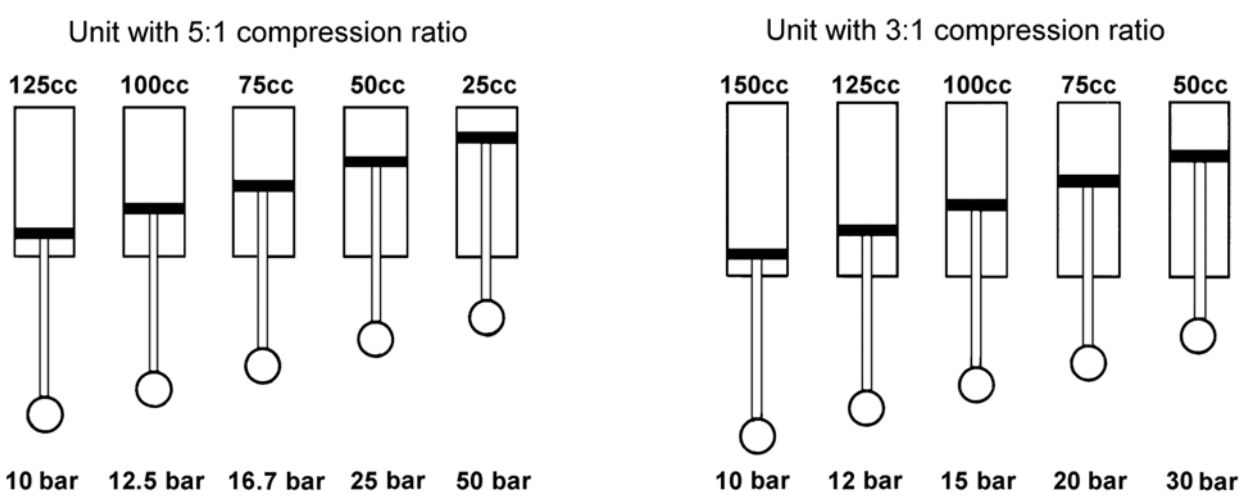
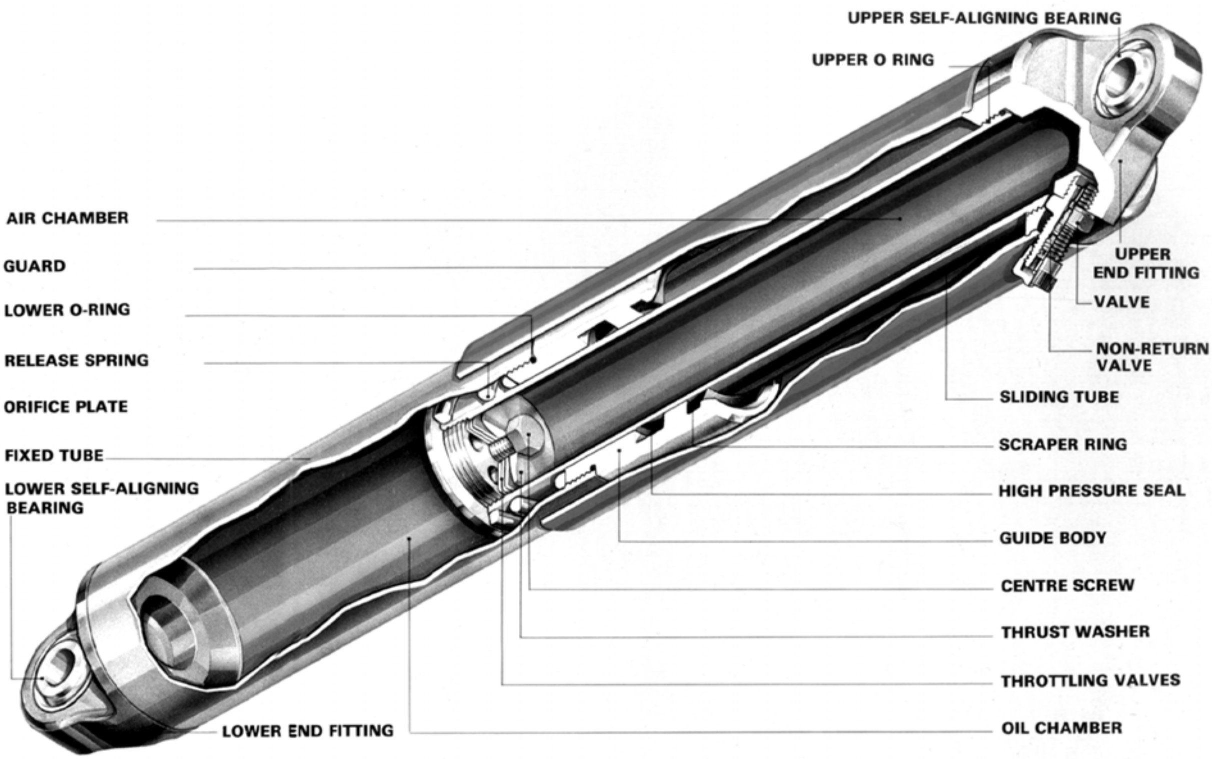



Fig. 6.3 **At full extension and with 10 bar pressure in each, the two units can support the same load. However at full displacement the one with a compression ratio of 5:1 now has a pressure of 50 bar and can support more load than the other which has only risen to 30 bar within the same suspension stroke. The only difference between these otherwise identical struts is that the lower compression one has an additional 25 cc. of internal volume.**

Apart from this progressive rate, air shocks have the advantage of easy adjustment to compensate for different loads on the bike. If a passenger and luggage for a trip, doubles the load on the back end, then just double the initial gas pressure. Then at every point in the suspension travel the load supported will double and the rate will be double. This gives perfect spring compensation for the increase in load. Of course to achieve perfect compensation we would also need to double the damping.

The preload/ride-height adjustment found on normal coil spring units, does not have any effect on the spring rate, only on the initial load capacity. Such suspension will show an increased tendency to bottom out when heavily laden, unlike the adjusted gas shock.




**Fournales gas suspension strut. Despite the apparent simplicity of construction, shocks like this must be made to very high standards in order to work at all.**

However, despite this adjustability the pneumatic unit can be at a severe disadvantage when it comes to tuning the unit for a particular application or to suit a particular rider's needs. For example, if the rider determines that for his use a softer spring rate would be desirable, then with a coil-over shock he need only obtain and fit an appropriate softer spring. On the other hand, the man with the gas unit is in a bit of trouble. His first thought may be to simply release some gas. But, as a given pressure is needed in the unit to support the static weight of the machine, all that happens is that the ride height is reduced to a level which compresses the gas back up to the required pressure. However, at this new ride height the volume in the unit will have been reduced and hence the spring rate at this position will in fact be higher, so not only has the ride height and hence available wheel travel been reduced but the rate has been increased too. If he took the opposite approach and added some gas, then the rate might well be decreased as desired but ride height will be increased. If too much gas is added then the preload may become so high that a bump is needed to even begin to compress the suspension, so negating the desired effect. Other than buying another unit with different characteristics, there is little that can be done. Some units allow for the addition or removal of small quantities of oil, this alters the internal volume and hence the effective spring rate, but the degree of progression will be changed also, because the compression ratio will have been altered.

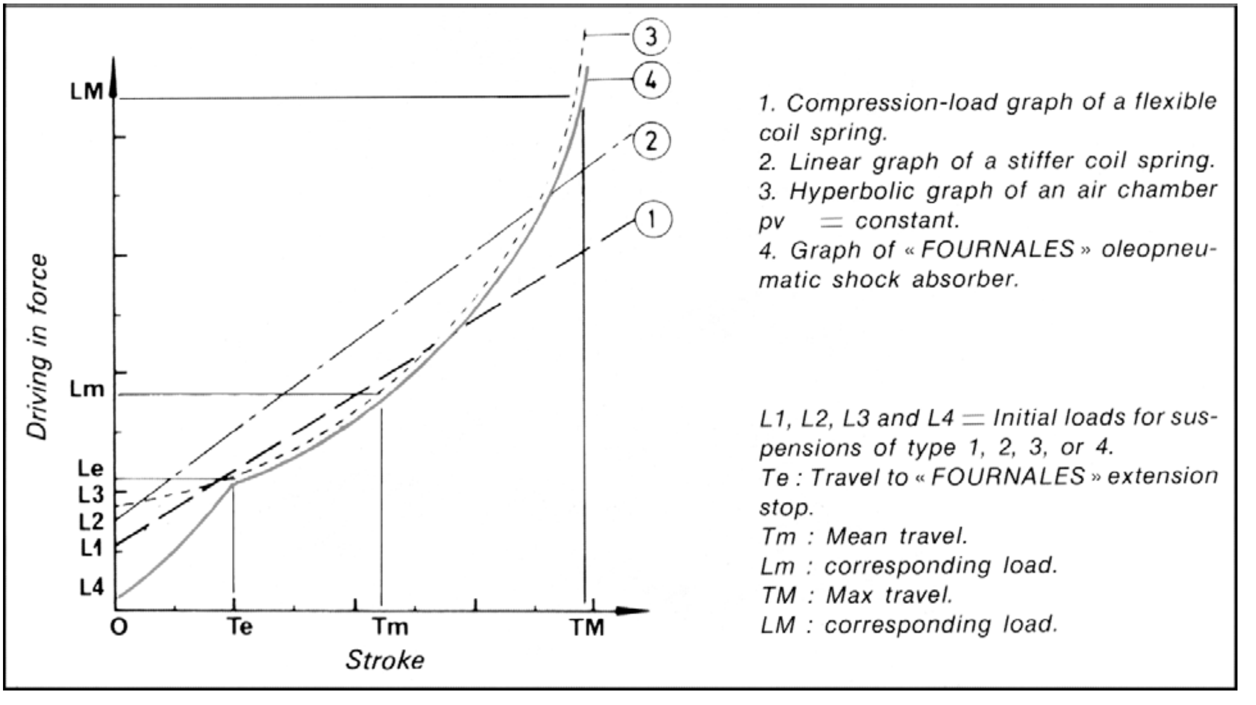



Fig. 6.4 **Load vs. displacement curves for a typical Fournales gas spring unit. The "natural" air spring graph (line 3) has been modified up to a stroke of Te by the addition of a "top-out" spring as shown by line 4.**

If a suitable setting cannot be obtained, then the only option is to get out the welding torch and change the leverage ratio. (This is not applicable to telescopic front forks, but these are not usually supplied as totally pneumatic.) For example, suppose that we move the suspension mount on the swing arm from the wheel spindle area to half way along, assuming that the frame mounting was moved also to compensate, the leverage on the suspension unit will now have been doubled. i.e. the static load on the unit needed to just support the weight of the bike will be twice the previous value. To achieve this the pressure in the unit will also need to be doubled, which in turn will increase the rate by a factor of two. The change of leverage will also have the effect of halving the movement of the unit compared to the wheel displacement. Now, the reduced movement and the increased rate of the unit combine to give the effect of halving the wheel rate, and the degree of progression will remain the same in terms of unit movement. But as the wheel will now move through twice the range as before we may be in trouble with not being able to accommodate this increase. Another problem with this approach is that although the increased pressure in the unit increased its spring rate it did nothing to increase the damping rate, which as a result will now be too small, assuming that it was OK. previously.

In my experience, most complaints levelled at after-market gas shocks are due to the rider's expectations of the adjustability benefits being raised excessively by the manufacturer's advertising hype. Once a suitable unit is matched to a particular application, then the ability to perfectly compensate for load differences by changing the pressure is a valuable benefit, although, unless the unit also has adjustable damping, then that cannot be optimum throughout the full range of loading conditions. Do not expect to match any old gas shock to your bike just by changing the pressure, it doesn't work like that.

To provide easy load compensation for large touring machines the American firm, S&W, used to offer air shocks with an on-board air-compressor. It was a simple matter of pumping the bike back up to the desired ride height after the addition of a passenger or heavy luggage.

## Other

Leaf springs and torsion bars have been tried occasionally (usually in steel), but have been superseded for various reasons, although in some applications, torsion bars may save valuable space. It is worth noting that a coil spring is really only a torsion bar wound into a helix, though that subjects it to bending stresses and stress concentration that are absent in a straight torsion bar, hence for a given load capacity the coil spring will be subject to higher stress or will have to be heavier to reduce the stress.

Titanium could be used for torsion bars, where its lower modulus of elasticity would result in a more compact installation.

Leaf springs have recently made a comeback in some sports cars, though not in metal. Modern composite materials have opened up new possibilities for this type of spring, although consistently producing springs with the same rate may raise quality control issues.

Yamaha and Ohlins patented an idea for a fibre-glass spring fitted inside the swinging arm itself. This is quite a clever design in that a wide range of characteristics can be provided by simple adjustment. Fig. 6.5 shows the layout.

- The initial preload adjuster determines the starting position of the spring on the especially shaped support cam, and so has some control over the initial spring rate.
- The support structure for the actuating roller can be adjusted back and forth to control the overall spring rate.
- The main preload adjuster controls the static ride height as on a conventional machine.
- The shape of the support cam controls the free length of the spring as the suspension compresses, which in turn controls the progressiveness of the suspension rate.

Thus the effect of different degrees of progression can be provided by simply changing the shape of the support cam. This done through the rear of the swing-arm. For those machines where this adjustment is not considered necessary the lower surface of the swing-arm could be shaped to remove the necessity of the separate piece.

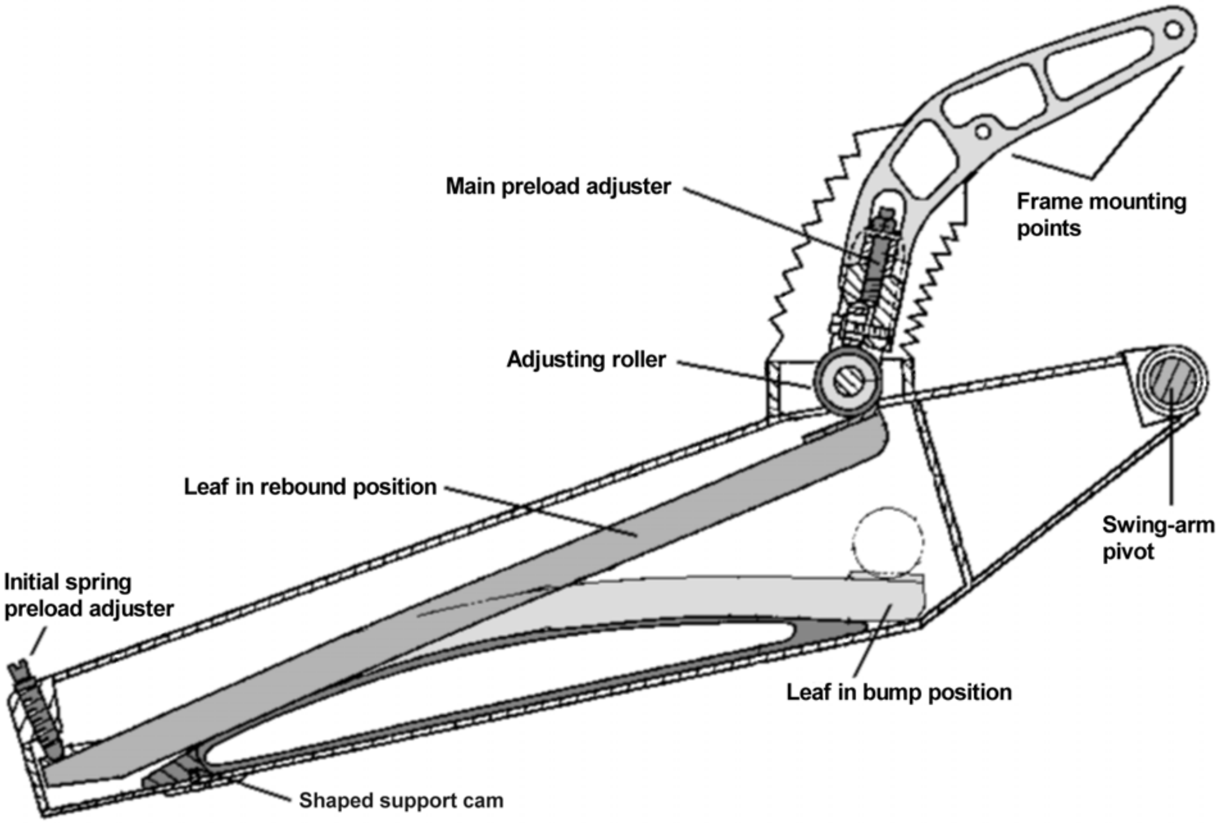



Fig. 6.5 **From a 1996 patent by Ohlins and Yamaha this sketch shows an idea for a composite (fibre-glass) leaf spring fitted inside the actual swing-arm. As the swing-arm rises the actuating roller forces the leaf to bend over the support cam. The detail of the shape of the support cam surface controls the active length of the spring and hence the overall spring rate throughout the range of movement.**

## Damping

A damper is simply an energy absorber. Movement of the damper causes mechanical resistance, but unlike a spring it has no natural tendency to return to its starting position, and hence uses up work or energy, which is ultimately dissipated as heat. This energy loss is necessary to prevent uncontrolled oscillations in the suspension. Imagine that a large bump has fully compressed a suspension strut, at that instant, energy is stored in the spring as potential energy. As the spring returns to its static length it gives up this energy, which if there were no damping, would be transferred entirely to the mass of the bike in the form of kinetic energy (energy of motion). This would cause the suspension to extend well beyond its normal position. This will have transferred the kinetic energy back into stored energy in the spring, which will then repeat the whole process again in the opposite direction. Thus after any disturbance, we would proceed down the road bouncing as if on a pogo stick.

The introduction of damping will absorb some or all of the energy imparted to the suspension by the bump and hence the oscillation will be reduced or eliminated, depending on the degree of damping. As the energy absorbed by the damper is changed into heat, hard worked suspensions, such as those on a motocross machine, will sometimes overheat. In any event the heating will change the damping characteristics of the oil and in some high quality units there are various measures to compensate for this.

As an energy absorber, any damper should be matched to the amount of energy to be dissipated. This depends on the springing, the bike's mass and the type of use to be expected from the machine. Before the introduction of hydraulics, dampers were of the friction variety and their characteristics were precisely the opposite of those required. The STatic frICTIONal force (often called STICTION) was high, but once the damper moved the friction dropped somewhat, but since a damper can only absorb energy when moving, obtaining adequate damping involved excessively high stiction, hence a large force was necessary to start the suspension moving. This meant that the ride was harsh and insensitive to small bumps. Any form of stiction is detrimental to suspension performance. In contrast, hydraulic dampers begin to move with a minimum of force, so providing little resistance at low rates of motion, while high damping forces are available as the speed of the damper rises. The seals and rod support bushes introduce some stiction and the manufacturers reduce this as much as possible, sometimes seals are made of PTFE., a low friction non-stick material.

In its simplest form an hydraulic damper comprises a piston forced to move in a cylinder of oil. There may be holes in the piston to permit oil flow from one side of the piston to the other. In a normal hydraulic unit there are two types of damping present -- viscous and hydrodynamic. Viscous damping arises from the shearing action of the fluid and the force produced is proportional to the speed of damper movement. Hydrodynamic damping is proportional to the square of the damper velocity (and so is often called quadratic damping), and is due to the mass transfer of fluid within the strut causing turbulence. This is the characteristic obtained by forcing the fluid rapidly through an orifice.

Viscous damping is a mathematical nicety and can be matched precisely to a single rate spring to give "critical" damping (or a desired percentage of critical) over a range of operating conditions. Critical damping is that amount which just prevents any oscillation or overshoot after hitting a bump. It would therefore seem sensible to design a damper as a viscous one in the first place, however, simple as this would be in an electrical context, the realities of practical hydraulics make this none too easy. Thick fluid would need to be sheared between moving plates or cylinders, and the damping would be very dependent on the fluid viscosity. Whilst the equivalent of viscous damping is easy to achieve in electrical applications it is not so readily available in mechanical ones. Some attempts at producing direct viscous dampers have been made which force oil through long thin tubes, but oil viscosity is very temperature sensitive and so too would be any such damper.

In practice dampers are made to force oil through small orifices. Without special design features, quadratic damping would be by far the highest proportion present in a normal damper. Temperature sensitivity is much reduced and is due to the change in fluid density not viscosity. However, this type of damper in unmodified form can give undesirable effects. Because of the "squared" effect the damping forces rise very rapidly with damper speed, and give little resistance at low speed. This means, that at low road speeds damping may be inadequate over small bumps, but grossly excessive on larger disturbances at high road speed. In order to make a satisfactory damper the manufacturers must modify this basic idea. To reduce the excess high speed force, the orifices need to be enlarged and then controlled by a blow-off or flow control valve which opens only at higher speed. This valve prevents the mid range damping from being reduced too much by the enlarged orifices, but will probably raise the low speed to excess. This can be overcome by introducing parallel bleed holes not controlled by the flow control valve. By juggling around with these techniques the manufacturer can straighten out the damping curve and match it to the application. Basically the response is generally brought closer to the nature of viscous damping.

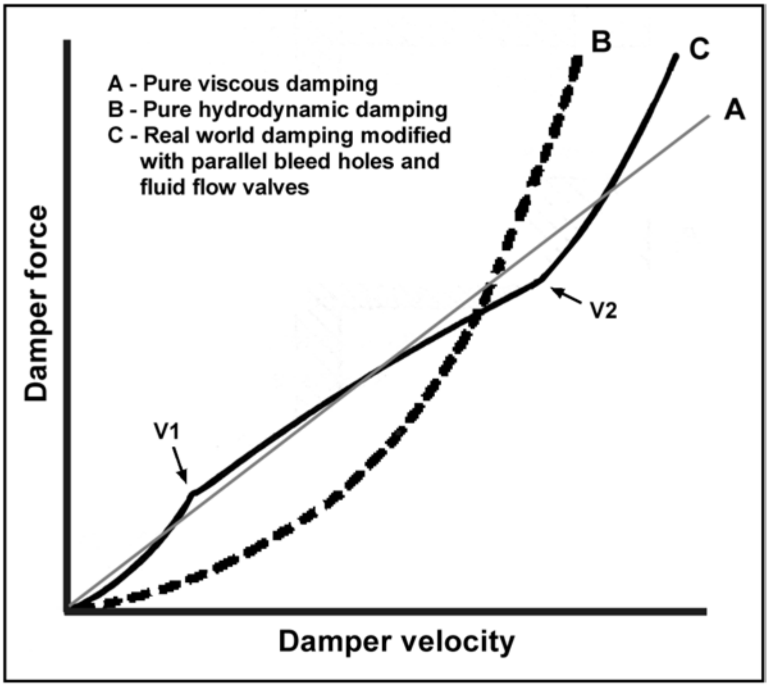



Fig. 6.6 **Characteristics of various damping types.**

**Curve "A" shows the characteristic of viscous damping, which is directly proportional to damper velocity.**

**"B" shows unmodified hydrodynamic damping which rises with the square of velocity. Note the very low damping at the low end combined with excessive resistance at high speed.**

**"C" shows how the damping curve might be modified with various tricks of the damper maker's art to approximate curve "A". This curve is just representative and the exact detail will vary depending on the characteristics desired. Up to the velocity V1 the flow valves are shut and the damping force is due to flow through parallel bleed holes. The region between V1 and V2 is controlled by the flow valves, and after V2 these valves are fully open and so the shape is once more similar to pure hydrodynamic.**

These flow control valves are critical to the design of the modern damper and several types are in use. Some are simple ball valves, others use a small spool valve which opens a progressively larger hole as the flow rate increases. Both these types may be spring loaded and can suffer from wear and lag, when the damper and hence fluid flow reverses direction the inertia of these parts causes a certain time lag and valve flutter. Most modern high performance dampers use what is called a shim stack to control the flow through the orifices. These shims act like a flat spring blocking the flow passages, but as the damper velocity increases the higher fluid pressure forces the shim to open away from the hole allowing greater fluid flow. The force/velocity relationship of the damper can be fine tuned by the shape and stiffness characteristics of these shims which may be stacked to achieve the required results. Fig. 6.7 shows how this might be done in practice.

The low speed damping (up to **V1** in fig.6.6) is controlled by the flow characteristics of the bypass circuit. It is usual to separate the bump and rebound flow channels, and on fully adjustable units separate adjustment by needle valves is provided to give independent control of each. As the damper velocity rises the pressure drop across the two sides of the piston increases to a level sufficient to force the shims to bend away from the piston allowing additional flow. The shims open farther at higher damper velocities, thus regulating the damping force between **V1** and **V2** in fig. 6.6. The details of the shim stack can be varied to provide a wide range of damper characteristics. The thickness, diameter and number of shims will control the slope of the force/velocity curve, but these shims can be mounted to have some amount of preload. This preload will exert a large influence over the velocity at which the shim stack will start to open. Adjustments to the shim stack normally require the unit to be dismantled, but some sophisticated versions provide for preload adjustment of a secondary stack in the remote reservoir.

At a certain point the shim stack will reach its maximum opening area and the damping will become dependent on the velocity squared again (**V2** in fig. 6.6). The flow rate through the shims may also be limited by the serial flow passages leading to the stack with similar, but not identical, results. Sometimes an adjustable valve is put in series with the shim stack to control high speed damping.

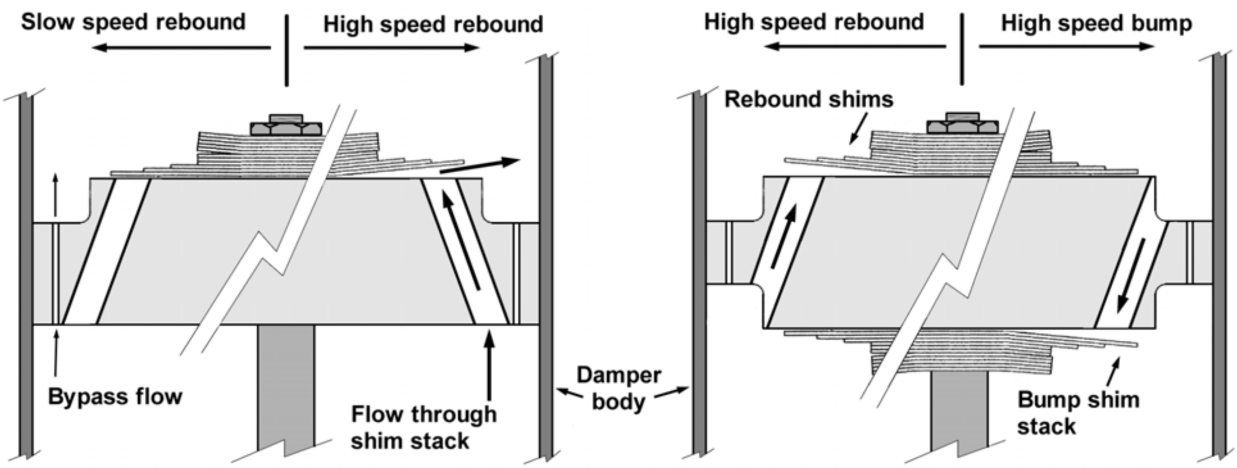



Fig. 6.7 **Diagrammatic representation of a damper piston with shim stack for the control of the mid to high speed damping. The left hand sketch is simplified to show only the rebound shim stack. The figure is split in two to show the shim positions for the low and high speed cases. At low damper rod velocities the main flow is forced through the bypass passages, but as the rod velocity increases the pressure builds up enough to open the shim flow valves preventing excessive damper forces. The sketch to the right shows how two shim stacks can be used on the same damper piston to control both bump and rebound damping.**

Many modern units based on the De Carbon principle, have some secondary valving, known as foot valves, in the fixed housing of the damper. Those with external reservoirs usually have the foot valves there. The advantage of such additional valving is that it is mechanically simpler to provide for adjustment mechanisms. Any adjustments to valving on the piston must be done by passing a rod down through the main damper rod. The disadvantage of having the adjustments on the foot valves is that only a relatively small oil flow passes these valves, the oil volume is that displaced by the damper rod as it enters the damper body. Thus adjustments made here have only a small range but nevertheless can provide a useful fine tuning aid.

Compensation for temperature variation is another challenge faced by damper designers. The damper's sole purpose in life is to absorb kinetic energy and dispose of it as heat, and so it is hardly surprising that under severe conditions high temperatures are common. Heat affects damping in two ways, fluid viscosity changes greatly, becoming thinner as the temperature rises. Fortunately, only a small part of the damping force is from viscous effects. Heat also changes the density of the fluid due to expansion, but the effect is much smaller than the viscosity change. The density affects the mass flow through the various channels and valves and so changes the damping force.

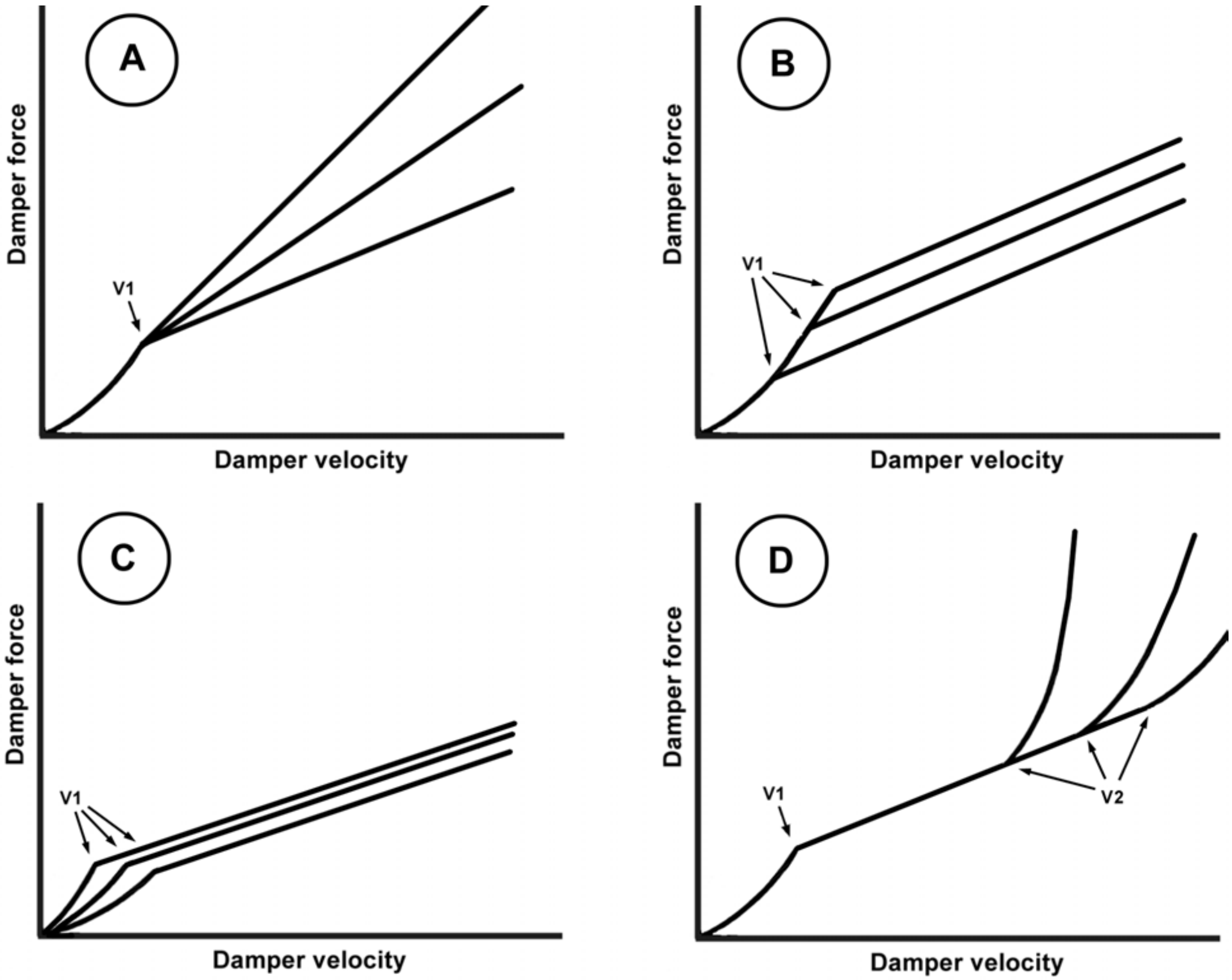



Fig 6.8 **The effects of different adjustment possibilities on the shape of the damping characteristics.**

**(A) Shim stack stiffness. The slope of the secondary stage is increased with greater shim stiffness.**

**(B) Shim stack preload. Greater preload delays the opening of the shims and so prolongs the use of the bypass flow, the secondary slope remains similar because this is controlled by shim stiffness.**

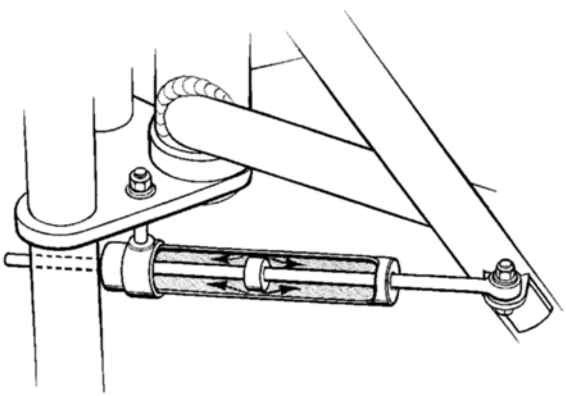
**(C) Parallel bypass. This is the principal slow speed adjustment but it also has a cross-talk effect on the mid and high speed range also.**

**(D) High speed serial flow. The maximum flow area and shim opening control both the location of V2 and the slope of the high speed part of the damping curve.**

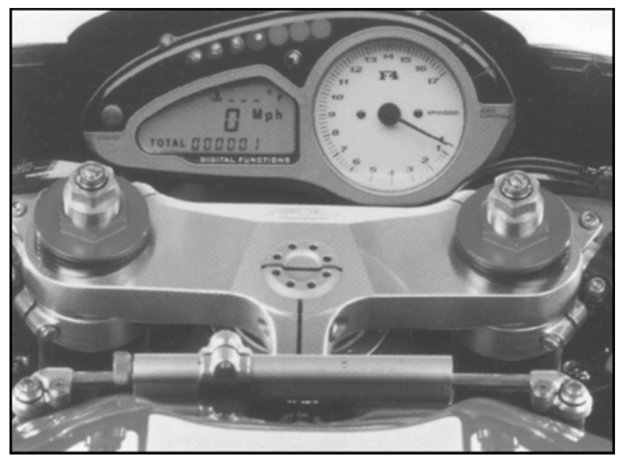
Rapid damper movements can lead to cavitation in the fluid. Cavitation is akin to local boiling or vaporization. The boiling point of a liquid depends on both its temperature and also its pressure. As the damper moves, one side of the piston is subject to an increased pressure but the other side feels a reduced pressure. At high damper velocities this pressure reduction may be enough for the damper fluid to boil or vaporize, the fluid now becomes a mixture of vapour and liquid with different properties to that of the liquid alone. The obvious answer to the cavitation problem is to increase the static pressure inside the unit so that the reduced pressure on one side of the piston is always above that needed to remain in a full liquid state. This is a principal reason for air or nitrogen pressurization common on high quality dampers.

As if the problems of obtaining a satisfactory damping curve were not enough the designer also has the problem of rod displacement to contend with. In other words, as the damper is compressed the volume of the rod entering the cylinder reduces the space available for the fluid. Hence, a compressible medium such as air, must be introduced to compensate, but as the damper is shaken on bumps the air and oil mix, drastically reducing the damping forces.

Many ingenious solutions to this problem have been tried over the years. One simple way to bypass the difficulty is to extend the rod through both ends of the cylinder so that there is no change in internal volume with piston movement. This does nothing to compensate for volume alterations due to temperature change, or just plain old fashioned fluid leakage, and this method is not currently used for suspension damping but is almost universally used for steering dampers.



**A hydraulic steering damper, with the piston rod extended to obviate the displacement problem. A typical installation.**



**The MV Agusta F4 mounts the steering damper across the frame just behind the top fork yoke. Unusually the damper rod is mounted to the bike at both ends and the fork clamp causes the cylinder to move sideways through a ball jointed linkage.**

**Note also the suspension adjusters at the top of the fork tubes.**

Many years ago, Girling adopted a clever solution on their twin tube strut. In this the free air was replaced with a sealed nylon bag containing freon gas (under the trade name of Arcton). This particular gas was used because it is composed of large molecules and they did not permeate through the bag. Production of this unit was short lived because of the production difficulties, although the idea has been used by other manufacturers. A later Girling solution was to revert to free gas but to substitute nitrogen at 100 psi. for air. An emulsion was formed between the oil and gas, the resultant fluid giving nearly constant damping characteristics.

Current design trends have been influenced by the DeCARBON principal, where the oil and gas are separated by a floating piston, either in the unit body or in a separate remote chamber. The gas pressure (up to 300 psi.) keeps all the seals pressurized to reduce leakage and to stabilize the damping and prevent cavitation, however rapid the recoil stroke. On some dampers of this type there is provision for adjusting the pressure as a fine tuning aid.

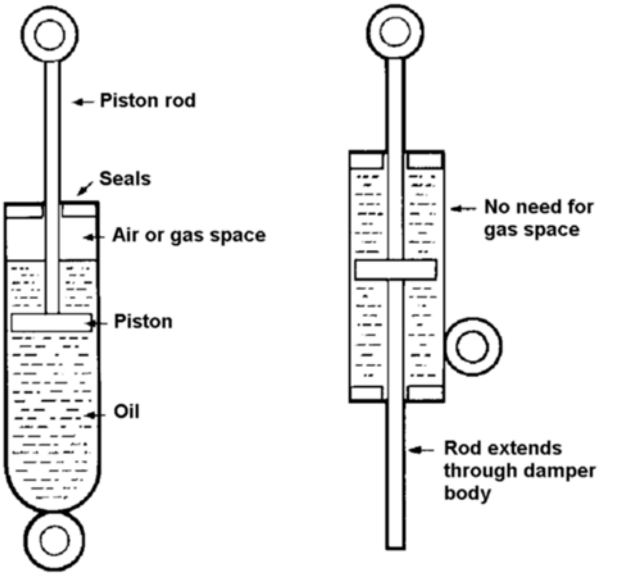
Fig 6.9 **Various possibilities exist for dealing with the rod displacement problem and general expansion. Some are shown here.**

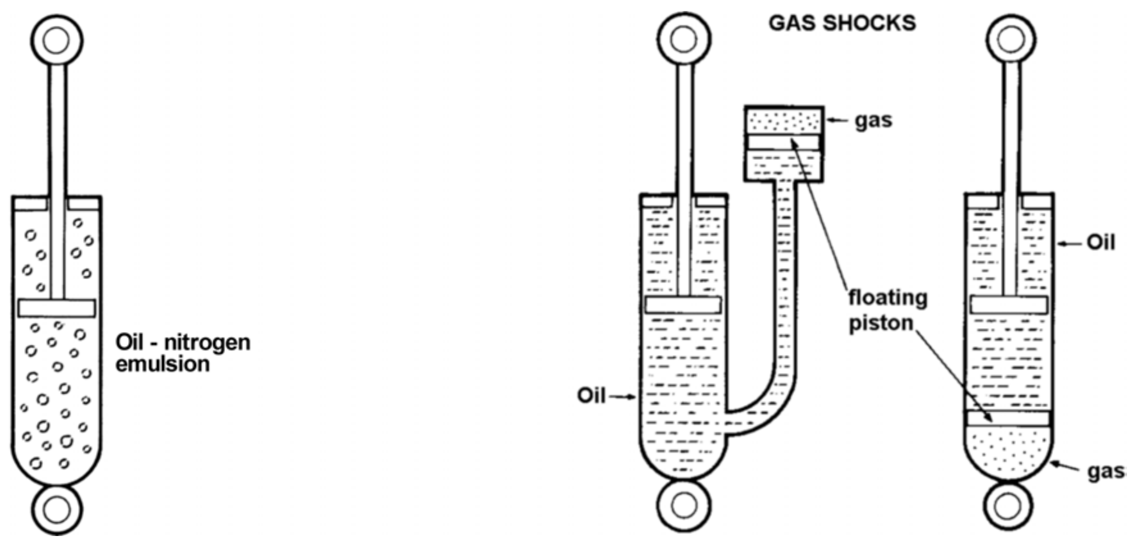
***Right,*** **a simple two-way hydraulic suspension damper, in which some air or gas space (compressible) is necessary to allow for the piston rod volume entering and leaving the cylinder.**

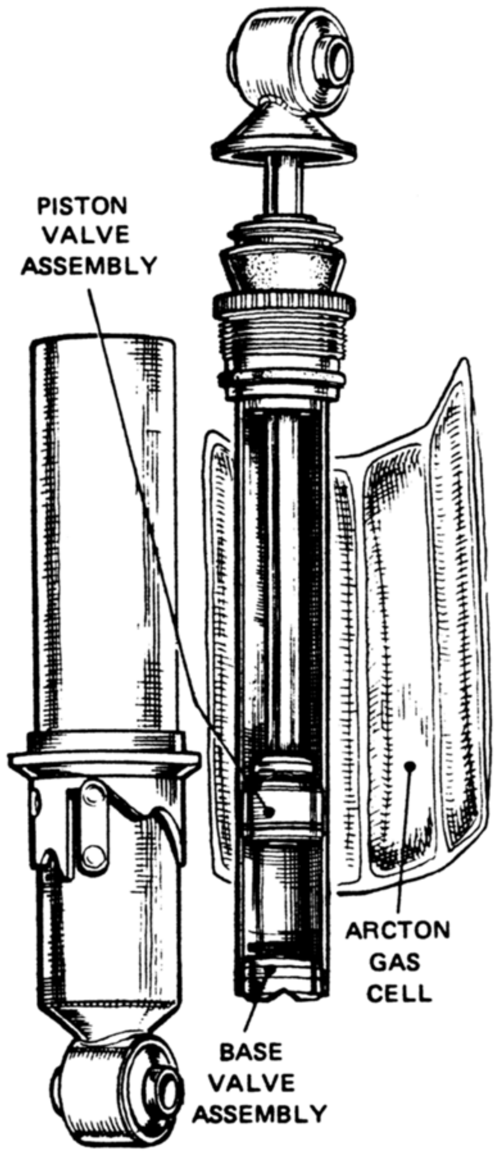
***Far Right,*** **a typical steering damper with a through-rod eliminating the physical displacement problem.**

***Below*****, As a substitute for free air, nitrogen at 100 psi. forms an emulsion with damping properties which remain substantially constant although lower than those of the neat oil.**

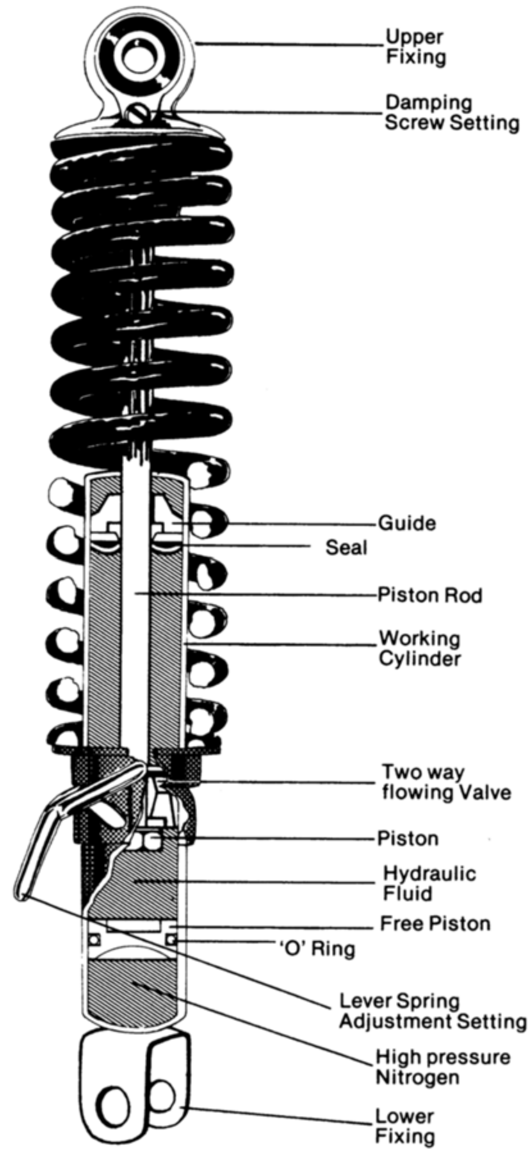
***Below Right,*** **With the DeCarbon principle, the hydraulic liquid and pressurized gas (usually nitrogen at up to 20 bar (300 psi.) are separated by a floating piston, which moves to accommodate the displacement of the piston rod. In some cases the pressure chamber is external and connected by a hose.**






**To prevent aeration of the hydraulic fluid, the compressible element (Arcton gas) in this Girling strut was contained in a nylon cell**




**In this DeCarbon "gas shock" the displacement of the piston rod is accommodated by a high-pressure nitrogen chamber separated from the hydraulic fluid by a free-floating piston. Spring preload and damping are both adjustable**

Another way to avoid the problem of rod displacement is to do away with the rod and this can be done with a rotary damper. This is nothing new to motorcycles of course, the old friction dampers were rotary in nature and Suzuki and Yamaha used them for steering dampers in the 1960s on some GP racers, and again in the 1990s on the road going TL1000 L as rear suspension damping.

More recently (1996) Yamaha and Ohlins patented a rotary design, fig. 6.10, with the claimed benefits of space saving and a lowering of the CoG. Separate bump and rebound valves are fitted in the oscillating damper piston, allowing individual adjustment just like a conventional linear unit. If required there is no reason why an external gas cylinder couldn't be used, as in normal high quality struts. This could compensate for thermal expansion. Sealing between the vane and the housing is likely to be the most critical aspect of this type of design.

Advantage of rotary units can include:

- The reduction in effective unsprung mass.
- Size not limited by the inner diameter of the spring.
- Rolling element shaft bearings could cut stiction.
- Rotating shafts are easier to seal than sliding ones and stiction would be reduced.
- The progressive or regressive nature could be separated from that of the spring. These are normally tied together through the use of a common actuating linkage.
- Offers alternative packaging options.

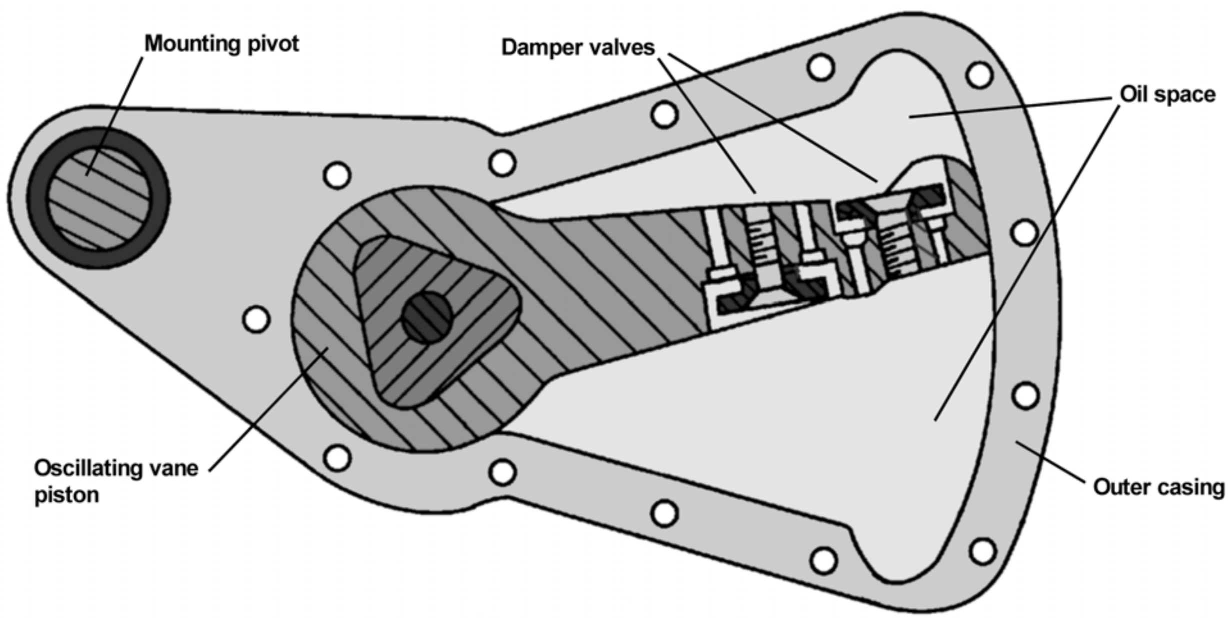
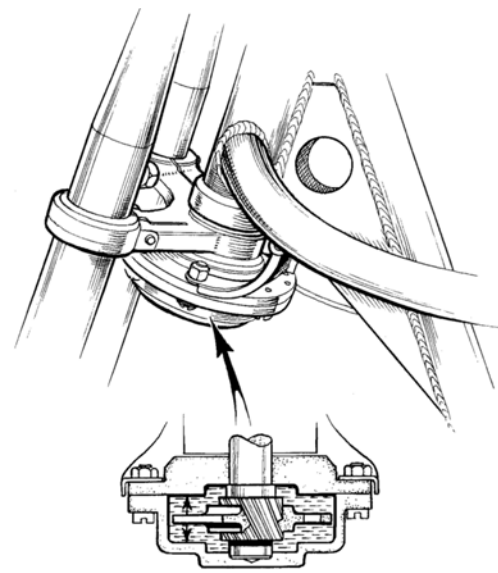



Fig. 6.10 **Yamaha-Ohlins rotary damper. As the oscillating piston or vane moves one way or the other then oil must be forced through one of the two damper valves, which can incorporate all the sophistication appropriate for the expected use. Leakage between the vane and the housing would need to be kept to an absolute minimum, to avoid uncontrolled influence over the damping forces.**

Suzuki fitted this rotary damper to the rear of the TL1000 L road bike. Aftermarket suppliers offered normal linear struts as a substitution.



This rotary actuated steering damper was used by Yamaha on some early racing models. An orifice plate was moved linearly by means of a quick start thread.

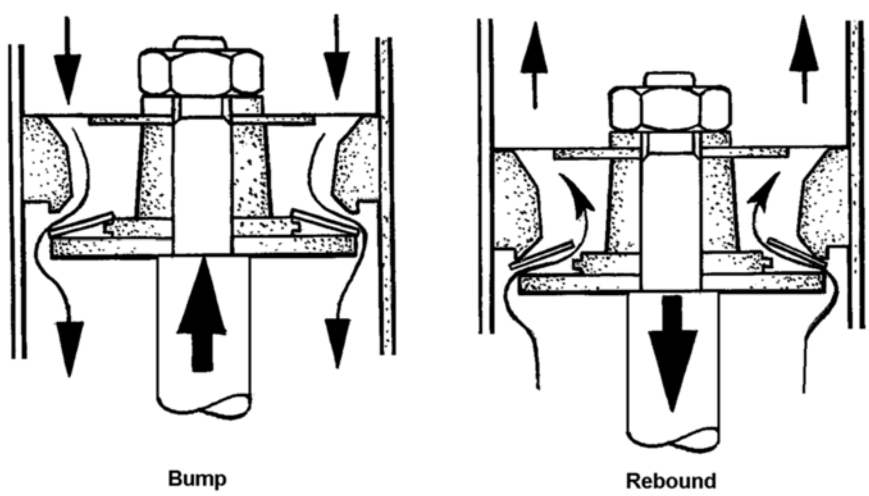



Fig. 6.11 **This shows one way of separating bump and rebound damping. The oil flow direction determines the flow path and hence different flow areas can be used to control the ratio between bump and rebound. Other designs force the oil through completely separate valves to give more control over bump and rebound.**

Hydraulic dampers can be endowed with various characteristics by their internal design --- e.g. one way damping only, two way damping with different rates for bump and rebound, dead spots in the movement and so forth. Early on, one-way damping only was quite common, on rebound only. The reasoning for this, was to present the minimum resistance to wheel movement after hitting a bump, transmitting the minimum force to the sprung part of the machine for the rider's comfort. Rebound was considered less important from the comfort point of view, and so all the damping was applied in this phase, requiring approximately double the damping forces that would have been necessary had equal damping been applied in each direction.

This approach has several disadvantages, the reasoning may have been sound from the comfort aspect when negotiating a single bump but could give rise to a serious problem when crossing a corrugated surface. Each bump compresses the suspension quickly (because of the absence of damping) while the subsequent recoil stroke is slowed by the heavy damping. This may prevent the wheel from returning to its static position before the next bump. The rapid repetition of this action may soon ratchet the suspension into a fully compressed state, so giving the effect of a solid frame, fig. 6.12.

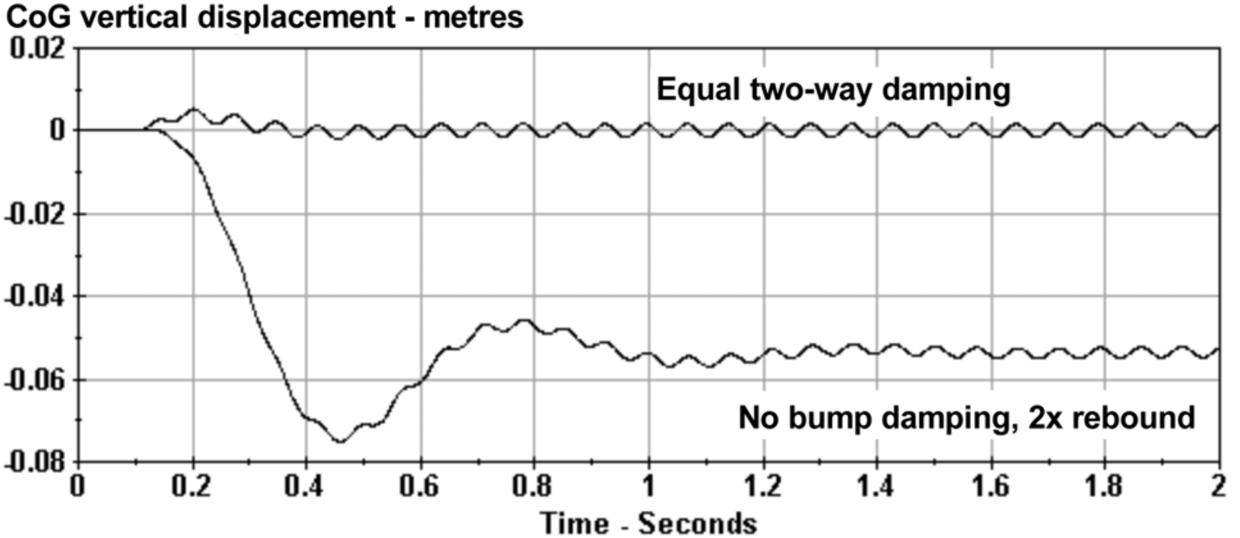
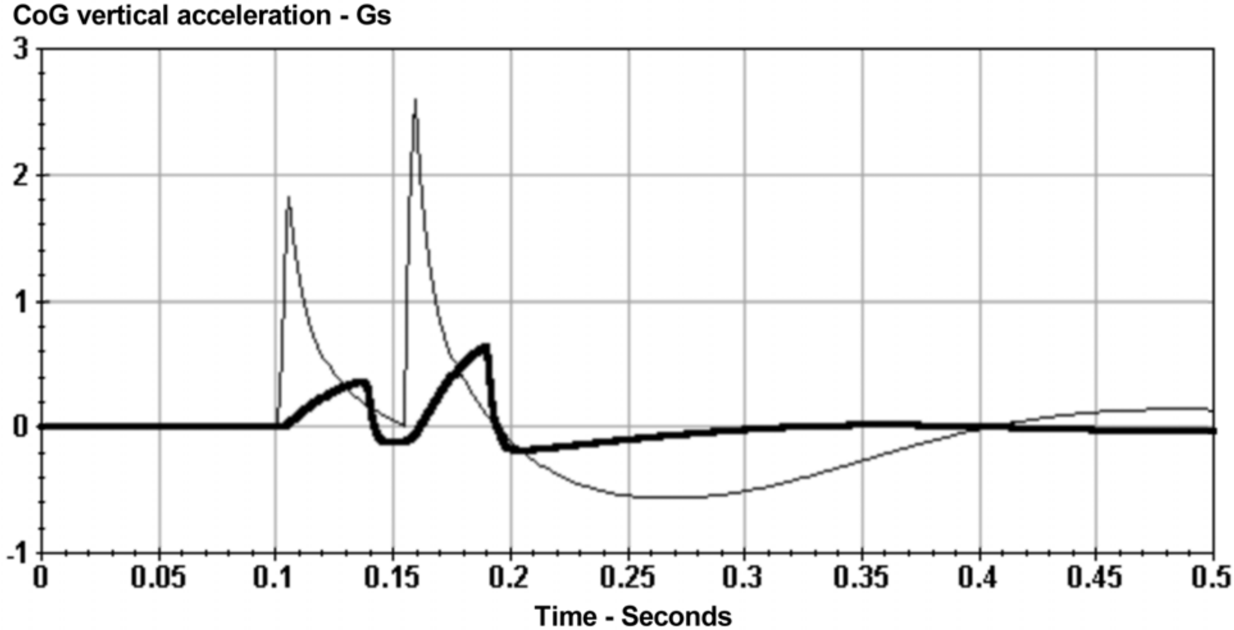



Fig. 6.12 **With equal damping in both directions the CoG oscillates about its static ride height when travelling rapidly over a corrugated road. The lower curve shows how, in this simulation with all the damping in rebound, the suspension jacks down. In some cases this effect can be severe enough to compress right down to the bump stops. Note also that the magnitude of the jacking down seems out of all proportion to the small magnitude of the oscillatory motion.**

Even single bumps and especially hollows can cause trouble with one way damping. At high speed the heavier rebound damping may prevent the wheel from maintaining contact with the ground as the wheel passes the crest of the bump, in a straight line this reduces traction and braking, which is bad enough but if this happens when cornering the result may be more serious. It is now almost universal to have two-way damping, but not equal amounts for the two directions. This would cause ride harshness and so a compromise must be struck. The ratio of bump to rebound damping used varies with the intended use of the machine, and some expensive units have provision for the independent adjustment in the two directions. For non-adjustable units the ratio may be fixed typically around the 1:4 to 1:2 region.



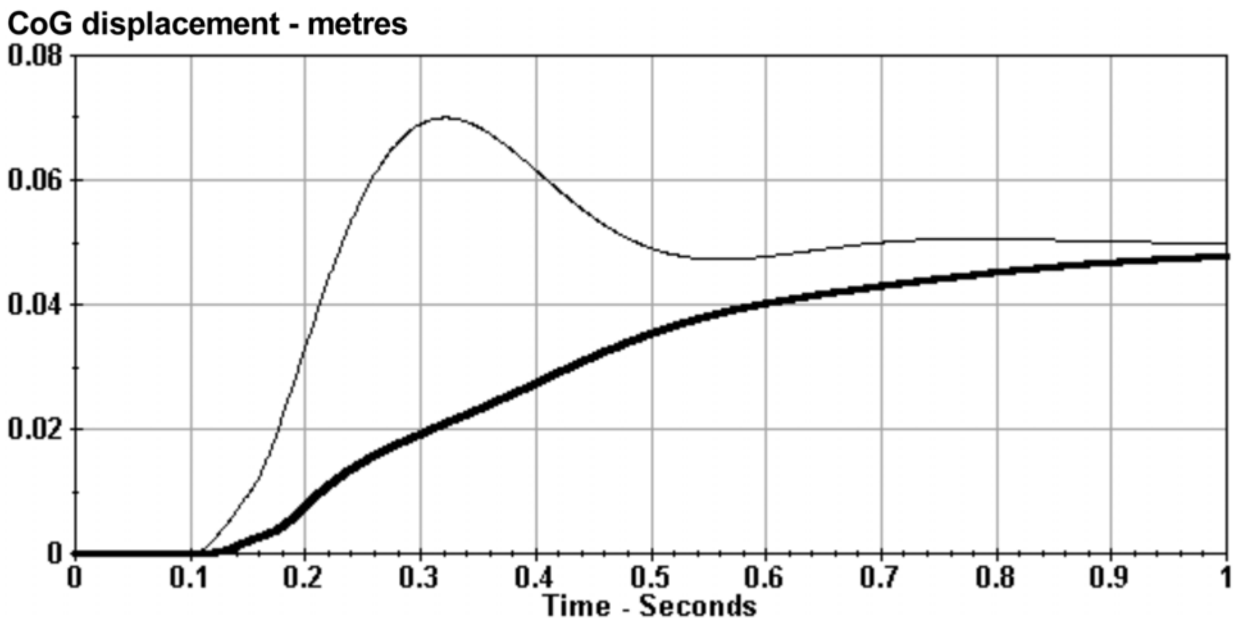




Fig. 6.13 **These graphs show different responses to the same bump, depending on the damping direction. The heavy line represents a simulation with all damping on the rebound and with no bump damping, the opposite applies to the thin line. The top set of curves shows the vertical acceleration at the CoG and clearly demonstrates the reduced shock that would be felt by the rider, with zero bump damping. The second hump in the curves occurs when the rear wheel hits the bump. The lower curves show the actual displacement of the CoG.**

**The bike encountered a 0.05 metre ( 2” ) high sharp edged step at 100 km/h ( 62 mph ).**

Fig. 6.14 shows the response to a drop or step down, which is the opposite case to that of fig. 6.13. In this case the ground to tyre contact force is plotted as this shows some road holding implications. A force of zero indicates that the tyre has lost contact with the ground. Studying the first part of the curves, between 0.1 seconds ( when the wheel reaches the step ) and 0.16 seconds, shows that in the case with only bump damping, the wheel is airborne for a shorter period, as expected. However, when we look farther along the time axis we see that the overall force fluctuations are greater and that the front wheel becomes airborne once again, the total time spent with no contact is actually greater than in the other case, with rebound damping only. With only bump damping, when the wheel hits the ground at about 0.14 secs. the higher bump suspension forces cause a heavier landing, shown by the higher peak tyre force, which in turn has caused more tyre bounce thereby lifting the wheel off the ground again at 0.23 seconds.

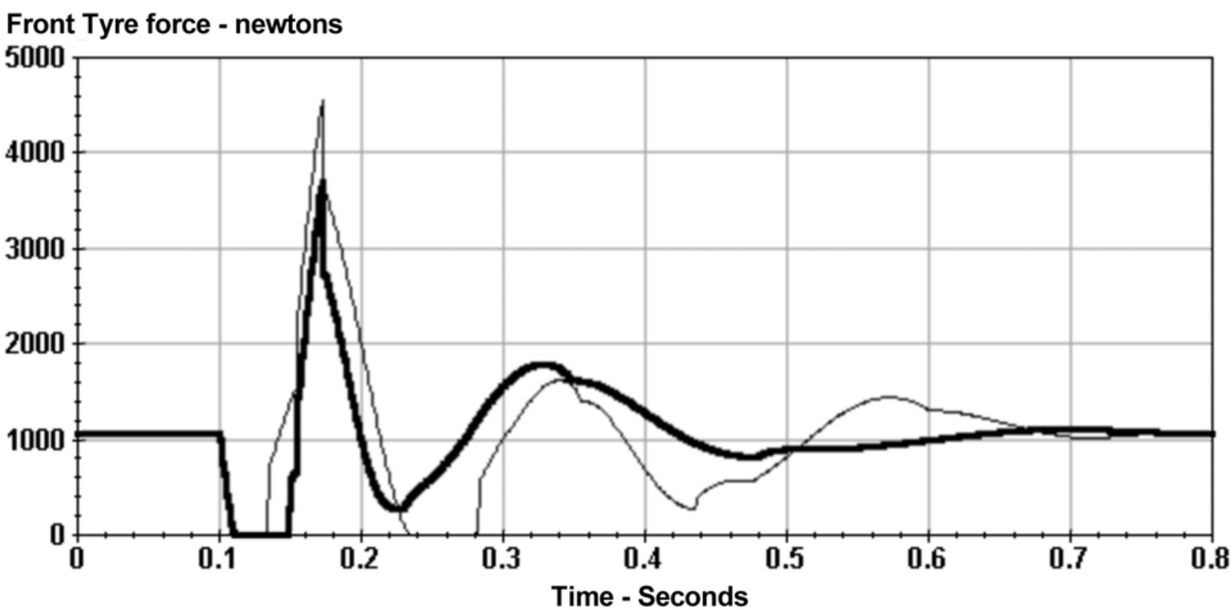



Fig. 6.14 **Simulation of the vertical front tyre / ground force as a bike passes over a 0.05 metre drop at 100 km/h. The dark line is with rebound damping only and the other is with bump damping only. Ultimate road holding requires the minimum variation in this force. This force can never go negative, because zero represents the point at which the tyre leaves the ground.**

The detail of what happens is very dependent on both suspension and tyre characteristics. Changing either will have quite large effects over the exact nature of the tyre force variation and fig. 6.14 should only be taken as an indication of the complexity of the situation. Under acceleration or braking there can be no traction when the wheel is airborne, and there will also be traction variation even when the tyre is on the ground. Driving and braking forces will have an effect on suspension forces also. This interaction makes the situation even more complex and in general will reduce acceleration or braking performance on uneven surfaces, in extreme cases severe juddering may occur. Maximum performance is obtained when we can achieve the minimum tyre force variation, but this is not so easy to do as there are so many parameters that interact. A different type of tyre would almost certain call for different suspension settings if we are searching for the ultimate road-holding as in racing. Showing once again why suspension setting occupies so much time at each race circuit.

## Sprung and unsprung mass

These terms are often a source of confusion, but the concepts are simple enough. Basically the sprung mass is the mass that is sitting on top of the suspension. For a motorcycle this is essentially all of the bike's mass less wheels, brakes and part of the suspension components.

The unsprung mass is therefore the whole mass minus the sprung mass, and so comprises mainly of the wheels etc..

There are a few factors which complicate the calculation or measurement of these parameters.

There is the mass of some of the suspension components themselves, the springs for example. The top parts of which are moving with the sprung part of the machine whereas the lower parts are firmly attached to the unsprung portions. Any elementary text book on mechanical dynamics will show that for a simple mass spring system the equivalent dynamic mass of a constant pitch spring is $^1/_3$ of the spring mass, not the $^1/_2$ that is often quoted. If we were to consider 2 equal masses oscillating in free space with a spring between them, then simple symmetry will dictate that the equivalent spring mass to add to each mass will be $^1/_3$ of $^1/_2$ , or $^1/_6$ of the spring. A vehicle suspension system is neither of these two extremes and so in most cases the equivalent spring mass to add to the unsprung masses will be somewhere between $^1/_6$ and $^1/_3$ of the total spring mass. Progressive or multi-rate springs add another complicating issue to these calculations, but the unsprung masses of the springs are not a major proportion of the total.

How do we assign appropriate portions of the total sprung mass to each end? This is often done by simply apportioning the mass in the same ratio as the static fore and aft weight distribution and this way certainly has its uses. However, it is not to hard to see that it also has its shortcomings. Consider for example the case when doing a wheelie. There is no front suspension active at this time and so the rear must support the total weight, thus the relevant rear sprung mass must be the total sprung mass but in addition what is normally regarded as the front unsprung mass must be included also.

Generally speaking, when we are looking at suspension performance it is the ratio of sprung to unsprung mass that interests us. As we shall see in the next section, it is desirable to have as high a ratio as possible, for roadholding, and to this end the lighter bikes are often at a disadvantage. With smaller machines it is usually difficult to reduce the unsprung mass by the same proportion as the sprung mass.

## Basic suspension principles

A road vehicle suspension is principally what is known in physics books as a *"mass, spring, damper"* system. In the above sections we've looked at some features of these components individually, now we'll see how they combine into one and we'll look at the overall characteristics and performance.

The principal historic reason for any form of compliant suspension is for the rider's comfort, to achieve this we ideally want a system that transfers none of the vertical wheel motion into movement of the sprung mass or rider. That is, we want the minimum possible vertical acceleration passed onto the rider. This is impossible to achieve completely, and another factor to take into account is the frequency of the disturbances felt by the rider.

As we have seen above roadholding is also greatly affected by suspension characteristics, and often the requirements of comfort and roadholding are in conflict, then we have to make inevitable compromises depending on the intended use of the machine.

## Suspension frequency

Humans are more tolerant of certain disturbance frequencies than of others, and this tolerance depends on the direction also. For example we are most comfortable when vertical movements are in the range of around 1 to 1.5 cycles per second (c/s or Hz) but this is a most unpleasant frequency when applied to our bodies in a horizontal direction. Vertical movements are mainly from road shocks and we experience horizontal motion from pitching of the bike. If the rider is not sitting with a vertical back, as when leaning in a racing crouch then there will be a cross coupling between these motions as far as the effect on the rider is concerned.

The following table gives a broad overview of the human perception of different vertical motion frequencies insofar as they affect vehicle suspension.

| Frequency (c/s, Hz) | Comments |
|---|---|
| 0.5 → 1.0 | Tends to promote motion sickness. |
| 1.0 → 2.0 | Generally considered the most comfortable. |
| > 2.0 | The ride is usually perceived as being hard or harsh. |
| 5 → 20 | These frequencies have various discomforting and sometimes damaging effects on different parts of the body. |

From this we can see that whilst we want to reduce all motion transferred to the rider, it is most important to design our suspension system to filter out, as much as possible, those vertical disturbances outside of the range of frequencies between approximately 1.0 and 2.0 Hz. To see how we might achieve this consider a simple mass, spring and damper system as in fig. 6.15.

If we apply a continuously oscillating disturbance to the lower end of the spring then we would expect at least some of this to be passed on to the mass above. This is akin to a wheel moving over a series of smooth bumps, and changing the vehicle speed will change the frequency of the disturbance. The graphs show how the mass responds over a range of frequencies, we see that with no damping there is a narrow range over which the movement of the mass is actually much greater than the input (wheel) motion. This frequency is known as the resonant or natural frequency. The introduction of damping will reduce or eliminate the excessive resonant response, and also reduce the frequency of resonance.

At disturbance frequencies much lower than the natural frequency of the suspension, we can see that the sprung mass will move with a magnitude near to that of the input disturbance, regardless of damping. This seems fairly obvious, if the wheel moves upward slowly then the rest is bound to follow suit. The graphs show an interesting and important fact about damping, below a frequency ratio of 1.4 (actually 1.414 or $\sqrt{2}$), the transmissibility decreases as damping is increased but is always at least 1.0. This means that when the road shock frequency is in this range the main part of the bike will receive no comfort advantage over an unsprung machine on this type of road.

In contrast to this the response is least when the applied frequency is several times higher than the natural frequency, thus in order to have the smoothest ride over the widest range of operating conditions we must aim to have a low natural suspension frequency, which requires soft springs. In this range more damping actually increases transmissibility and decreases comfort, for example at a frequency ratio of 3 with no damping the transmission ratio is 0.125 but increases to 0.608 at critical damping. This is logical because higher disturbance frequencies with the same displacement generate higher wheel velocities and damping force increases with velocity, hence passing more force through to the sprung mass.

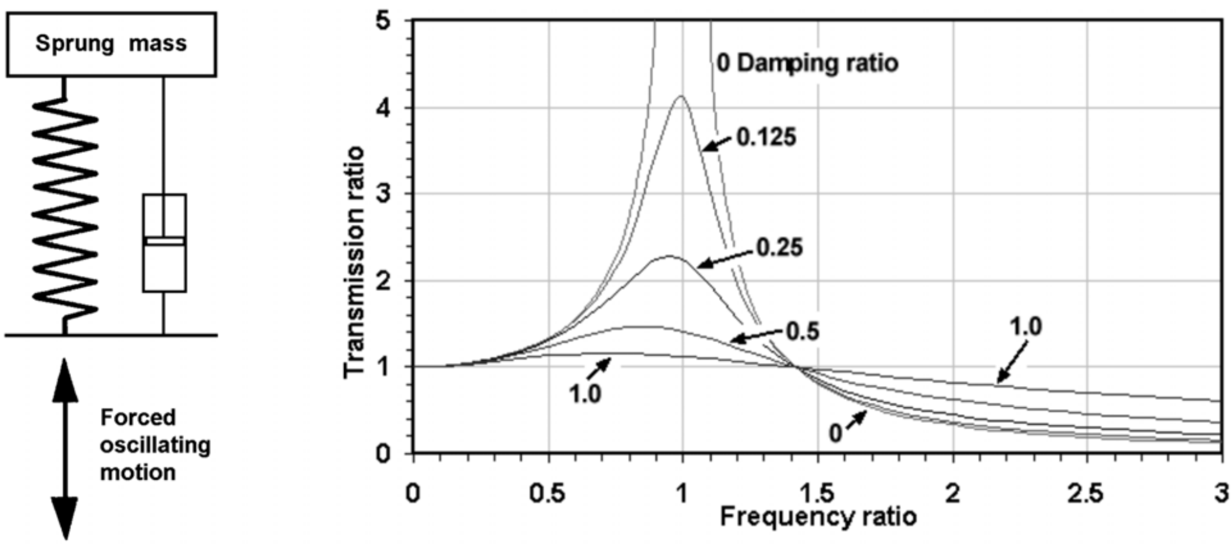



Fig 6.15 **Response of a damped mass spring system to a forced oscillation, such as the vertical movement of a bike compared to the road surface height variation. The vertical scale on the graph is known as the transmission factor and shows the peak amplitude of the movement of the mass in relation to the movement of the input. A value of 1 means that the two movements are equal. The horizontal scale is the frequency ratio, a value of 1 is the resonant or natural frequency, a value of 2 means that the input movement has a frequency of double the natural frequency. The individual curves show the effect of damping ratio, which is the degree of damping compared to critical damping. A value of zero is with no damping and a value of 1.0 is critical damping.**

The calculation of undamped suspension frequency can easily be made using a simple formula:

$$F = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

Where: F = frequency in Hz, K = spring rate and M = sprung mass

This is shown graphically in fig. 6.16.

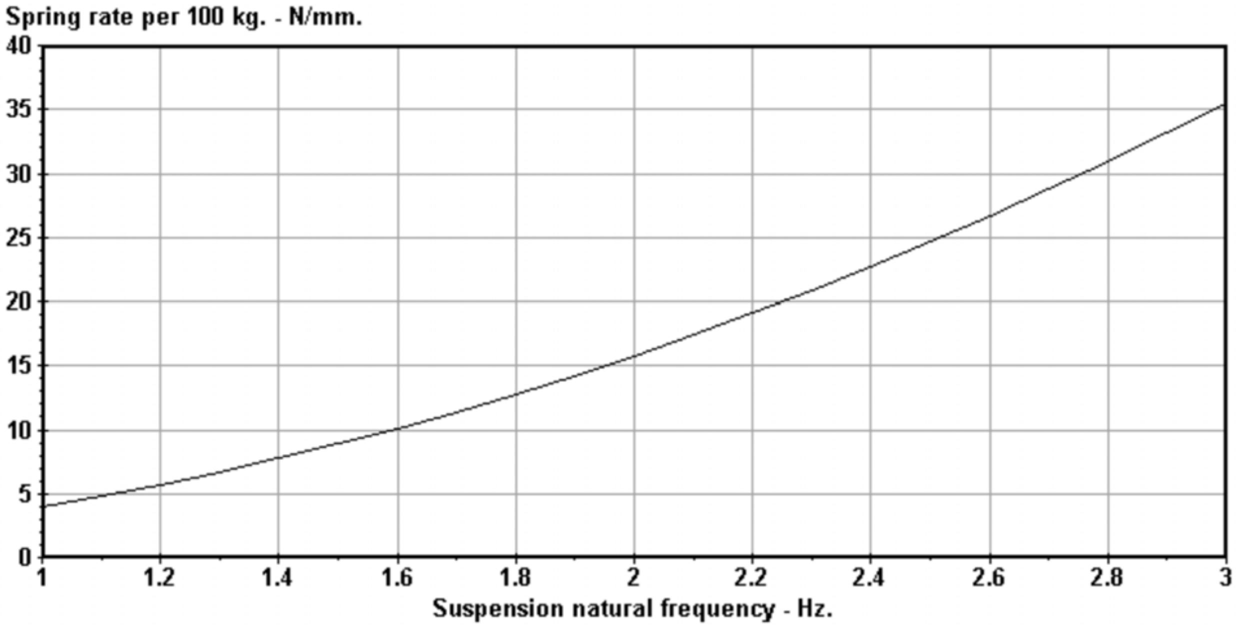



Fig. 6.16 **Valid for any vehicle, this graph shows the spring rate required for any suspension natural frequency – Hz., per 100 kgs. of sprung mass.**

## Spring sag

Some simple manipulation of the frequency formula will show that for a given frequency a spring will sag by a fixed amount regardless of the sprung mass.

This is really saying nothing more than: if we double the sprung weight we must also double the spring rate if we wish to maintain the same suspension frequency.

So the difference between the statically loaded length and the free length of the spring will not change regardless of the size of the bike, if we want the same suspension characteristics. The following graph, fig. 6.17 shows the relationship between frequency and sag.

From this we can easily calculate the spring rate needed for a given requirement. For example for a required suspension frequency of 2.2 c/s we would need a spring sag of 51.4 mm. If the bike has a laden front sprung mass of 100 kg. (weight = 981 N.) then the spring rate required is simply 981/51.4 = 19 N/mm..

It is even easier if we use a graph such as in fig. 6.16 which shows the rate required per 100 kg. of sprung mass. Say we want a natural frequency of 2.4 c/s, from the graph we can see that requires a rate of about 22.7 N/mm for every 100 kg. of sprung mass.

This spring rate, and those referred to when discussing frequencies above, are the effective vertical rates at the wheel. The actual front fork rate will depend on the fork angle, and the rear spring rate will depend on the linkage design or mounting position. This is explained more in the appropriate chapters.

*From*

$$F = \frac{1}{2\pi}\sqrt{\frac{K}{M}}$$

we get

$$\frac{M}{K} = \left(\frac{1}{2\pi F}\right)^2$$

or

$$\frac{W}{K} = g\left(\frac{1}{2\pi F}\right)^2$$ Where W is the sprung weight and g is the gravitational constant

but $\delta = \frac{W}{K}$ Where δ is the spring displacement or sag

*therefore* $\delta = g\left(\frac{1}{2\pi F}\right)^2 = \frac{248.62}{F^2}\,mm. = \frac{9.788}{F^2}\,inches = \frac{0.8156}{F^2}\,feet.$

Thus δ is directly related to frequency. Fig. 6.17 shows this in graphical form.

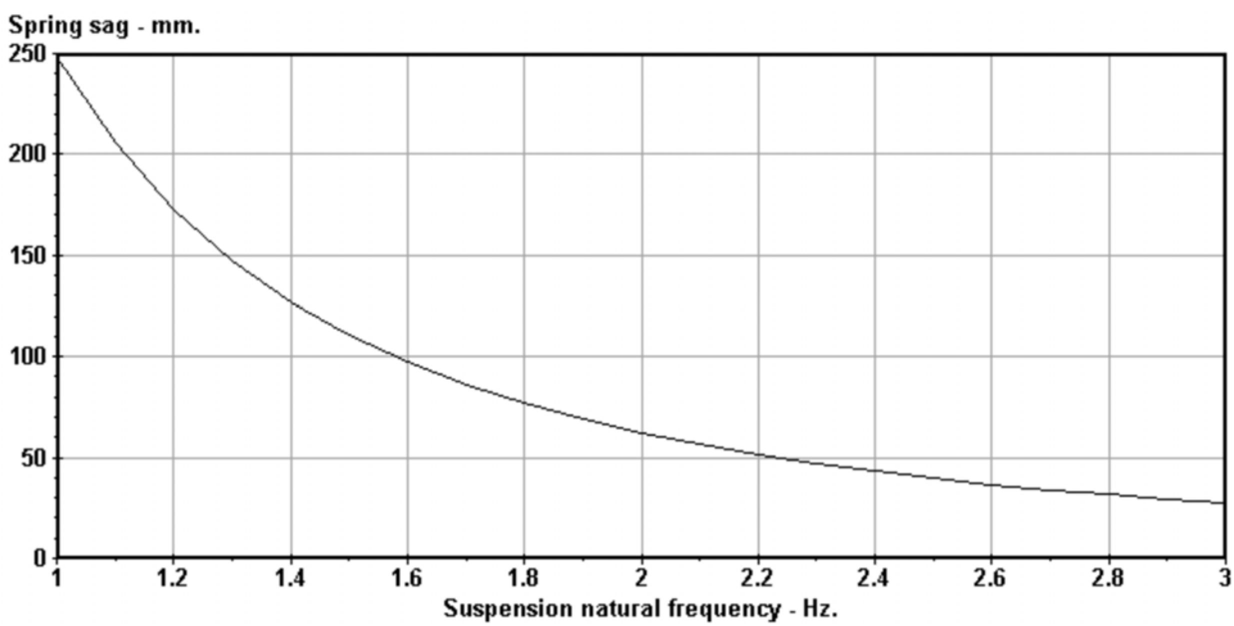



Fig. 6.17 **Applicable to any vehicle, this graph shows how the statically loaded spring sag, from the length of a free spring, is related to the suspension frequency in the range most likely used for suspension.**

The actual range of frequencies imposed on the suspension system is determined by the bike's speed and nature of the road. If we consider the road surface to vary in a sinusoidal manner, then we can relate the stimulus frequency to the length of a complete bump, known as the wave length.

$$F = \frac{V}{\lambda}$$

Where *F = frequency in Hz, V = Velocity in m/s,* $\lambda$ *= Bump wavelength in m.*

Fig. 6.18 shows this in graphical form over a wide range of speed and wavelengths. However, this only tells us the range of frequencies to be expected, another very important aspect of ride comfort is the actual amplitude of the displacements that occur at different frequencies. this varies from road to road but in general we can say from experience that the amplitude reduces as the wavelength reduces.

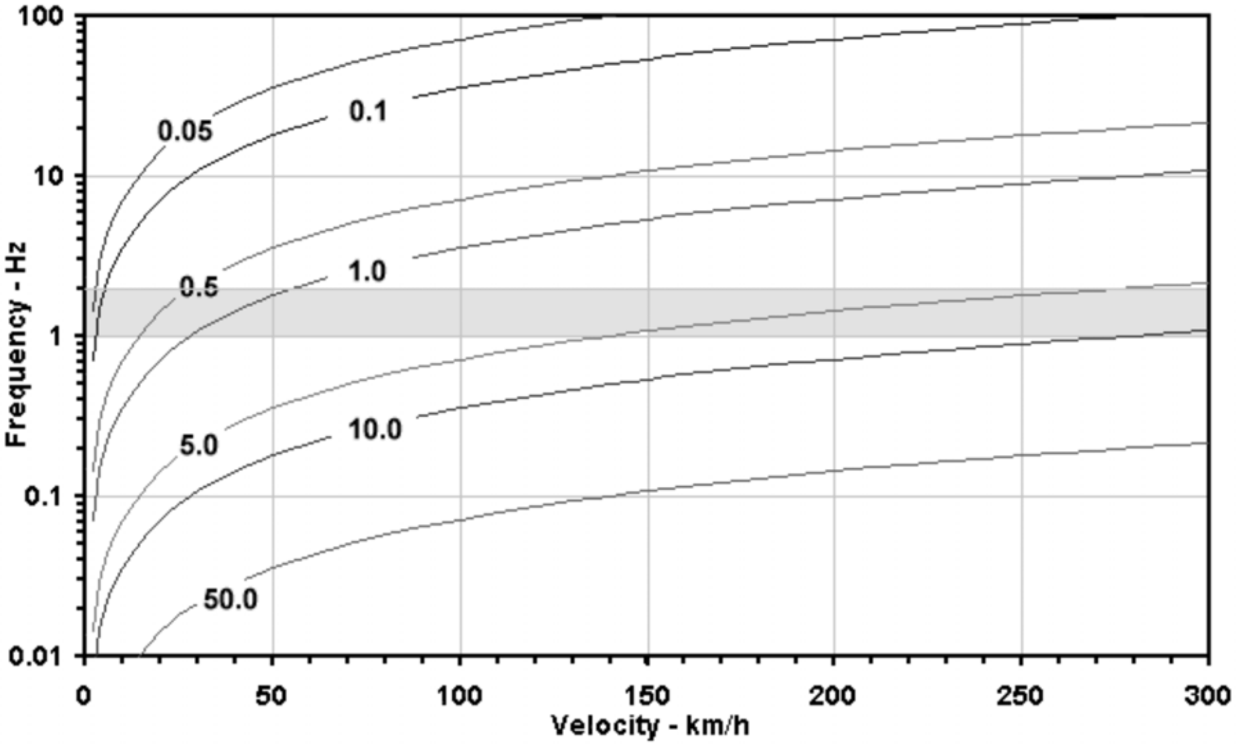



Fig. 6.18 **The relationship between velocity and suspension forcing frequency for various bump wavelengths. Note that the frequency is plotted on a logarithmic scale. The numbers on the various curves are the bump wavelength in metres. The shaded area is the most comfortable frequency range of between 1 and 2 Hz. We can see that for speeds between 30 and 300 km/h the most comfortable frequencies occur with wavelengths between 0.5 and 10 metres.**

Standards organizations such as ISO have developed standardized models of various types of road surface for use in suspension analysis. These define the amplitudes to be expected at different wavelengths, the inverse of which is often called spatial frequency to differentiate it from the normal time based frequency.

The above discussion is based on the notion that the road disturbances were a continuous sequence of evenly spaced smoothed bumps, implicitly assumed to be sine-wave. Whilst washboard surfaces like this exist, some Australian country roads being just one example, they are far from the only or even the most common type of suspension input. In daily use we are likely to come across a whole variety of obstacles ranging from potholes to sharp edged steps, and the effects of acceleration, braking, cornering and landing after very high jumps in the case of some off-road machines. All of these things must be considered and may require modification of the main suspension parameters away from the optimum for comfort over one type of road surface.

In the tyre chapter we saw just how important that tyre compliance was to the suspension process. So tyre characteristics are important to the selection of suspension settings. With all of these often conflicting requirements it is not surprising that racers have to spend a lot of time testing in order to get the overall optimum settings for a particular track.

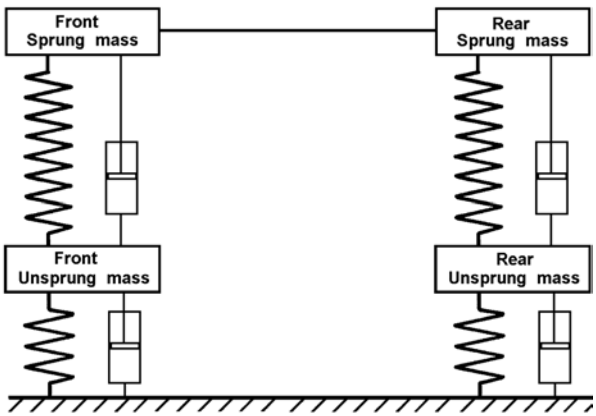
## Suspension models

The previous discussion has so far only considered the vehicle as being supported by one wheel, whereas in reality there are two. This gives us another opportunity to adjust the response to bumps, if we make the suspension frequencies different at each end, then the extent of the motion felt by the rider can be further smoothed out over a wider frequency range. Indeed it is usual to have a higher suspension frequency at the rear. Figs. 6.19 and 6.20 show a slight complication compared to the basic system in fig. 6.15. Both wheels are considered and unsprung mass and tyre stiffness have been added to make the model more realistic. As we shall see later the relationship of the wheelbase to wave-length is also of vital importance to the suspension response and rider comfort.

The concept of suspension frequencies being separated front to back relies on the assumption that we can divide the bike's total sprung mass into two lumps of mass, each centred above the respective wheel, this is shown diagrammatically in fig. 6.19 and implicitly demands that what happens at the front doesn't affect the rear and vice versa. The antithesis of this is also shown in fig. 6.20, This assumes that all the mass is concentrated at the CoG, here we can see that any bump at the front will cause the bike to rotate about the CoG (pitch backwards) and hence also tend to compress the rear suspension. In this case, considering the overall suspension system as being two separable parts can only be considered as an approximation. These are two extreme models of a motorcycle suspension, each is useful it their own right but neither fully explains the true dynamics.

In reality the sprung mass of bike and rider is not concentrated at each end nor about the CoG, but is distributed over the length of the machine and its (sprung) pitch moment of inertia will have to be taken into account along with the total sprung mass in order to accurately evaluate suspension performance. A more accurate model for this is shown in fig. 6.21 in this case the sprung part of the bike is considered to comprise of a point mass at the CoG plus a pitch moment of inertia. Using this as the basis for suspension calculations will give more realistic values for the separate front and rear natural frequencies, and also properly allow for the interaction between front and rear.

However, despite the shortcomings, the use of separate front and rear suspension frequencies is still a useful first guess tool for setting suspension parameters and for comparisons between various types of motorcycle and other road vehicles.



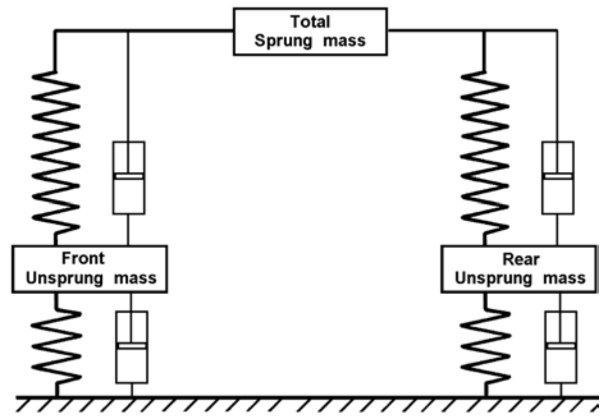




Fig. 6.19 **Model of the suspension system of a motorcycle. In this simple layout the sprung mass is assumed to be divided into two separate front and rear parts. This is the implied model used when calculating simple suspension frequencies. The rear suspension will not be affected by disturbances at the front.**

Fig. 6.20 **Here the sprung mass is grouped around the centre of gravity. A bump at the front will tend to rotate the machine backwards about the CoG, thus also compressing the rear. This is somewhat closer to reality but would predict an over quick response as there is no provision for pitch inertia.**

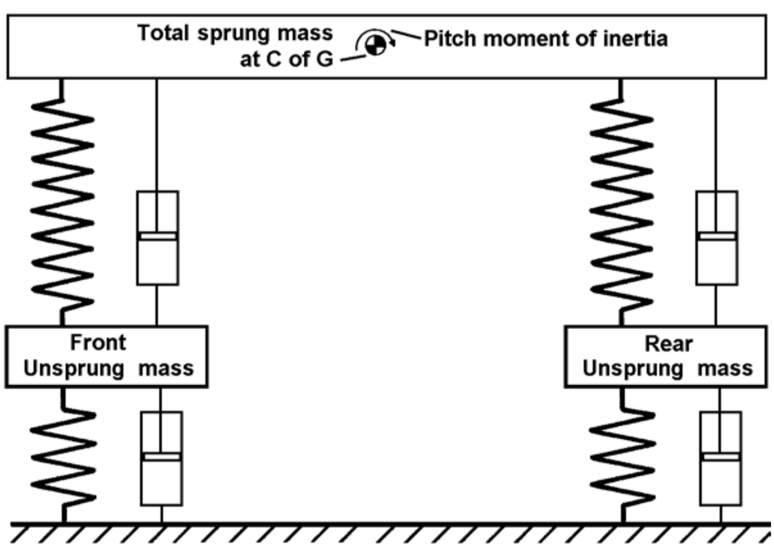



Fig. 6.21 **This is a reasonably accurate representation of a practical suspension system. Models like this are used in computer simulations of suspension performance.**

**When looking at car suspension the compliance of the seat is sometimes added for greater accuracy, but this is of less importance on a motorcycle as the seats are usually more rigid.**

**The spring and damper below the unsprung masses represent the tyre, and the ones above are the actual suspension units. The damping from the tyre is different from hydraulic damping and is mainly hysteresis as explained in the tyre chapter.**

Using models like this we can evaluate suspension performance against time or frequency for a wide range of loading conditions as shown in some previous and following examples, for some purposes it may be more useful to plot the frequency response of the complete motorcycle, rather than just a simplified single wheel model as in fig. 6.15. There are many resonances that will give various peaks in the curve rather than just the simple single peak shown in fig. 6.15.

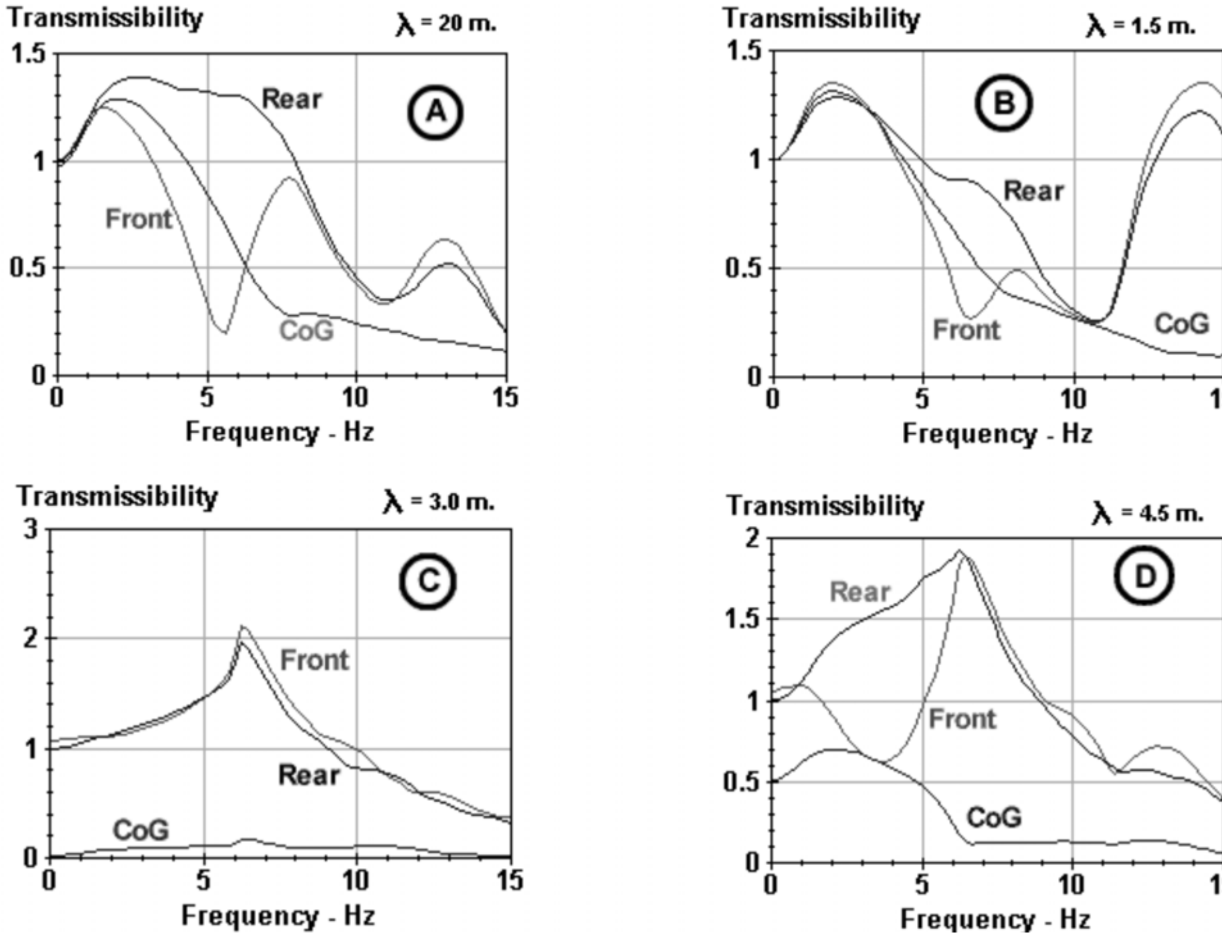



Fig. 6.22 **Four sets of frequency response curves with the same bike parameters plotted from simulations at different bump wavelengths (λ), as shown. (A) is for λ=20 m. which is several times the wheel base of 1.5 m. (B) is when the wave-length is equal to the wheelbase and (C) & (D) show the cases where the bump length is 2 and 3 times the wheelbase respectively. Note that the vertical scales differ in some cases. It is clear just how much the relationship between wheelbase and bump length affects the transmissibility of the bump displacement to the sprung part of the bike. However, these curves don't tell all the story by any means. Compare (C) with the upper plot of fig. 6.32 , both show that there is little vertical movement of the CoG, but the front and rear parts are subject to quite high peak amplitudes but out of phase with each other. The front moves up as the rear moves down. In other words there is considerable pitching motion which is generally more uncomfortable than simple vertical heave motions.**

Each end of the bike has its own natural frequency on the suspension springs, there are separate resonances for pitch motions and each wheel has its own natural frequency largely determined by tyre stiffness. Dampers are rarely pure viscous in nature and are invariably set to give more rebound than bump damping, tyre damping is complex and reliable information is not readily available. Add to this the effects of the wheelbase to wave-length ratio (shown in more detail in fig. 6.32), and you can see that the whole subject is extremely complex, and the response at any given frequency can be changed by small modifications to a whole range of parameters, as shown in figs. 6.22 and 6.23.

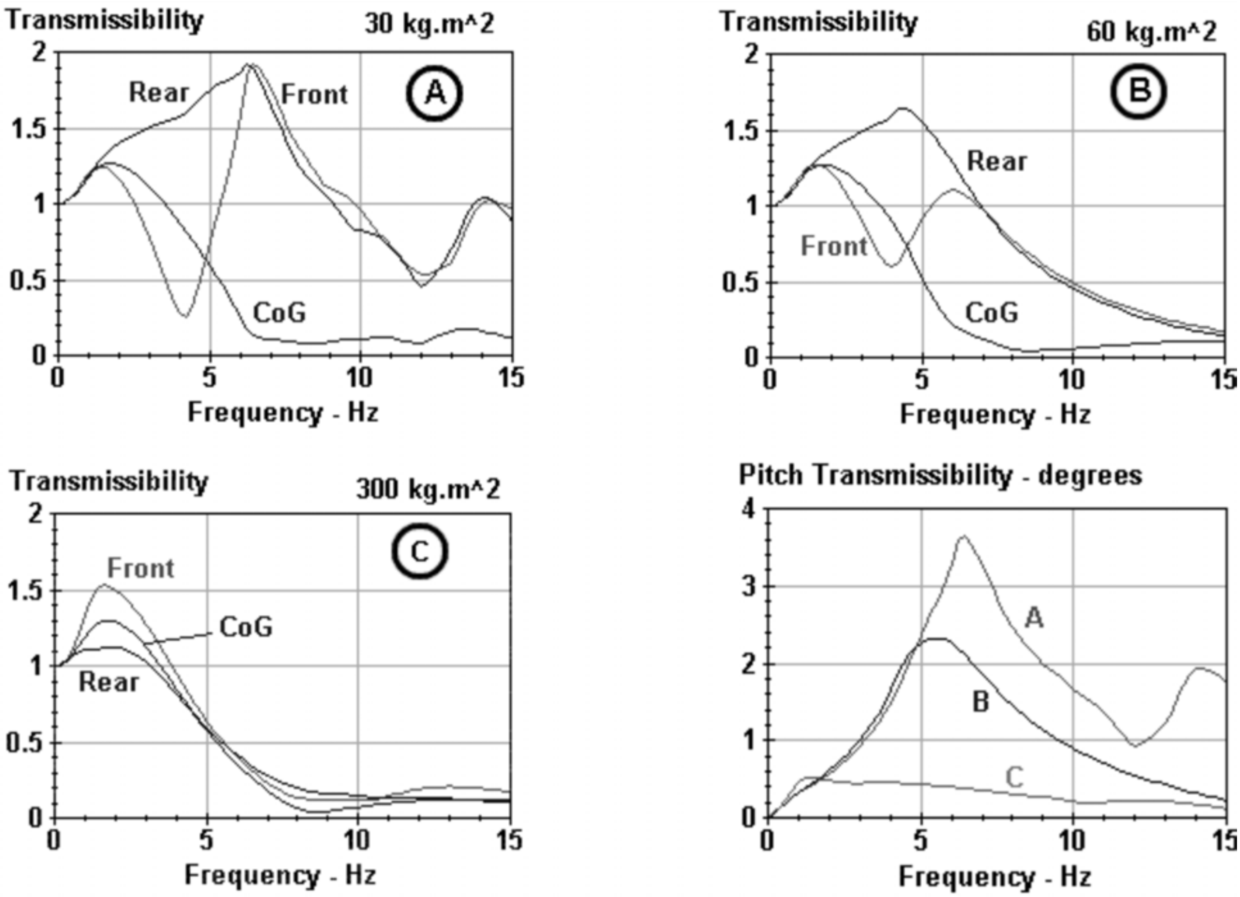



Fig. 6.23 **In these plots the velocity is kept constant at 100 km/h and the stimulating frequency was varied by changing the bump wavelength. (A), (B) & (C) differ only in the pitch inertia as shown. (A) & (B) cover the range of realistic values but (C) is an unrealistic test case with a pitch moment of inertia 10 times that of (A). This shows that with a large pitch inertia there will be much less pitch displacement and so we'd expect the front, rear and CoG to move up and down in concert. This is shown clearly in (C). The fourth set of curves show the transmissibility in terms of pitch movement for the three cases. Note the low transmissibility for case (C) with the very high pitch inertia.**

**For all these cases the main natural frequencies were: Main pitch about CoG: 4.2 Hz. (except (B) = 2.9 Hz., (C) = 1.3 Hz.)**

**Front: Unsprung 2.0 Hz, wheel on tyre 14.2 Hz., wheel on springs 4.3 Hz.**

**Rear: Unsprung 2.3 Hz, wheel on tyre 12.7 Hz., wheel on springs 4.7 Hz.**

**Careful study of the above plots will shown some influence from all these resonance frequencies.**

# Other factors

## Sprung and unsprung mass ratio

It is often stated that we always benefit from having the highest ratio possible. This is not true in all cases as the demands of roadholding and comfort are often opposed. Roadholding (tyre grip) requires the minimum dynamic variation in the tyre to road vertical load, and this calls for the minimum unsprung mass. For comfort we want the minimum force transmitted to the sprung part and as we shall see, this is helped by a high unsprung mass when hitting a raised bump, but not for hollows.

Imagine that a wheel leaves the ground or travels over a hollow in the road. When the front or rear suspension is partly or fully compressed, it exerts equal and opposite forces on both the sprung and unsprung parts of the machine – i.e. it tries simultaneously to raise the bulk of the machine and return the wheel to the ground. We want a system that returns the wheel quickly to the ground, for good road-holding, without unduly disturbing the sprung mass. The acceleration of any object depends on its mass and the force acting on it. Since, in our case, the same force is acting on both the sprung and unsprung masses, their ***relative*** accelerations depend on the ratio of their masses, while their ***absolute*** accelerations are determined by the suspension force and the absolute values of the two masses.

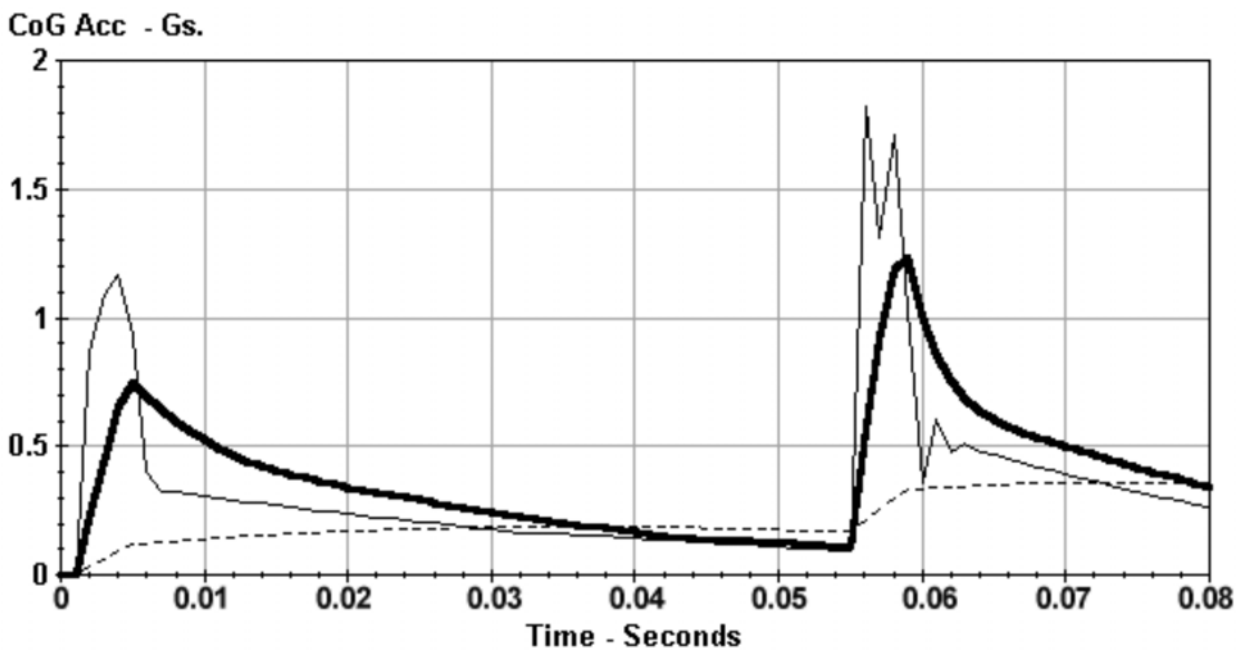



Fig. 6.24 **These plots are the result of a computer simulation showing the effect of various sprung to unsprung mass ratios. The motorcycle is travelling at 100 km/h and encounters a step of 25mm. (0.025 m.). The front suspension frequency is 1.9 Hz. with the rear a bit higher at 2.2. The graphs show the vertical acceleration of the CoG expressed in Gs. for three values of sprung to unsprung mass ratio. This ratio is changed by a factor of five between each curve. The heavy line is for a typical ratio of 4.5, the solid thin line has 1/5 of the previous unsprung mass with a ratio of 22.5, and the dotted line is for an unsprung mass of about 5 times normal with a ratio of 0.9.**

**The first hump in the curves occurs as the front wheel meets the bump, and the second is when the rear hits. It is clear that in this case the reduction in shock passed on to the sprung part of the machine and rider is greatly reduced with the increased unsprung mass.**

For roadholding (quick response of the unsprung mass) we need the highest possible ratio between the two. Unfortunately, there is a limit to how light we can make the unsprung mass and so an increase in sprung mass and spring rate enhances roadholding on rough roads and also quickens the response of the unsprung mass, so keeping the wheels in better contact with the surface.

However, the existence of a compliant tyre between the road and wheel (unsprung mass) leads to the situation that more unsprung mass will increase rider comfort when the wheel hits a bump. Fig. 6.24 shows the reduced shock passed on the CoG for higher unsprung mass. Imagine an infinitely heavy wheel, when this hits a bump the tyre will deflect to absorb the shock but the wheel will not move. If the wheel doesn't move then it can't impart any disturbance to the rider. Now consider the other extreme with zero unsprung mass, then the road shock will be passed up to the rider through the tyre stiffness in series with the suspension stiffness.

The price to be paid for improving the rider's comfort in this way is the greater tyre and wheel loads that result. When the tyre has reached the top of the bump, the increased wheel mass will tend to keep going and may actually leave the road, in any case roadholding will suffer.

## Cornering requirements

Another consideration, often overlooked, is the effect of cornering forces, particularly on road-racing and sports machines, where banking angles may exceed 45 degrees. This represents a 41 percent increase in the static suspension loading. Fig. 6.25 shows how the weight and cornering force combine to increase suspension loading.

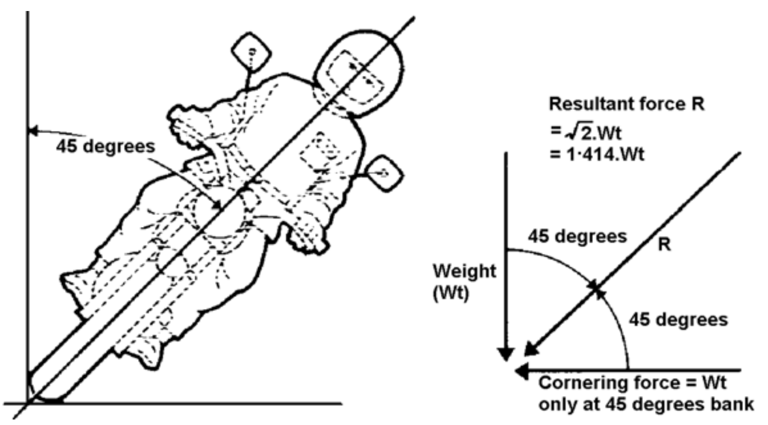



Fig 6.25 **At a cornering angle of 45 degrees (i.e. 1g lateral acceleration) the load on the suspension is 41% higher than when the machine is upright**

Fig. 6.26 shows the degree of increased loading for different values of cornering acceleration. In ice racing, where machines are banked through 60–70 degrees, the static loadings could be increased two or threefold, which partially explains why rear springing has not been adopted in that sport.

Imagine negotiating a series of opposite-hand bends at high cornering speed. As we bank into the first bend the suspension compresses under the increased loading, but when we straighten up and heel the other way it first extends as the cornering load is removed and replaced by an outward centrifugal force (due to the rapid roll velocity), and then compresses again, fig. 6.27.

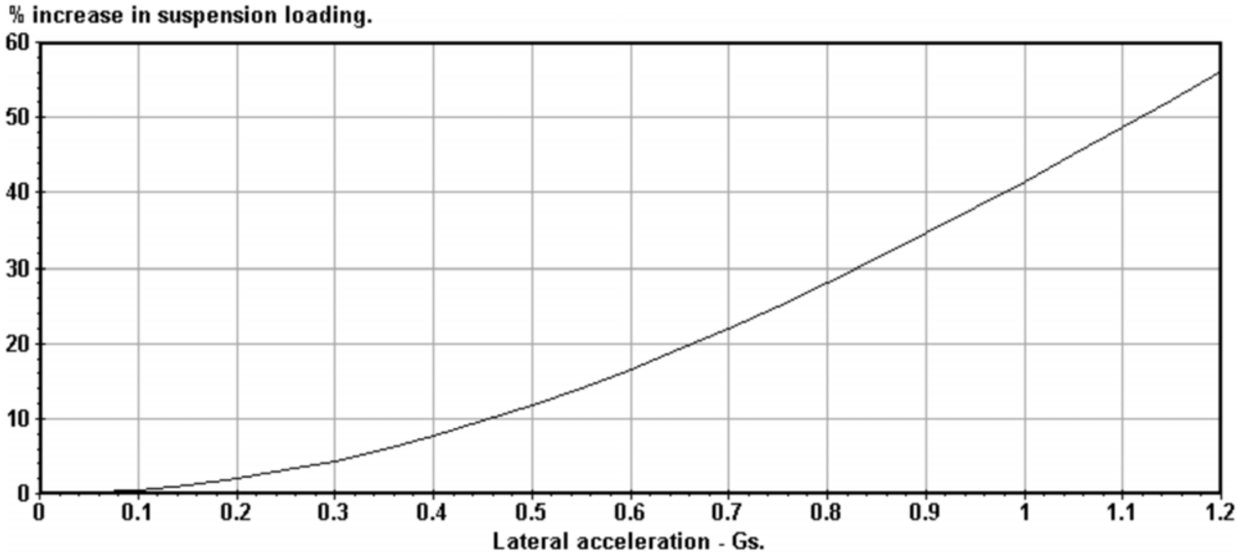



Fig. 6.26 **Showing the percentage increase in both front and rear suspension loading when cornering.**

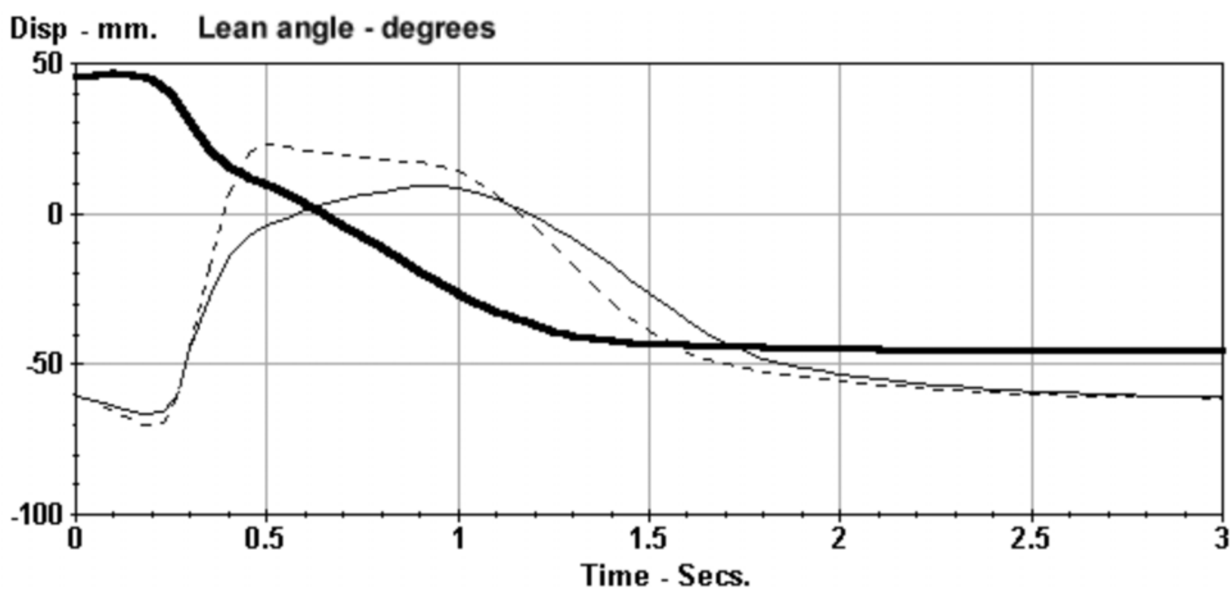



Fig. 6.27 **The effect of changing from a 45 degree bank to a similar lean angle towards the opposite side (centre part of an "S" bend). The dark line shows how the lean angle has changed and the other lines show the suspension compression and extension for two cases with different values of damping. The dotted line has less damping than the solid line, allowing the overshoot during the early part of the suspension extension. It might seem that the stronger damping would be preferable as the total suspension movement is slightly less but it is likely that the centrifugal force will then reduce the tyre load by a greater factor.**

This not only reduces the suspension movement available in the bends for absorbing bumps but also subjects the rider to an up-and-down movement that does nothing to improve his control. If the machine has telescopic forks there is a change in wheelbase too. This is another example of how suspension settings can affect the handlings characteristics.

This cross-coupling between suspension movement and cornering roll can have very important implications for stability and control. For example, if the damping is insufficient then the spring compression due to cornering may start a small suspension oscillation as we lean in, this in turn will create varying tyre/road loads altering the relationship between the lateral tyre forces actually produced and those required. This can thus cause cyclic yaw and roll movements, both of which will further add to the tyre force variation. These interactions can get extremely complex and may become self sustaining or worse, actually increase in severity. In practice these effects can manifest themselves as nothing more than a brief damped wobble during corner entry or exit, through undesirable suspension "chopping", to a fully developed uncontrollable weave resulting in a crash. In any case the varying tyre loads will reduce the ultimate cornering capability to the detriment of racing lap times or road safety.

In the mid 1970s I often used to spend each Wednesday at Brands Hatch either testing or helping other riders sort out their bikes. This was the period when rear suspension struts with damping and ride height adjustment where just coming into common use. Riders were less aware then of how to go about achieving good settings, and it seems that if you give some people adjustability then they'll manage to arrive at the worst possible setup. It was quite common to see bikes chopping badly on their rear suspension all the way around the bottom bend, and I remember that in countless cases I was able to help cure this just by changing suspension settings.

Figs. 6.28 → 6.31 hi-light how reducing only the rear damping can introduce weave motions that show up as variations in most of the dynamic variables. The bike manoeuvre was leaning into a corner from a straight approach with a final lean angle of 44 degrees. This is another example of the integrated nature of motorcycle dynamics, a change in one parameter usually has repercussions throughout the whole machine. For these simulations the rear spring frequency was set lower than the front as this was often a setup that I encountered in practice, when asked to help cure similar problems. The various graph sets cover suspension displacement, tyre forces, lean angle and rear slip angle, which is more or less the yaw angle of the bike. In each case the "wobbly" curve of a pair is the one with the reduced damping. In some parameters there is more variation at the front than the rear, this is due to the effect that the other variables have on the steering angle (not shown).

We can see that not only is there a damped cyclic motion of the suspension as we might expect, but this has cross coupled to a yaw weave, tyre force variation and lean angle oscillation as well. Sometimes there is a very fine line between what is stable and what is not. For the case shown the weave was stable and gradually died away after about six seconds, but if the rear damping was reduced by just a few percent more then the weave would increase rapidly and the simulated motorcycle would crash within two seconds.

This type of instability can also be initiated by hitting a bump when cornering. Frame and wheel flexibility also influences the reaction to such disturbances.

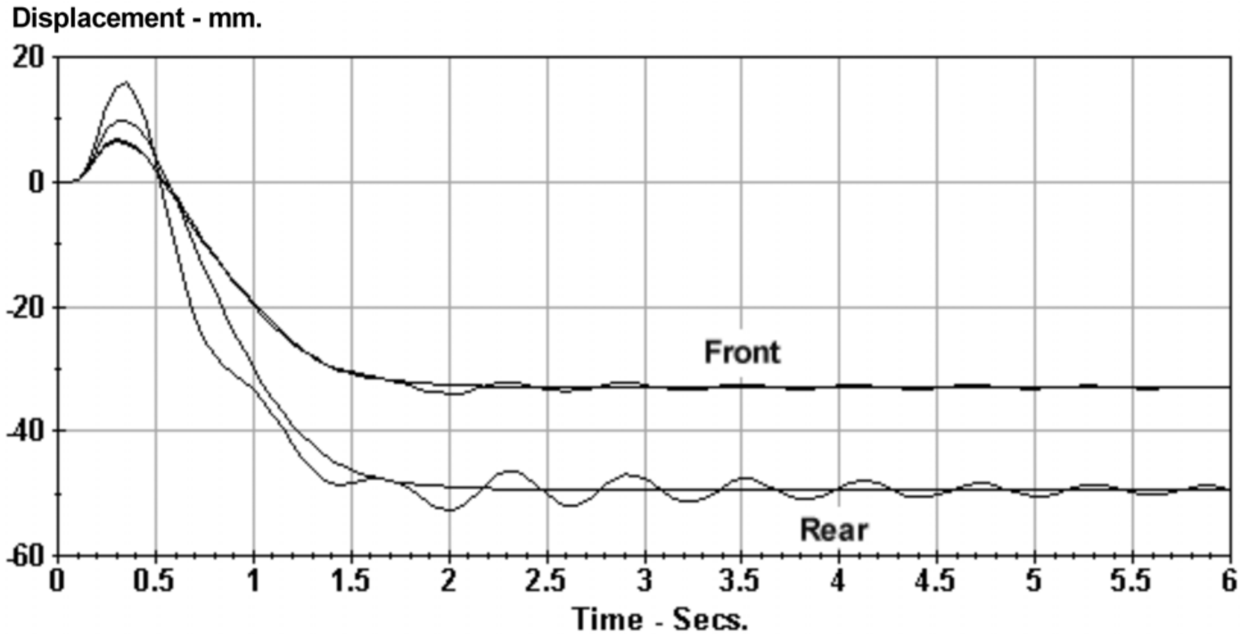



Fig. 6.28 **Front and rear suspension movement, showing the variation in the under-damped case. Note also the suspension extension during the first 0.5 second, this is due to the centrifugal force from the initial build up of roll velocity before the cornering force has overcome it and caused the compression. The damping value also affects the amount of this extension.**

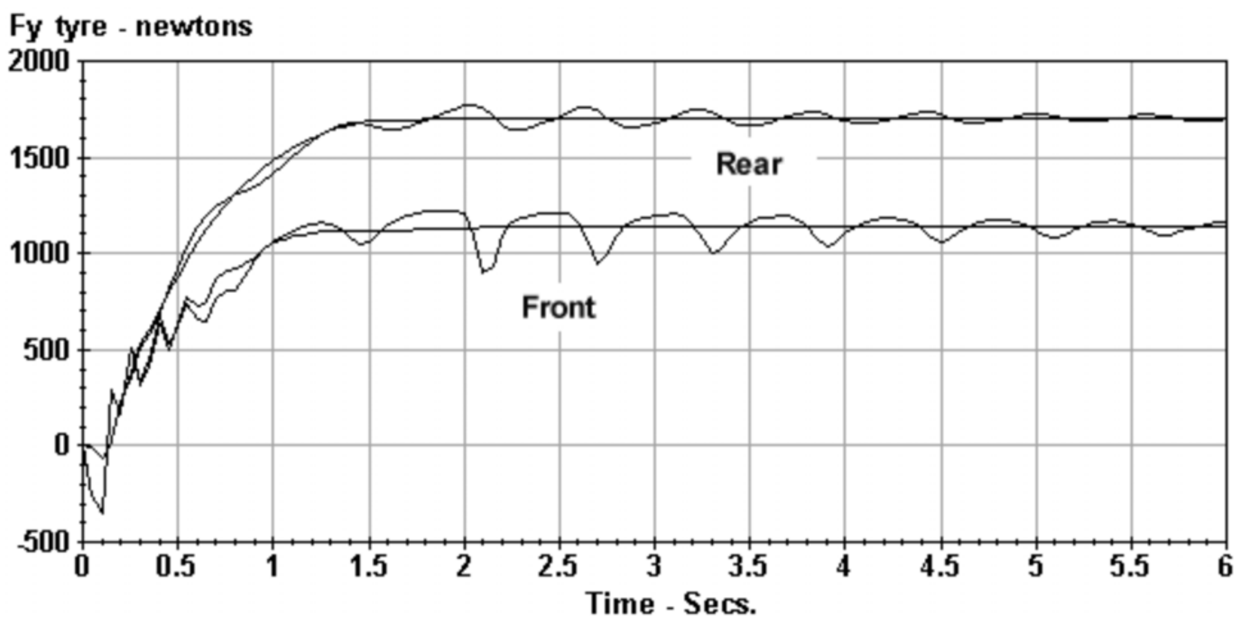



Fig. 6.29 **Lateral tyre forces. Even though it is the rear that is under-damped there is more variation on the front.**

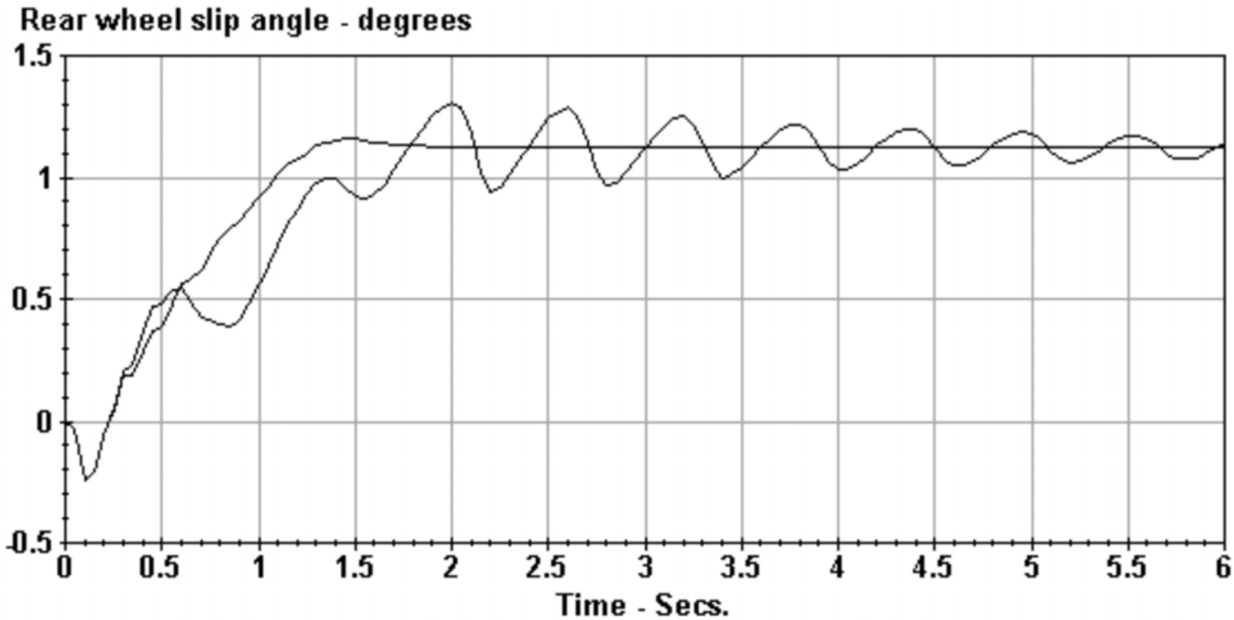



Fig. 6.30 **Rear wheel slip angle which is similar to the yaw angle of the bike.**

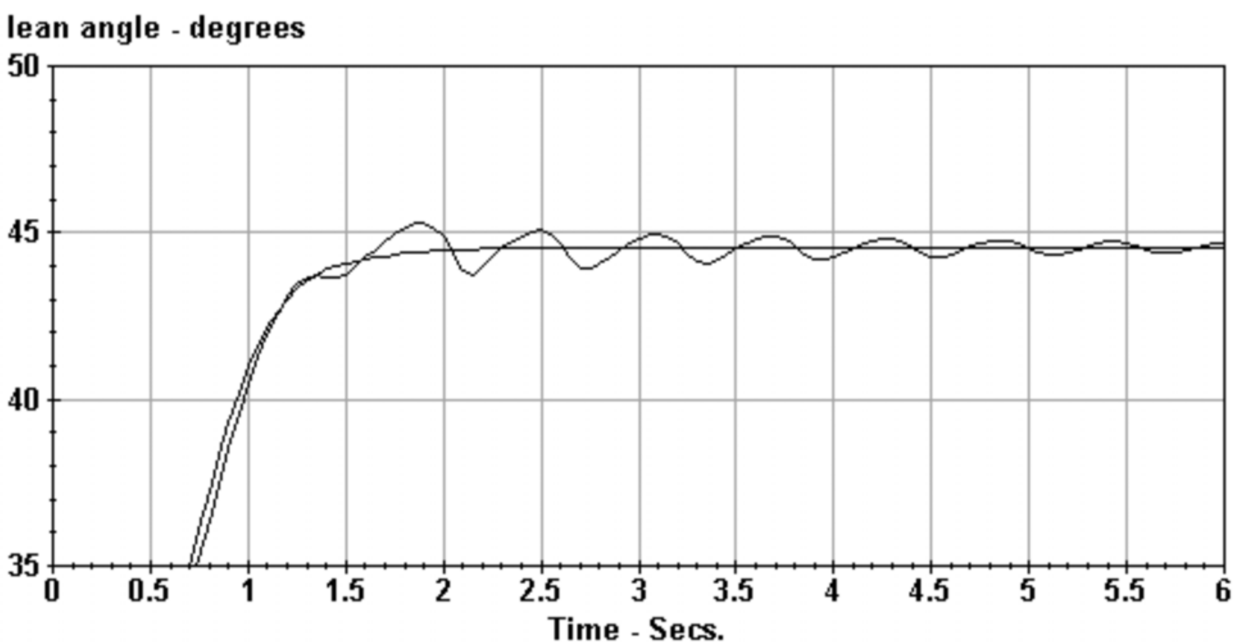



Fig. 6.31 **Lean angle, the initial peak variation is about 5% of the steady lean angle of 44 degrees. The weave has nearly died out after 6 seconds.**

## Spring rate and total wheel travel

We have to consider these together for the simple reason that the more wheel travel we have, the softer the springing we can use. Over the years, wheel travel has increased steadily to the point where motocross machines have 250 – 300 mm.. This has enabled riders to negotiate rough terrain much faster for two reasons:

The greater displacement can absorb larger bumps while softer springing imparts less movement to the sprung part of the machine, so enhancing control and comfort and thus reducing rider fatigue.

The wheels are kept in closer contact with the ground, enabling more power to be transmitted (at the rear) and giving better steering (at the front).

However, extreme wheel travel may entail both mechanical and functional problems: in the first case, too much variation in chain tension or drive-shaft angularity, in the second, instability as a result of gross variations in steering geometry, and ride-height changes with different loads. Since variation of the load (rider's weight, passenger, luggage, fuel load) occurs mainly at the rear, this alters the rake and hence the trail.

In an attempt to improve the situation, some designers use progressive-rate springing, by means of either variable-pitch coil springs or a linkage that progressively increases the ratio of strut movement to wheel movement, so stiffening the effective rate. This works well in motocross (where the main hazards are rough terrain and jumps) but may be a mixed blessing for road racing because the low initial rate leads to a disproportionate amount of the available wheel travel being taken up by cornering loads, leaving less travel and a higher spring rate to cope with any bumps. Similar considerations apply to a tourer, where the appreciable extra load of a passenger and luggage uses up the soft springing, leaving only the hard to handle bumps. In this case, though, the increased static load calls for a higher spring rate – but this can still be achieved only at the expense of movement and a change in attitude, unless the static ride height or spring preload are adjusted.

## Ride height and preload

Most suspension struts incorporate an adjustment for the initial loading on the spring (preload). If there is some displacement in these struts under static load, then this adjustment will alter the ride height and can be used to compensate partially for different loads. Note that this displacement is not the same as the spring sag mentioned above when discussing frequency, because some initial compression is given to the spring when it is fitted to the suspension strut itself. Some years ago, it was common to preload the suspension so that there was little or no displacement under static load and some force was needed to start the movement. The reason for this was that many riders found an improvement in handling, albeit with a harder ride, when they adjusted the preload to the hardest setting. This was a classic case of treating the symptoms rather than the illness, for the problem was twofold:

- Inefficient dampers, unable to control the available movement properly, hence it was helpful to restrict the movement (a self-defeating approach).
- Insufficient stiffness in the rear or front fork, in which case preload tended to stiffen these components.

It was these two shortcomings that gave rise to the popular misconception that hard springing is necessary for good handling. Besides giving an uncomfortable ride, this approach restricts usable suspension to ***raised*** bumps, with the springs preloaded, we are no better off in dealing with ***hollows*** than we would be with a rigid frame. Mercifully, the current approach is to have stiff pivoted rear forks, efficient damping and softer springs adjusted to allow some extension of the struts on hollows. Ken

Sprayson, designer of the famous Reynolds racing frames, used to specify one-third of the available wheel travel for extension (sometimes called sag or static sag) and two-thirds for compression, which seems as good a starting point as any for most purposes. See the end of chapter 9 for more details of some additional effects of spring preload.

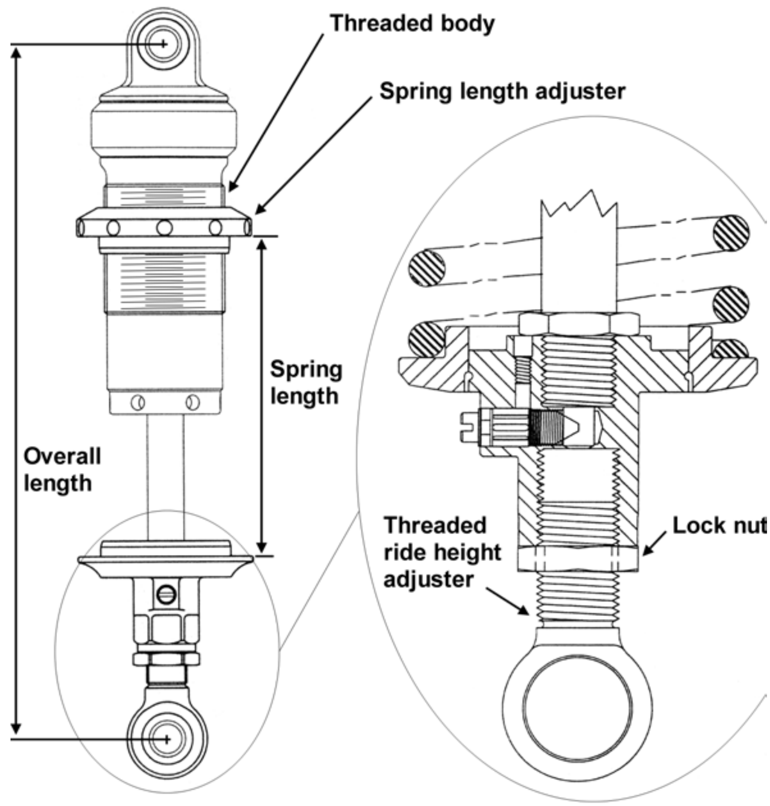
**Typical of modern rear damper units, this Penske shock shows the sag and ride height adjustments commonly available.**

**Decreasing the installed spring length with the threaded adjuster ring will both increase rear ride height and reduce the static sag (or increase pre-load).**

**On the other hand the ride height adjuster will only change the overall shock mounting length, and so adjusts the ride height independently of any thing else.**

**Various combinations of ride height and sag are possible by appropriate settings of these two adjustments.**

**Damper stroke length is a parameter that usually requires a different damper unit.**




Modern rear suspension systems generally have independent adjustment for installed spring length and over-all shock length which are used for fine tuning the attitude of the bike. Raising the rear ride height has several effects on the bike.

- Reduces rake angle and trail, thus quickening the steering.
- Raises the CoG which increases load transfer during braking and acceleration.
- Alters the relationship between rear axle height, swing-arm pivot height and drive sprocket height which affects the anti-squat performance of the bike as explained in detail in chapter 9.
- It is often claimed that the frontal weight bias is increased, whilst this is true, the effect is often exaggerated. For practical changes in rear ride height the effect on static weight bias is quite minor.

## Wheelbase

Wheelbase can have quite an effect on suspension behaviour in certain circumstances, as we saw in fig. 6.22. In conditions that cause pitching of the machine then in general the pitching angle will be inversely proportional to the wheelbase. That is if we double the wheelbase then we will cut the pitching effect in half, additionally the longer machine will almost certainly have a higher sprung pitch moment of inertia further dulling the response.

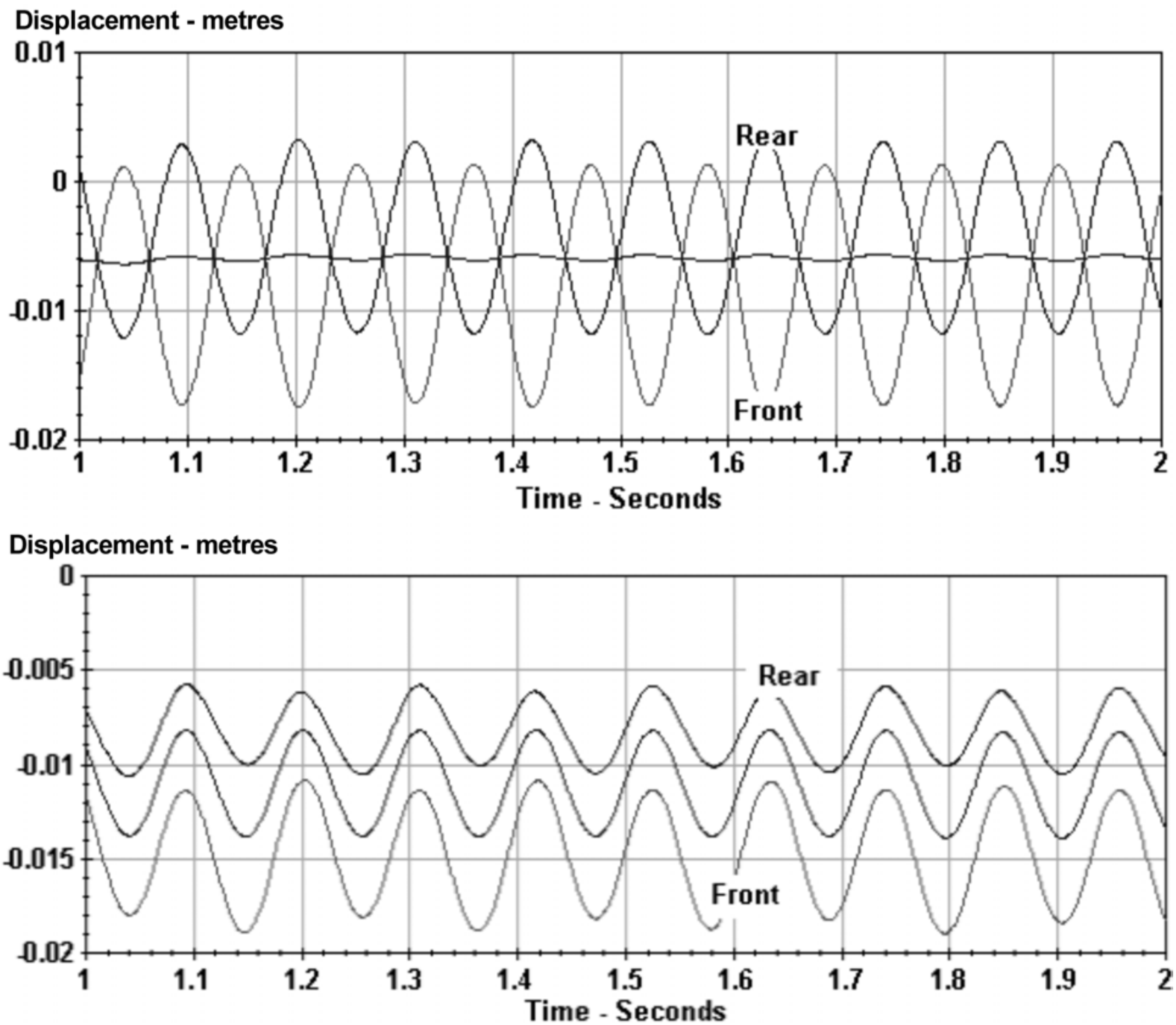



Fig. 6.32 **These two sets of curves demonstrate wheel base effects on a corrugated road. In each case the middle curve is the vertical displacement of the CoG, the other two lines are the vertical displacements at the front and the rear.**

**The first set of graphs is with a 1.5 metre wheelbase and 3 metre bump wavelength. The second set is with a wheelbase equal to the bump wave-length at 1.5 metres. The road speed was adjusted to keep the bump frequency identical in each case.**

There is another less obvious wheelbase effect which combines with surface conditions to greatly influence ride quality. If we are travelling over a series of bumps as on a corrugated dirt road for example, then the movements transferred to the rider will depend heavily on whether the wheel-base is a multiple of the bump wave-length, or an odd multiple of half of the wave-length. Fig. 6.32 demonstrates this perfectly.

The curves with equal wheelbase and bump wave-length show the bike with the back, front and CoG moving up and down in concert. In this case each wheel is travelling over the same part of adjacent bumps at the same time. However, when the bump length is twice the wheelbase then when the front wheel is at the top of a bump the rear wheel is at the bottom, hence the vertical movements at the front and rear are in opposition. At the CoG, roughly halfway between front and back there is very little vertical movement.

From this it would be easy to conclude that this case would be much more comfortable for the rider, but we rarely get anything for nothing and the price is a pitching motion. If the front is going up when the rear is going down, and vice-versa, then the vehicle must be pitching back and fore. These are the two extreme responses determined by the wheelbase to bump length ratio, other values of this ratio will produce a ride quality somewhere in between. That is, some pitching combined with some vertical bump. When designing a motorcycle and its suspension we have no control over the wavelength of any series of bumps that we might hit in the future and so this is a not a parameter that we can use as input for selection of wheelbase.

## Load compensation

Passengers and luggage can amount to a sizeable percentage of the all-up weight of a motorcycle and so it is desirable to be able to compensate for varying static loads to ensure that both comfort and roadholding don't suffer.

An ingenious development for touring machines – designed to maintain ride height and suspension travel constant regardless of load – was the Boge Nivomat self-levelling strut, introduced on BMWs in the early 1970s. This is an extremely sophisticated hydropneumatic design using nitrogen as the springing medium and oil for damping. Unlike earlier self-levelling systems on cars, it requires no hydraulic pump and attendant pipework. Instead, an internal pump is energized by the suspension movement itself; this raises the internal pressure to bring the ride height up to a predetermined level. The pressure increase also causes an increase in the effective spring and damping rates. This system can be used either on its own or in conjunction with auxiliary springs. The internal volumes and areas can be tailored to suit the characteristics desired in any particular machine.

It is to be hoped that this type of system will receive increasing consideration, as it solves many of the suspension problems arising from the relatively large variations in wheel loading common on a road motorcycle, particularly at the rear.

S & W, a former American manufacturer of suspension units, specializing in gas shocks, used to market systems with gas springs and an onboard air compressor to allow for simple adjustment to cater for changes in load or terrain.

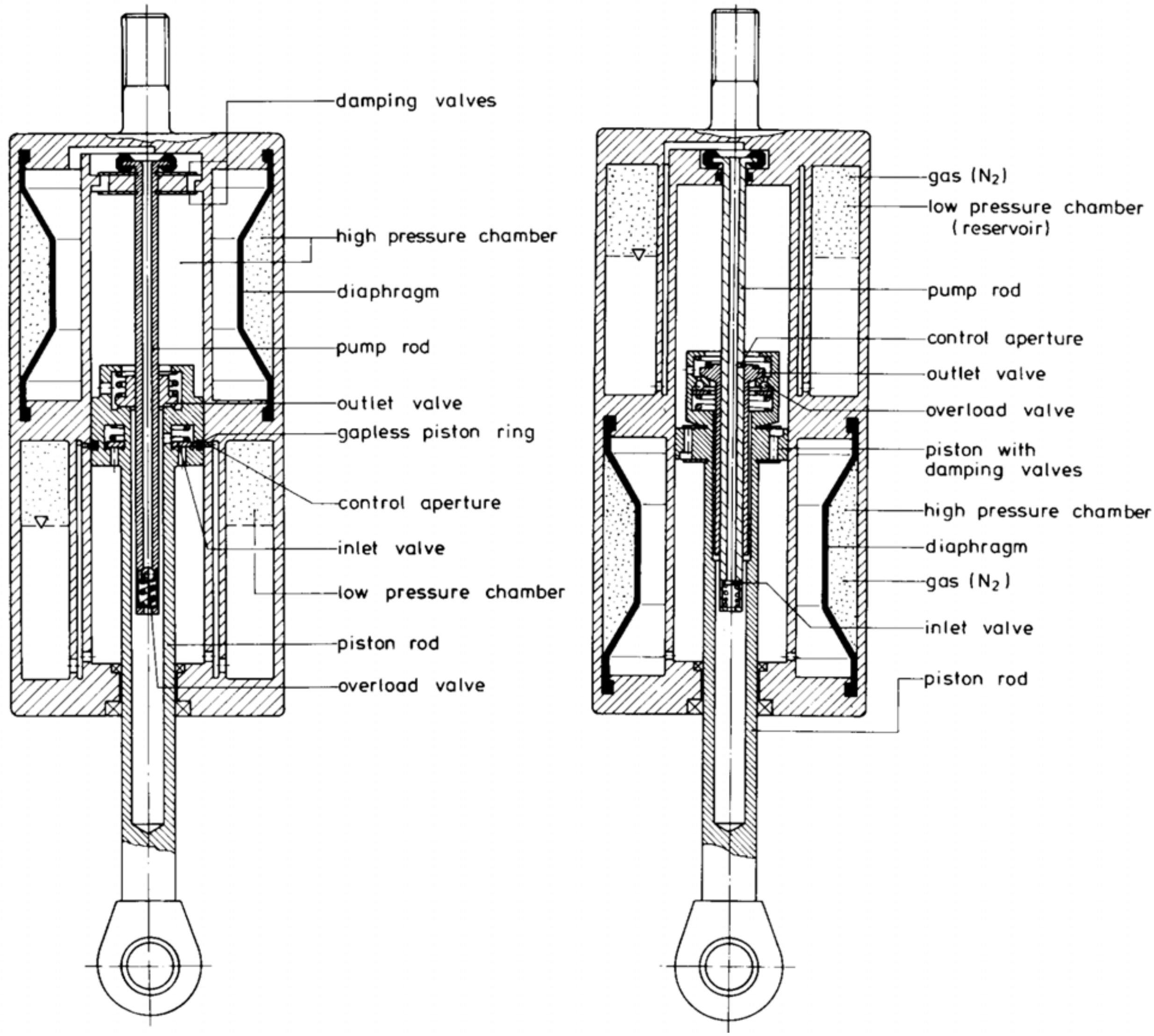



Fig. 6.33 **Boge Nivomat self-levelling suspension strut. *Left* is a twin chamber version with a single chamber to the *right*.**

Cruder methods of load compensation have been used almost since the introduction of rear springing. Less attention has always been applied to the front end in this regard. Cam adjusters which just simply raise the lower spring seat have probably seen the most wide spread use of all systems. Basically all these do is raise the rear slightly to partially counter any added load. Unfortunately they do nothing to raise the spring rate to maintain the desirable suspension frequency. Progressive rate springing can be useful here, because the increased load will take the spring into a higher rate position.

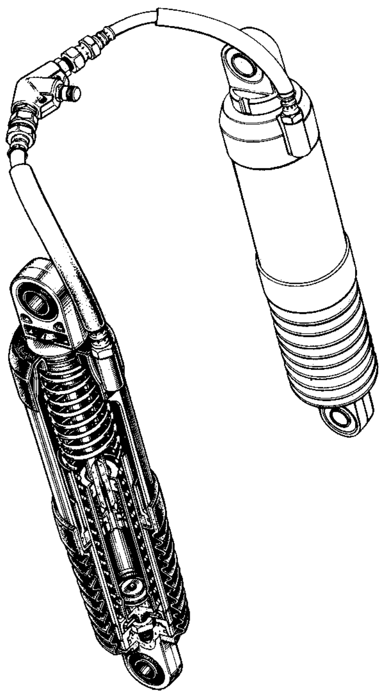


Fig. 6.34 **Used on various touring models in the mid 1980s. This Kawasaki design used twin air-adjustable rear shocks on the rear. The internal pressures were equalized by means of the connecting tube, the centre of which held a valve so that pressure could be adjusted to allow for differing loads. Normal coil springs were also fitted to carry some of the load.**

## Braking and acceleration

Discussed in more detail in the chapter on anti-squat, these longitudinal accelerations cause a load transfer from one end to the other. Depending on geometric aspects of the suspension design, which affect anti-dive and squat, it is usual for these load transfers to extend or compress the suspension medium. As shown in chapter 9 it is not unusual for the front forks to be subject to a load under braking of three times the static load. Loading variations like this place heavy demands on the suspension system, which as we've already seen has to also cater for increased compression when cornering as well as its more obvious function of bump absorption and trying to maintain an even tyre force on the road. With such diverse requirements it is a wonder that suitable compromises can be made to enable the suspension system to function as well as it does.

# Lateral suspension

When we hit a bump whilst lent over in mid corner, the bump force will probably be approximately vertical. Therefore, only a component of this force will act in line with the suspension and the rest will act at right angles or laterally with respect to the bike. Fig. 6.35 shows the case for a 1 G. turn with a lean angle of 45 degrees. It is not just single bumps that do this, of course, but a bike is always subject to a variety of disturbances, which when leant over will produce continuous variation in the tyre to road force. We have seen earlier that this load variation is detrimental to traction and so is of the utmost concern in racing circles.

As tyres and hence cornering speeds have improved over the years this problem has assumed greater importance, especially as power outputs have increased also. Transmitting this power effectively requires that the tyre maintains good contact with the ground, this is achieved by having the minimum possible variation in the tyre/road contact force. Ironically the gradual trend towards much stiffer chassis over the past two decades, to generally improve handling, has made this problem more obvious and we see experiments with “controlled flex” or lateral suspension as a counter-measure.

At the moment this seems to be a little understood and very controversial topic, without any real consensus about whether a chassis should be as stiff as possible or have some built in compliance designed to reduce the tyre force variation. Some laboratory and track testing has indicated a reduction in this variation, to the benefit of traction in general, by the introduction of some extra flexibility. However, races are not won by technical considerations alone, the rider has to be accounted for also. It is a fact that some riders feel more at home on a bike with a rock solid feel to it, whereas others are happiest with a machine that appears to “give” a little. What is certain though, is that any deliberately introduced compliance must be below a level that would introduce other instability problems, such as wobbles or weaves. Prior to about the 1980s many chassis fell below that level and benefited greatly from increased chassis stiffness, however beyond a certain point it is doubtful if increased stiffness would be noticeable. It appears that some modern chassis may have exceeded that point possibly to the detriment of traction.

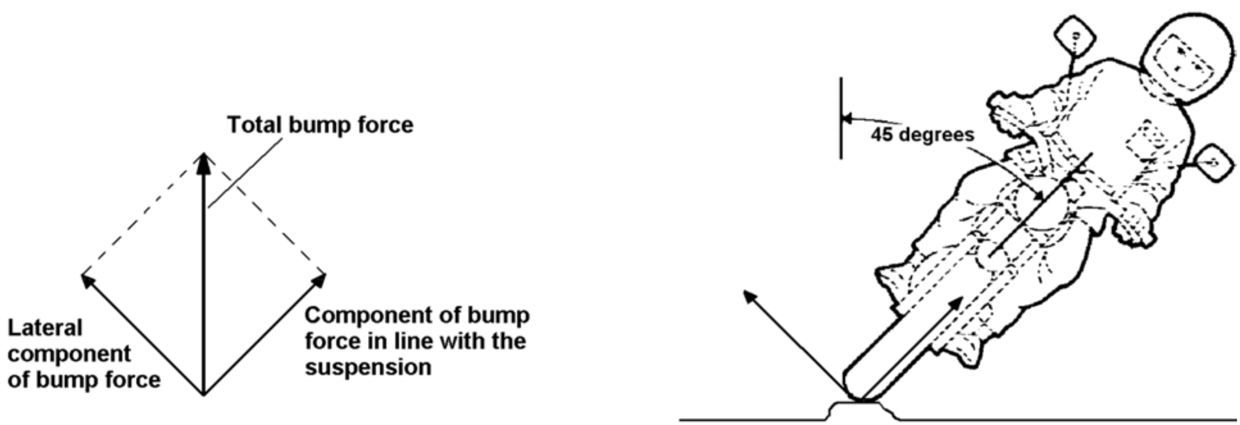



Fig. 6.35 **At 45 degrees lean a vertical bump force will create equal forces , one in line with the suspension movement and the other at right angles to it. These forces will each be 71% of the vertical bump force.**

Let’s look a little closer at this idea of lateral compliance and see if it has anything to offer.

At the 45 degree lean angle of the example in fig. 6.35, the extra force (due to a bump) in line with the suspension will also be equal to the lateral force, each being 71% of the total bump force. This fact leads us to the conclusion that the suspension will now compress by a lesser amount than that which would occur if the bike were upright when traversing the same bump, other factors being equal.

Other factors are not equal as fig. 6.36 shows. For a given vertical tyre displacement, if we introduce some lateral movement at the tyre then we will reduce the tyre motion in-line with the bike by an equal amount. The lateral motion primarily comes from a roll movement of the whole motorcycle, as shown, and lateral and torsional chassis compliance, which in practice depends on numerous factors.

Thus there are two factors which tend to reduce the displacement of the suspension unit, one being the reduction of the in-line component of the vertical force and the other due to the lateral displacement.

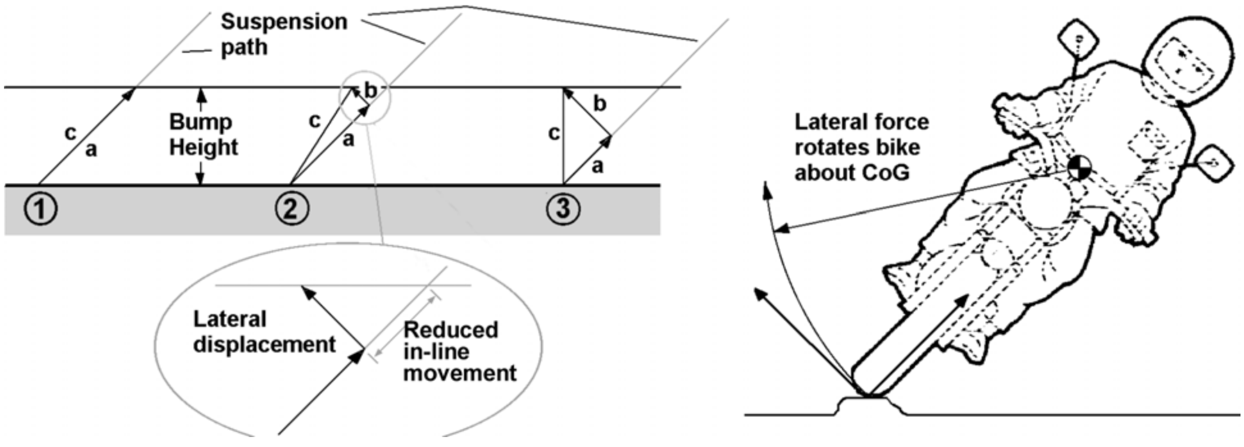



Fig. 6.36 **The sketches to the left show how the tyre movement needed to surmount a bump is changed by introducing lateral movement.**

**1. Shows the case where all movement is along the path "c" which is co-linear with "a" the suspension compression direction.**

**2. Shows how a small amount of lateral displacement "b" changes the tyre path, and the expanded view shows that the in-line movement reduces by the same amount.**

**3. When the lateral motion is equal to the in-line movement ( "a" = "b" ) then the wheel path is vertical when at a lean angle of 45 degrees, .**

*Right.* **The lateral impulse at the tyre tends to rotate the bike about its CoG, thereby giving some lateral displacement even without any lateral structural compliance.**

In fact calculations show that this roll response "absorbs" most of the bump disturbance when lent over in a curve. The lateral motion at the tyre due to this roll, and compliance, is extremely important to the whole suspension action when hitting a bump whilst leant over, as we shall see.

Fig. 6.37 shows the end view of an idealized simple model of a motorcycle at a lean angle of 45°. It has a mass of (**M**) centred at the CoG and a roll polar moment of inertia about the CoG of (**I**). The CoG is a fixed distance of (**r**) from the tyre (that is; there is no suspension movement). If this idealized machine is subject to a vertical bump force (**F**) then two types of motion are created;

A vertical movement of the CoG shown as (**v**).

An angular motion about the CoG shown as (Θ) which leads to a lateral displacement at the tyre of (**l**)

Therefore the overall motion of the tyre will be as shown by the grey line in sub-sketch **(A).** The detail of this motion is controlled by the values of **M, I, r** and the angle of lean (This is just a standard physics problem and a full explanation can be found in many text books).

For our current purposes the ratio of the lateral displacement to the vertical is of interest and for a given angle of lean it can be shown that:

$$\frac{l}{v} \approx \frac{Mr^2}{I}$$

Substituting motorcycle values for these parameters we find that the lateral motion could typically be around five times that of the vertical, even more when chassis flex is accounted for.

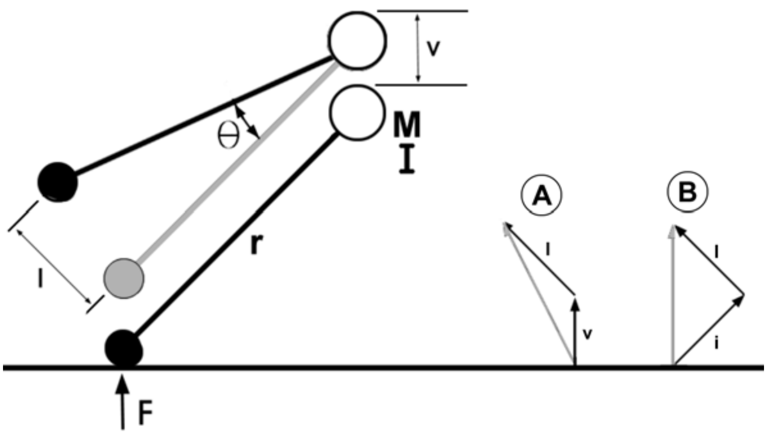



Fig. 6.37 **An idealized motorcycle leaning at 45° is represented by a tyre connected by a rigid rod of length (r) connected to a mass (M) with a moment of inertia of (I). A bump is shown as a vertical force (F).**

**The sketch at (A) shows the motion of the tyre, (v) is the vertical movement and (l) the lateral. (B) shows the motion if a sufficiently compliant suspension was introduced to give a totally vertical path, (l) being the lateral component and (i) the in-line component.**

So it would seem that the problem is not so much one of needing to introduce more lateral displacement as often suggested, but one of restoring lost damping and thus in-line movement. **(B)** shows how enough in-line movement converts the wheel motion to purely vertical. This in-line motion is of course provided by the normal suspension system. The ratio of the suspension movement to lateral motion is affected by many parameters, e.g. **M, I, r** as before but also the unsprung mass and compliance of the suspension and chassis. In most practical cases, at high lean angles, the lateral motion will exceed that due to suspension movement. We have already seen that this lateral motion will reduce the suspension movement, for a given size of bump. If we consider the basic requirements of dissipating the effects of a bump it will become clear that reducing the suspension movement is highly detrimental.

At the most basic level, when we hit a bump some of the forward kinetic energy of the motorcycle is converted into energy acting in a vertical direction. Without any damping at all, this will cause a continuous oscillation on the tyre and suspension system, the introduction of damping will dissipate this vertical energy as heat and kill the oscillation. In practice this damping is mainly provided by tyre and suspension damping, although the suspension damping normally overwhelms that due to the tyres.

Fig. 6.38 shows the response to a small sinusoidal shaped single bump (with the bike upright), in terms of tyre to road contact force, with three different values of damping. The bump is only 0.01 m. (10 mm.) high by 2 m. long and at 100 km/h. it takes approximately 0.07 seconds to pass over it. The physical parameters are fairly typically, a wheel load of 1250 N (127 kgf.), an unsprung mass of 20 kg., suspension frequency of 2.5 Hz. and a wheel hop frequency of 14 Hz. The damping values considered are:

- Tyre damping only.
- Tyre damping plus a typical degree of suspension damping, with a rebound to bump ratio of 4:1.
- Tyre damping with the suspension damping reduced to 0.25 of that above.

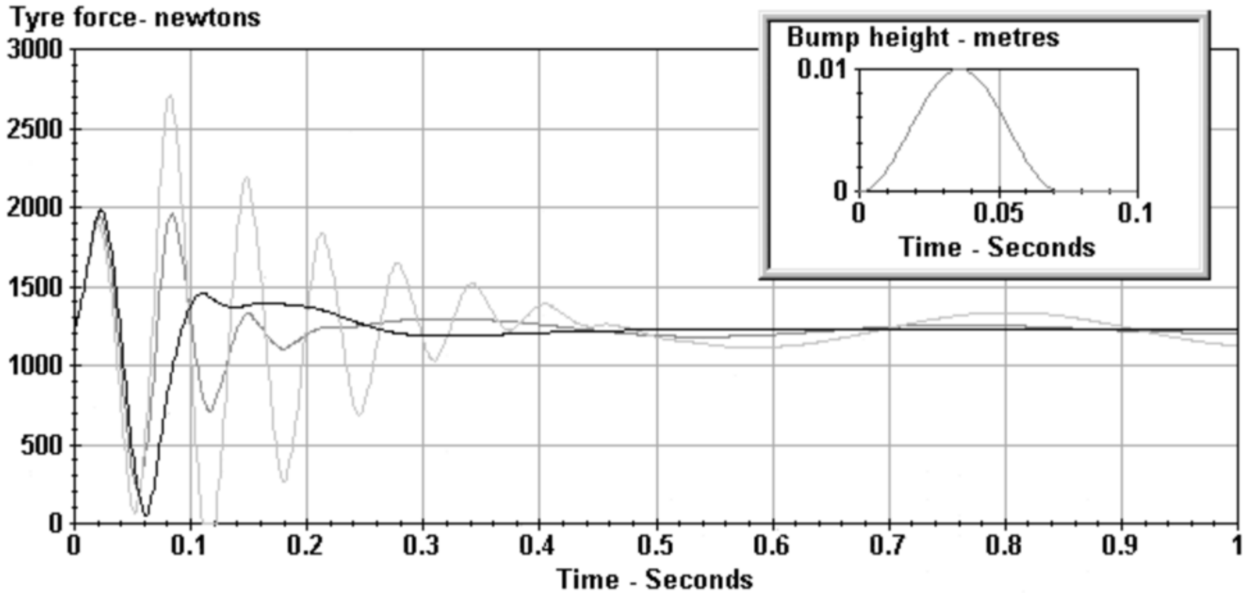



Fig. 6.38 **Tyre force response to a small sinusoidal bump (shown in the inset) for various values of damping. The light grey curve is with tyre damping only, the darkest is with typical suspension damping and tyre damping, and the other is for a quarter of the previous suspension damping. The oscillation visible prior to 0.5 seconds is due to the unsprung mass bouncing on the tyre (wheel hop) and the much lower frequency motion after 0.5 seconds is due to the sprung mass moving on the suspension springs.**

We can see that the initial force peak is similar in all three cases, because, as we have seen in the tyre chapter, the initial impact of small bumps is mainly taken up by tyre deformation. However, there is considerable difference after the bump has passed. With the minimal damping due to the tyre alone there is considerable force variation at the wheel hop frequency lasting for close to 0.5 seconds. In the case of typical suspension damping this is almost smoothed out by about 0.1 seconds, and with a reduced suspension damping, by 0.25 seconds. This cyclic tyre force variation is bad for traction, and so it follows that reducing damping from an optimum level is also bad for traction.

Figs. 6.36 and 6.37 have shown that any lateral motion at the tyre, when leant over, will reduce the in-line or suspension movement. If the suspension movement is reduced then so too is the damping energy. In general the energy absorbed by a damper depends on the square of the displacement and so if the suspension movement is cut by half then the overall damping energy will be reduced to a quarter. In fact the movement will probably be reduced much more than by one half. So if the suspension system is adjusted for good traction when upright the damping is certain to be far too low when leant over on the same terrain. Allowing more lateral movement, for example by introducing lateral frame compliance would therefore seem to be counter-productive, as it would further reduce the in-line motion and damping along with it.

However, if by increasing chassis compliance we also introduced additional damping, then the overall effect would be beneficial provided that the chassis flexibility was still low enough to avoid instability problems. In fact, practical laboratory measurements show that a motorcycle chassis does not act as a simple elastic structure, but shows a hysteresis characteristic similar to that shown in the tyre chapter. That is; during a loading – unloading cycle there is a nett energy loss, or damping, as shown in fig. 6.39.

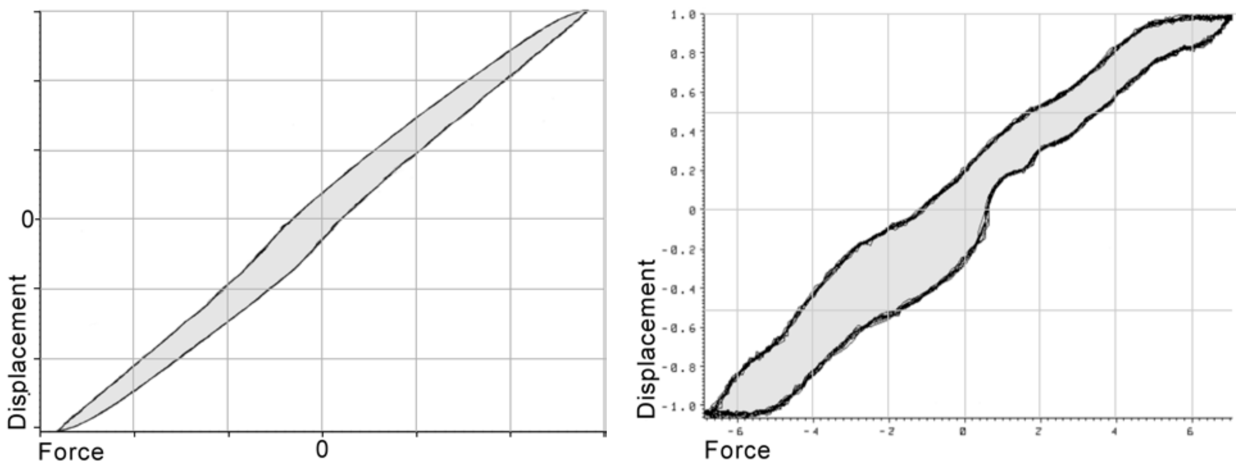



Fig. 6.39 **These force-displacement curves are measured from actual motorcycles. At left is a complete loading and unloading cycle covering positive and negative values. This shows the torsional characteristics from axle to axle. The second curve to the right show the stiffness of a complete wheel and swingarm assembly from a used sport bike. Despite the very irregular shape these characteristics are very repeatable from one loading cycle to the next. Slop and stiction etc. are causes of the strange shape. In both of these cases the main point to note in the current context is that they exhibit a hysteresis effect as explained in the tyre chapter. The shaded area represents a loss of energy that occurs during each complete loading cycle, damping in other words.** (data courtesy of Dr. Robin Tuluie and MTS)

This then, is where any benefit of increasing frame lateral flexibility comes from, an increase in the total system damping to help replace the lost damping in the suspension system. This will probably follow the same square law relationship mentioned above. If the chassis is allowed to dynamically flex twice as much then the damping energy dissipated will likely increase by a factor of four. Looking at it from the opposite perspective; if we increase chassis stiffness, beyond the value needed to avoid instabilities and any other poor handling traits, then we rapidly decrease the amount of damping from this source. Tyre grip when leant over will suffer as a result. Any practical chassis will have various sources of compliance and this can lead to various and different distributed resonances, the relationship between these resonances and the wheel hop frequency can cause interference effects which may increase or decrease the wheel hop. We are currently at the very early stages of beginning to understand this in sufficient detail to apply it to actual designs, without a lot of testing of alternative frame configurations.

So if some lateral compliance is necessary we need to consider how best to achieve it. There are countless design possibilities for the manner in which lateral suspension could be implemented, but the easiest and most obvious are also probably the worst. For example, at the front end we could just introduce additional lateral flexibility to the forks, but as shown in fig. 7.1 this will introduce an additional camber angle to the tyre. Spurious and probably undesirable steering impulses would result with implications for handling and stability. At the rear some flexibility could be built in to the swinging arm mounting area, but as well as lateral movement at the rear we would get wheel yaw attitude changes, also leading to spurious steering inputs.

At the rear we also have the problem of the large chain forces waiting to take unfair advantage of any extra compliance. We really need to ensure that only true lateral motion is allowed. This could be done by just allowing the wheel or rim to move laterally, with the remaining chassis parts as rigid as possible, fig. 6.40.

We have seen that the fundamentals of dealing with bumps whilst cornering mean that the suspension units experience considerably less bump movement than would be the case if the bike remained in an upright position. This greatly reduces the rate of energy dissipation in the dampers. To counter this effect we need to take steps that will maximize suspension displacement, these include:

- Soft suspension springs – but this is compromised by all the other demands on the suspension, such as handling bumps when upright, braking and driving dive and squat etc.
- Low unsprung mass – this is beneficial in general for most suspension demands.
- Increased rider lean-in – this will keep the bike and suspension travel more vertical.
- Lower the CoG height – the formula on page 6-45 shows that this has a squared effect. As shown throughout this book there are many conflicting demands on whether the CoG should be high or low and this one is just another in the mix.
- Use active suspension – this would be the ultimate way of reducing tyre force variation, and the subject is covered in more detail in chapter 18.

Once we've tuned the above parameters to ensure the maximum suspension movement and damping from the suspension units, there are a few more methods to increase the overall damping.

- Increase tyre damping – an obvious suggestion but it means higher tyre temperatures.
- Dynamically adjustable dampers – a low power alternative to full active suspension, also explored in chapter 18.
- Introduce lateral damping – as explained above this can be a result of allowing an optimum degree of lateral structural compliance. We need to consider how to construct the structural elements to maximize the inherent damping.

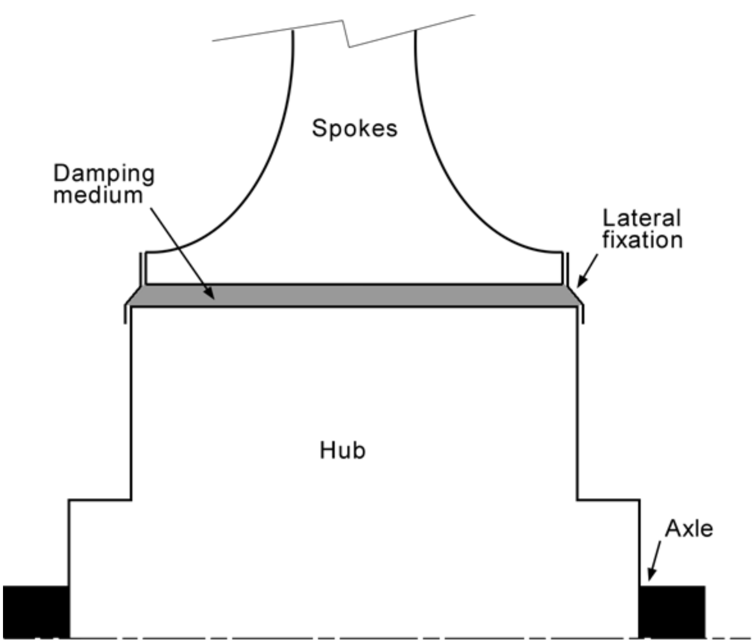



Fig. 6.40 **A preliminary proposal by the author for a two piece wheel with built-in lateral damping, which may be worth some consideration.**

**This cross section shows a central hub which contains the bearings and provides disk and sprocket mounting in the normal fashion. The outer part of the wheel consists of the rim, spokes and a central ring with an inner diameter slightly larger than the inner hub. These two parts are joined together by some fixation method which allows some lateral compliance but gives a rigid radial and torsional support. The annular gap is filled with some form of damping medium. This manner of providing lateral compliance and damping only entails a small weight penalty, but on the other hand gives a minimal effective lateral unsprung mass, which is important to the reduction of dynamic tyre loads.**

**Such lateral damping may also have beneficial effects on weave stability, but analysis and testing would be needed to confirm its worth.**

In this section we have been considering the problem of suspension under cornering conditions as being one of tyre grip, obtaining the maximum traction. There is also the issue of ride comfort and as we have seen earlier in this chapter, ride comfort and the need to reduce tyre force variation are sometimes at odds with each other. Ride comfort requires a slow response from the wheel so as to pass the minimum possible accelerations through the suspension to the sprung mass of bike and rider. On the other hand, we need a rapid wheel response to reduce the dynamic loading on the tyre.

## Summary

Motorcycle suspension is a coupled (back and front) dynamic system comprising springs, dampers and so-called sprung and unsprung masses. Whilst the basic layout is quite simple the dynamic interactions with the overall handling, stability and comfort of the machine are extremely complex. As shown in a previous chapter, tyres are the most important element in the suspension system.

In general we benefit from softer suspension but this must be balanced against the available movement and geometry changes. The need for appropriate behaviour under braking, acceleration and cornering has to be taken into account also. The many diverse requirements make it impossible to design a "perfect" setup for any particular bike, compromise is inevitable. The ideal for comfort may for example lead to weaving under rapid corner lean-in or perhaps excessive dive under braking.

Even though suspension movement takes place in the centre plane, the tight integration of motorcycle dynamics can lead to responses about other axis, yaw and roll. Poorly set suspension can reduce roadholding and/or allow these responses to become dangerous instabilities.

The necessity to lean when cornering introduces bump forces at right angles to the plane of suspension movement, which conventional suspension is ill equipped to handle. Damping is thereby reduced leading to increased wheel hop. Some attempts have been made to introduce a degree of lateral structural compliance to address this issue, but as yet there is no generally accepted solution to this problem. Realizing that the basic problem is one of insufficient damping, and not one of insufficient lateral motion might help point the search for improvement in the right direction.

# 7 Front suspension

Over the past years we have heard much about a new generation of front suspension systems, usually with the promise that soon all future motorcycles will be built this way or that. However, as with most engineering ideas in general and motorcycle ideas in particular, there is little that is really new. Interest in a wide range of steering designs is as old as motorcycling itself. Although, only a small number of designs have ever been accepted for quantity production. The girder fork was the early favourite, being used by most manufacturers at one time or another.

There were always dissenters though, the Neracar used a hub centre design and O.E.C. had an unusual virtual steering-axis system, both were said to give excellent results by contemporary standards. By the early 1950's, hydraulically damped telescopic forks were becoming well established, mainly due to their improved ride over the undamped or friction damped girders. Some manufacturers, however, were even then aware of the deficiencies of this design, and used some form of link forks, e.g. BMW, who had been quite prophetic in their use of telescopic forks from 1935 until 1955 went for the Earles fork. Although they returned to telescopic forks for marketing reasons in 1970. Ariel on the other hand used pressed metal trailing link forks on their Leaders and Arrows. NSU and Moto Guzzi both achieved great success in GP racing using leading link forks.

Despite the real or imaginary defects of telescopic forks, it must be said that they have been fitted to the greatest number of motorcycles yet produced, and have been virtually unchallenged in production terms since about 1950. Although many scooters and other low cost machines have been fitted with either leading or trailing link designs.

In the recent past (and since the first edition of this book) only BMW and Yamaha, among the major manufacturers, have been willing to test the market with an alternative. Yamaha tried a double link design, but on one model only, the GTS. On the other hand BMW have changed over completely to the telelever, which retains the neat appearance of telescopic forks.

There are many different types of motorcycle front suspension that have been tried, with varying degrees of success or failure. Broadly speaking these can all be separated into two main groups.

- Those that require a conventional head stock to define the steering axis, and mount the fork.
- Those that do not use such a frame mounted head stock.

Included in the first group are the ubiquitous telescopic fork, the leading and trailing link forks and the old style Girder fork. The second group contains a very diverse range of designs. Many fall into the category of what are commonly called hub-centre steered, a term which I personally don't like because it is very non-descriptive and is often used for any and all designs without a steering head. Also in this group must be included a type which not only scorns the head stock but also has no fixed mechanically defined steering axis. The old OEC is an example of this genre.

## Head stock mounted forks

All these types of front suspension/steering systems have a common feature, they all mount on and steer through a steering head stock. The sketch in fig. 7.1 shows how any lateral flex in the fork legs allows the tyre contact patch to move away from the steering axis. This really matters, because wobbles can be caused or greatly increased by this misalignment, both on fairly smooth roads at a particular speed, and at any speed on a more bumpy surface. We have all experienced this to some degree, sometimes it amounts to nothing more than a minor handle-bar shake, but in far too many cases it can develop into a frightening tank slapper. There are many other problems that stem from the use of head stock mounted forks, but this potential for lateral displacement of the tyre contact patch is possibly the single most important one. Another disadvantage is the large amount of leverage that can be exerted on the steering head itself, particularly from braking. This results in very large forces which have to be resisted by a strong and hence relatively heavy frame.

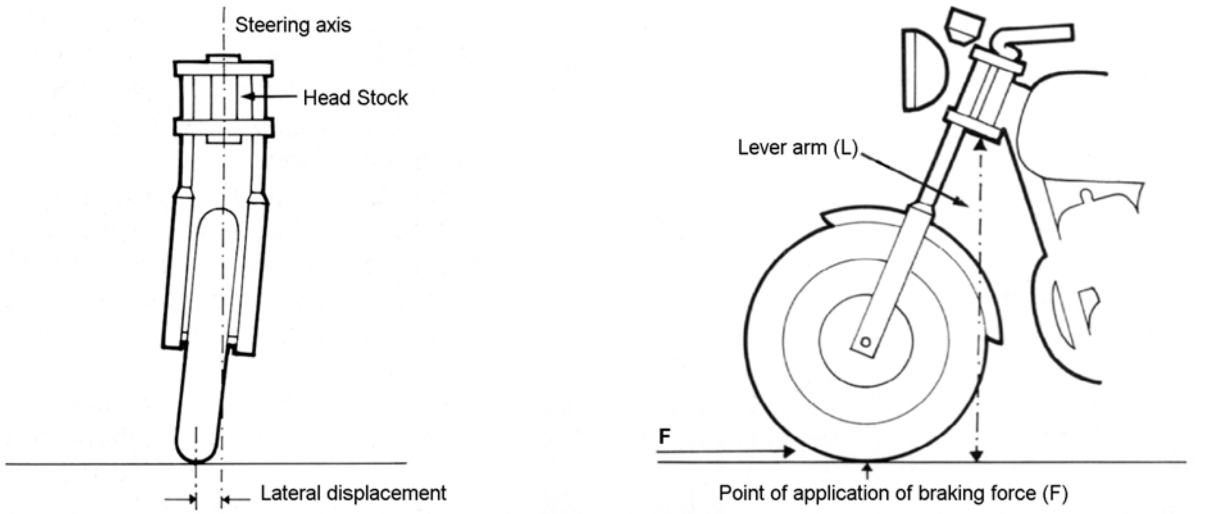



Fig. 7.1 **With such a large lever arm these types of deflection are present to some degree in all head stock mounted forks.**

## Telescopic forks

Except on some scooters and small utility machines, the telescopic front fork is almost universal. Its popularity cannot be justified on engineering design grounds, however, because it has many technically adverse features. Nor is low cost a valid explanation, as is sometimes claimed on the grounds that the main components are amenable to mass-production techniques. If cost was the decisive factor, it would be difficult to understand why many manufacturers have fitted link-type forks on their mass-marketed small commuter and shopping bikes, where a low selling price is even more important. No, the chief reason for the telescopic's long reign is more likely to be a collective fear in the marketing departments of the major manufacturers that the fashion-conscious enthusiast is not yet ready to accept a change from such a visually appealing fork. There are other factors of course, the telescopic fork has under-gone a very long period of development and evolution which cannot be matched by any of the alternative designs. In this modern world of high compensation payouts in liability litigation, this long experience establishes accepted practice which in a legal sense could not be claimed for any new design. This alone would make any manufacturer cautious about the basic design of such a safety related item.

The traditional telescopic fork comprises a pair of aluminium or steel sliders fitted over chromium-plated steel stanchion tubes clamped in yokes at top and bottom of the steering column. Although there was once a fashion for relatively large-diameter springs surrounding the stanchions, today's springs are usually of smaller diameter, longer and fitted inside the stanchions. An hydraulic damping mechanism is incorporated in the sliders and the damping oil also serves as a lubricant.

Over the past 15 years or so there have been huge improvements in the performance of high quality versions of these forks. Low friction seals, improved manufacturing tolerances and surface finishes and special coatings have all contributed to improving the breed. Tube diameters have increased, thus improving general rigidity which has also been helped in some cases by a changed design to what has become known as the upside-down fork. In this case the smaller diameter tube is the slider and is connected to the wheel axle. The outer tube is clamped in the supporting yokes. This design has the advantage that the slider can be longer, extending up inside the area between the fork clamps, hence giving more overlap and better support.

Although not new, this design was used by the BSA Bantam (amongst others) many years ago. The Frenchman, Eric Offenstadt, was probably the first in more recent times to apply this form of construction to a racing machine in the 1970s. His fork had the outer sliders and fork clamps all made as one piece, thus improving rigidity and reducing weight. The disadvantage here would be more difficult machining.

Let us consider some of the basic engineering problem areas inherent in telescopic forks:

- When the fork is fully extended there is minimum support for the sliders (because of the reduced overlap), so that the effect of the working clearance is considerably magnified at the wheel spindle.
- The sliders can move independently of one another except for the bracing effect of the wheel spindle at the bottom and perhaps a mudguard bracket or fork brace at the top.
- Considering the loads and leverages imposed on them, the stanchions are quite small in diameter (typically this was 35 – 38 mm on large machines, although that’s increased in recent years).
- Under braking this type of fork is usually subject to a great amount of nose dive.
- Because of the rake angle, bending loads are applied to the fork legs under static loading and this gives rise to stiction, which hardens the response to small bumps (see fig. 7.3).

These features add up to a fork that is relatively flexible in most directions, and as we have mentioned earlier, lateral flexibility can impair stability. However, this aspect have been vastly improved over the past decade. The upside-down (USD) fork in particular has improved stiffness in all directions, larger wheel spindles have become more common, often hollow to save weight. Pre-USD forks often had a mudguard bracket or fork brace above the wheel which added overall stiffness to the assembly, but the USD fork only has the axle to hold the two legs in alignment. Thus the size of the axle is more important in this case. High quality USD forks perform very well but can be quite expensive. For fashion reasons some cheaper motorcycles are fitted with USD forks of dubious quality with small diameter axles, often resulting in greater flexibility than the older types. Confirmation of this is easily obtained by holding the front wheel between your knees and twisting the handlebars.

The problem of the difference in overlap between the compressed and uncompressed states became much more acute when motocross machines started to drastically increase wheel movement. Over a relatively short period of time typical movements increased from a maximum of 100 – 125 mm. to the 305 mm. or so that we see today. A partial solution to this problem was to use an offset axle location which allowed the slider and stanchion to be longer. Fig. 7.2 shows how the longer stanchion can extend farther into the slider.

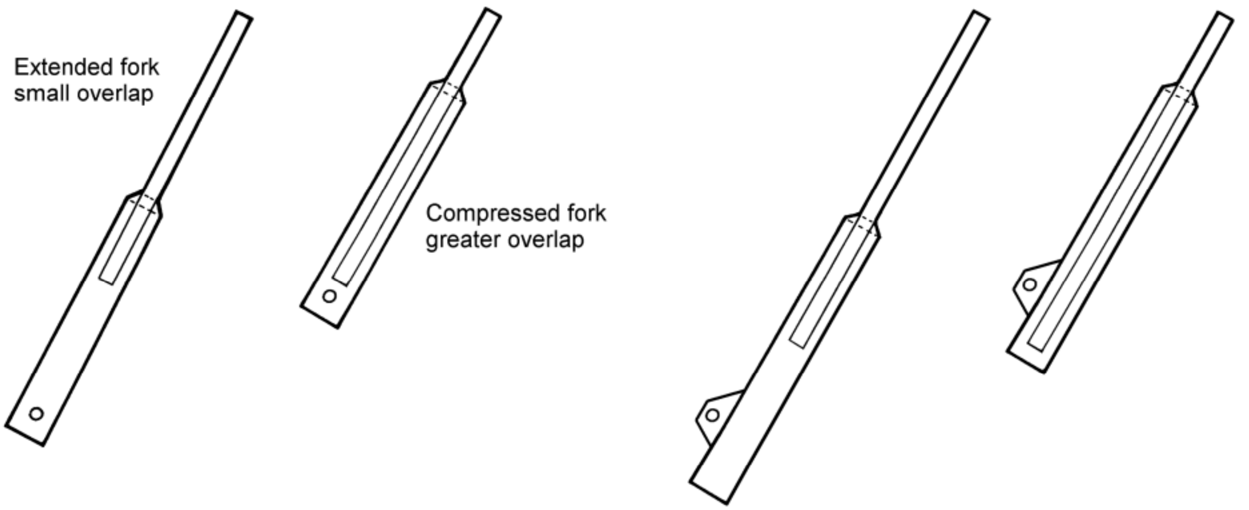



Fig. 7.2 **When a telescopic fork is fully extended the sliders are poorly supported because of a reduced overlap on the stanchions. On full bump, the overlap is at a maximum. Offsetting the axle allows a longer stanchion to extend farther into the slider for better support.**

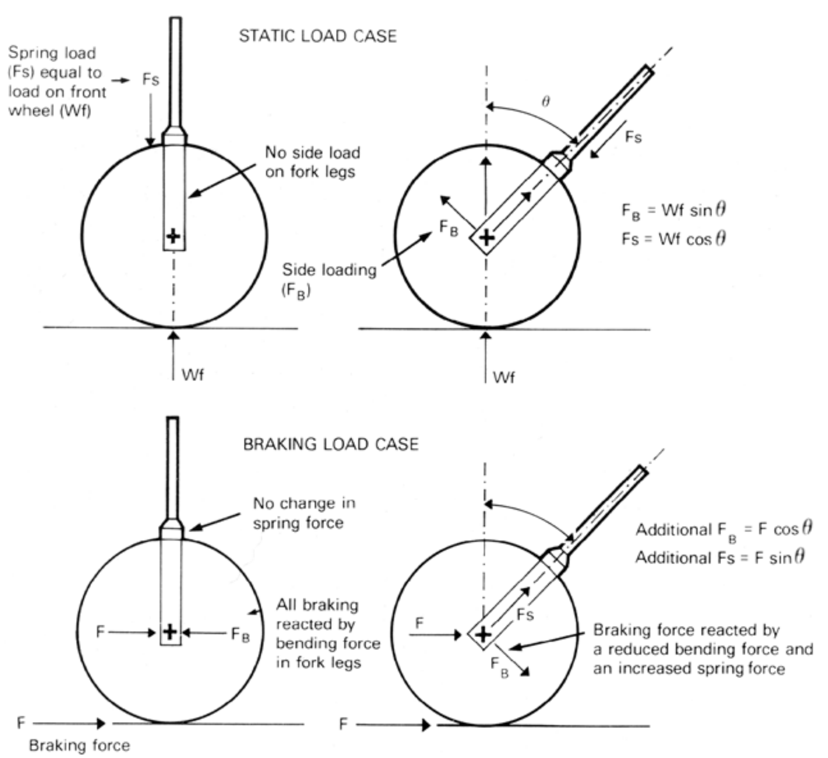
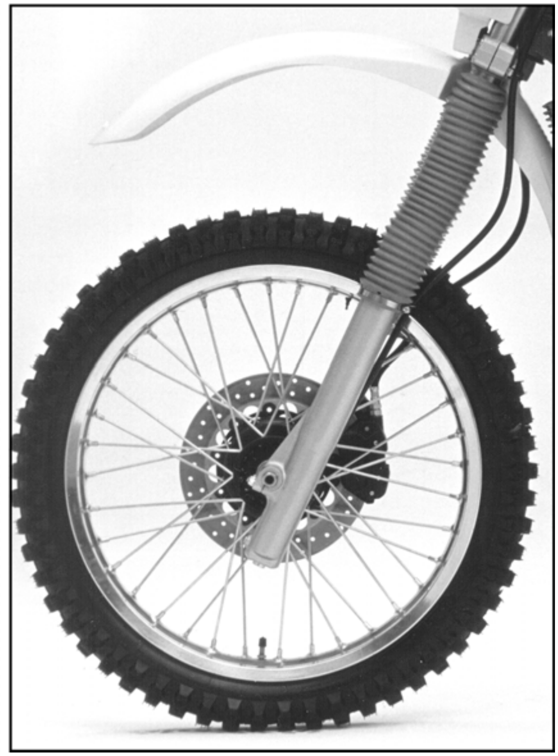



Fig. 7.3 **With a normal rake angle, just supporting the weight of the machine imposes a side load on the slider which in turn results in increased friction and decreased sensitivity to small bumps.**

**Under braking the rake angle causes some of the braking force to be reacted as an increased force in the spring. Therefore the spring compression and dive are increased over that due to weight transfer alone.**

**The load on the suspension can be tripled over the static load conditions, under heavy braking.**



**Two examples of offset wheel spindle forks. The one on the left has the spindle offset behind the slider whereas on the right we see the more usual system with the axle in front. Surprisingly both of these designs are from contemporary models by the same manufacturer, Cagiva. The first design allows the fork to angle back more, and so is better able to deal with the rearward component of bump forces. (See fig. 7.6) This must be set against the increased wheelbase variation, more brake dive and probably greater steering inertia.**

Under braking, telescopic forks are well known for diving, and although this is commonly attributed solely to forward weight transfer, fig. 7.3 shows that, for a normal rake angle, there is an additional factor at work – that is, a component of the braking force also acting to compress the legs. From the figure this component is equal to $F.\sin\Theta$, hence for a 27 degree rake this additional force is 45 per cent of the front wheel braking force. This can approximately triple the load on the suspension over the static case, under maximum braking, as explained in detail in the chapter on anti-dive.

Often it is claimed on behalf on teles. that trail remains constant throughout the full range of wheel travel. This assumes, firstly, that constant trail is desirable (which is by no means certain) and, secondly, that there is no change in the attitude of the rest of the machine while the fork is compressed by level ground under the front wheel, fig. 7.4.

It is difficult to visualize many conditions under which this situation arises – or when both wheels move the same distance vertically. In practice, other than when cornering, the fork is usually compressed either by braking nose-dive – in which case the rake angle is decreased, hence the trail shortened – or by hitting a bump, which initially moves the contact patch forward, also reducing the trail, fig. 7.5. As the wheel moves over the bump, the trail first returns to its static value, then lengthens as the contact patch moves rearward, only to return to normal again as the wheel regains the level road. Considering these changes, we can hardly agree that a telescopic fork (nor any other type) maintains constant trail.

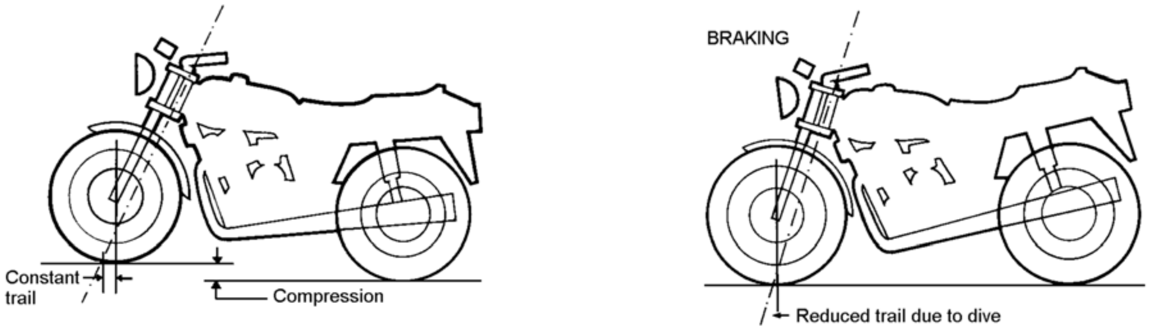



Fig 7.4 **On the left we can see the conditions needed for the trail to remain constant under the action of fork compression. Nose dive as experienced in braking reduces the trail as shown on the right.**

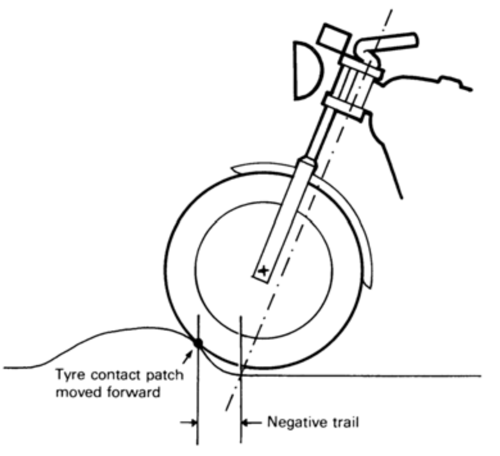



*Fig. 7.5* **By moving the tyre contact patch forward, a sharp bump reduces trail, it may even go negative as shown.**

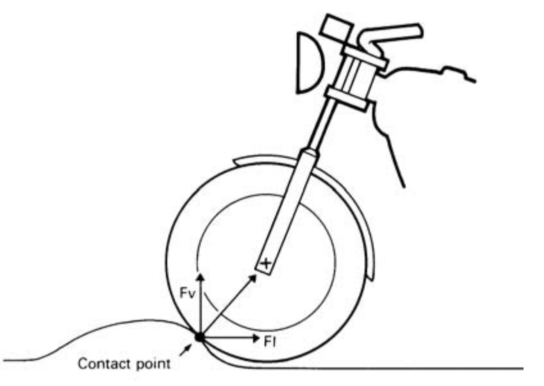
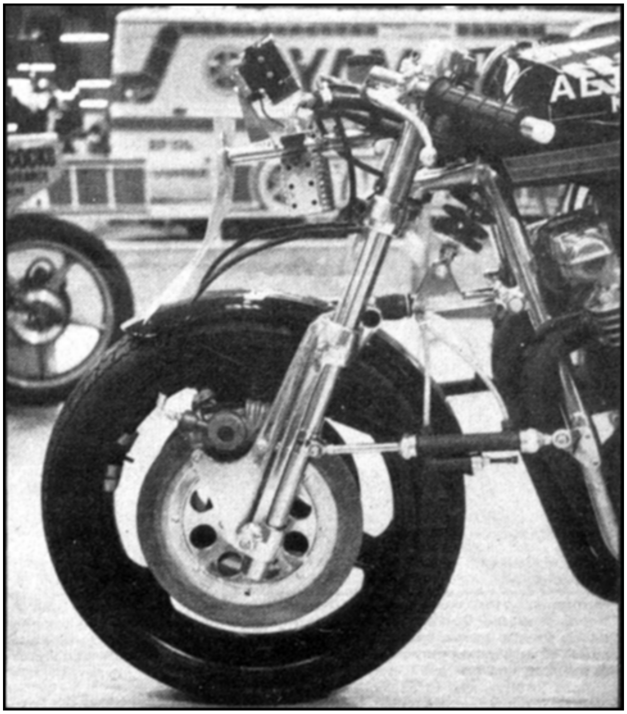


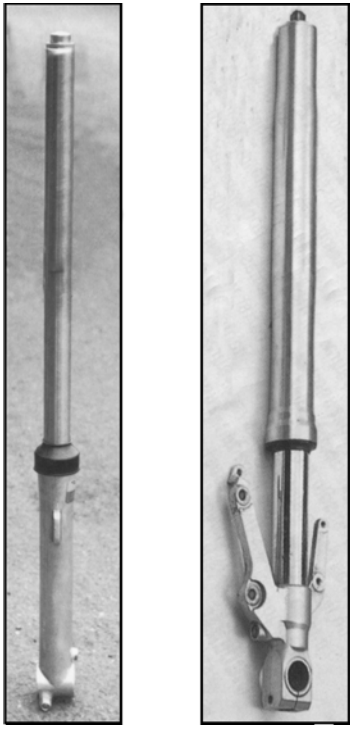

*Fig. 7.6* **The resultant wheel load from a bump acts radial inward from the contact point. Fv is the vertical component and Fl the longitudinal component of the tyre force.**

Road bumps tend to impart a longitudinal force to the wheels as well as the vertical force expected. The rearward movement of the front wheel under fork compression helps absorb this force, enhancing both comfort and control, fig. 7.6.

The Altec fork showed a valiant attempt by Alloy Technique, of Dartford, to eliminate some of the shortcomings inherent in more conventional telescopic forks. The stanchions are extra long and extend through both ends of the offset sliders, thus supporting them fully throughout the full range of suspension travel and minimizing the effects of the working clearance. The sliders are rigidly joined, not only by the wheel spindle but also by a strong bridge that virtually prevents the independent movement that is otherwise the chief cause of the tyre patch moving sideways. A pair of interconnected hydraulic cylinders link the stanchions to the main frame in such a way as to permit free steering while resisting fore-and-aft flexure under heavy braking. (This also reduces the horizontal loads on the steering head, so reducing flexure in the frame and possibly prolonging its life.) A single suspension strut, anchored to the frame, is connected to the sliders through a linkage system at the lower end, in this way the strut forms no part of the steering inertia.



**The Altec fork. Note how the slider is supported at each end throughout the whole range of movement. The hydraulic cylinder for the fore-and–aft bracing is clearly visible.**



**Typical modern "upside-down" fork leg on the right. The large diameter upper part is subject to the greatest bending moment. On the "conventional" leg (left) this bending moment is less ably carried by the smaller diameter stanchion.**

## Damping

Damping in telescopic forks is usually internal with the mechanism mounted inside each fork leg. There have been a small number of examples that have used a single external unit similar to those common on the rear, there was the Altec above and the photo below shows such a design by Ceriani, applied to USD forks. Note also the brace over the wheel. This looks like the designers thought carefully about the problems with this type of fork and tried to achieve better solutions than the norm. The external unit makes for easy setup and replacement, useful for racing. Except for a less tidy appearance this design would seem to have much to offer over the more accepted form of USD.

**An unusual interpretation of the USD theme, made by Ceriani. Suspension is provided by a single spring and damper unit connected to a bridge piece connecting both legs together.**

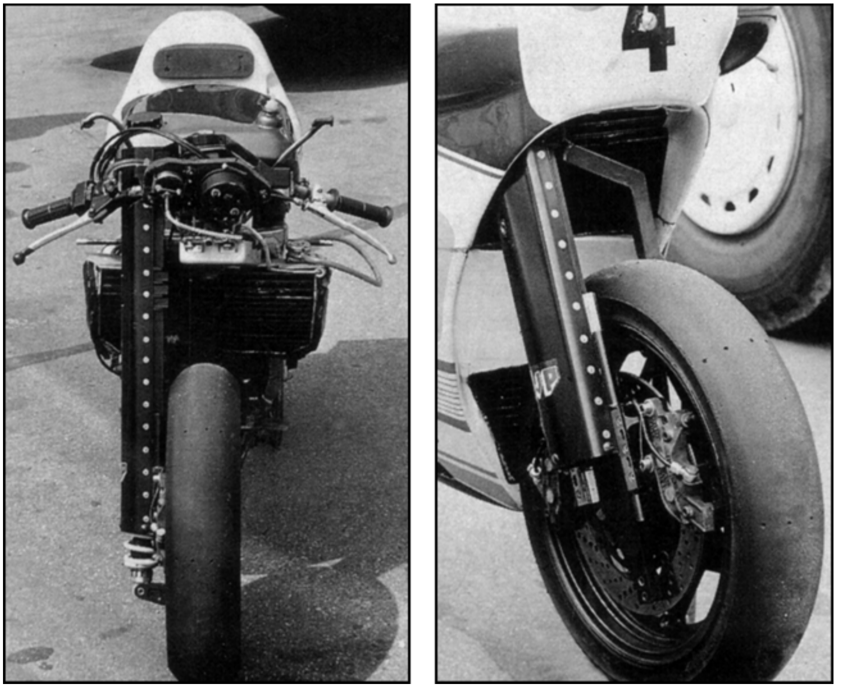
**Gilera produced this system. Essentially this is a single leg telescopic suspension. Steering torque is transmitted via a folding linkage. This whole design is very much like that used on large aircraft landing gear.**

For a long time the damping mechanism used in front forks has been inferior to the technology employed at the rear, cost seems to be the only rational explanation for this, although even in racing the front still lagged behind. Currently there are two principal types of dampers in use, the so-called damper rod type and the more modern (from around the late 1980s) cartridge type. The later is more expensive and is more likely to be installed in high end sports bikes. The damper rod type is quite crude in general, basically consisting of a tube with one set of holes for bump damping and another set for rebound. In general the damping of these forks is of the quadratic type, so giving insufficient low speed combined with excessive high speed damping forces. The common way to adjust these forks is to change the grade of oil used. The cartridge type is so called because the damper assembly can be removed complete as a single cartridge. Internally these employ similar technology to that used for rear dampers, with the internal flow being controlled by shim stacks and adjustable bypass passages etc. Some specialist suspension firms offer conversions to change the internals of older type forks to a cartridge system. There also what are know as “cartridge emulators” which can be fitted to some older forks, these emulators are fitted to the top of the damper tube and add more sophistication to the bump damping, with shim stack control.


**The current peak of fork development is amply demonstrated by these Showa USD telescopic forks fitted to the MV Agusta F4.**

**Large diameter fork tubes and the large hollow wheel spindle all contribute to a much stiffer structure than the forks used only a decade ago.**

**Sophisticated damping, low friction seals and coatings all contribute to good performance.**




**Experimental forks by White Power. These used linear rolling bearings to reduce the friction inherent in traditional telescopic forks. The company has not brought these into production.**

## Leading link

An alternative to the telescopic fork is the leading-link variety, which has the distinction of having been fitted to what were probably the best-handling racing machines of their period – the world champion Moto Guzzis of the mid 1950s. The Earles fork with longer links, used on some early MV Agusta racers and for many years by BMW, is a variation on the same theme but has the disadvantage of considerably higher steered inertia.

Several examples of both varieties are shown in the photographs. Generally, they comprise a tubular or pressed-steel structure connecting the steering column to the link pivots and incorporating anchorages for the suspension struts. The links may be independent or formed by a single U-shape loop around the back of the wheel. If the links are separate, then their resistance to independent movement, as in the case of the telescopic fork, depends on the rigidity of their attachment to the wheel spindle. In the better designs, this is of larger-than-usual diameter (hollow for lightness) and secured by extra-wide clamps. However, a large-diameter spindle means large wheel bearings and it may be that the most weight-efficient solution is a loop behind the wheel and a smaller-diameter spindle. The benefits of this type of fork depend greatly on the quality of detail design. In some respects, a well thought-out leading-link fork can have some benefits over telescopics. Greater rigidity is possible, with benefits in stability and precise control. The lack of stiction considerably enhances sensitivity to small undulations and any degree of anti-dive under heavy braking can easily be designed in. The precise path of the wheel travel depends on the relative heights of the link pivots and wheel spindle (see fig. 7.7). Due to the curved nature of this path, these forks are generally unsuitable for the large movement used on modern off-road machines. Some attempts have been made with multiple link designs but in general telescopic forks are firmly established for these applications.

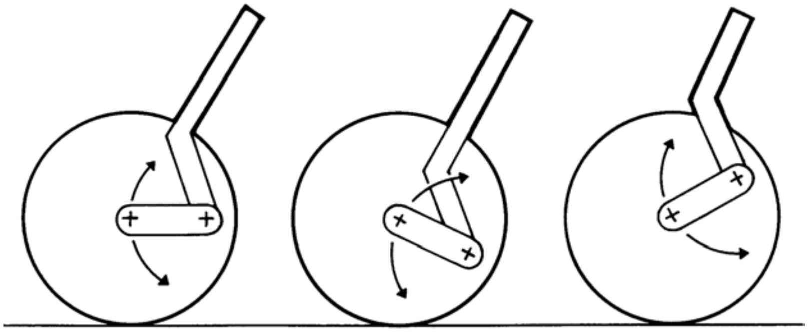
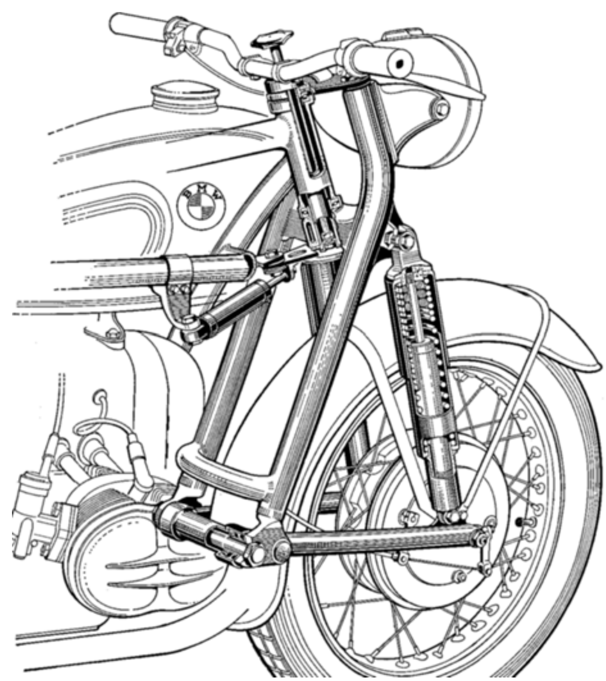


Fig 7.7 **With leading link forks, the relative heights of wheel spindle and link pivots determine the path of wheel travel.**

**A leading link fork which helped to create an enviable reputation for handling, as fitted to the 1954 NSU 125cc Rennfox world championship-winner. The stanchions were fabricated from steel pressings and the links had wide spindle clamps.** (MCW)



**From 1955 until 1970 BMW used Earles leading link forks. Two pivot positions were available. The rear was for solo use and the forward one gave less trail for sidecar duty. The brake back-plate was attached to the link and there was enough antidive effect to lift the bike under front wheel braking only.**

**A leading link fork design by the author. These were used on large capacity road bikes, the links were separate for styling reasons and relied on a relatively large diameter wheel spindle to provide sufficient stiffness.**

**A rather complicated interpretation of the trailing link theme by Frenchman Eric Offenstadt. The wheel spindle clamps could be slid along the link tubes for trail adjustment. The long upright link outside of the main assembly was there to transfer suspension loads to a high mounted single spring/damper unit.**

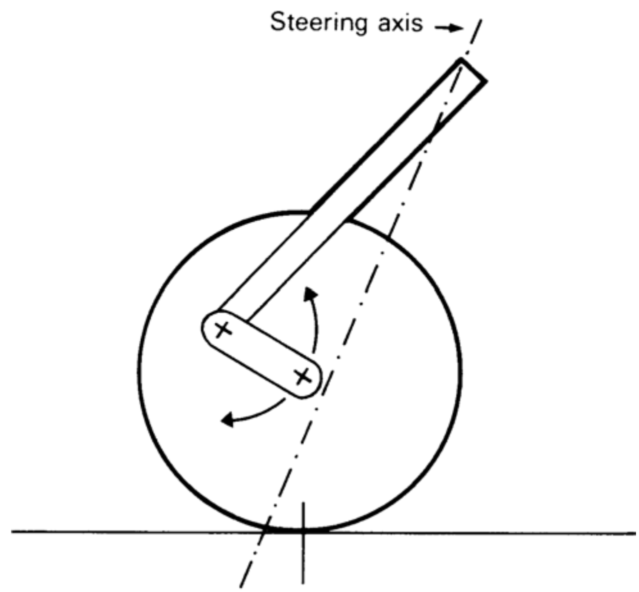
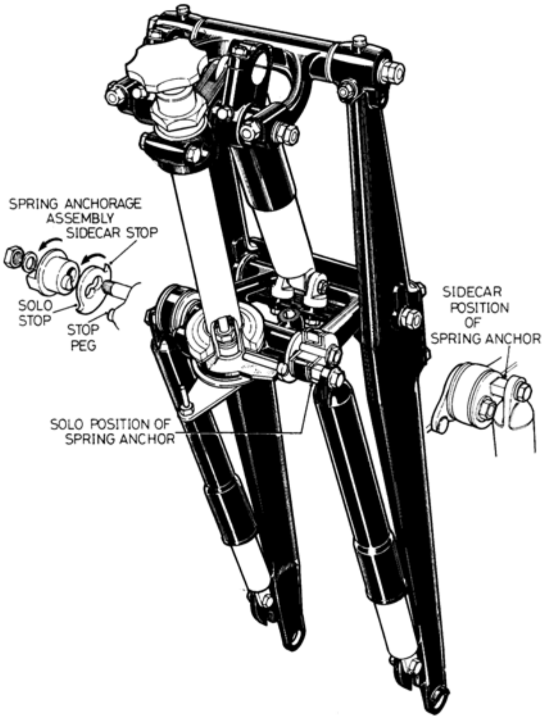



Fig 7.8 **In a trailing link layout, steering inertia is increased by the relatively long distance of the mass of the fork from the steering axis.**

## Trailing link

Trailing-link forks differ from the leading-link variety only insofar as the link pivots are ahead of the wheel spindle, not behind. Their disadvantage is higher steering inertia, since the bulk of the mass is relatively far from the steering axis – an effect that is only partly off-set by the smaller amount of material needed to reach the pivots, fig 7.8.

## Girder forks




**Most sophisticated of the girder forks was this Vincent "Girdraulic", with forged light-alloy blades and one-piece upper and lower link assemblies. Trail was readily adjustable. Springs were in long telescopic tubes, behind the uprights, but the hydraulic damper was separate, mounted in front of the head stock. Lateral stiffness was enhanced by a plate bridging the front of the blades. Interestingly, although hydraulic damping was employed against suspension movement a friction damper was used to damp out steering excursions.** (Haynes)

Girder forks in their day were almost as widely used as telescopic are today, and some were known for their excellent steering. Structurally there is much to commend this type of fork if executed well. Performance was generally limited by friction dampers, very crude by current hydraulic standards. The links that allowed suspension movement were quite short and would not have been very suitable for anything but a small total amount of suspension movement.

# Alternatives to the head stock mounted fork

There are several ways in which these can be categorized, but here I shall use four broad groups.

- Hub centre steered. This type has the main support and steering mechanism mounted within the actual wheel hub.

- Double link. These in one way or another can be considered akin to a car double wishbone system rotated by 90 degrees.
- McPherson strut based.
- Virtual steering axis.

## Hub centre steered

Generally a large diameter, steerable but non-rotating hub is mounted on a king-pin located within it. Another hub, of larger diameter, and forming part of the wheel, is mounted onto the first hub via large diameter ball races. The centre line of the king-pin defines the steering axis, and so the only flexure that can allow the tyre to deflect away from this axis, is in the wheel and the hubs themselves, but as we must also have these components with a conventional design, it can be seen that virtually all the other sources of compliance, in the forks, have been eliminated at a stroke. It must be remembered though, that any wear or play in the king-pin bushes or bearings will permit the dreaded lateral displacement, play in bushes has a different effect from that due to flex, and is generally more detrimental to stability. Hence the importance of good detail design. Consider now, some actual examples and let's study their pros and cons.

### Ner-a-Car

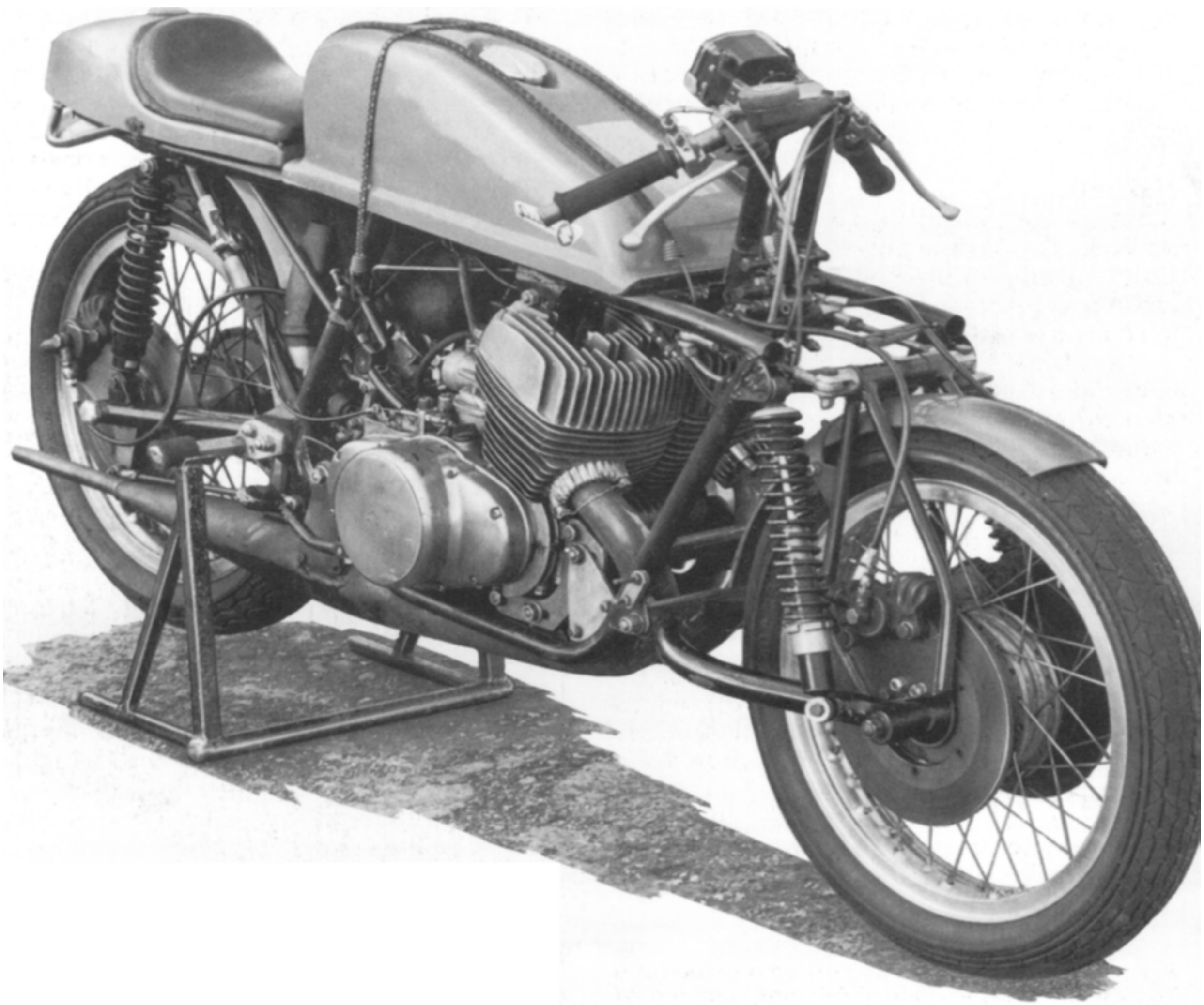
Illustrated in chapter 1, this early example of hub-centre steering had what amounted to a pivoted front fork (actually a pivoted U-shape axle, closed end forward) supporting an inclined kingpin in the centre of the wheel. The wheel swivelled on the kingpin and the swivel arm on the hub was connected by a link to a 20-per-cent-longer arm at the bottom of the vertical steering column, so gearing-up the steering. In its day the Ner-a-Car was renowned for its outstanding stability, though credit for that must be shared also by its long wheelbase (1500 mm. with an unsprung rear end, 1740 mm. with pivoted-fork rear springing) and its exceptionally low centre of gravity. A possible drawback in the design was that the inclination of the kingpin (i.e. the rake angle) and hence the trail varied considerably with suspension movement, showing that constant trail is not all-important in some cases. With today's long, soft suspension that would probably be unacceptable but suspension travel on the Ner-a-Car was very short.

There was no front brake on the Ner-a-Car but, if there had been, the effect on the suspension would have been the same as with more conventional leading-link front forks: that is, if the shoe plate (drum brake) or caliper (disc brake) was anchored directly to the links, brake torque would have extended the suspension fully despite forward weight transfer, so causing juddering, but if the brake was anchored by a pivoted linkage to a sprung part of the chassis, then brake torque could have been isolated or any chosen degree of anti-dive built in.

### Difazio

From 1968 a hub-centre conversion designed by Jack Difazio was available to riders of standard roadsters seeking to get away from the shortcomings of the conventional telescopic fork. This is the design that most readily comes to mind when hub centre is mentioned.

The axle, with a kingpin in the middle (raked between 16 and 22 degrees), is carried in a pivoted fork and to allow for suspension movement with minimal change in rake angle, the horizontal arm of the kingpin is bushed to rock on the axle. Swivelling on ball bearings at the top and bottom of the kingpin is a steerable but non-rotating drum supporting the large wheel bearings (130 mm outside diameter, 85 mm inside diameter); the drum is slotted for steering. Bolted to the outer flanks of the drum are two upright A-frames (one each side) whose apexes are united by a bridge-piece above the wheel.



**Difazio hub-centre steering is shown on this 500 cc Suzuki-powered racer. Suspension is by pivoted fork and steering by a parallelogram linkage. The steering links had a double duty and were also needed for the upper location of the A-frames, and hence were subject to reactions against braking torque. Range of kingpin angles was from 16 to 22 degrees.** (MCW)

To prevent free rotation (except the small amount needed for suspension movement) the inner hub is connected upward to the steering links via these "A" frames on each side. The brake calipers are mounted on these A frames and their torque reaction is taken by the steering arms. The axle is held between the open ends of a forward facing swinging arm, which must be wide enough to give tyre clearance as the steering is turned. Suspension units are connected to this swing arm, and the suspension loads are carried through the king-pin bushes.

Overall this is quite a clever design mechanically if not aesthetically, and remember that it was patented nearly forty years ago. As mentioned above the source of lateral compliance (relative to the steering axis) is reduced to that of the wheel, wheel bearings, and the king-pin bearings. Braking loads are spread between the bottom swing-arm and the top steering links, and because of their distance apart and proximity (compared to a head-stock) to the tyre contact patch, the point loading at the frame mountings is much less than that normally experienced with a head-stock.

Disadvantages also exist, as might be expected, we seldom get anything for nothing. As all the sideways rigidity is provided by the king-pin, this part is quite highly loaded when the wheel is subject to a lateral force, e.g. hitting a bump whilst cornering. The potential strength of the side 'A' frames is not used to relieve this loading, which also tends to bend the axle into an S shape, and twist and bend the swing-arm. However this axle and swing-arm distortion does not allow the front tyre to misalign from the steering axis (because the axis moves with the wheel). Perhaps another problem is that it is a fairly complicated system, both in terms of the number of required components and also in terms of wheel changing ease.

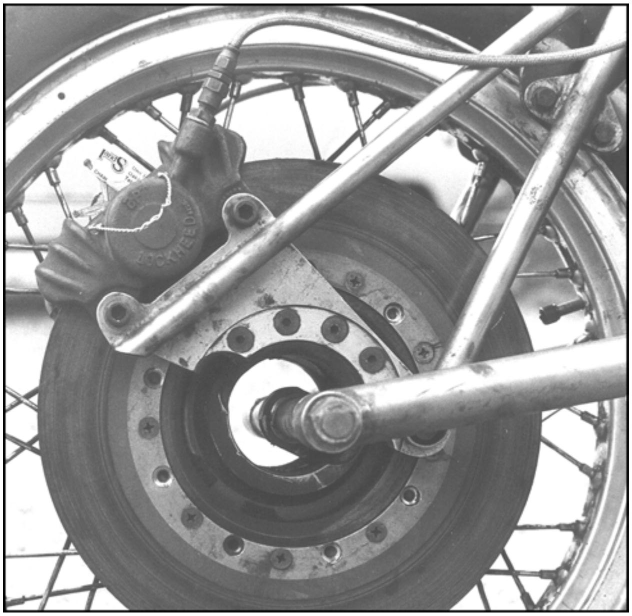
## Mead and Tomkinson

On some of their endurance racing bikes (The famous Nessie amongst them.), this team experimented with variations on the Difazio theme. One such design replaced the king-pin with a spherical joint that was free to pivot in all directions, to prevent the sideways rotational movement thus made possible, a triangulated wishbone was connected to the middle of the cross piece joining the tops of the 'A' frames. A single suspension unit was mounted on the top of the wishbone, relieving the hub pivot of suspension loads. Although the axle was still subjected to bending loads from braking, this system removes those due to lateral and suspension forces, and in addition, the possibility of swing-arm twist is eliminated. So far so good, but the potential for misalignment between the wheel and the steering axis is increased, flex in the 'A' frames can now add to that. Despite this, these modifications to the original had considerable potential for an overall improved design.

Steering is effected through a parallelogram linkage connecting the A-frames to the steering column. Rake and trail are determined by the length and location of the wishbone. The brake calipers are mounted on the A-frames so that torque reaction is taken by the wishbone. This design has the following advantages:

- the degree of antidive is governed by the geometry of the fork, A-frames and wish-bone (in-side view) and can be varied as required;
- wheel movement is substantially upright, so maintaining an almost constant wheelbase, it is easy to arrange for the rake angle to remain constant too;
- the trail can be readily altered (with no change in wheelbase) by changing the lengths of the wishbone and steering links, so catering for different machines, road conditions and personal preferences.

A snag with any suspension system (front or rear) giving approximately vertical wheel movement is that this does not absorb the rearward component of the force due to road bumps, as does a telescopic fork. But this is probably a small price to pay for the benefits of hub-centre steering. Certainly, overall rigidity is much superior to that of a fork mounted on a conventional steering head. Also, since the loads fed into the main frame are much smaller, that component can be made lighter. On the debit side, hub-centre layouts of the types described above are often criticized on the grounds of appearance, weight and limited steering lock.



**One of the Mead and Tomkinson design variations of the Difazio theme. Of particular note is the rearward offset of the steering pivot within the hub, this allowed steering geometry to be closer to that found on head-stock mounted forks. The hub pivot was a spherical joint in place of a kingpin.**

## Bimota Tesi

Probably the currently best known of the true hub-centre designs, there have been several versions of this over a long development period. The original was part of a college design project and was probably best remembered for its use of hydraulic steering. Heralded in many press reports at the time as a major innovation, the hydraulic steering appears to have been just another source of potential problems. As Bimota seemed to have discovered. It is difficult to see any benefits which outweigh the disadvantages. Anyone with experience of such closed circuit hydraulics knows that provision for expansion and piston rod displacement is necessary. The solution adopted was the provision of a pressurized gas chamber, separated from the oil by a flexible diaphragm, similar to many suspension units. Unfortunately, the compliance thus introduced may cause sponginess in the steering.

Bimota’s original system had some features in common with the Difazio, although the king-pin was fixed to the axle and not free to rotate on it. Consequently the braking forces were taken through the wheel spindle to short brackets, fixed to each end. These brackets were in turn located by two torque arms. Because these arms were relatively close to the swing-arm the forces were high compared to the Difazio setup. The king-pin/axle is much more highly stressed, and cannot be rigidly clamped in the swing-arm, instead it must be free to rotate within it, to allow for vertical wheel movement. Hence, the spindle cannot add to the torsional stiffness of the swing-arm, which needs to be bigger as a result.

The design of later Bimota front ends changed considerably from the first version, improving the areas of doubt mentioned above. For one, the hydraulic steering was replaced by mechanical links.


**A 1983 Milan Show special – the student designed Bimota Tesi, with hydraulically operated steering. Geometrically similar to the Difazio, this design loaded the wheel spindle heavily in torsion when braking.**

## Double link

This type generally, is mechanically simpler, comprising basically an 'upright' (car terminology) to which the wheel and its bearings are attached. This upright is held in place by two forward facing pivoted arms or wishbones, the front ends of which allow for steering and suspension movement.

Structurally, the various designs within this family differ mainly in the locations on the upright to which the two pivoted arms are attached. Considering the lateral stiffness characteristics, we see that this is dependent only on the stiffness of the upright and the wheel. Let's look at some actual systems.

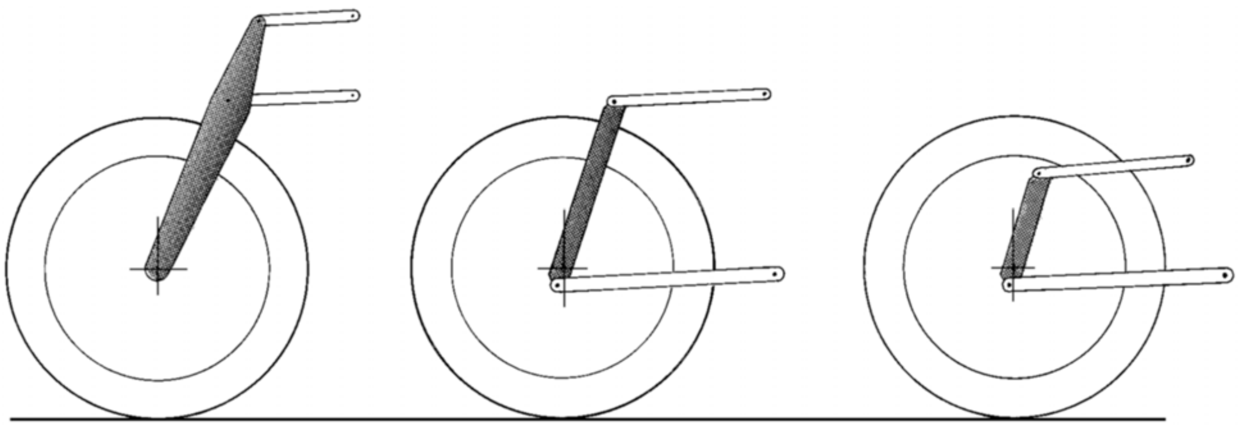
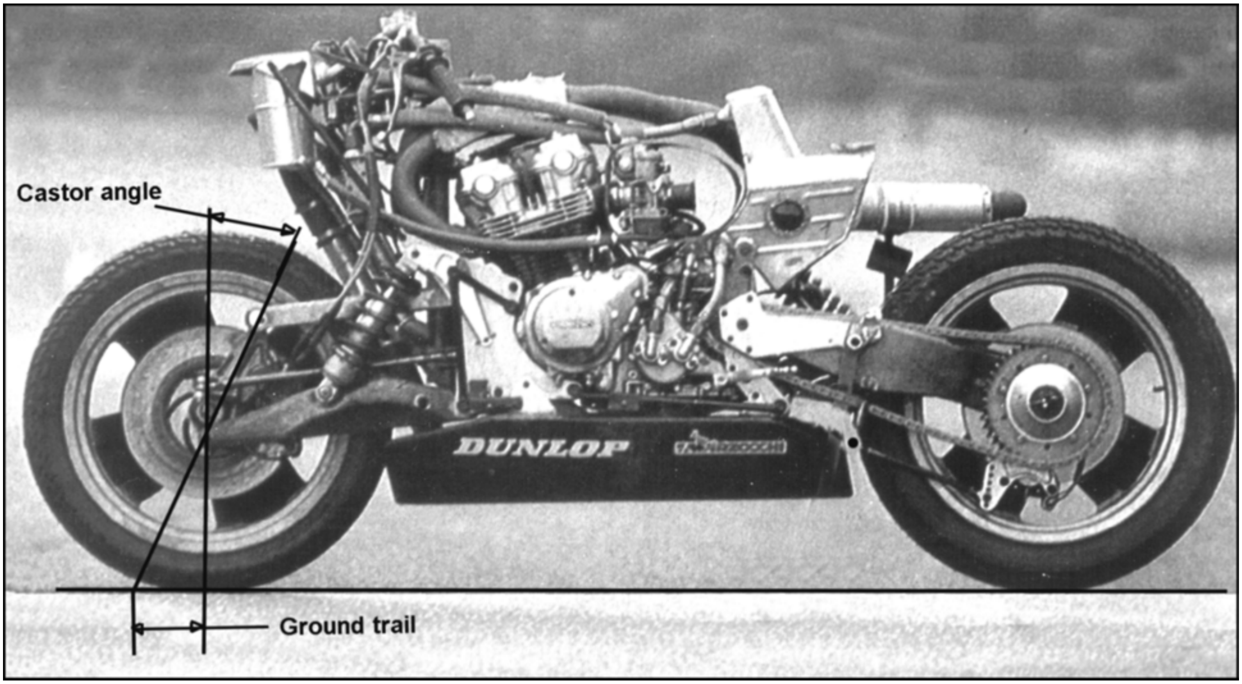


Fig 7.9 **Three types of double link front suspension. Basically only different in the location of the links, although the layout on the left is usually made with the upright as a fork so that a conventional wheel with double sided mounting can be used. The other two require a single sided upright and special single sided dished wheels.**

## Elf. e (also similar to Didier Jillet)

Two French designs of the double link family are those of Andre de Cortanze and Didier Jillet. Sponsored by Elf, Cortanze's design was first raced in Formula 750 with a Yamaha engine, then in endurance events with Honda power. The general layout of this type of design is as follows: A car-type upright (a light-alloy casting or forging) supports either a fixed stub axle or a live rotating axle for one-sided wheel mounting, the brake calipers are bolted to the upright. Two one-sided longitudinal arms connect the upright to the main chassis via spherical ball joints. For unbiased steering (effected by a single drag link) the centre of these joints must lie in the centre plane of the wheel. This necessitates that the two pivoted arms have their forward ends within the wheel circumference, which in turn means that they have to be curved outward for tyre clearance on full steering lock (to one side), the steering drag link must also be widely spaced for the same reason. So, even though it would be hard to design a more efficient upright, the pivoted arms have somewhat less than the optimum shape. A fact aggravated by the need to carry the suspension forces through one of them. This introduces torsional and vertical bending loads into that arm, which must then be large enough to handle them.

A wide variety of steering geometries is possible. Scrutiny of the photos indicates that the first Elf (with the TZ750 engine) had a fairly conventional steering geometry, there being some positive offset between the steering axis and wheel spindle combined with a normal rake angle, to achieve the desired trail. However, the later Honda-engined design appears to use little offset, with a rake angle of approximately 23 degrees (although this might be varied for different circuits); this gives a trail of about 127 mm. This is higher than normal and probably accounts for the reported heavier than usual steering with good stability.




**There were many Elf sponsored racers with alternative front suspension designs. This one was a Honda powered endurance racer. Scaling from the photograph indicates that the ground trail was more than average at about 127 mm. In the forward end of the top link the pivot bearing is mounted in an eccentric block which could be easily adjusted to change the trail. Because the lower ball-joint is close to the wheel axle this trail adjustment only changes the wheelbase by a small amount. Note also the single sided cast rear swing-arm, with external sprocket, setting a style followed by others.**

The Jillet design has no offset and a steepish rake angle. Although the suspension strut could, in theory, be attached to either arm, its actual attachment to the lower one helps keep the weight low down; it is, of course, not steered.

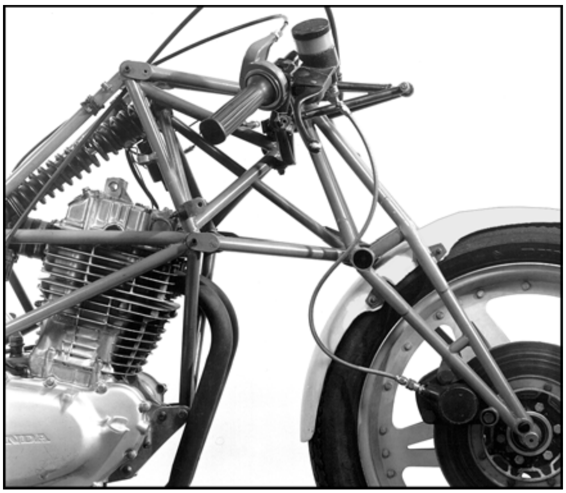
This steering design is hard to fault and can have the following points in its favour;

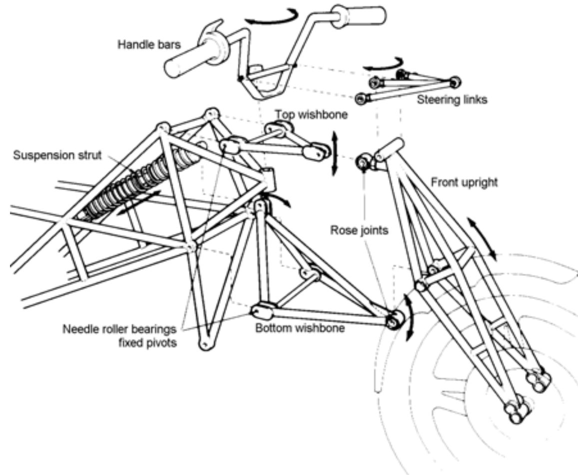
- low unsprung mass.
- low overall mass.
- high rigidity.
- low steered inertia.
- easy wheel changing;
- reasonably neat appearance.
- many constructional inaccuracies can be simply adjusted out by varying the lateral location of the pivoted arms or the pivot points of the uprights; hence wider tolerances are permissible in manufacture.
- rake angle, hence trail, can be easily changed with minimal effect on wheelbase.

A valid criticism is that steering in one direction is restricted by the pivoted arms (albeit bowed for tyre clearance) and the steering drag link.

## Hossack / Fior / Foale / Britten etc.

This type is the antithesis of the above, within the double link family. Instead of a very small upright with large curved arms, this setup has a larger upright which is located by very light and stiff wishbones. These are mounted above the tyre. Although the upright is usually constructed in the form of a fork, which allows for the use of normal wheels and brakes, there is no reason in principle why this could not be a single-sided component, to give the quick wheel changing of the Elf.e. but the stiffness to weight ratio is probably better with the fork.






**Original Hossack tubular construction, showing front fork, steering linkage, wishbones and actuation of suspension strut. The steering linkage allows for suspension movement without introducing bump-steer in the straight ahead position. A small amount of bump steer would have been introduced when the steering was turned by a large angle, but this is of little importance on such a machine.** (Dave Jenner)

Compared to the Elf. this design has both advantages and disadvantages. The steering joints are mounted higher, the increased leverage causes greater loads in the spherical pivot bearings, and the potential for using the engine structurally as the main frame (exploited by the Elf.) is reduced, simply because of the required physical location of the pivots.

In common with the hub centre schemes the Elf. has limited steering lock, but on the other hand the Hossack type is free from that constraint and as such is the only system, described here, that has any real potential for off-road use. Indeed Hoyt McKagen in America developed a system for off-road use, which uses a suspended headstock, allowing the main suspension loads to be fed directly into taper roller bearings. This design has the added advantage that steering friction is reduced compared to steering through a spherical bearing subjected to suspension loads.

At a casual glance, the Norman Hossack design (1980) shown in the sketch was often mistaken for an obsolete girder-fork layout, but any similarity is strictly superficial and the design is quite sophisticated and well thought-out. Whereas, with a girder fork, the links are short and steered along with the spring, the only steered part of Hossack’s suspension (apart from the wheel and brake) is the actual fork, which we'll call the upright. There is no steering head. Instead, the fork pivots on spherical bearings at the front apexes of upper and lower forward-facing wishbones. The lower wishbone is triangulated upward from its pivot to actuate the suspension strut. Rake and trail are easily altered by screwing the top spherical joint into or out of its lug at the top of the fork. A system of links designed to eliminate significant bump-steer connects the fork to the handlebar. The fork itself is rigid and well triangulated, so providing strong resistance to lateral deflection of the tyre contact patch from the steering axis.

**Hoyt McKagen built this system for MX use. The suspended headstock allows for the use of normal low friction taper roller bearings. Minimizing steering friction is particularly important on off-road machines. This design is somewhat similar to the 1983 proposal of John Wright-Bailey.** (facing page)

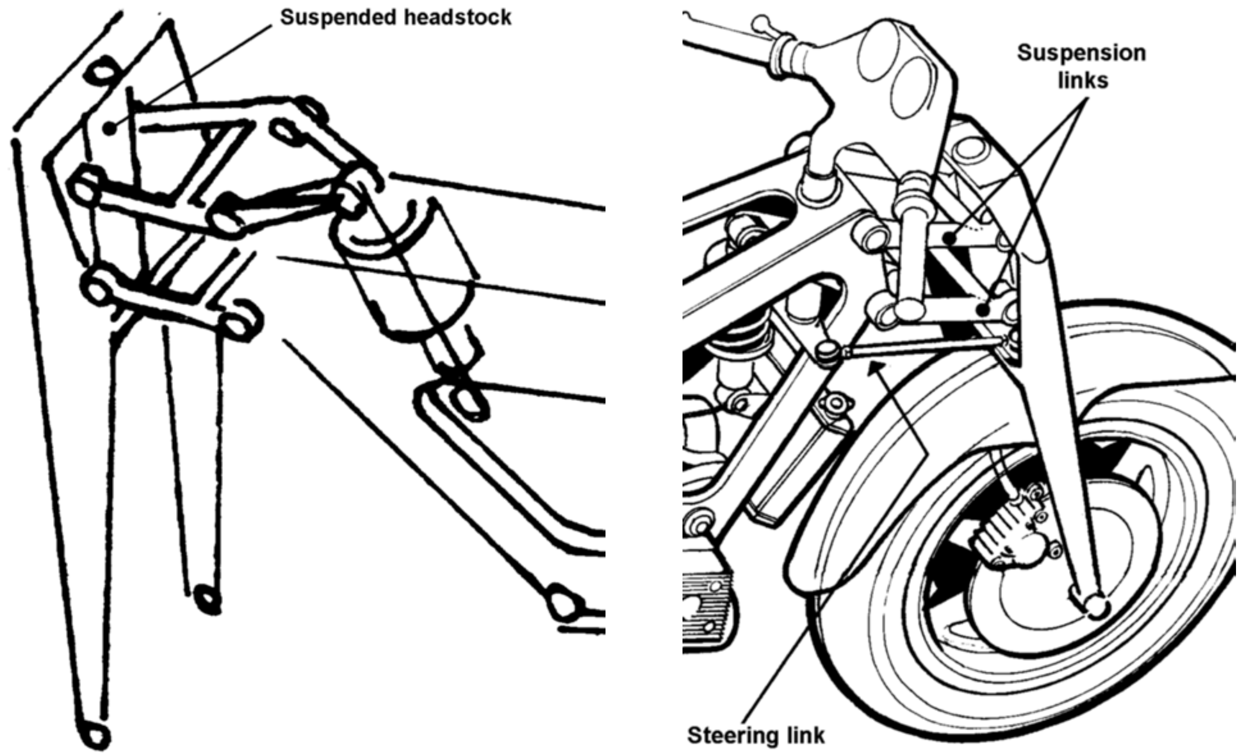
As in the Difazio and Elf layouts, the precise geometry of wheel movement is governed by the lengths and angular dispositions of the upper and lower wishbones, so allowing the designer much more flexibility in the geometric layout than with a telescopic fork or one with leading or trailing links. Indeed, the Hossack design may almost be regarded as an Elf with the pivot axes moved from within the wheel to above it. In this light, the Hossack has both advantages and disadvantages compared to the Elf.

Among its advantages we can include:

- the wishbones can be triangulated, so combining strength with lightness.
- the layout of the wishbones is not dictated by tyre clearance, hence they do not restrict steering lock.
- a conventional wheel can be fitted, with twin brakes if necessary, so reducing costs for low production models and one-off specials.
- appearance is more conventional, especially when a fairing is fitted, and air drag is reduced.

On the debit side:

- the higher location of the wishbones, fork, suspension strut and steering links raises the centre of gravity.
- the lateral and longitudinal loads on the pivot bearings are increased by the extra vertical leverage.
- adjustment of rake and trail effects a greater change in wheelbase.
- the main frame needs to be more complex, and possibly heavier, because of the longer load paths.




**A 1983 proposal by John Wright-Bailey. Similar to the Hossack genre this differs (like the McKagen dirt bikes) in using a suspended head stock, which allows the fitting of taper roller bearings, or similar, for the steering pivot, thus reducing steering friction.** (MCN)

Though different in appearance, the Hossack system is similar in principle to a design by the author in which the upright, rather than being a triangulated fork, was fabricated in 51 x 25 mm. box-section tubing, mainly for ease of construction. Some later Hossack models followed this same form of fabrication. For a given standard of rigidity and strength, the triangulated construction can be lighter. Another, more functional difference, is in the steering geometry. To achieve the required trail, the Hossack design has a fairly normal rake angle and offset, whereas the Foale layout had no offset, hence a steeper rake of around 15 degrees.

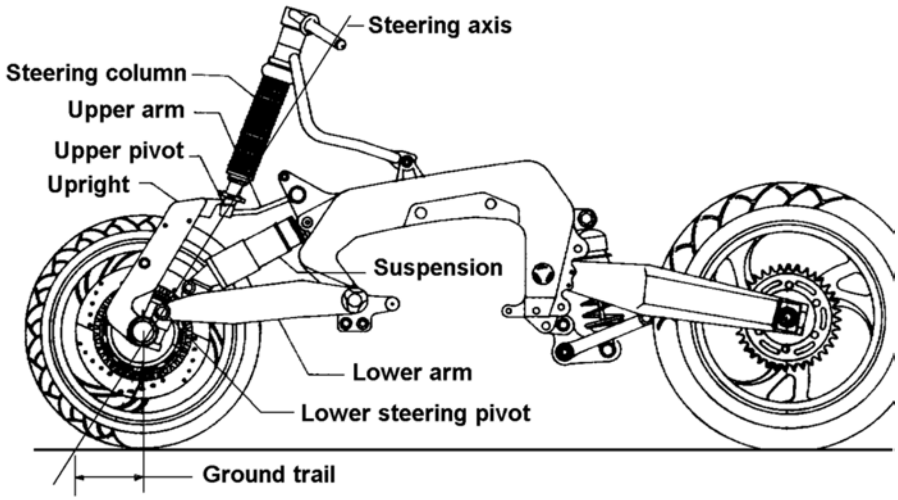
Many of Fior's racing motorcycles also featured very similar designs as did the V-twin engined racers designed by the late John Britten.

## Foale / Parker / Yamaha GTS

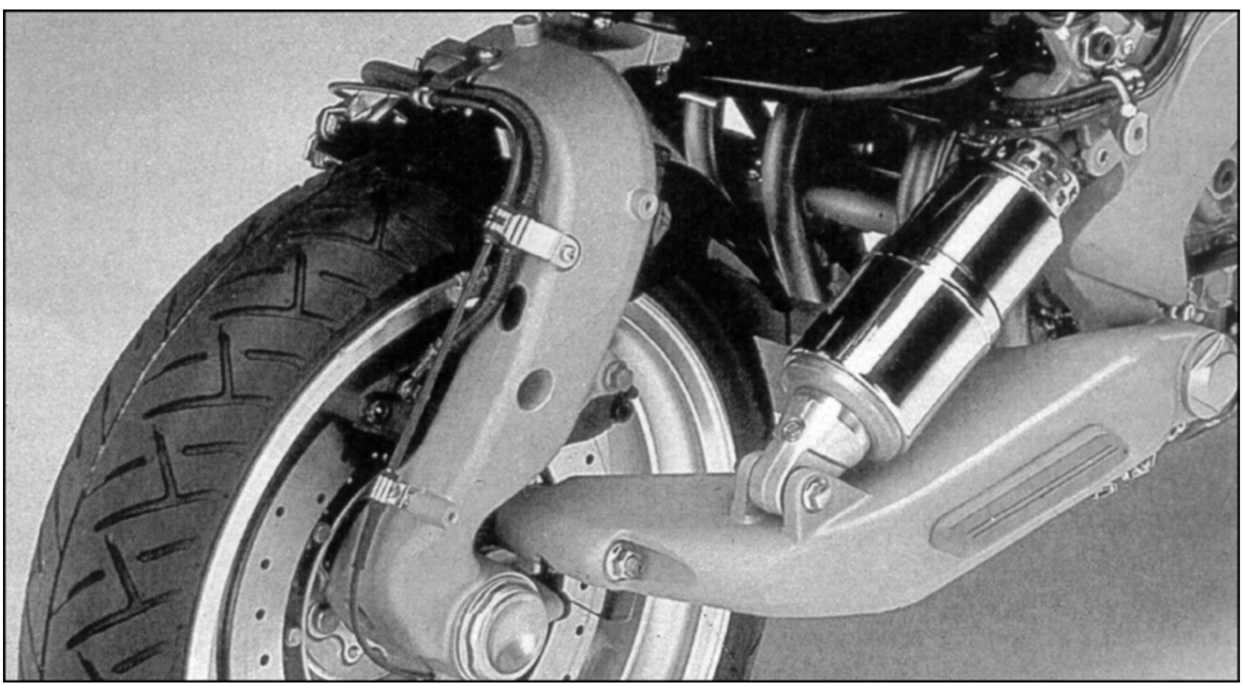
This design combines features of both the two previous systems. The lower arm is curved and at hub height, like the Elf., but the upper one is a wishbone above the tyre, like the Hossack. By increasing the distance between the upper and lower arms (compared to the Elf.), the loads fed into the pivot bearing are considerably reduced. The suspension unit can now be fitted to the top arm near the upright, thus eliminating torsional and vertical bending loads in the lower curved arm. This component can now be made thinner and lighter. The price paid for this, is that the upright must be longer, and hence heavier and/or less stiff, than the Elf.e. On balance it is my opinion that it is a better compromise, as it also allows the steering drag link to be above the wheel where it is closer to the handle-bars, thus simplifying linkages, and need not be mounted wide to clear the wheel on full lock. The Parker system differs from the author's in that the steering mechanism does not employ a drag link, instead it uses a telescoping torsion tube to pass the rider's messages on to the wheel.


**The author's interpretation of the double link theme. Shown here fitted to a Suzuki powered Q2. The lower arm is a simple aluminium casting, a shallow section for this was made possible by mounting the suspension unit above the wheel and so the curved arm only had in-plane stresses to deal with. The upper-link is a fabricated tubular "A-arm". Steering is via a drag link just visible above the top link.**

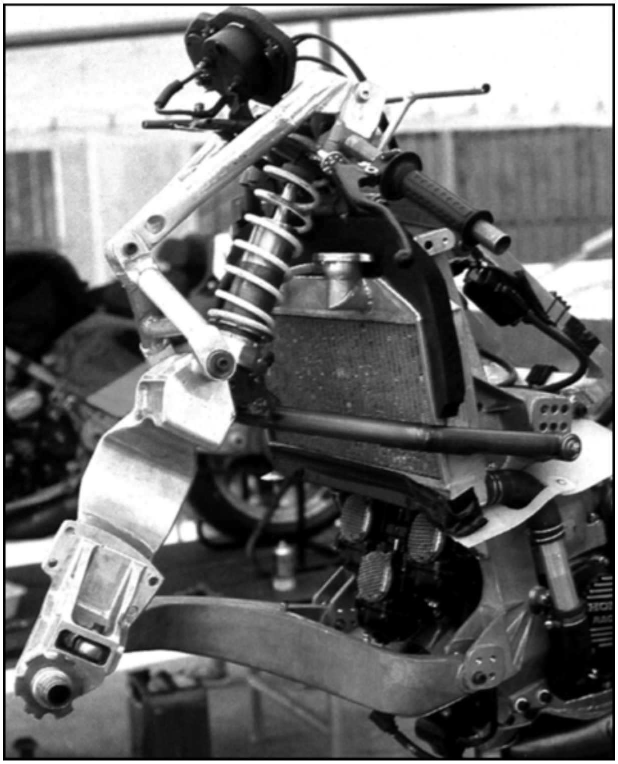
Yamaha enlisted the help of Parker with the design of their GTS road bike. Still the only large scale modern production bike that dared depart radically from the normal head-stock mounted forks. Even though it had a limited production life, from 1993 to 1999, it attracted a lot of aficionados amongst owners very impressed with the front end. The GTS was a heavy weight tourer and many speculate as to whether greater market acceptance would have occurred had the front suspension been used on a top of the range sport bike.




**Yamaha GTS. The layout is very similar to Parker's own bikes and retains the telescopic steering column to eliminate any possibility of bump steer.**



**Yamaha GTS in the flesh. The suspension components were of generous proportion to say the least, or in other words, downright heavy. It was a marketing risk for Yamaha to produce such a machine and they must be applauded for that. Presumably they didn't want to take more risks with structural failure, with its attendant bad publicity and liability concerns.**



**Never afraid to experiment this Elf machine followed the lines of the Foale/Parker design and raised the top arm to above the wheel. The steering mechanism is unlike either and follows the lines used by Hossack. Note the large number of alternative mounting holes at the rear of both upper and lower links. This allows for adjustment to rake, trail, wheelbase and anti-dive characteristics.**

## Bump-steer

More familiar to car enthusiasts, the term bump-steer refers to the results of a conflict between the suspension and steering geometries. With telescopic forks and indeed any head stock mounted suspension, the handlebars are connected directly and suspension movement has no effect on the steering angle.

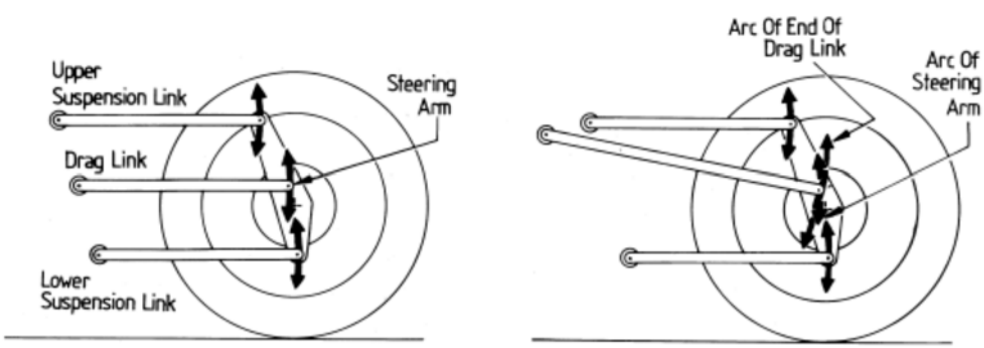
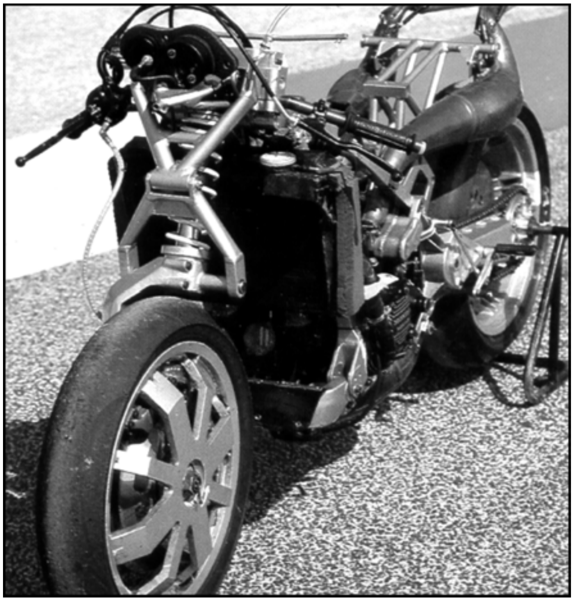



Fig. 7.10 **The left-hand sketch shows a layout that gives no bump steer. The exaggerated case to the right shows clearly the origin of bump steering.**

However, those systems without a headstock all need some form of indirect steering mechanism, and the kinematics of this need careful designing to avoid the introduction of steering motions as the suspension moves. Fig. 7.10 shows how an incorrect geometry with a parallelogram link system can introduce such bump-steer. The layout of the suspension links defines the locus of the end of the steering arm on the upright, to avoid bump-steer the steering drag link must be of an appropriate length and be located such that it describes an arc coincident with the end of the steering arm. If not, it will force the wheel to steer.

The possibility of bump-steer has been completely eliminated by the telescopic steering column of the Parker design and GTS, but both the systems using a drag link and also those like the Hossack with a hinged linkage have the potential for such problems. In practice it is usually impossible to entirely eliminate bump steer over the full range of suspension and steering movements from these designs. Before constructing the Q2, shown above, the author did a series of tests that deliberately introduced large amounts of bump-steer to observe the practical riding effects. The results of these indicated that whilst it was desirable to minimize the bump-steer in the straight ahead position, it was much less important at large steering angles, which are only used at very low speed when manoeuvring. This compromise is not usually difficult to achieve.

## McPherson strut based



**Yet another approach on the Elf bikes (post Cortanze). This design owes much to the McPherson strut, common on modern cars. The typical lower arm is retained but the upper location is done entirely by the suspension unit which has a larger than normal sliding rod.**

The McPherson strut is widely used on modern street cars and its chief advantages are more to do with cost and packaging than they are to do with the best technical solution. Essentially this design locates the wheel by a pivoted link at the bottom and by damper strut and rod at the top. The upper mounting of the damper rod and spring must be free to rock and to steer with the minimum of friction. The rocking motion is necessary to allow the bottom of the strut to move as defined by the lower pivoted link. The pictures of one of the Elfs above gives a good idea of how it can be applied to motorcycles, but in this case it is hard to see much justification for its selection. It seems to combine some of the disadvantages of the link type suspension with those of telescopic forks, without offering compensating advantages.

## Saxon/Motad

At first site the Saxon/Motad system bears little resemblance to the Elf above, but it can still be considered as an adaptation of a "McPherson strut" for bike use. It uses just one pivoted link and a fixed pivot at the top. In place of a single strut the wheel is held in two sliding legs similar to a telescopic fork, although the suspension function is handled externally by a separate spring/damper unit. The steering axis is defined by the line drawn through the upper fixed pivot and the lower moving one, which is attached to the fork sliders, although the overhang from the wishbone to the wheel is much greater than is usual in the car version of strut suspension.

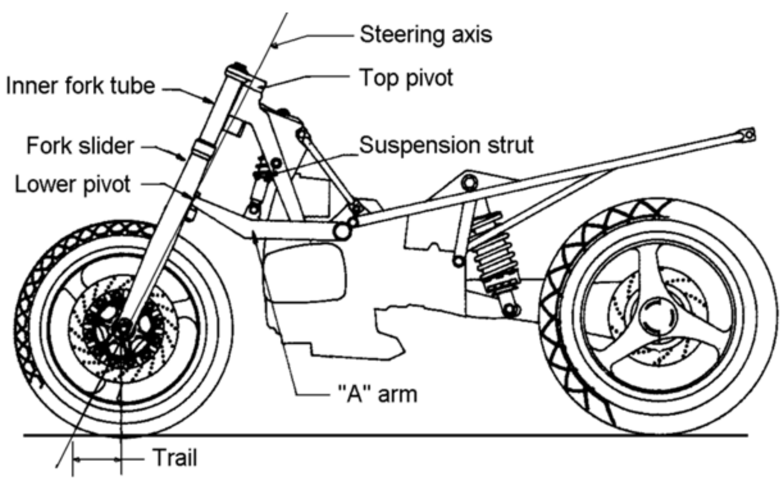
**This Saxon/Motad design is a forerunner to the BMW telelever. Although it features double sided wheel mounting and the lower arm has been replaced with an A-frame above the wheel, it is geometrically similar to the McPherson strut type. An attempt to retain the appearance of telescopic forks to keep market acceptability was one of the aims of this design and in that direction it succeeds well..**

One of the main design criteria, was that it must look similar to telescopic forks, in the belief that the market would respond better to such a design. Viewed in that light, it has considerable potential for improvement over normal forks.

## BMW telelever

Along with the Yamaha GTS, BMW is the only other major manufacturer to have introduced an alternative to the telescopic fork on standard production models. Unlike Yamaha though, BMW made the change throughout their model range, hardly surprising because BMW have often been prepared to go against the tide with front suspension. Named “Telelever” the design is very similar to the Saxon/Motad. An “A” arm or wishbone is mounted to the sliders just above the wheel and there is a piece like a conventional upper fork yoke that mounts to a dummy steering head through a floating bearing.

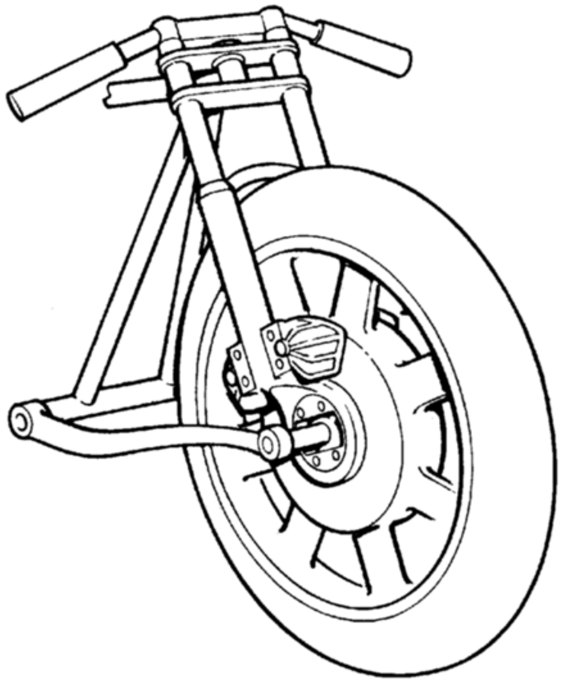
BMW have also been strong advocates of ABS braking systems and this may have had an influence over the decision to ditch telescopic forks. The cycling frequency of the ABS system can be close to the longitudinal resonant frequency of the fork and wheel assembly, resulting in unwanted vibration. The telelever is stiffer in this direction and so its natural frequency would be somewhat higher and less likely to suffer similar problems. Additionally, as shown in the chapter on squat and dive, the geometry of the telelever layout can be selected to reduce nose-dive under braking.




**BMW Telelever. Maybe not in appearance but this type of suspension can be seen as a variant of the McPherson strut. The wheel is mounted to a sliding assembly which is located by a “A” arm.**

## Killeen

An interesting combination of telescopic fork and hub-centre steering was proposed and patented by Tom Killeen, who called it head-to-hub steering. “The telescopic fork”, according to Killeen’s description, “is mounted on a kingpin or spherical bearing in the hub centre and on a self-aligning or spherical bearing at the steering head on the main frame. The arms of the pivoted fork are bowed outward for tyre clearance on full lock and the pivot on the main frame is so positioned as to minimize the rocking of the telescopic fork at the steering head on bump and rebound.” Except in appearance, it is difficult to see what advantage this system offers compared with, say, the Difazio – but it is superior to the conventional telescopic fork in several ways. In all conditions other than braking, the sliders are relieved of fore-and-aft loading, hence stiction is minimal. Even during braking, these longitudinal loads are only those resisting the caliper forces; the main loads are taken by the pivoted fork. The fork also resists differential movement of the telescopic legs and much enhances lateral stiffness.



**Kileen "head-to-hub" steering. This also is a member of the McPherson strut based family, by virtue of its movement geometry. Similar to the Saxon and BMW telelever except that the support arm is mounted at axle height instead of above the wheel.**

It is sometimes claimed for hub-centre steering that the elimination of the conventional steering head reduces frontal area. In the case of a roadster, this is not so because of the upright riding position. But it is clearly true for a low record breaker such as the fully streamlined, 2 ft-diameter (610 mm.) Harley-Davidson projectile in which Cal Rayborn raised the world record beyond 426 km/h. (265 mph.) in 1970.

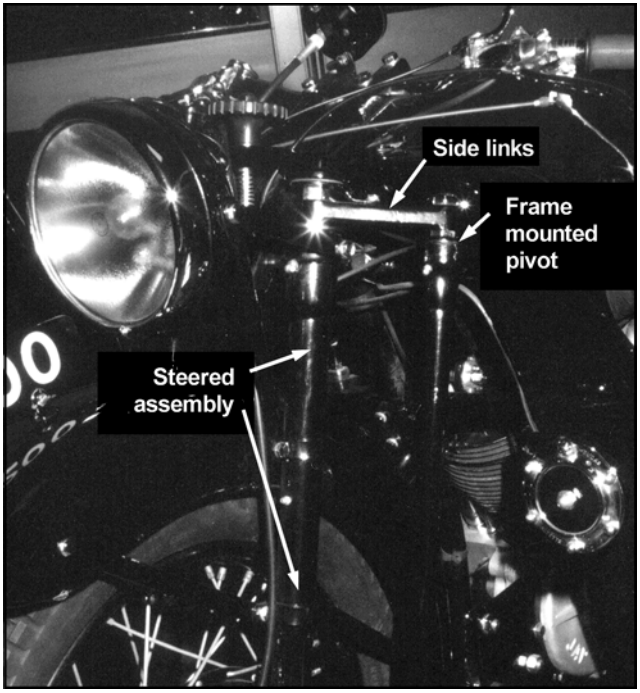
**By eliminating the conventional high steering head, Carl Rayborn's double-link steering layout enabled the streamline shell of this land speed record machine to be kept down to 610 mm. in diameter. Note also the cable steering.** (Weed)

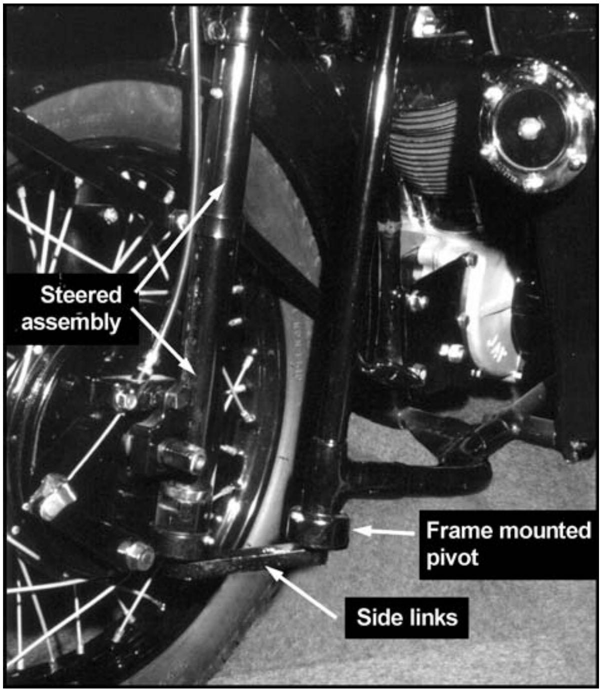
## Virtual steering axis

The idea of using a steering system in which the steering axis is not mechanically fixed in the centre plane of the motorcycle has found few adherents. An alternative to the norm is a system in which the instantaneous steering axis is defined by the virtual centre of a four-bar linkage, as explained in appendix 6. The old OEC had two such mechanisms, one below axle height and the other well above the wheel, just below the handle-bars. The steering axis was defined as the line through the upper and lower virtual centres of these mechanisms. This gave a steering motion such that as the wheel turned to steer, the steering axis also moved sideways and there was a small longitudinal movement as well. This machine was in production around the 1920s or 1930s and so is very difficult to compare with today's machines, but in its day it did have a reputation for steady steering.

The four-bar steering/suspension linkage is composed of :

- The wheel axle.
- The main frame of the motorcycle.
- Two side links, often called radius arms.




**This OEC is resident at the National Motorcycle Museum near Birmingham, England. The side links only moved for steering and were not pivoted vertically for suspension movement. These links held two upright steered "tubes" which held the wheel and suspension.**

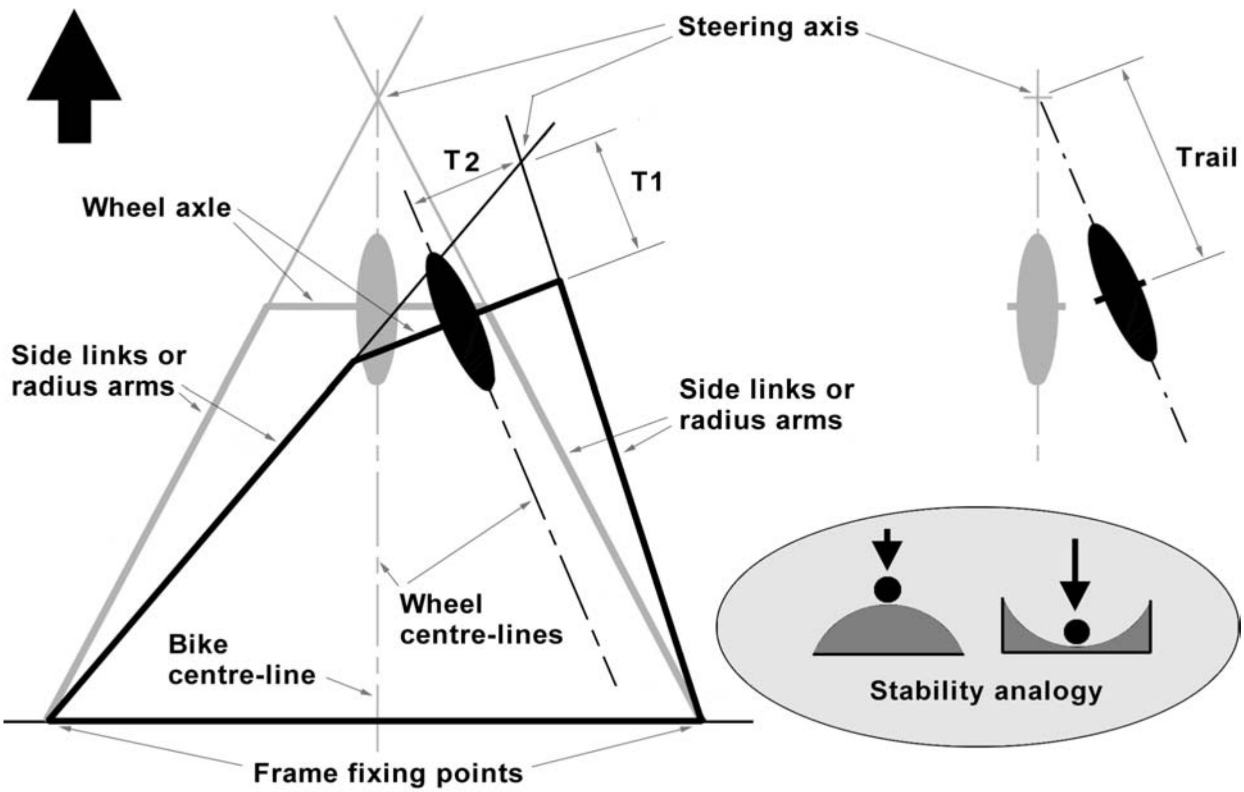



Fig. 7.11 **The geometric aspects of a four-bar linkage steering/suspension system. The sketch to the right represents a normal head stock steering design. Both drawings show the projection onto the ground. The grey coloured lines show the steering in the straight ahead position and the black shows it steered approximately 20° to the left. The tyre contact patches are shown as ellipse shapes. The inset shows an analogy to the directional stability characteristics of each type.**

Fig. 7.11 shows the kinematic layout of such a mechanism as projected onto the ground plane. There are several aspects of the steering movement that differ from a normal head stock design, also shown for comparison.

The instantaneous steering axis moves both laterally and rearward.

The centre of the contact patch moves rearward as the steering is turned, this is in contrast to the head stock design where the patch moves forward. As any drag and road disturbance forces are likely to act in a general rearward direction, the link system will have a tendency to turn away, whereas the head stock design will be more stable and tend to straighten up. This will be especially important under braking. An analogy using a ball on a mound or in a hollow is shown in the inset to fig. 7.11 demonstrating that when the ball (analogous to the tyre) moves against the force, as in the hollow, the force acts to return the ball to the original stable position.

Change of trail. The trail of a conventional design hardly changes with moderate steering angles but it is easy to see that it reduces considerably in the case of the virtual axis system. In fact the situation is more complex and we can consider the trail as two components, shown as **T1** and **T2** in fig. 7.11.

Imagine the case where the front wheel is slightly steered whilst braking, at the tyre contact patch there will be a resultant force in a general rearward direction, partly comprised of the braking force and a lateral force due to the slip angle. We can resolve this force into two components, one in line with the wheel and one acting laterally along the wheel axle. These two components each act to produce torques about the instantaneous steering axis, depending on their respective lever arms. These lever arms are the two trail components **T1** and **T2** from fig. 7.11. We can see that the braking force in line with the wheel acts on trail **T2** and this tends to increase the steer angle, i.e. it is directionally unstable. The lateral force acts on **T1** and is a stabilizing reaction, as it is in the normal fixed steering axis system.

Under non-braking conditions, the drag force in-line with the wheel will be greatly reduced, consisting mainly of the force to accelerate the rotary inertia of the front wheel, residual bearing, brake and aerodynamic drags. So the overall effect is likely to be stable due to the lateral force acting on **T1**, but note that this is reduced considerably when compared to the value of **Trail** in the "normal" case.

It is hard to see any compensating advantages that such four-bar linkage systems have over either, the traditional fixed head stock design, or the various hub centre and double link systems discussed above. The only known production example, the OEC, had a relatively rigid structure for holding the frame end of the links and maybe its reputation for stable steering stemmed more from that rigidity than any inherent advantage in the kinematic layout. In any case it is certain that the brakes on bikes of that era were grossly inferior to today's, and so the braking instability problem would be better hidden.

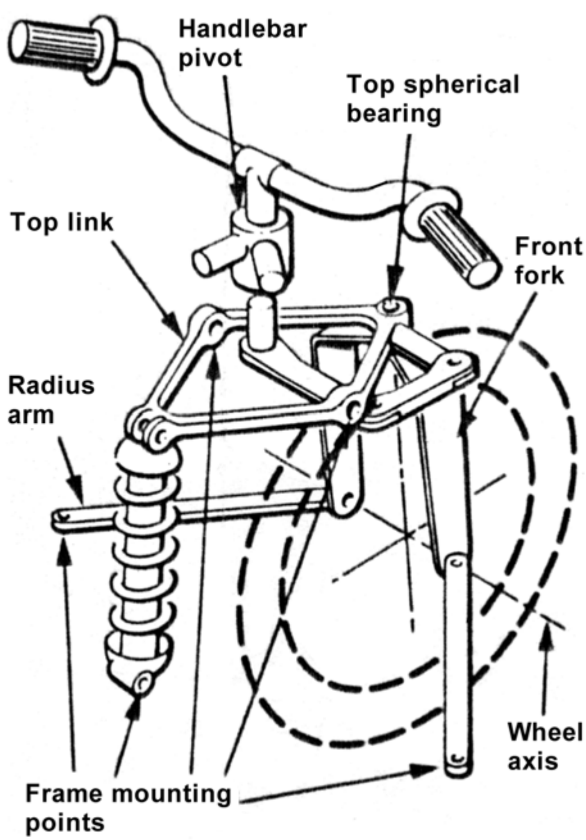



Fig. 7.12 **This design by Mike Loasby, a car engineer, features a combination of the four-bar linkage at axle height and a single ball joint centrally disposed above the wheel. So the steering axis passed through a fixed location above the wheel and a virtual centre at the axle. Due to a leverage effect, the ground projection of the motion will be approximately double that at the axle. However, another interesting motion takes place, as the steering is turned the lower part of the tyre moves sideways more than the top. Therefore increasing the camber angle into the direction of a road curve. This may help speed up corner lean-in during the initial countersteering phase.**

**As drawn, this design would suffer from camber bump steer. Unless the steering is in the straight ahead position there will be some camber change as the suspension moves through its range. Thus causing a changing camber thrust.**

# 8 Rear suspension

In one form or another, the trailing pivoted arm or fork (commonly known as swinging arm) has long been established for rear suspension, although it is far from perfect it is difficult to think of a better alternative. The swing-arm rapidly superseded the widely used but technically poor *plunger* type rear suspension in the 1950s. With the notable exceptions of the Vincent and some early Moto Guzzi racers, it was long customary for the fork to comprise simply a cross-tube to house the pivot bearings, and a pair of side tubes to support the wheel and suspension struts. Clearly this elementary layout lacks torsional stiffness and preferably needs two suspension struts to avoid torque from suspension forces.

The most weight-efficient way to eliminate these defects is to triangulate the fork and connect the apex to the springing medium, as patented by Vincent-H.R.D in 1928. Nearly half a century later, Yamaha 'reinvented' the idea (initially for motocross), which acquired a semblance of novelty through the terms monoshock and cantilever. Possibly to avoid a charge of copying, as well as to keep the weight low down, Suzuki triangulated the fork on the RG500 road racer below pivot level (as did Moto Guzzi in the mid-1930s), though retaining two shock absorbers. These were steeply inclined forward and fitted with either pneumatic suspension units or progressive-rate springs. (Refer to fig. 8.5 and illustrations in the first chapter.)

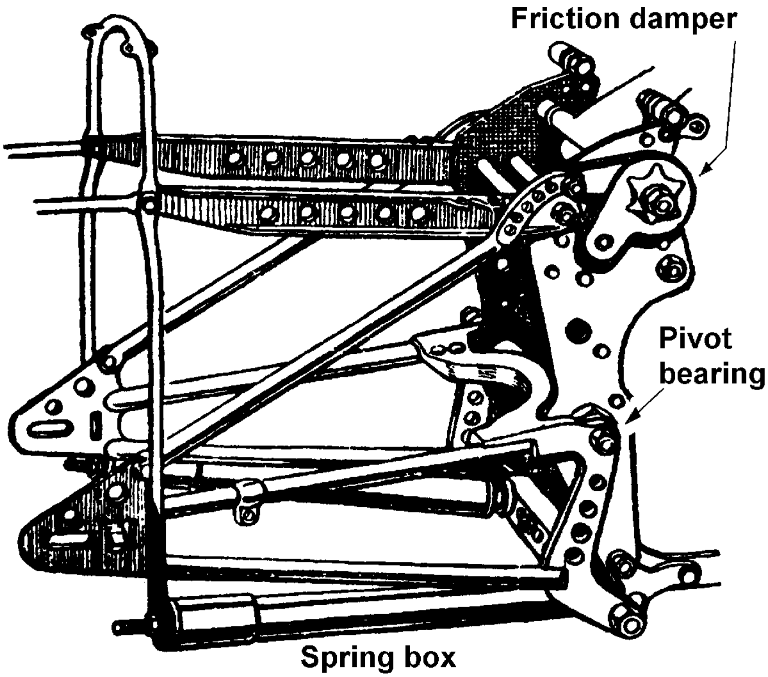



Fig 8.1 **Original (mid-1930s) Moto Guzzi pivoted rear springing. The fork is triangulated below pivot level and actuates long springs in horizontal boxes; upper links connect to friction-type dampers.**

More recently there has been a trend towards rocker-type rear suspension, with a whole new jargon such as Kawasaki's ***Uni-track***, Honda's ***Pro-link*** and Suzuki's ***Full Floater***. etc.. An early example of a link system for racing, in the 1970s, was the French Godier-Genoud endurance-racing Kawasaki, with the pivoted fork triangulated downwards and connected to the suspension strut via a bell-crank. The aim of all these designs is normally to obtain progressive-rate springing and damping by geometric means. If progression is desirable, then this is maybe a good way to achieve it because the springing and damping rates vary together. To achieve a progressive effect, we need to turn a link or lever through a large angle for a given linear movement, and this necessitates a short lever. All the rocker systems have this in common. Provided they give similar changes in effective spring rate (measured at the wheel spindle), and due regard is paid to stiffness and weight, then none of the layouts has any particular advantage over the others, despite the makers' claims. Thus the choice of design is best determined on structural and space considerations. It is also beneficial to use the minimum number of joints in the system.

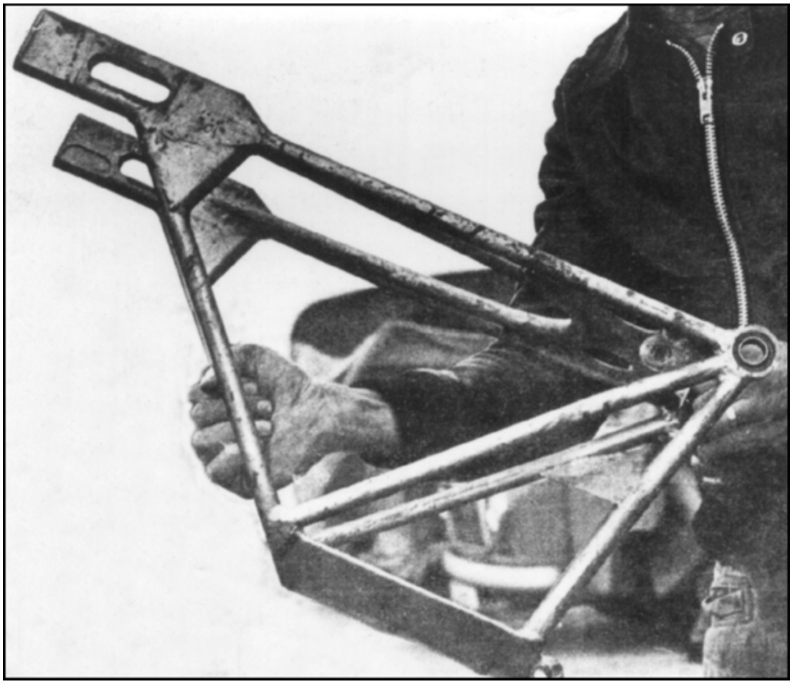


Fig 8.2 **This rear fork for the Godier-Genoud endurance racing Kawasaki from the 1970s is triangulated below pivot level. The link to the suspension rocker arm was connected to the lug at the front edge of the plate shown** (MCN)

At the time of writing there are more variations, in use, of the detail of rear suspension design than there are with front suspension. Basically they are all some form of pivoted arm or fork (swinging arm), but there are single sided and dual, there are single and dual suspension struts, a wide variety of rocker designs and some incorporate drive shafts whilst others have to accommodate the pull of chain drive.

Another consideration for a designer of pivoted rear suspension is pivot-bearing loads. With a simple traditional swing-arm, controlled by a pair of struts mounted near upright at the rear, suspension forces place very little load on the pivot. However, the Vincent layout and all the rocker-arm systems considerably increase these loads. Fig. 8.3 shows what happens in some selected designs. In most cases, this may not constitute a serious problem, but bearing life may be shortened and the design of the pivot mounting on the main frame must take these increased loads into account. Generally, on a chain driven machine the chain pull creates more load on the pivot than the suspension, but both must be added to get the total effect.

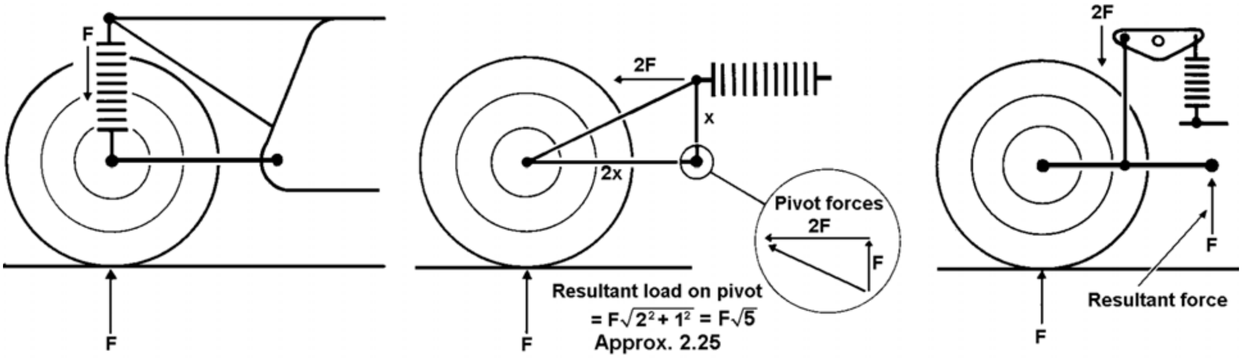
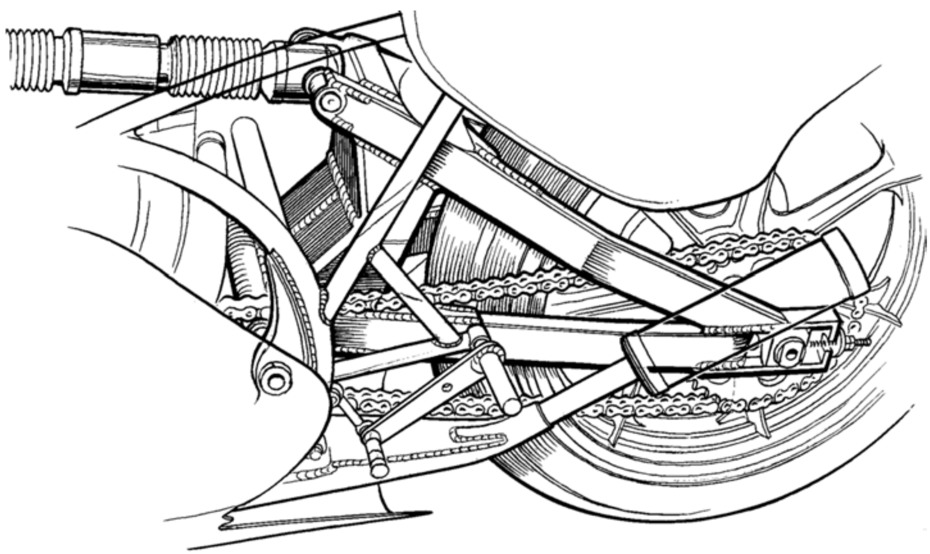



Fig 8.3 **The simple traditional design on the left balances the rear weight directly by the suspension struts and so there is no weight-induced load on the fork pivot. The centre sketch shows how the pivot load can be increased with a mono-shock triangulated fork, in this case the pivot is subject to 2.25 times the wheel load. On the right we see how there must be load on the pivot to balance the forces with a rocker system. Each layout must be considered separately for there are an infinite variety of possible designs. The chain pull on a large machine is likely to create much greater pivot forces.**



**One of Yamaha's many interpretations of the 'cantilever monoshock' theme. The swing-arm is fabricated in aluminium alloy, note the heavy boxing in from the pivot up to the suspension unit mounting. A very long suspension unit extended up to a mounting behind the head stock.**

**This feature was often erroneous claimed in press articles as helping to put more weight over the front wheel. This is complete nonsense of course.**

## Effective spring rate

Unless the suspension unit is mounted as in the left side sketch of fig. 8.3 the spring rate at the wheel will be different from the spring rate of the suspension unit, usually less. In fact the relationship between these two rates follows a square law, that is, if the leverage ratio is such that the wheel movement is double that of the suspension unit then the actual spring rate will have to be four times that of the required effective wheel rate. Fig. 8.4 shows why.

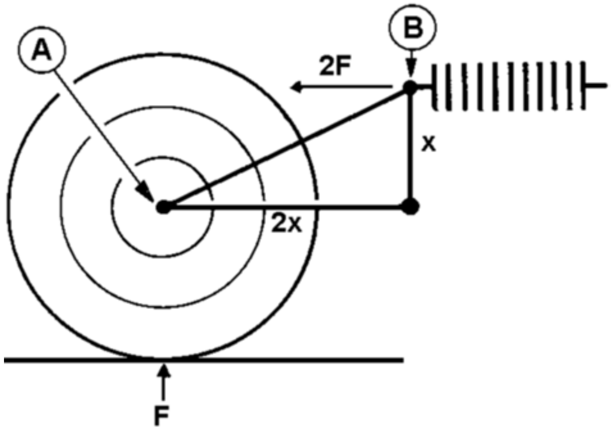



Fig. 8.4 **Typical 2:1 leverage system. The wheel load (F) is half the load on the spring (2F) but the vertical movement at the wheel (A) will be double the spring movement (B).**

**Let the wheel movement be represented by $\delta$,**

**then the wheel spring rate = $F/\delta$**

**the suspension spring rate = $2.F/(\delta/2) = 4.F/\delta$**

**That is, the suspension spring rate needs to be 4 times the required wheel rate, for the case of the wheel to spring movement ratio equal to 2. In general, this relationship follows a square law.**

In this age of the widespread use of rocker and linkage type suspensions it is often forgotten that the more traditional systems can produce progressive or regressive characteristics also. Using a 1970s. RG500 of the type used by Barry Sheene as an example in fig. 8.5, we can see that geometric changes lead to a progressive rate.

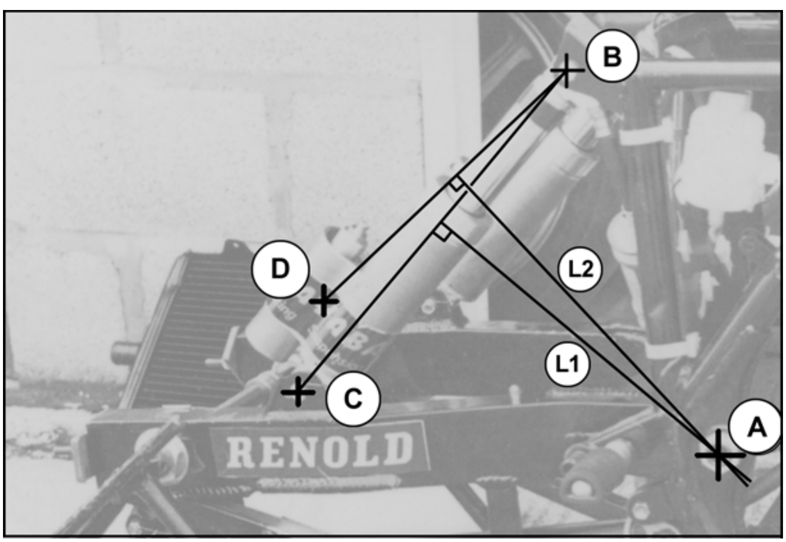



Fig. 8.5 **The rear end of a Suzuki RG 500 racer showing the angled suspension units. Point 'A' is the swing-arm pivot, 'B' is the upper suspension mounting, 'C' is the lower suspension mount when extended and 'D' is the lower suspension mount with the swing-arm in the compressed position. L1 is the extended suspension moment arm and L2 is the moment arm in the compressed state. L2 is greater than L1 and so the effective spring rate will be higher when compressed. Scaling from the photo, L2:L1 = 1.05. The ratio of the two rates is equal to the square of this value or in other words the compressed rate is 11% greater. This is purely the geometric rate to which we must add any additional progressive effect from the suspension units themselves.**

When the suspension is compressed, the line of the suspension force acts at a greater distance from the swing-arm pivot and so has a greater effect on the wheel motion. The difference in effective wheel rates will obey the square law and, forgetting about the inaccuracies inherent in measuring from such a picture, is about 11%.

If the suspension units are angled too much or too little the variation in rate over the range of suspension movement can actually be made regressive, that is the rate decreases with increasing compression. The range of inclination that gives rise to a progressive action can be easily defined geometrically as shown in fig. 8.6, which shows three special case positions for mounting a suspension unit of equal length. ‘A-D’ shows the unit mounted aligned as close as possible to the arc described by the movement of the lower mount over its range of movement ‘D’ to ‘E’. Alignment like this reduces any geometric rate change to the minimum possible. With the strut mounted between ‘B-D’ any suspension movement will reduce the leverage and hence give a regressive characteristic. When the upper mounting is between ‘B’ and ‘C’ the action will be mixed, the initial movement will progressive changing to regressive at full bump.

This diagram was drawn assuming a given length of unit, but for a given required wheel movement the locations ‘C’ and ‘B’ need relatively smaller suspension unit movements and so it is likely that the units themselves would be shorter, but equivalent ‘C’ and ‘B’ locations can easily be found by similar methods.

As well as being shorter the spring rates would need to be higher in these positions.

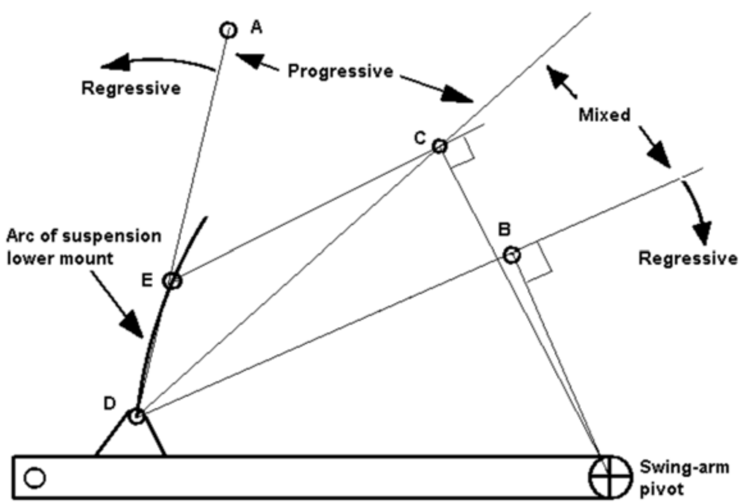



Fig. 8.6 **Defining the range of suspension unit mounting locations to produce particular characteristics. ‘A’, ‘B’ and ‘C’ are alternative upper suspension unit mounting points. ‘D’ and ‘E’ are the range of movement of the lower mounting. When the unit is mounted at ‘A’ there is an absolute minimum change of leverage and so the effective spring rate remains constant. If the upper mounting is to the left of ‘A’ the rate change will be regressive, to the right will be progressive. ‘B’ and ‘C’ are defined as the upper mounting points such that a line from these through the swing-arm pivot is normal to the axis of the suspension unit, ‘B’ is for the extended case with ‘C’ for the compressed situation. When the upper mount is below ‘B’ then the action is regressive through all the range, but if between ‘B’ and ‘C’ then the initial portion of the movement will be progressive changing to regressive as the suspension compresses more.**

## Rocker and link systems

From the early 1970s. a lot of attention started to be applied to improving suspension systems, particularly in the moto-X and enduro fields where there was a rapid trend to vastly increased wheel movements. These increased from a norm of around 100 mm. up to about 305 mm. in a short space of time. Such large movements were hard to accommodate with the traditional placement of the suspension unit, upright near the end of the swing-arm and so it is little wonder that it was in the dirt bike field that attention initially became focussed on designs that applied some kind of leverage to the suspension unit to reduce its longitudinal movement. Yamaha introduced a monoshock system with a triangulated swing-arm and other manufacturers followed with a wide variety of rocker systems. In addition to just being able to use single suspension units with reduced movement these rocker systems allow tremendous geometric control over the spring rate properties. Progressive, regressive and combinations of both are easily achieved. Fig. 8.7 shows how the leverage ratios can change drastically when a short rocker is rotated by a relatively large angle, giving rise to a strongly progressive rate.

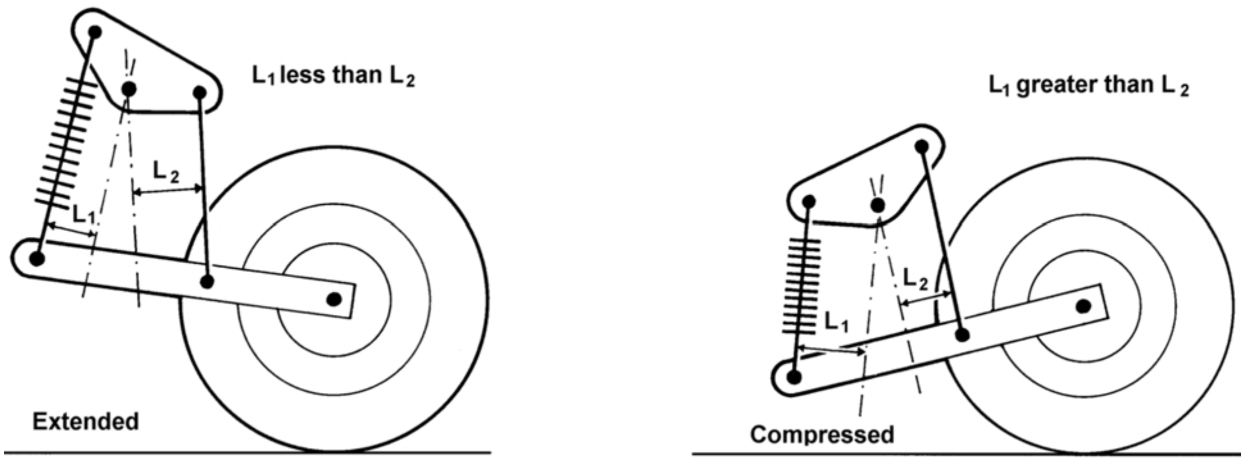


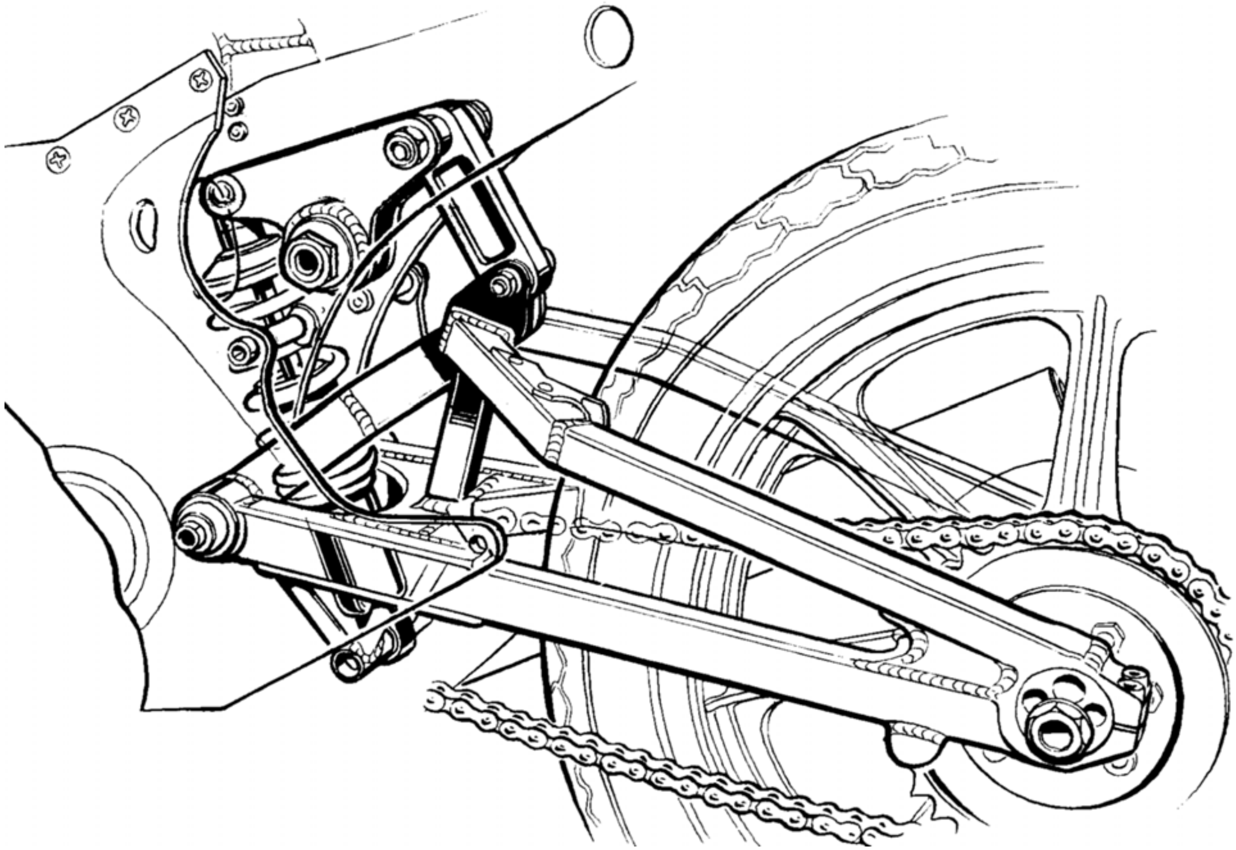

Fig 8.7 **Example rocker system. Because the rocker ratio $L_1$ : $L_2$ varies with wheel movement, this rear suspension system gives progressive rate springing and damping. The effective wheel spring rate varies as the square of the above ratio. In the extended position The side connected to the swing-arm has the most mechanical advantage and so the rate will be softer than in the compressed case.**

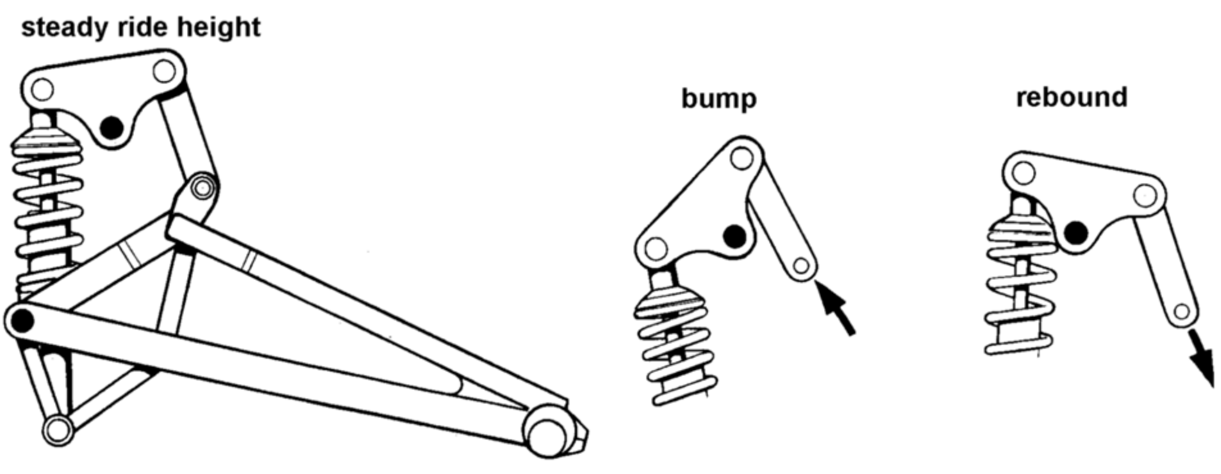
These systems were a godsend to the manufacturers' marketing departments, which must have worked overtime to think up countless new names and acronyms, this was harmless but they also thought up various performance properties to try and counter those from their rivals. A large number of these claims have no basis in fact but have been responsible for much confusion and misinformation. None of these systems have characteristics very different from each other, but in many cases though, manufacturers are forced to use a slightly different design feature to avoid legal problems with a patent on some trivial feature owned by a competitor.

When comparing one design with another there are only a few relevant features to consider:

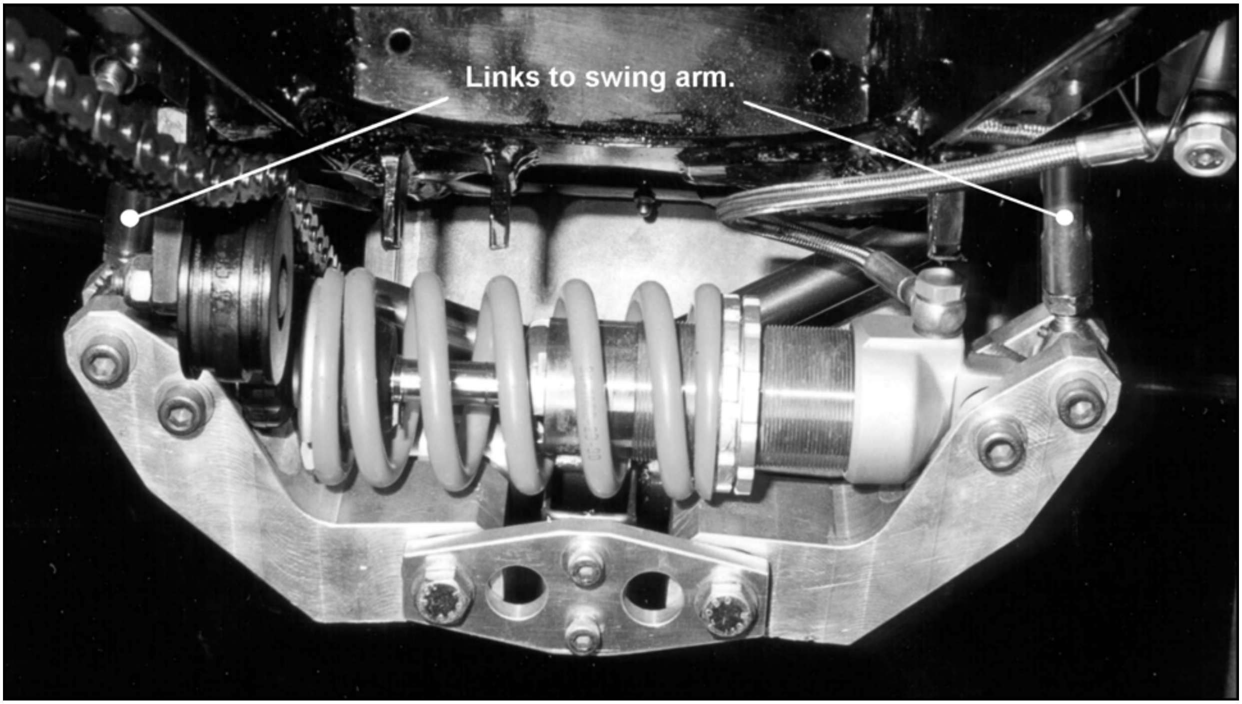
- Weight of the system.
- Unsprung mass.
- Structural integrity. Is it strong and rigid enough?
- Spring and damping rate characteristics. How does the rate vary with suspension compression?
- Packaging. Space is often of prime importance, especially in these days of large capacity air-boxes and the like. The suspension unit often needs some cooling airflow and it would be foolish to have it mounted close to a hot exhaust system. An often conflicting requirement is mount it away from road dirt and flying gravel. Many designs are the way they are because of packaging, although we are rarely told this, it doesn't have a hi-tech ring to it.
- The number of pivots and joints. Each one is a source of friction and slop, both detrimental to good suspension performance. It is thus desirable to minimize the joint count.

The following illustrations show various design interpretations of the rocker system in general.



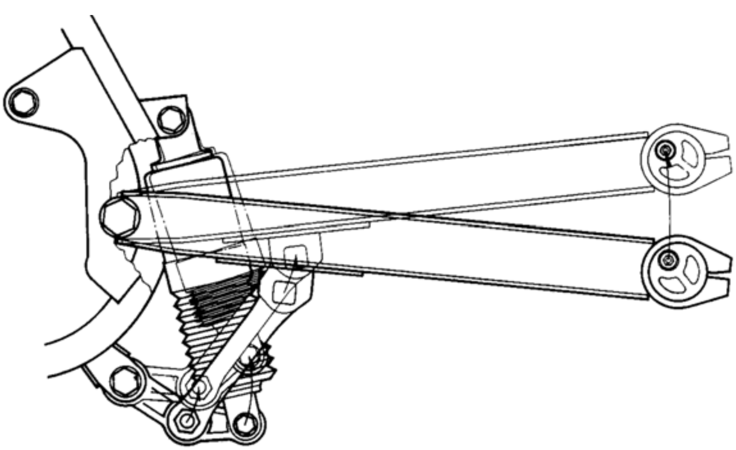



**Kawasaki were one of the first to use a rocker and link system for road racing. Illustrated here fitted to the KR500 with 'monocoque' chassis. The lower suspension mount is attached to the underside of the swing-arm, whilst the mainly horizontal movement here has some effect over the variable rate properties its main advantage is that it eliminates the need for additional frame structure to support the unit.**



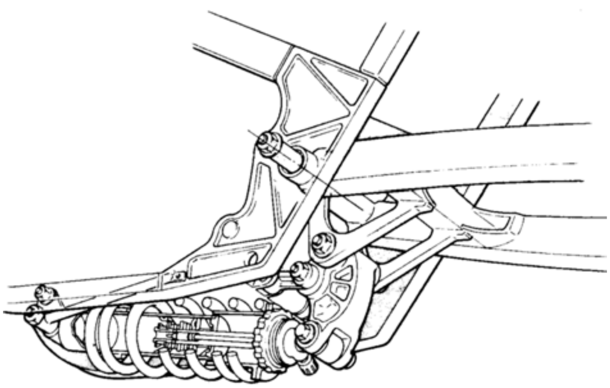

**On the Australian built Drysdale V8 of 1999, space was at such a premium that the suspension unit was mounted across the frame under the swinging arm. Viewed from underneath we see how two vertical links drop from the swing arm to operate the two cranks at each end of the spring unit, compressing it in concertina fashion. Compare this with the 1982 Yamaha OW61 illustrated in the first chapter.** (photo: Greg Parish, ACS.)

A possible advantage of compressing a suspension unit from each end is a reduction in effective unsprung mass. Due to the square law relationship each end of the spring will only contribute one eighth of the unsprung mass, adding the two ends means that the effective unsprung mass of the spring will only be one quarter that of a the same spring compressed equally, but from one end. The effective unsprung mass of the damper is harder to determine because each end does not have equal mass.

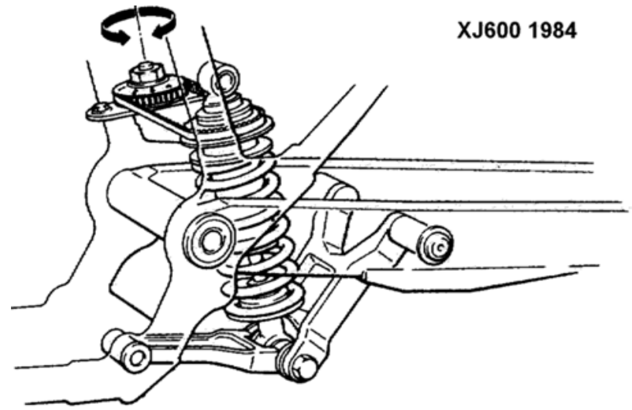


**Kawasaki 'Uni-Trak' from 1985 fitted to the GPz750. The rocker is pulled up by the link from the swing-arm connected at a very small radius from the rocker pivot. The suspension unit is connected at a greater radius and so moves upwards at a greater rate.**

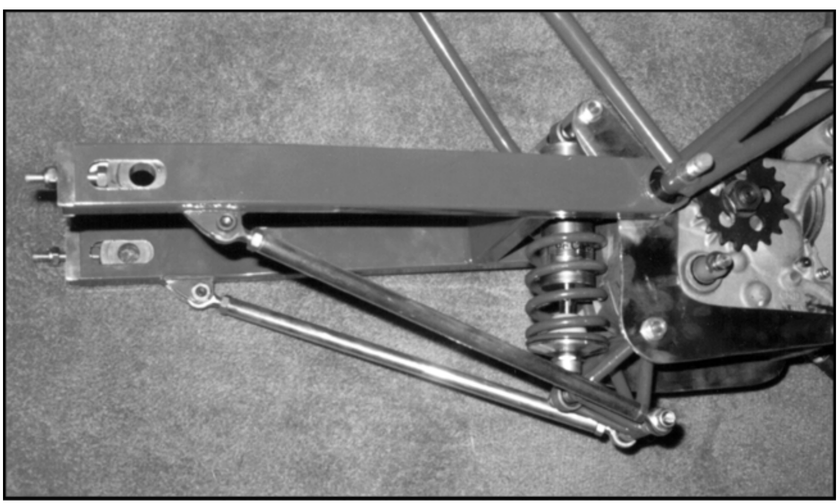
**Despite the passing of nearly two decades from the original use of such designs, there are many current models that use very similar systems. The 2000 Suzuki GSX-600 for example has an almost identical layout.**



**This Yamaha RD500LC of 1984 is interesting because the swing-arm connects directly to the rocker without an intermediary link, this means that the rocker must be connected to the main chassis by way of a short link. As the swing-arm moves upward the lower end of the rocker moves forward compressing the unit.**




**Although made in the same year as the example to the left this Yamaha design has a more usual vertical mounting for the unit. Used on several different models this layout has no rocker in the normal sense, basically it just uses two links to define the movement path of the suspension unit.**



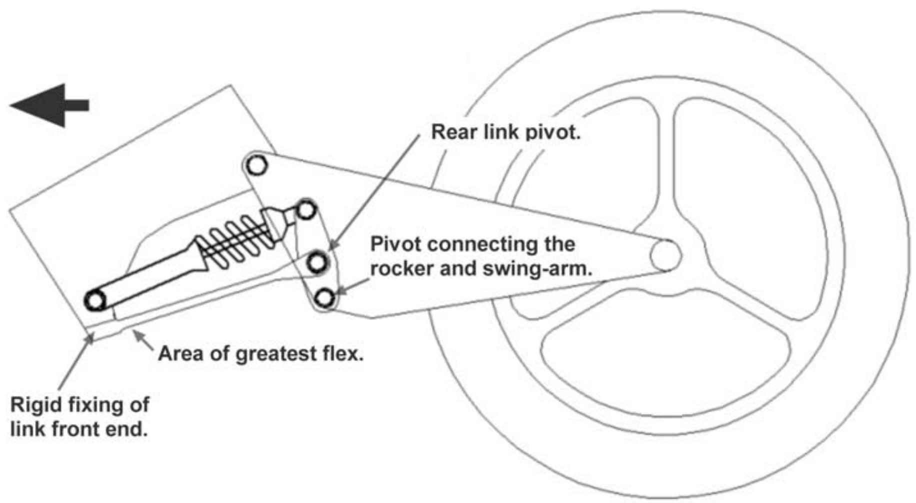
**Space saving in the under-seat area, to leave room for the fuel tank, was the motivation for this design by the author. A racing machine powered by a Rotax 250 engine. The rocker was fabricated from steel tubing as a small strongly triangulated structure, in conjunction with the links from the swing-arm it served as an important structural feature for increasing the torsional and lateral stiffness of the swing-arm.**

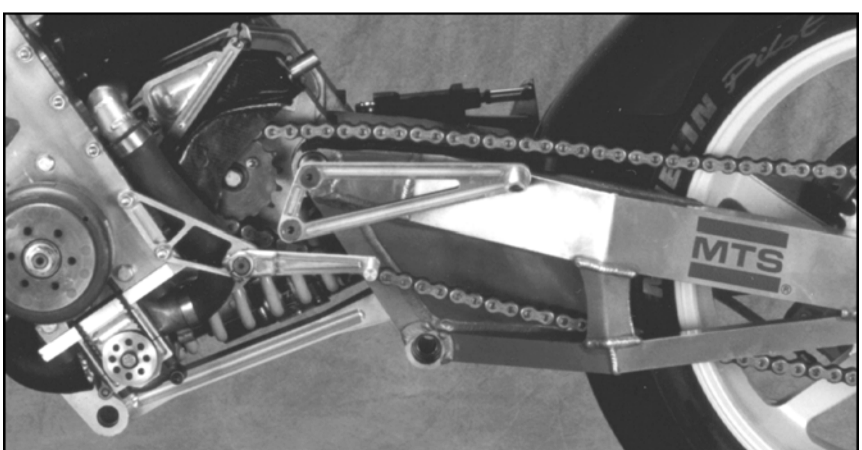
Once the idea of using rockers and linkage became accepted, it seems that most permutations of the design were tried out within a fairly short period around the beginning of the 1980s. Few of the current designs are significantly different, even on GP racing machinery. A notable exception can be seen on the Tul-aris designed by Dr. Robin Tuluie. This uses a titanium “flexure link” which eliminates one pivoted joint in the system, reducing possible backlash and friction. The following figure shows how such a link can be used. The rear end of the link only moves vertically by less than 2 mm. throughout the full range of suspension travel, if the link is sufficiently long this movement can be provided by flex along its length. The front end therefore does not need to have a rotating joint and can be bolted directly and rigidly to the chassis.

It is only over the past couple of years that F1 cars have started to use one dimensional flexure links in their minimal movement suspension systems but the Tul-aris goes one better with the use of a two dimensional version. In other words the link can flex in a horizontal direction as well as a vertical one, this allows for any misalignment caused by swing-arm and/or chassis flex. As shown in chapter 10 flexure can cause failure by metal fatigue and so the design of the link is of extreme importance, as failure could have serious consequences. It is only with the use of modern testing and computer analysis methods that such components can now be designed with confidence. I suspect that we'll see increasing use of this type of link in bikes from the major manufacturers. For racing use there is the possibility of improved performance, and for road machines, manufacturing costs can be reduced. Constructors without the knowledge and facilities for data collection, testing, simulating and fatigue analysis are strongly advised not to attempt the use of designs which rely on the deliberate flexing of suspension components.

**Representation of the Tul-aris patented rear suspension system. The flexure link helps reduce the number of pivoted joints in the design. Each joint is a source of undesirable backlash and friction, degrading suspension performance, which is of great importance at the top levels of competition.**

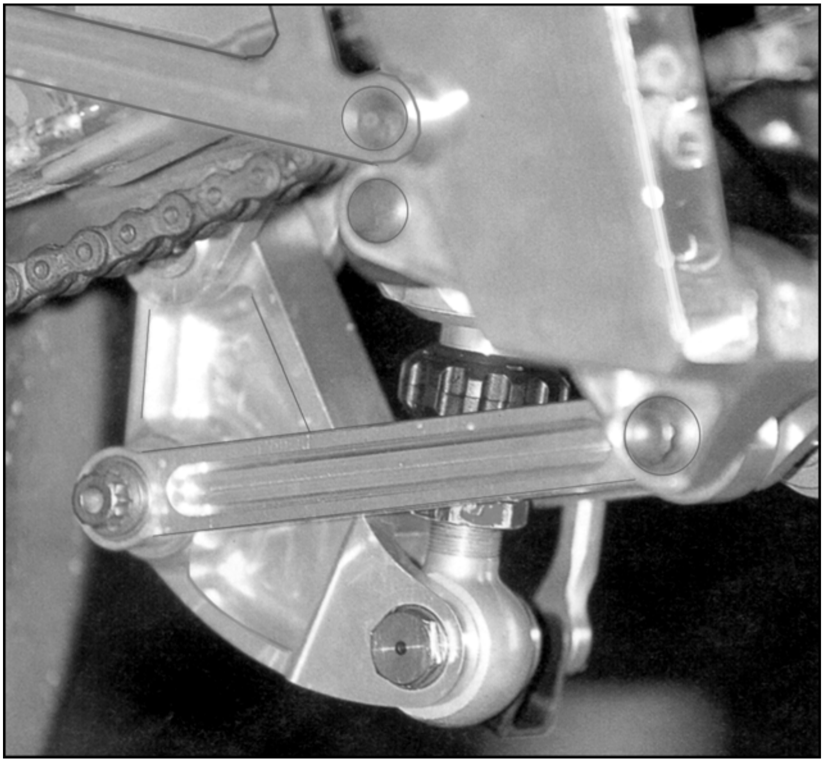
(Dr. R Tuluie)




**The compact nature of the Tul-aris rear suspension can be seen in this photograph. Note that in this stage of its development the titanium suspension flexure link is of a different form to that shown above. The flexure link has now (2002) had a full season of racing without problems.**

(Dr. R Tuluie)



**Even modern GP racing machinery use rocker and link designs which generally have their origins in the early 1980s. This layout mounts the rocker directly to the swing-arm as did the Yamaha RD500LC shown above.**

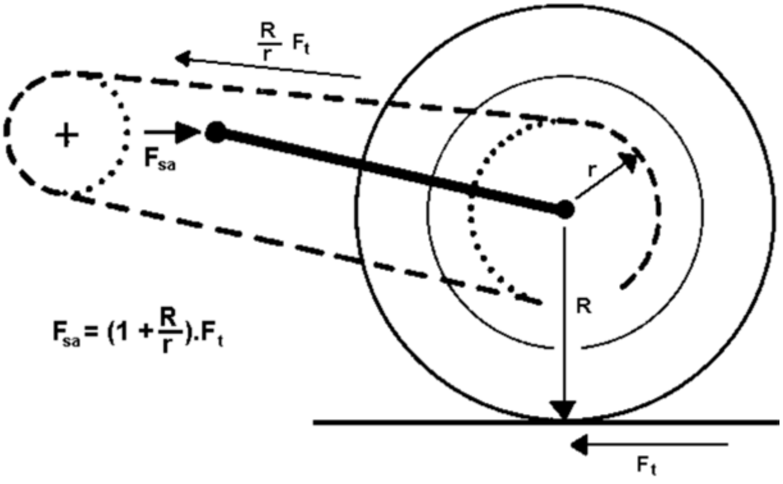
## Chain effects

Fig. 8.8 **Both the tyre force and the force in the chain act on the swing-arm and its pivot. To a first approximation, $F_{sa}$ the swing-arm force is equal to the expression shown in the sketch. Where :**

**r = rear sprocket mean radius.**

**R = rear wheel radius**

**$F_t$ = tyre driving force**




**This ignores the angularity of the swing-arm and chain, but gives a good average indication over the range of suspension travel.**

There are three main aspects to the chain's effect on the rear suspension design:

- Structural.
- Chain slack.
- Anti-squat. This is covered in a separate chapter.

## Structural

The force in the chain is often much higher than generally realized. Typically, the radius of the wheel is between about three and four times the radius of the rear sprocket, and as shown in fig.8.8, the chain force will also be three of four times the driving force of the tyre on the road. The swing-arm and its mountings are subject to both the chain pull and the driving force and so will be in the range of four or five times the force at the tyre.

Imagine a loaded sport bike with enough power to drive the tyre to its limit, if the all-up weight is say 270 kgf. and the tyre coefficient of friction is 1.0 (it could actually be higher or lower than this), then under acceleration most if not all of the weight will be supported by the rear tyre and so the driving force will be 270 kgf. The load on the swing-arm pivot will be four or five times this value, that, is up to about 1350 kgf. or over 1.3 tons. This is likely to be one the highest local point loads anywhere on the bike and so has great structural importance. The swing-arm is loaded in compression but this is unlikely to present any problems, most swing-arms being more than capable of supporting that kind of load. The wheel adjusting system and the pivot bushes or bearings must be designed with these loads in mind. It is the detail of the frame design in the pivot area that has sometimes been poorly done, leading to very bad handling in some cases. Any fore and aft flexure in this area will allow the swing-arm to move toward the chain-side of the bike, when power is applied, thus introducing unplanned and unwanted steering inputs.

When riding in conditions which need changing throttle opening, like a race track or curving road, this steering effect will be constantly changing. In the late 1960s and 1970s when factory development centred more on power output rather than chassis design this was often a major contributor to bad and dangerous handling of both racing and road machines.

The trend to rubber mounted engine/gearbox units also created problems due to chain pull, which was many times the value of the static load that each rubber bush needed to support. Thus, the characteristics of the bushes often had to be selected on their ability to withstand the chain pull instead of on the optimum for vibration isolation. The designers of the Norton Commando recognised this problem and rubber mounted the engine/gearbox and swing-arm assembly as one unit. naturally this had the disadvantage that the swing-arm was now only connected to the main frame and front wheel through a flexible medium. Good handling was only achieved when the lateral preloading (via shims) was correctly maintained.

## Chain slack

Unless the swing-arm pivot (virtual or real) and the front sprocket are coaxial, and in the absence of additional idler sprockets or rollers, then the slack in the chain will vary over the range of suspension travel. This comes about because of the different paths followed by the wheel spindle and the required path to keep the distance between the sprocket centres constant. Fig. 8.9 shows how a normal swing-arm describes a tighter curve than that centred on the front sprocket, this leads to the chain becoming slacker as the suspension moves away from the centre position. The obvious solution is simply to mount the swing-arm pivot concentric with the front sprocket, however, this is easier said than done. It would normally entail a very wide swing-arm at the front end.

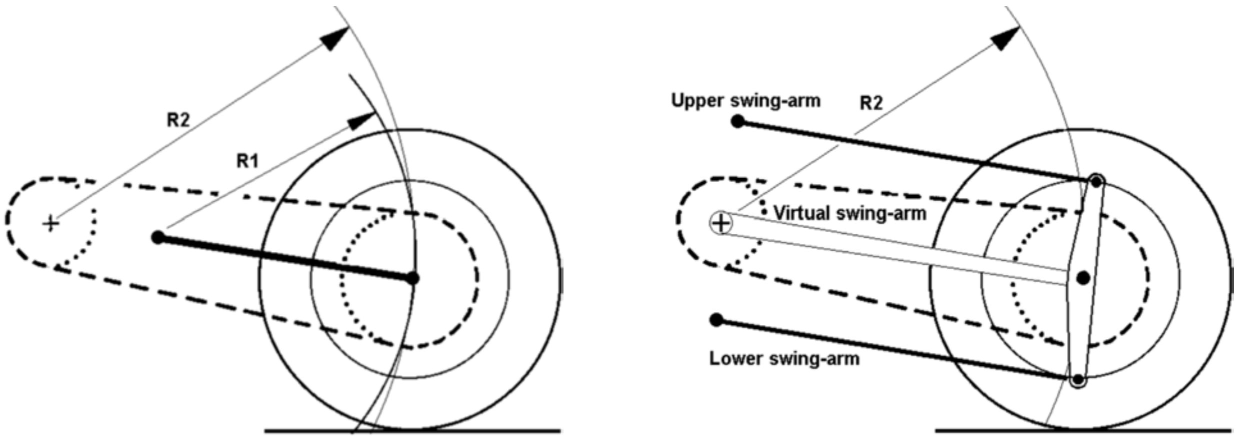
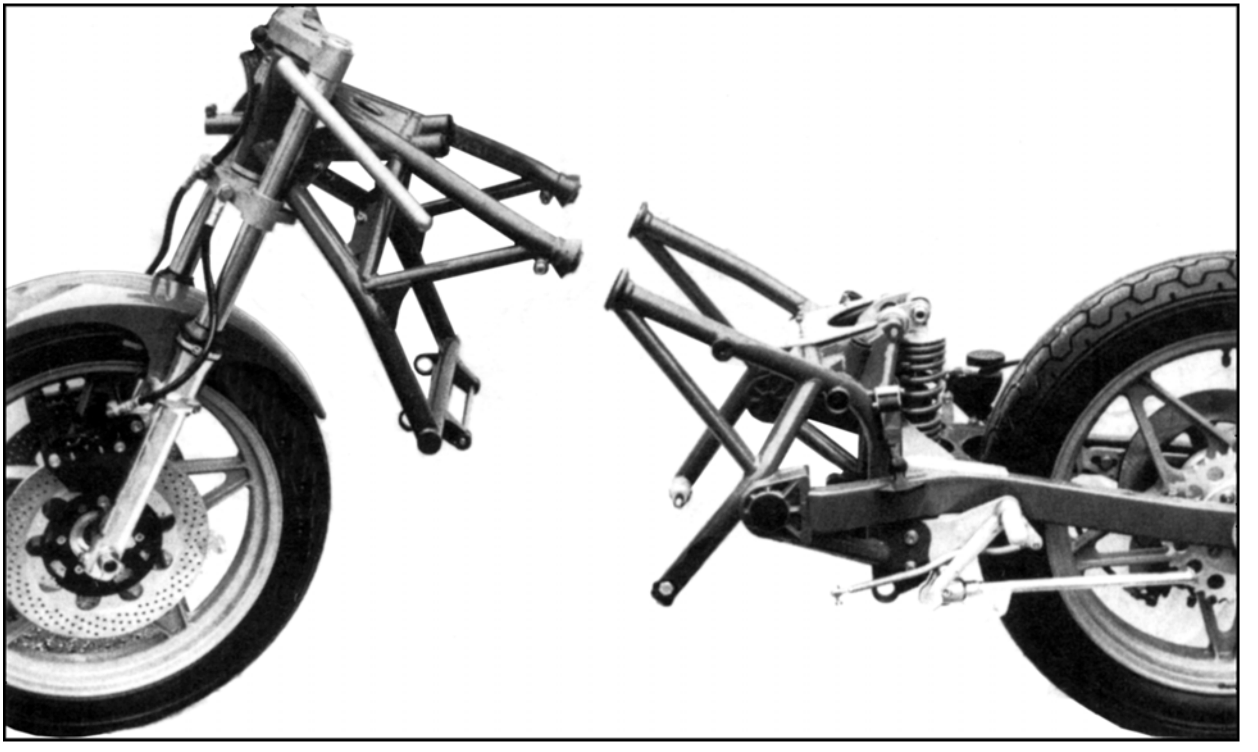


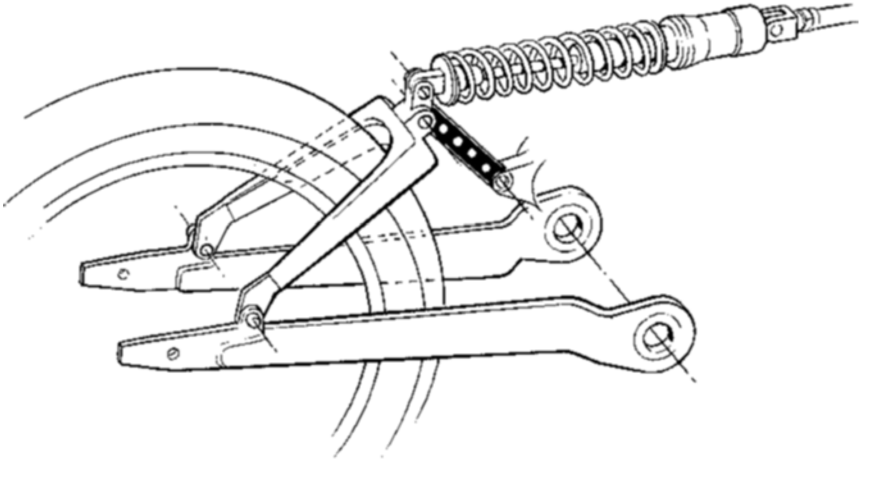

Fig.8.9 **The sketch on the left shows how the path of the wheel movement follows a smaller radius (R1) than that required to maintain constant chain slack (R2). If the swing-arm pivot is concentric with the front sprocket then the two radii are the same and the chain slack will not vary. In most cases this form of construction is not practical and another way to achieve the same end is by the use of a parallelogram double swing-arm system as shown on the right. If the two swing-arms have the same length as the centre distance between the two sprockets then the motion of the wheel axle will follow the same curve as a single swing-arm pivoted about the front sprocket. That is, a virtual swing-arm is created.**

Another method of maintaining more constant sprocket centres throughout suspension travel is to duplicate the swing-arm to form a pivoted parallelogram on each side of the wheel, which is mounted between plates joining the rear ends of the upper and lower forks. The possible advantage here is that positioning the fork pivots further forward may not involve any increase in width. If these swing-arms are the same length as a required single arm then a virtual swing-arm of the required parameters is created as also shown in fig. 8.9. The parallelogram design is only of advantage if it enables the forward width of the arms to be narrower than possible with a single concentric arm. Whilst this has been tried in both road racing and moto-X it has never demonstrated sufficient benefit to be worth the extra weight and complication, despite the often extravagant claims by its proponents. On the other hand a little more success has been attained with the concentric pivot design.

Bimota, on one of their early models used a single arm fabricated from box-section tubing and pivoted on the same axis as the final-drive sprocket to obviate changes in chain slack. Although this is a laudable objective, it involved an extra-long fork, of considerable width at the front end. Unless an engine is specificially designed for this system, the extra width across the gearbox may be unacceptable, but the Bimota had rear-set footrests mounted where the width was less. The Bimota was a high quality, low production machine and its market appeal was determined to a large extent by the use of unusual technical features, we can only speculate as to what degree such marketing considerations influenced the design. It is interesting to note that later Bimota designs dropped this system in favour of putting the pivot behind the gearbox, in the normal fashion.

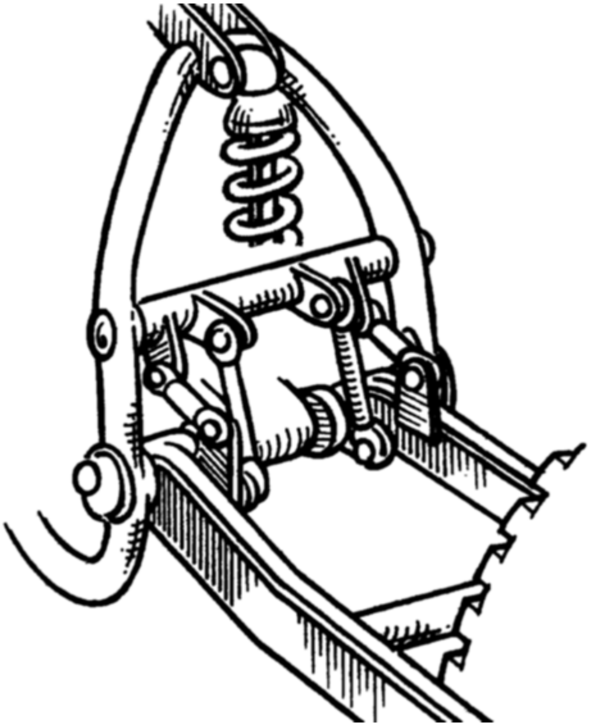


**This early Bimota featured a two-part chassis with the rear fork pivoted on the gearbox sprocket axis. The extra width and weight seem a high price to pay to eliminate relatively small variations in chain slack. Other Bimota models have used the conventional system of mounting the swing-arm pivot behind the gearbox.**



**The Honda NR500 used a coaxial swing-arm mounting as shown. The fairing was structural, 'monocoque', and the pivot was attached to the rear end of this body-work. Space was at a premium and dictated the pivoted U-link and tension link for the attachment to the suspension unit. This U-link and the wheel axle were important to the torsional rigidity of the swing-arm assembly.**

It is on moto-X and enduro machines, with their long suspension travel, that chain slack variation is really likely to be a serious problem. Despite numerous patented designs of varying complexity to counter this, the most generally used method is also probably the simplest. That is, a spring loaded roller or idler sprocket on the underside of the chain run. This simply takes up the excess slack with the swing-arm at the extremes of its range. Of the alternative methods the most practical is probably the use of two chains and an intermediate double sprocket mounted on the swing-arm pivot. The first chain is short and connects the gearbox sprocket to the intermediate one which in turn connects via the second chain to the rear sprocket. The A-Trak system, described in the next chapter, fig. 9.14, has apparently been used with success also, and is easy to retro-fit to some existing machines.



**One of many suggestions to maintain constant chain slack. This system had the swing-arm pivots mounted on an eccentric, as the arm responded to bumps etc., the eccentric would rotate slightly to move the pivot closer to or farther away from the gearbox sprocket to maintain the slack. The links to the swing-arm and the eccentrics controlled this motion. It is not known if this was just a proposal or whether a prototype was built.**

The geometry of swing-arm mounting and its relationship to chain runs is also very important to the anti-squat behaviour and these aspects are covered in a separate chapter. It is important to remember that any design must satisfy many diverse requirements.

## Wheel trajectory

Fig. 7.6 in the previous chapter shows how the rearward motion of the front wheel when hitting a bump is important to the efficient absorption of the bump forces. The same considerations apply to the rear suspension also. To achieve a wheel trajectory like this, the swing-arm pivot must be mounted considerably higher than the wheel spindle as shown in fig. 8.10.

Because of their high build, moto-X machines do approach this sort of layout to a limited degree, but the Track-Lever system as described in the following chapter, follows a much more rearward path similar in angle to that of telescopic front forks. The bump force acts radially inward and the ideal wheel trajectory is in-line with that force, then the force is all directed into the suspension unit before being passed on to the rest of the machine. If the bump force direction is not aligned with the wheel motion then a rearward component of the force has to be reacted by horizontal forces fed directly into the bulk of the machine without first being cushioned by the suspension. Bumps come in different sizes and so it is not possible to design a layout that is optimum in all circumstances, but larger bumps create larger forces and so it is more important to consider these.

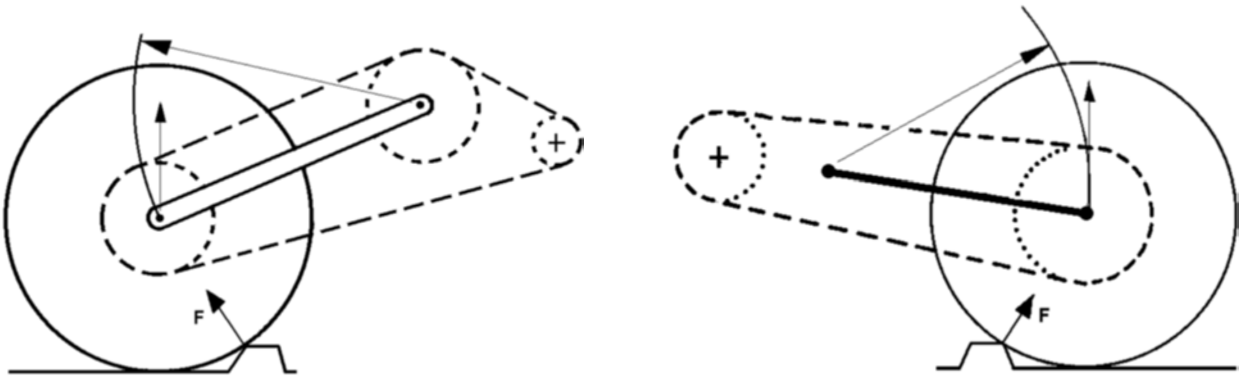


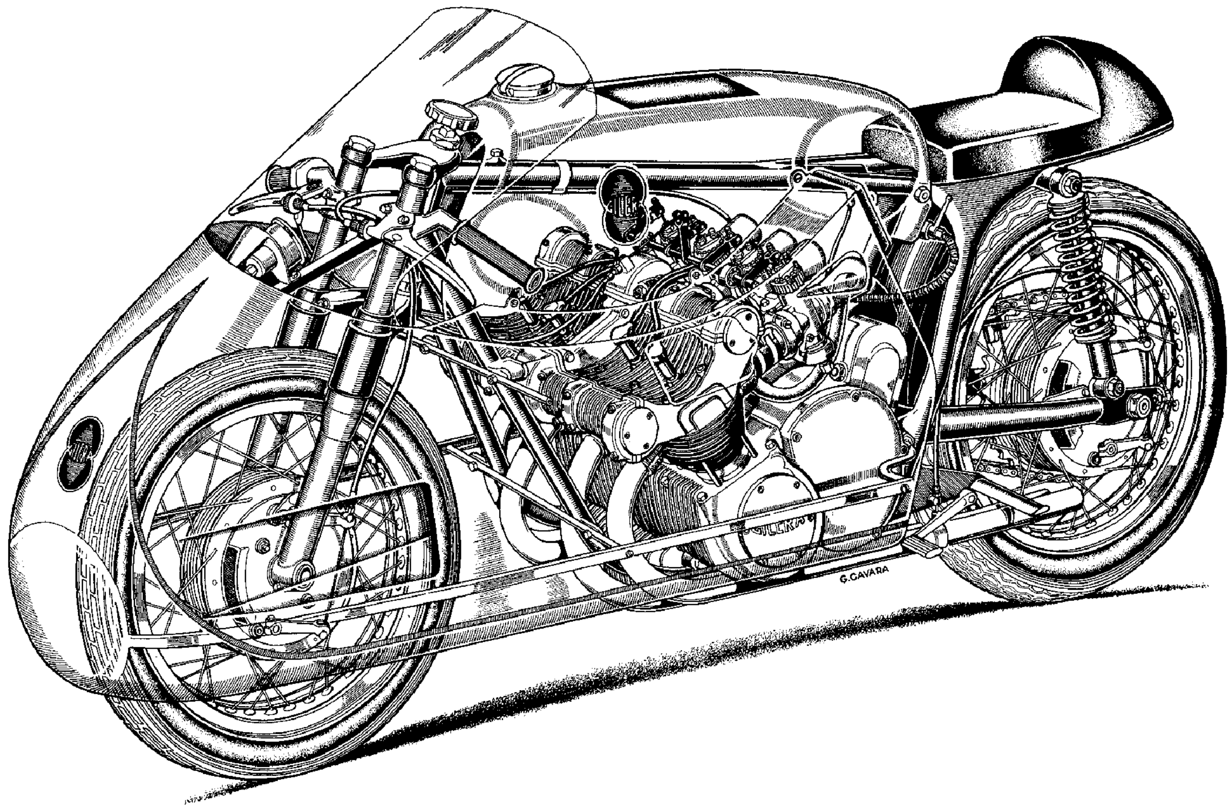

Fig. 8.10 **The sketch to the right shows a fairly normal swing-arm layout, clearly displaying the near vertical path of the wheel. The bump force (F) acts radially inward from the tyre contact point and requires that the wheel moves backward as well as up in order to minimize the impact. The high mounting of the pivot, as shown left, of the Track-Lever system gives this type of trajectory, rearward from the vertical.**

## Structural

Even though several very early designs of swing-arm suspension, such as the Moto-Guzzi and Vincent, were triangulated to give structural stiffness, this aspect seemed to have been largely ignored in the 1950s. and 1960s. when swing-arms began to become universal. Led no doubt by the Norton featherbed, most designs simply used a cross tube for the pivot with two relatively small diameter side arms. These lacked torsional stiffness to such an extent that Girling sold matched pairs of suspension struts as an after market premium option, to reduce additional twisting from unbalanced suspension.

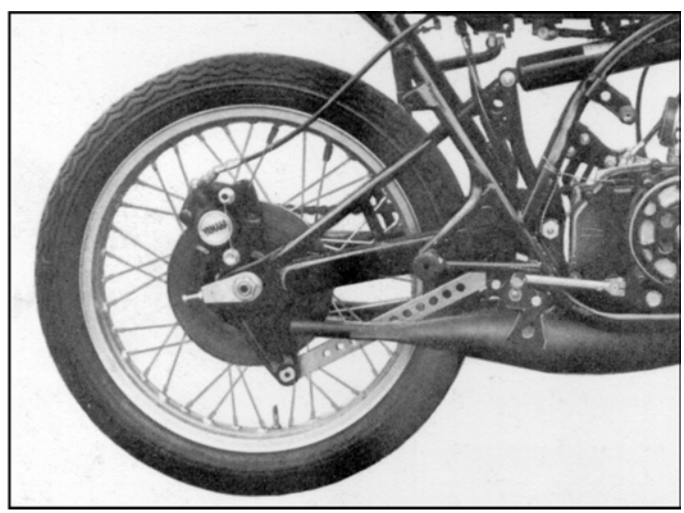
It wasn't until Yamaha resurrected the triangulated design, around the end of the 1960s. and early 1970s., that due attention was paid to structural considerations. Since then swing-arm stiffness has gradually increased up to the present time.

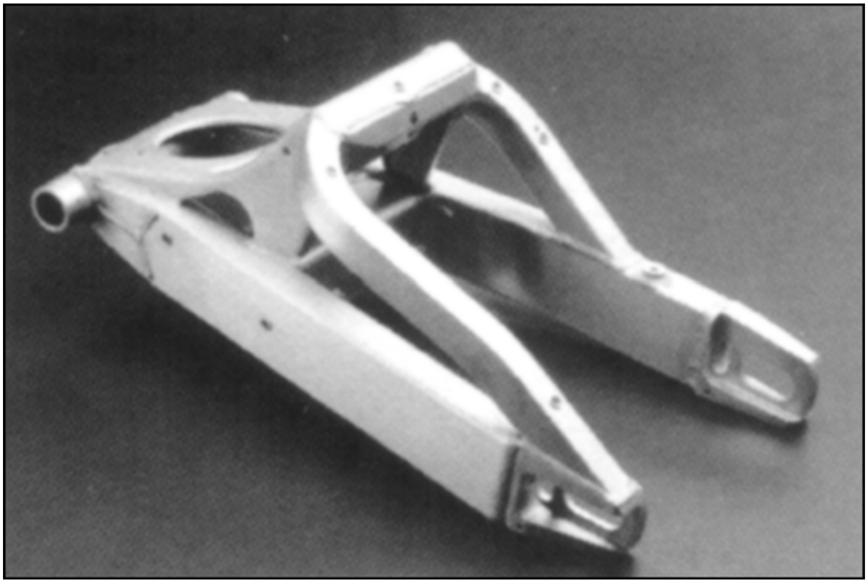
Flexibility in the swingarm allows the rear wheel to move sideways, creating both slip and camber angles and hence steering forces. Fig. 8.14 shows these deflection modes and their effect on the attitude and position of the wheel. As these steering actions are not under the direct control of the rider they may be thought to be detrimental to handling and stability, but to some extent they are similar in action to front wheel trail. Both the camber change and steering deflection act to oppose the disturbing force, and so in some cases may actually produce less steering disturbance. As described in chapter 6, structural stiffness of chassis components in general has now reached the level that may cause problems with bump absorption when cornering, and some manufacturers have been deliberately reintroducing some targeted compliance.



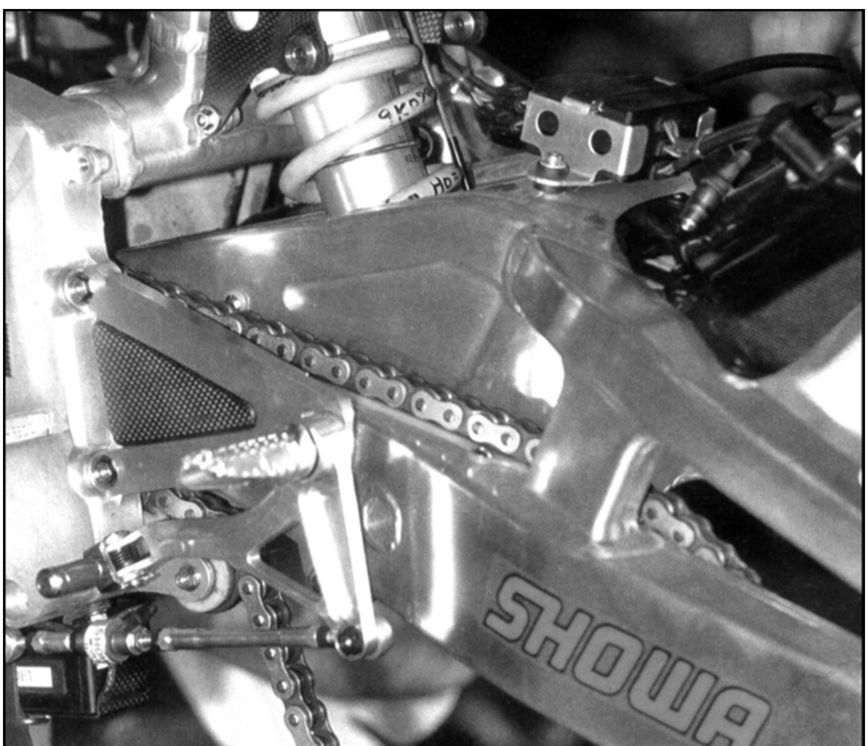

**The simple swing-arm on this 1956 GP Gilera 500-4 was typical of those used by most racers, road and moto-X cross machines during the 1950s. and 1960s. Simple triangulation could have improved rigidity with a minor addition of weight.**

**1976 Yamaha TZ250 used a triangulated structure of small diameter tubing atop an otherwise simple swing-arm. Stiffness was considerably improved and enabled the use of a single unit. One advantage of a single suspension strut is that in general it will be larger in diameter giving more space for the damping mechanics.**





**Fabricated in aluminium this arm from a Yamaha R6 is typical of modern designs. Deep section main arms are braced heavily above. Many racing swing-arms have the whole top section boxed in with sheet material to achieve the maximum stiffness.**




**Typical modern racing swing-arm. It is double sided, made of aluminium alloy and heavily braced. Constructions like this are many times stiffer than swing-arms of old.**

## Single or dual sided

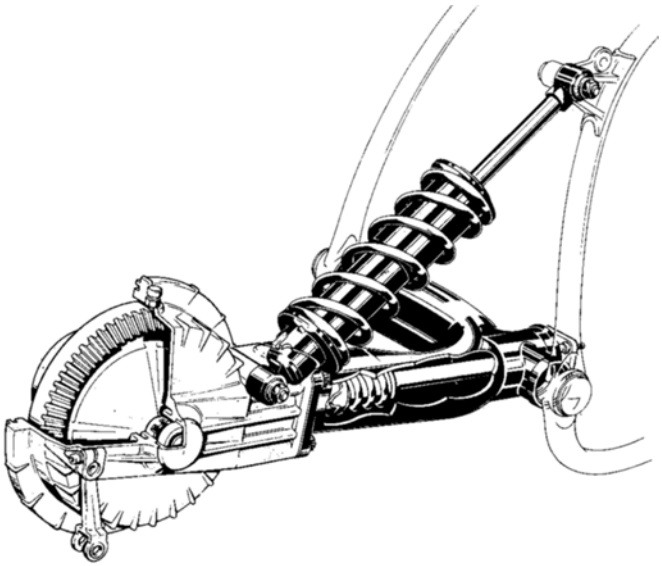
Although single sided swing arms have long been used on small mopeds and scooters, it wasn't until the Cortanze designed, Elf sponsored endurance racing bikes of the 1970s. that the removal of one of the two traditional arms became generally considered as a serious possibility for large sport and racing machines. Since then there has been a mixed reaction to its introduction. Honda have used it on a variety of sports and racing vehicles, both chain and shaft drive, Ducati have succeeded on the racetrack whereas BMW have changed over exclusively to single sided suspension on their shaft drive models, firstly with a simple rigid arm and latterly with an articulated "paralever" system to control the rise and squat. MV have decided to use a cast single sided arm on their new F4 superbike. In fact the use of single sided arms is now much more widespread than commonly thought.


**Single sided swing arms are quite common on low priced mopeds such as this Vespino, and the majority of scooters. The swing arm doubles as a transmission cover and support for the engine unit.**

However, the majority of manufacturers still retain the standard of the past 50 years, the dual sided swing arm, in one form or another. Is this because of tradition, product liability considerations, styling or does the symmetrical two sided arm have inherent technical advantages? This book is only concerned with the technical aspects of that question and to find the answer we need to look dispassionately at what characteristics make a good arm and how the two approaches meet those requirements.

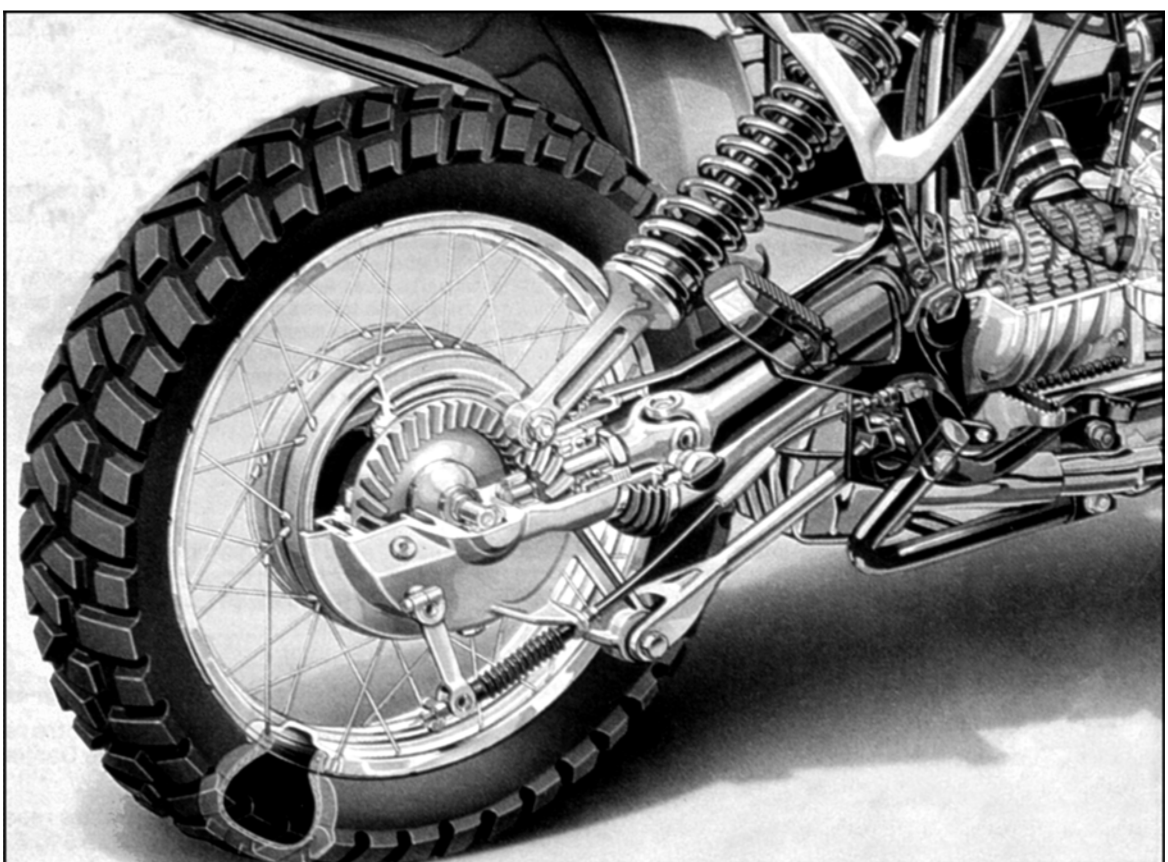
Assuming that the two variants operate with the same geometry (equal lengths and identical pivot mounting points on the frame), and that the suspension springing and damping are equal then there are three main technical criteria to compare, viz:

- Structural stiffness.
- Overall weight.
- Unsprung mass.



*Left*, **Early version of the BMW single sided swing arm for shaft drive. The arm was fabricated from large diameter steel tube.**

*Below*, **Later model "paralever", the arm is cast in aluminium but connects to the rear bevel drive case through a horizontal pivot. This requires a separate "torque stay" under the main arm to prevent free rotation of the drive unit. The frame, swing arm, bevel box and torque stay form a four bar linkage which gives more design control over squat and rise. (See chapter 9 on anti-squat)**



## Structural comparison

As real designs of both single and double sided arms vary so much, it is difficult to compare like with like. The single sided arms for chain drive machines as produced by Honda, Ducati and others, use quite complex shapes, due to the design feature that places the sprocket outboard, and the structural properties of these arms can be evaluated with the use of FEA (Finite Element Analysis) techniques. This is a computer based method that considers the structure as being composed of many small elements joined together, according to set rules, to form the whole. Fig. 8.11 shows an example of how a Ducati cast magnesium arm was divided up for such analysis. Single sided arms for shaft drive machines are generally of simpler construction.

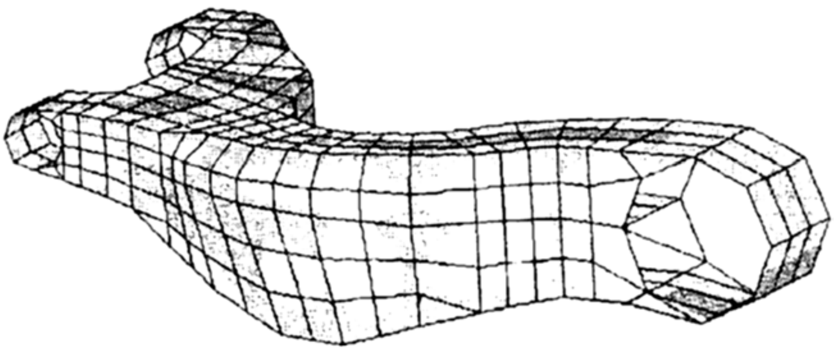
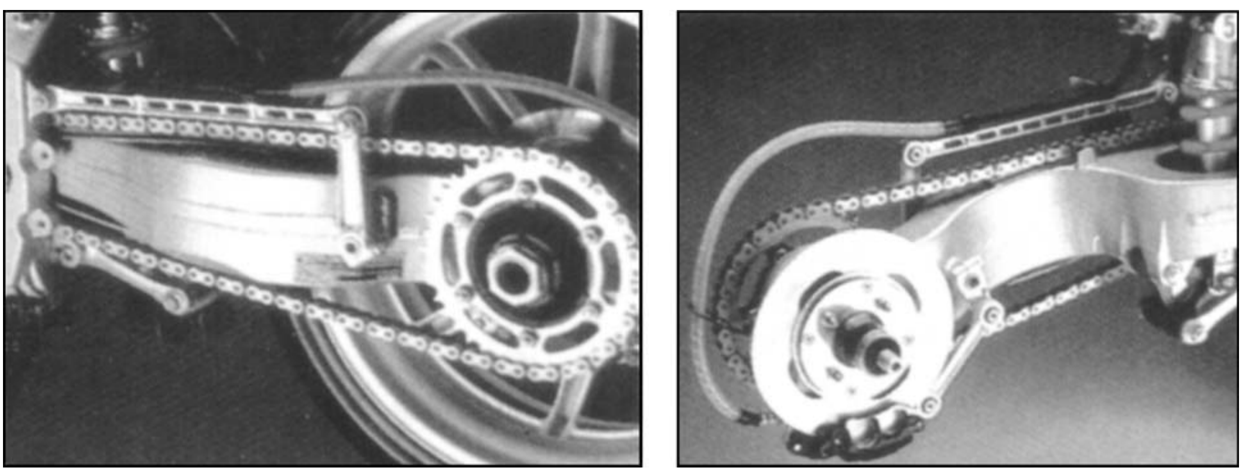


Fig. 8.11 **This example of an FEA model shows how a cast magnesium single sided arm was divided up for analysis by Ducati.**



**Chain drive single sided arm, interpretation by Honda. Note the levers each side of the arm just in front of the sprocket and brake disk, these take the torque from a floating brake caliper back to the main frame. Just like a normal floating caliper this gives control over the degree of rise. In order to mount the wheel bearings between the sprocket and disk the arm is quite complex in shape. This design necessitates a rotating axle which is large in diameter and hollow.**

The deflection characteristics of the two sided swingarm depend greatly on the nature and rigidity of the wheel spindle and its clamping method. The worst, typified by arms with a thin plate at the end slotted for chain adjustment, approach the case of a pin jointed axle. The best are probably found on shaft drive machines with a substantial axle clamp. Fig. 8.12 shows the deflection pattern in bending of the two extremes. The first is with no stiffening effect from the spindle assembly and the second is when it provides complete end fixation. A real world swing arm will be somewhere between these two extremes and the results shown in fig. 8.14 are an average of the two extreme values.

The torsional deflection of the dual arm has three main sources of flexibility that we must consider.

- Twist in the pivot tube.
- Vertical bending in the arms.
- Twist in the side arms, this depends heavily on the spindle assembly as does the lateral bending.

To get an idea if either the twin sided or mono-arm designs have any overwhelming structural advantage over the other let's consider an FEA of simple examples of each, in steel, and look at how they compare. Each type is analyzed in two forms as shown in figs. 8.12 & 8.13. The twin sided example is used with and without a second cross bracing tube, and the mono-arm with and without the gusset behind the pivot cross tube. Both cases use 457 mm. long arms and both have identical pivot tubes of 203 mm. width, 44.4 mm. diameter and 3.17 mm. wall thickness. For the traditional two sided design the arms are 51 x 25 x 1.59 mm. This was a typical configuration long used by many aftermarket manufacturers of replacement arms. The single sided arm chosen is that used on the Q2 (pictured below), which employed a 76 mm. diameter round tube with a 1.59 mm. wall. Thus both are from practical examples.

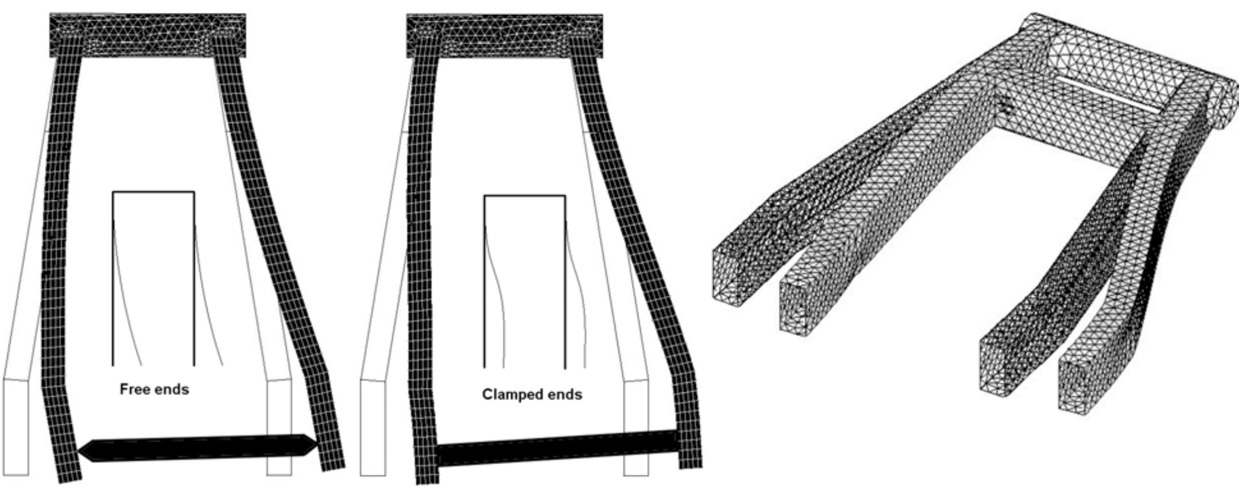



Fig. 8.12 **The lateral deformation modes of the two extreme possibilities for the dual arm design, depending on how rigidly the spindle assembly attaches to the arms. On the left is the equivalent of a pin-jointed connection, and centre is the case with a completely rigid fixing between the axle and side arms. For clarity the two examples are shown with equal lateral displacements, but in reality the pinned case will deform by about 3 or 4 times more. The swingarm with a second cross-tube is shown to the right.** (David Sanchez)

Fig. 8.14 shows the deflections of the wheel that were considered and the results of the analysis. These are shown as being relative to the stiffest in each case. We can see that for the two examples analyzed there is no clear advantage of one over the other. The gusseted mono-arm and the twin-sided arm with the second cross tube have almost equal stiffness against camber change. For lateral dislacement the mono-arm is roughly twice as stiff, but the situation is reversed for the steer deflection. However, the steer resistance of the twin-sided example is very dependent on the layout of the design. For example if the two side arms are parallel then little steer deflection will occur regardless of the lateral flexibility of the arms. On the other hand, if the side arms are splayed out toward the rear, as in most practical cases, then some steer deflection will occur, as can be seen in fig.8.12. Therefore, it is hard to generalize about this characteristic of the double arm design as a type. It is necessary to look at the detail of specific examples.

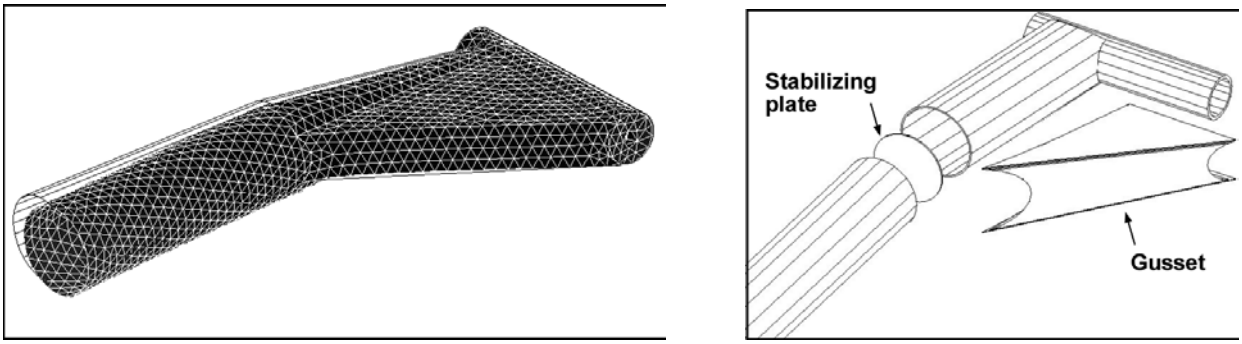



Fig. 8.13 **FEA model of the single sided swing-arm. The illustration to the left shows how the structure is divided up into small elements for analysis. The detail of the internal construction is shown to the right. The stabilizing plate is an example of the attention to detail that should be taken when using large section thin walled members. This plate largely prevents localized deformation at the point where the gusset joins to the side arm, reducing lateral displacement and stress concentration. In this case the plate increased the lateral stiffness by about 30%. The analysis considered two cases – with and without the gusset.** (David Sanchez)

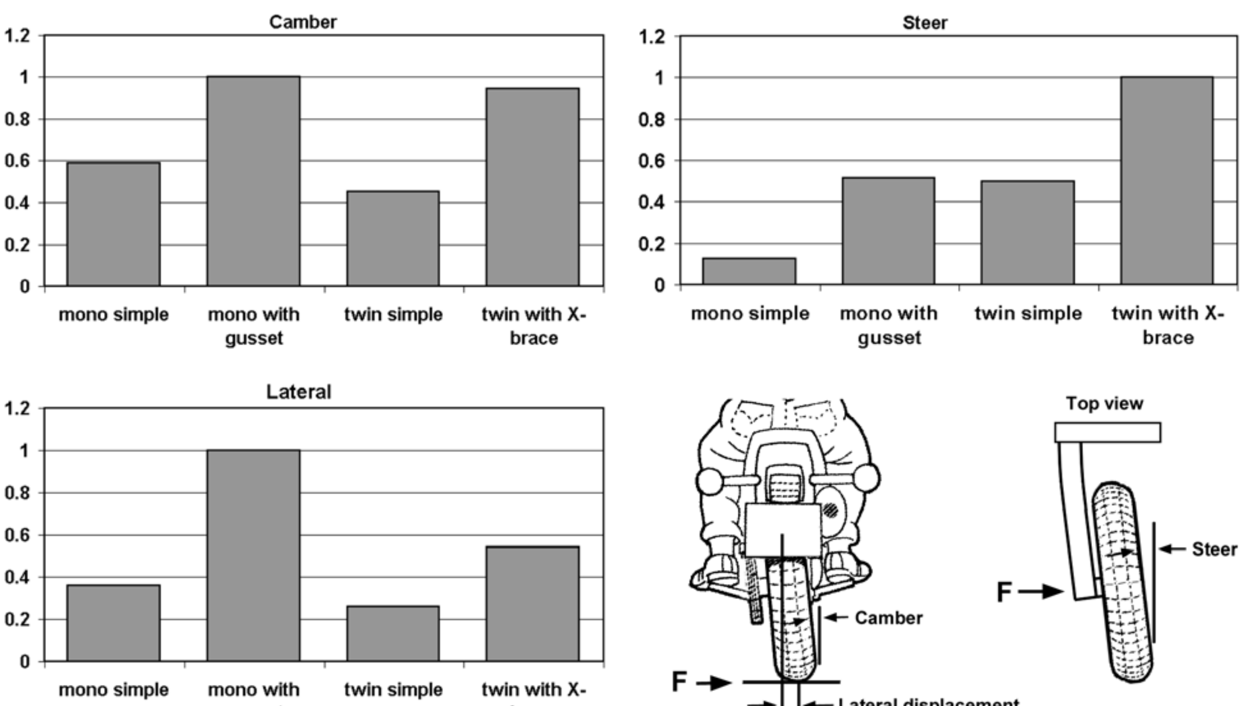
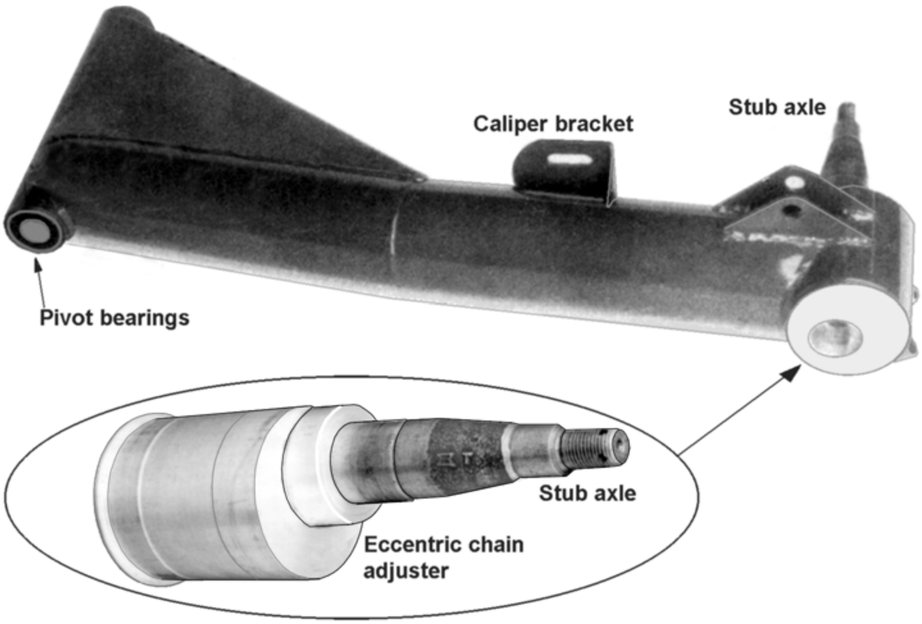
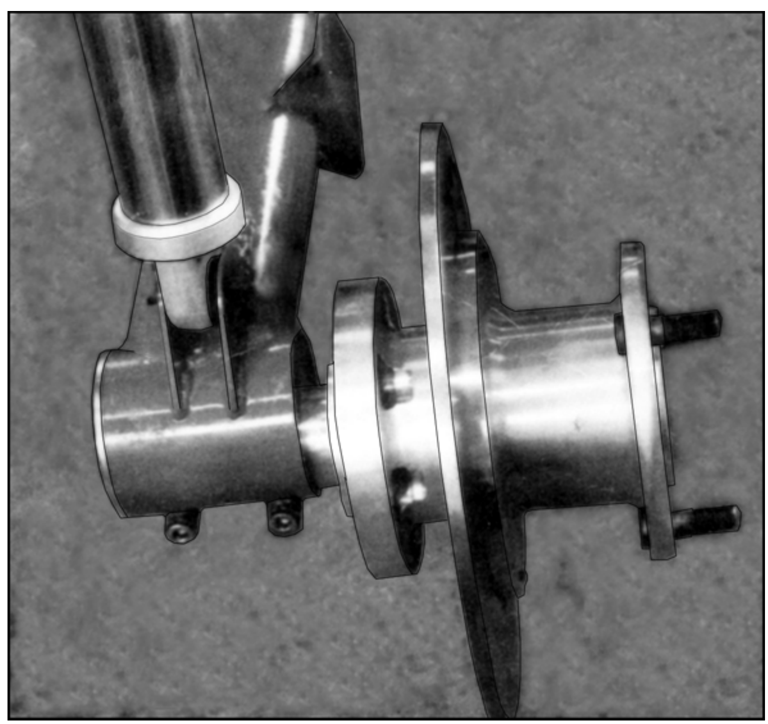



Fig. 8.14 **The results of the FEA comparison. The stiffness values are shown relative to the best in each case. The** *lower right* **sketch illustrates the three modes of displacement considered in the analysis, when subject to a lateral force (F) at the road/tyre junction . Camber and steer are angular motions and the lateral displacement is linear.**




**The mono-arm example used above is similar to this single arm used on the author's Q2 design. The gusseting behind the pivot tube added to the overall stiffness by quite a significant amount as shown in the analysis.**

**Rear hub detail of the Q2. This rotated about a fixed stub axle, shown above, similar to that used on the front of many cars. In fact the axle was taken directly from an XJ6 Jaguar car.**

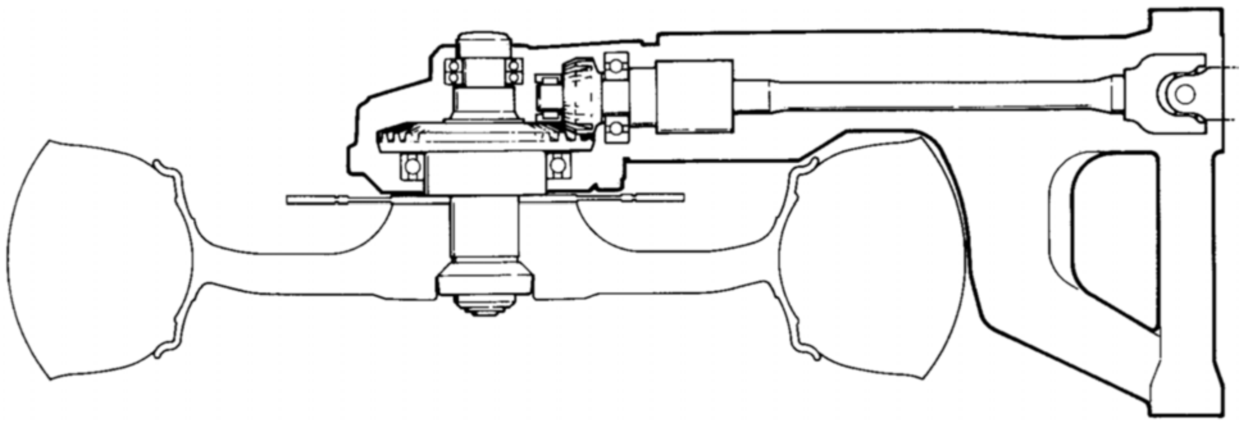


**Showing the above single sided arm fitted to the Q2. Suspension was by means of a single Fournales air shock attached close to the rear of the arm.**

It is clear that the addition of the gusset to the mono-arm and the second cross tube to the double-arm is very structurally efficient. In each case the overall stiffness is approximately doubled for a minor addition of weight.

So it seems that structurally, either style of swing arm can be made to perform quite satisfactorily. The range of stiffness values noted between the two examples being less than the range of values that could be experienced by either type depending on the detail design. Thus the decision of which to choose would normally be based on other considerations. Ease of wheel changing would be a definite point in favour of the single arm, especially for endurance racing, but other considerations such as cost, ease of chain adjustment, ease of maintaining required manufacturing tolerances, styling, tradition and others would be taken into account.

As a general rule, the relative stiffness of chassis components needs to be balanced, for example it is pointless to use extra weight to stiffen a good swing arm that's used with a flexy frame, it would be better to use that extra weight to stiffen the frame first.



**Honda's shaft drive Revere. When the implementation is this easy there seems little point in considering a double sided configuration. Chain drive designs are seldom so simple.**

## Summary

Swing-arm suspension is now universally used design at the rear. Over time structural stiffness has tended to increase to the benefit of handling in general. With one or two exceptions, twin units, one each side of the wheel mounted nearly upright, was the favourite suspension system for a long period. In fact this system is still extensively used, although for dirt bikes, racers and road sport-bikes the system of choice is some form of mono-shock compressed by a rocker and link mechanism. In addition to structural advantages this gives great freedom at the design stage to incorporate progressive or regressive rate characteristics.

There is currently much discussion, but without any consensus, on the worth of single-sided arms compared to the more usual double sided version. In reality both can probably be made to work equally well depending on the quality of the detail design. The ease of wheel changing should be of great advantage in endurance racing. It has been on shaft drive bikes where volume manufacture of single-sided arms has been seen.

Whilst there are various possible solutions to the variation in the centre distance between gearbox and wheel sprockets they are very seldom used. Changes in chain slack are generally just accepted, although the large travel of dirt suspension causes variations too large to be ignored and this is controlled by the simple solution of a spring loaded roller. Any attempts to control chain slack will likely have effects on the anti-squat behaviour and this is explained in the next chapter.

# 9 Squat and dive

## Load transfer

This is normally referred to as weight transfer, but that is really a misnomer. Weight is the gravitational attraction of all the particles in the bike towards to the centre of the earth, and for convenience we usually consider the sum of these forces to act through the CoG. Neither acceleration nor braking can cause this weight to transfer elsewhere. As a result the use of the term ‘load transfer’ is preferable.

We’re all familiar with load transfer. Every time we apply the brakes or open the throttle we can feel the tyre load lightening at one end whilst increasing at the other. The amount varies from bike to bike but one thing that is the same for all bikes, but often forgotten, is that the total vertical load supported by the two tyres remains the same under steady conditions. In other words, if under braking we transfer 50 kgf. to the front wheel then we also unload the rear by 50 kgf., the total steady load on both tyres must sum to the total bike weight. So if we lift the front wheel under acceleration then the rear tyre will be supporting the whole weight. Sometimes it is suggested that a certain type of anti-squat or anti-dive system will force the tyre into closer contact with the ground enabling greater braking or acceleration. Except for momentary transient conditions, the total vertical load supported by the tyres will always be equal to the total weight of the bike and rider (plus any aerodynamic down force). As we shall see these transient conditions are very important to the dynamic performance of the bike in general.

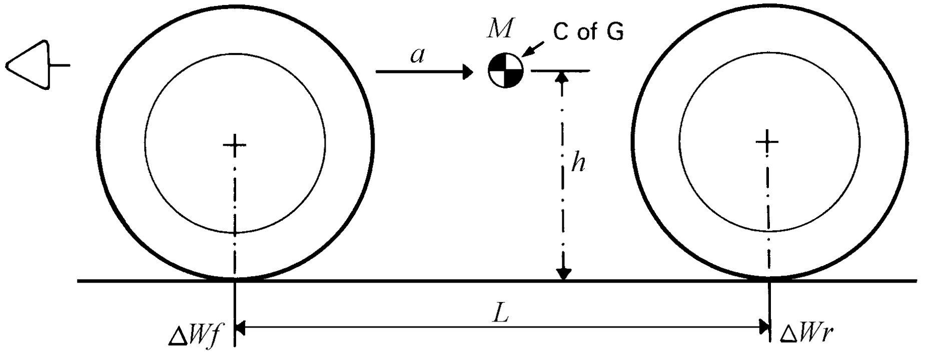
Motorcycles experience these effects to a much greater extent than most other road vehicles due to their relatively high CoG in relation to their short wheelbase. On average this ratio is about 50%, which is considerably higher than on passenger cars, formula racing cars will have a lower ratio still. Road and racing bikes will have similar values to each other due to their similar proportions whereas motoX machines are built higher and so will have a higher ratio, and thus be more inclined to wheelie.

As we shall see, the detail of the suspension geometry can affect the internal distribution of forces within the bike, extending or compressing suspension, but these internal reactions will not affect the external non-transient loads. This is just like trying to pull yourself up with your own shoe laces. There is tension in the laces and counterbalancing forces in the muscles of your body, but this does nothing to affect your weight on the ground. Motorcycle load transfer mainly has four sources:

- Inertial, this comes from the forces necessary to accelerate and brake the machine.
- Aerodynamic, in a previous chapter we saw how the drag force tends to lift the front and load the rear.
- Attitude, when going down hill more weight is supported by the front and vice-versa, actually in this case it would be reasonable to use the term “weight transfer” as well as “load transfer”.
- Torque reaction from accelerating the crankshaft and clutch etc. This only applies to across the frame engines, a forward spinning and accelerating engine transfers some load to the rear and a backwards spinning and accelerating engine relieves load off the rear.

Considering the effects of acceleration on global load transfer, we see that the only parameters that matter are the wheelbase, the height of the CoG, the actual acceleration and the mass of bike and rider. The longitudinal position of the CoG does not affect the load transfer but it does affect the actual load on each wheel. For example, when the CoG is rearward there is less load on the front wheel to start with and so it will need less load transfer before it lifts.

**Load transfer calculation.**




**Where:**

| | | |
|---|---|---|
| $L$ | = | **Wheelbase** |
| $h$ | = | **CoG height** |
| $M$ | = | **Mass of machine** |
| $a$ | = | **acceleration** |
| $\Delta Wf$ | = | **Load transfer, front** |
| $\Delta Wr$ | = | **Load transfer, rear** |

**Then:**

**Horizontal force at CoG** = $Ma$

**Moment due to this force** = $Mah$

**This moment must be resisted by an equal couple from the load transfer acting over the wheelbase L.**

**Therefore:**

$\Delta Wr.L = -\Delta Wf.L = Mah$ **thus load transfer** $\Delta Wr = -\Delta Wf = \dfrac{Mah}{L}$

**We can see that steady state load transfer is proportional to the bike's mass, CoG height and acceleration, and inversely proportional to the wheelbase.**

The following illustration, fig. 9.1, shows how we can construct a line on a side view of bike and rider to determine a limit position for the CoG height to avoid a backwards rotation for any given value of steady acceleration. To avoid looping at 1 g (32.2 ft./sec$^2$ or 9.8 m/sec$^2$) acceleration then the overall CoG must not be any higher than a line drawn at 45°, as shown. So we see that the CoG can be raised if it is also moved forward. In practice many modern bikes have laden weight distributions of approximately 50/50 with CoG heights close to half of the wheelbase and hence are very close to our 1 g limit line. No wonder that it is so easy to wheelie most sports bikes, these days. Under the right conditions modern rubber allows us a co-efficient of friction a bit higher than 1, and if we constructed our limit line for an acceleration of say 1.2 g then it would have to be drawn lower at 40°. Of course this assumes that the bike has sufficient power to produce accelerations of this level. Dynamic transient effects also have a large influence on looping etc..

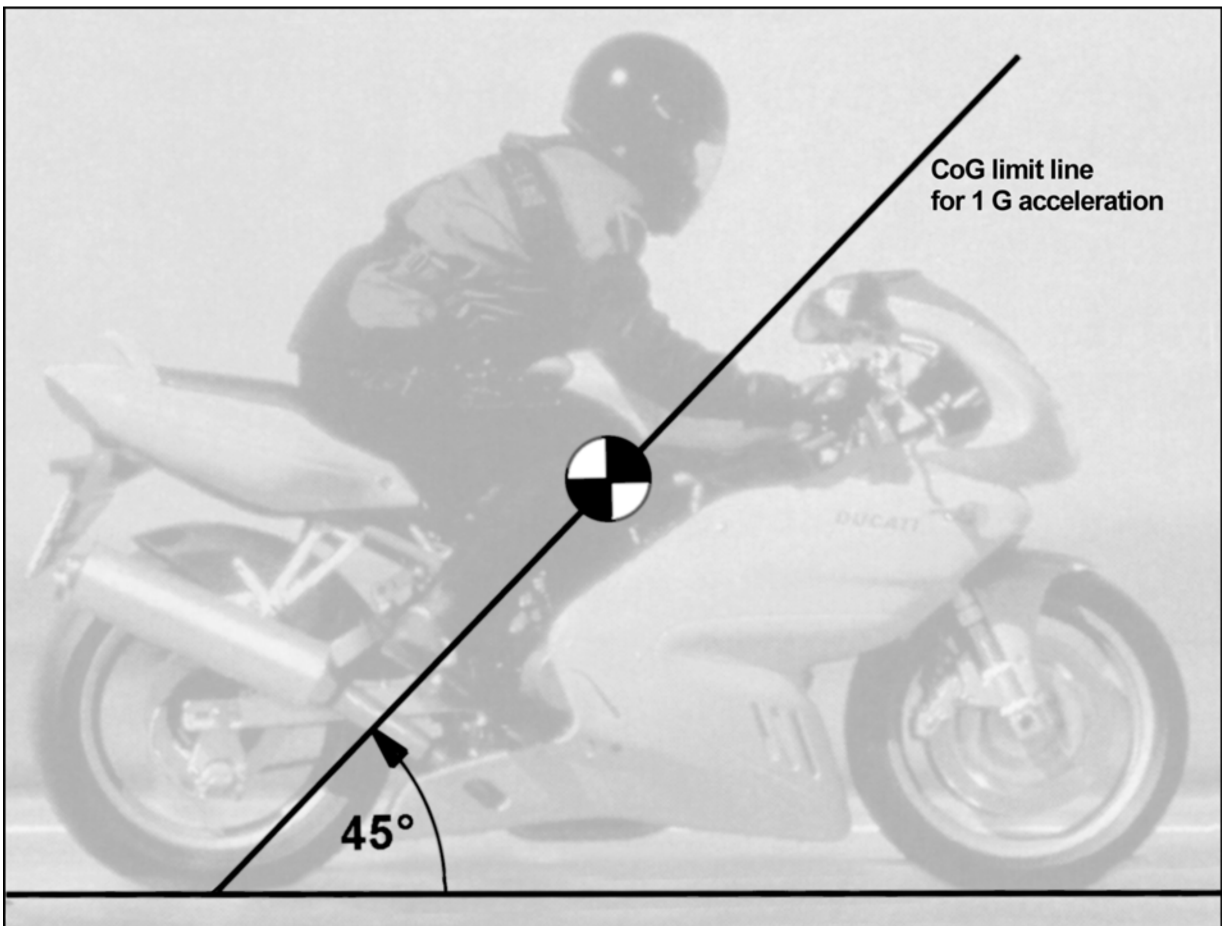



Fig. 9.1. **To avoid looping at a steady acceleration of 1 g the CoG must be in front of and below a line drawn at 45°, through the rear tyre contact patch as shown. A similar line drawn through the front tyre backwards at 45° will define the CoG height limit to avoid forward looping with 1 g braking. These lines should be drawn with the bike in the attitude adopted under these conditions.**

A similar line can also be drawn backwards from the font tyre contact patch to indicate the limit line for braking. If the CoG is above or in front of this line then the bike will lift the rear wheel when braking above the deceleration level represented by the slope of the line. As before, 45 degrees represents 1 g.

## Squat and dive

These are terms that refer to pitch and height changes of the sprung part of the motorcycle. Dive is a forward pitching motion caused usually by braking, whereas squat refers to a rearward rotation normally due to acceleration and aerodynamic forces.

Without some mechanism to the contrary the load transfer under acceleration will cause some squat, that is, the front will rise and the rear will sit down. In practice the amount of rear suspension compression is partially or totally compensated for by various reactions from the rear swing-arm, chain and other geometric features. Depending on design, braking and driving forces and their internal reactions may cause the suspension to either extend or compress.

This happens with both chain driven and shaft drive bikes, but firstly we'll look at shaft drive as this is somewhat easier to understand.

## Shaft drive

If we accelerate hard from rest on some shaft-driven machines, the rear end can be felt to rise as the driving pinion tries to climb up the crown wheel. To put it another way, the crown-wheel housing is subject to an equal and opposite torque to that of the crown wheel itself. This torque acts on the pivoted arm or fork, which therefore tries to rotate backward, so lifting the main-frame via the pivot and extending the suspension. As well as this lifting force the swing arm also pushes forward on the frame, this is the force that goes to drive and accelerate the bike. These forces are shown in Fig. 9.2.

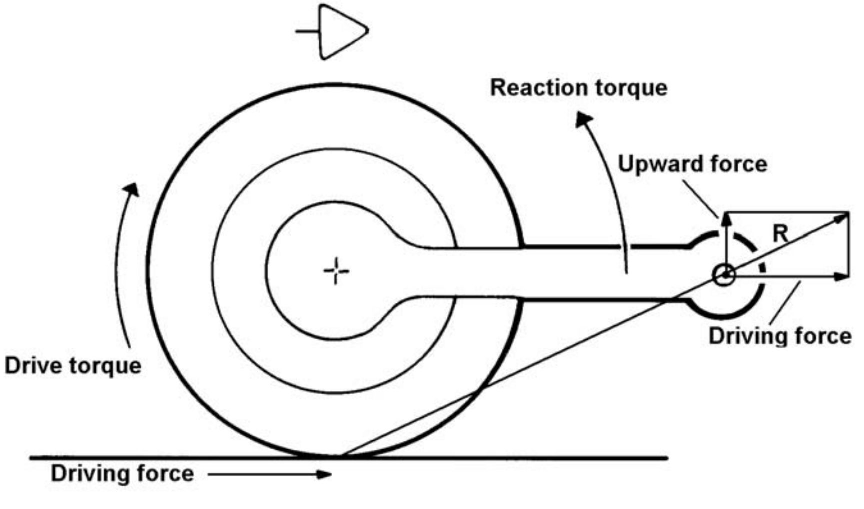



Fig. 9.2. **With conventional shaft transmission, the wheel driving torque produces an opposite torque on the crown wheel housing that tends to rotate the swing-arm backward, so lifting its pivot and rear of the machine. The driving force at the tyre ultimately acts through the pivot with both vertical and horizontal forces. These forces can be considered as a single force R acting in the direction shown. The upward force lifts the rear of the machine and hence works against the squat due to load transfer.**

The vertical and horizontal forces at the pivot can be resolved into a single force ('R' in the sketch) that acts in the same direction as a line drawn through the tyre contact patch and the swing arm pivot (This is easy enough to prove, but I'll leave that as an exercise for the interested reader), let's call this the "force line".

Now we will apply these forces to the sprung part of the machine and see what effects it has. Fig. 9.3. shows how suspension and driving forces act to support the sprung part of the machine, which for clarity is stripped of its wheels. A line of action of the resultant driving force (as per Fig. 9.2.) is drawn for each of two imaginary locations of the swing arm pivot (lines 1 & 2). Line 1. is drawn through the combined centre of gravity of bike and rider and line 2. is drawn through the intersection of a vertical line through the front axle and the CoG height. Neither of these swing-arm pivot positions necessarily reflect the actual pivot position for this bike and in fact the position for line 2. would certainly be impractical. However, these lines represent two important special instances of the general case and are well worth considering, practical or not.

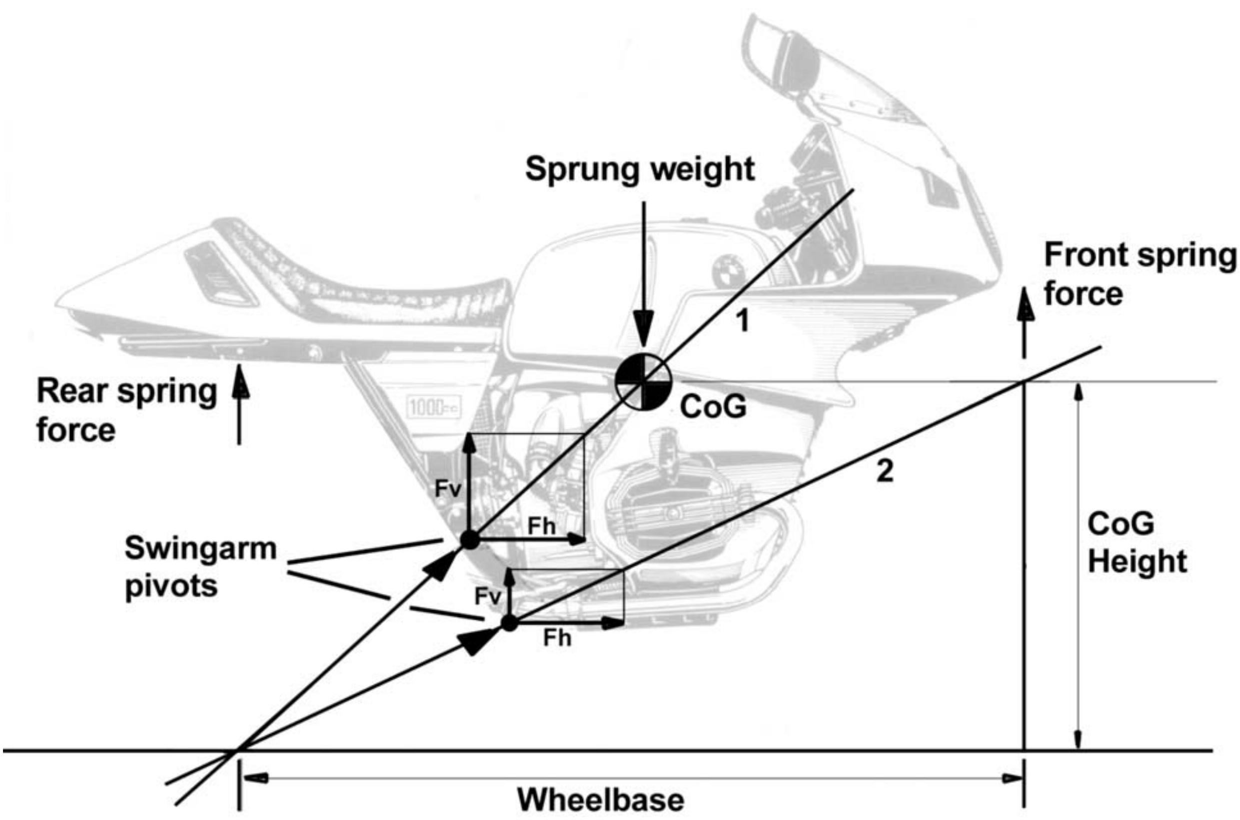



Fig. 9.3. **Balance of forces on sprung part of the bike. Line 1 shows the force line passing through the CoG, and line 2 has one half of the slope of the other. Fh is equal in both cases as this is equal to the driving force.**

To keep things simple let's imagine that the weight distribution is 50/50 and that the CoG height is half of the wheelbase, and to further simplify things let's assume that the suspension forces act vertically in line with the axle locations. *(Even if we allowed for the true fork angle and suspension unit location, and* *even if some form of rocker was used, we would still arrive at identical answers but the real issues would be hidden under unnecessary detail.)*

Consider line 1. The horizontal force at the swing-arm pivot (Fh) goes to accelerate the mass of the machine, but it also creates a moment or torque about the CoG tending to rotate the bike backwards, cause squat in other words. On the other hand, the vertical force (Fv) tends to lift the machine, relieving the suspension spring load, but also creates a torque about the CoG tending to rotate the bike forward, an anti-squat effect. In this case where the force line passes through the CoG the squat and anti-squat moments balance each other out and the remaining vertical force just tends to lift the sprung part of the machine without any pitch moment. If the forward acceleration is 1 g. then this vertical force will be more than enough to support the entire sprung weight of the bike and the suspension will be unloaded. At the front this seems reasonable as we saw earlier that this is the condition for the front wheel to lift. At the rear though, it seems harder to accept as we know that the rear wheel stays firmly planted on the ground and in fact has to carry the full weight of the bike, but how can it do this if the suspension has been unloaded? Well, just as we can consider the driving force acting along our force line providing both vertical and horizontal forces at the pivot, this must ultimately be balanced by forces between the road and the tyre. That is, the same vertical load, will be passed through the tyre contact patch, which in this case is equal to the machine weight. The total vertical tyre load is the sum of the suspension load plus the downward component of the force along our force line. As the value of acceleration increases from zero, the vertical lifting force increases, which reduces the total front and rear suspension load by exactly the same amount that it increases the rear tyre load, and everything remains in balance.

To make this a little clearer I suggest a simple experiment. Hold a pencil loosely at its top end and lower it onto some kitchen or shop weighing scales, until it is at an angle of about 45 degrees. Now with a second pencil held horizontally, push on the bottom of the first pencil. Even though you are only pushing horizontally you will note that the scales register a download force. This is just like a tyre pushing forward producing a vertical force also. Some vertical force will always exist except for the case where we have the swing-arm pivot at ground level. If we did have it in this unlikely location then there would be no anti-squat moment and the rear suspension would just compress in accord with the changes due to load transfer alone. The rear would sit down and the front would rise up.

Now let's look at force line 2. which represents a much lower and/or forward swing-arm pivot. This line has 1/2 of the slope of the previous, which means that the vertical force will also be reduced to a half. Again let's consider 1 g. acceleration, the front will just be lifting off and so the total load will be on the rear tyre. However, the tyre horizontal driving force will now only produce half of the total vertical force needed to support the weight, the other half comes from the suspension unit. So we can see that the rear suspension unit supports half of the weight of the bike, but this is exactly the same as it is called upon to support at rest. Thus in this case the rear suspension neither compresses nor extends under acceleration. The front spring gets unloaded as before and the wheel becomes airborne, although the lack of suspension lift at the rear keeps the CoG a bit lower and so we can accelerate a bit harder before the wheelie onset.

Just to keep things from getting too complicated the above explanations implicitly have assumed that the wheels and suspension weigh nothing. This is not true of course and so will affect the fine detail of anti-squat performance. The rear wheel mass will need some of the horizontal tyre force to accelerate itself and so the force passed along our force line will be reduced by an appropriate amount. In reality suspensions have some preload and so instead of the front unloading gradually and lifting off when the spring force becomes zero, it reduces down to the preload value (tops out) and then applies a lifting force to the front wheel, the weight of which then effectively gets added to that of the sprung part. The mass of the front wheel must also have a horizontal force passed to it to be accelerated too. This can only be done through the front forks. If the front wheel etc. is about 7% of the total mass then for forks set at 25 degrees rake there will be approximately another 6% of fork compression (ignoring stiction) compared to the static value, this will offset to a small extent the extension due to load transfer.

As mentioned before, lines 1 and 2 represents two special cases of anti-squat. The first creates no residual pitch moment and tends to lift the whole machine in a level attitude. It doesn't actually keep the machine dead level because it is usual to have a proportionally higher spring rate at the rear compared to the weight distribution (higher suspension frequency, see chapter on suspension), and so in practice the rear will lift a bit less than the front. To balance the lift equally we would have to mount the swing-arm pivot a little higher or more rearward than line 1 to compensate. The second case exactly balances out any suspension movement at the rear but the front will rise as before causing an angular pitch or squat attitude without lowering the rear end. We also mentioned a third special case that occurs when the swing-arm pivot is at ground level. There is no anti-squat effect and the pitch angle will be approximately twice that of the previous case with compression of the rear suspension as well as extension of the front.

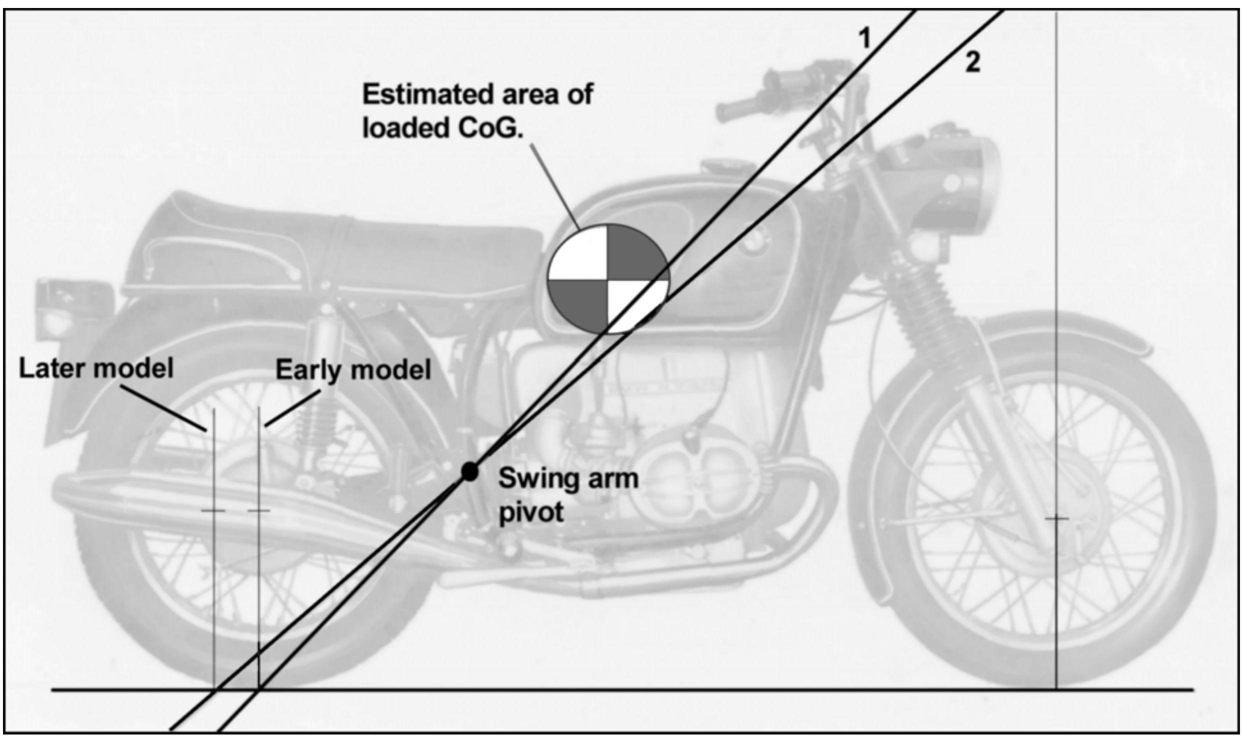



Fig. 9.4. **R75/5 BMW. The force line passes close to the estimated loaded CoG thereby causing a very high anti-squat effect. Later models featured a lengthened swing arm which reduced the anti-squat to about 84% of the former value.**

So what anti-squat characteristics should we aim for, which is best?

Starting from the front we can't do much there, we pretty well have to accept that the front is independent of the rear and will rise and may actually lift off depending on the CoG location as shown in fig. 9.1. So it can only be the behaviour of the rear that is relevant to our decision.

Anyone who's ridden early BMWs (R75/5 or earlier) will be familiar with the near level lifting of the bike as you start from rest and accelerate. Fig. 9.4 shows how the force line passes very close to the laden CoG of an R75/5 and so this machine is close to our case 1. Most people find this action to be unsettling. I've ridden various types of BMW boxer continuously since the early 60s. and can vouch for the fact that this is not their most endearing feature, especially when riding hard. There is a continual up and down motion as you slow and then accelerate again, when swinging through bends this can be very unsettling to the handling. Aside from this effect there are a couple of less noticeable but very important disadvantages of high anti-squat geometry, which become more serious with more powerful machines.

Firstly, if we apply enough power to support all or most of the weight by swing-arm torque then the rear suspension will top out becoming effectively solid. There is obviously no need to explain, in more depth, why this is less than good.

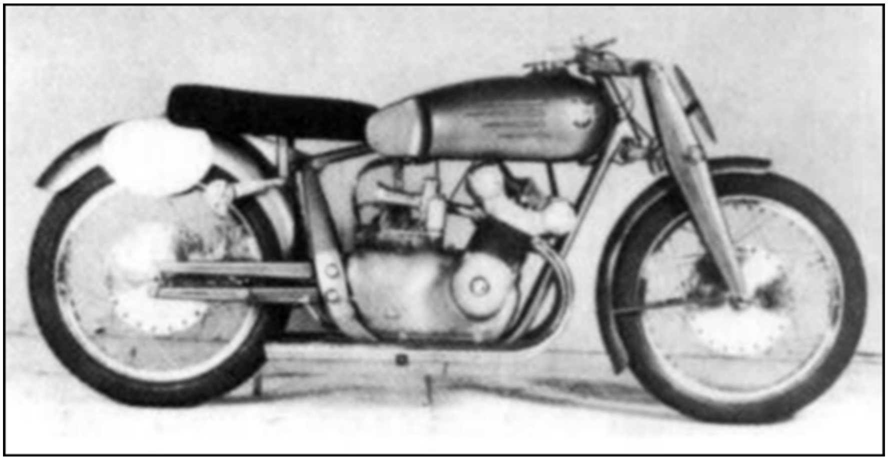
Secondly, when the throttle is snapped open and a lot of power suddenly applied, the effect will be for the rear wheel and rear of the bike to be pushed away from each other very quickly. Now because the mass of the bike is considerably greater than the mass of the wheel assembly, it will be the wheel that tries to move away from the bike more. This causes a momentary increase in the vertical tyre load, compressing the tyre at its contact with the road. Tyres have some damping but not much and so this sudden compression can be the precursor to tyre hopping or chattering, which is not a prime requirement for good handling nor drive. This problem is made worse by the topping out of the suspension unit, if the suspension unit is effectively solid then it also offers no damping to help reduce the chattering.

If we go for the physically difficult to achieve location of the swing-arm pivot at ground level, then we have no anti-squat and the rear suspension will compress according to the load transfer. At maximum acceleration this may double the static load on the spring and so use up an unacceptable amount of available suspension movement, reducing the bike's capacity to handle bumps, again with detrimental effects on handling, comfort and traction.

The case described by line 2 in fig. 9.3 seems quite desirable as the load on the rear suspension is unchanged under acceleration. This will create the minimum possible disturbance to the bike's handling as well as not tossing the rider about. The rear suspension becomes independent of driving force, under steady conditions. (See the discussion the dynamic effects near the end of this chapter)

We see from figs. 9.3 & 9.4. that avoiding an excessively high anti-squat figure is fairly difficult with a standard shaft drive machine. The swing-arm pivot position required is just not practical. Later models in that BMW range had the swing-arm lengthened by about 50 mm. which as we can see from line 2. in fig. 9.4. does decrease the excessive anti-squat slightly. Scaling from photographs indicates that the newer machine has about 84% of the anti-squat effect of the earlier models, but this is still too high.

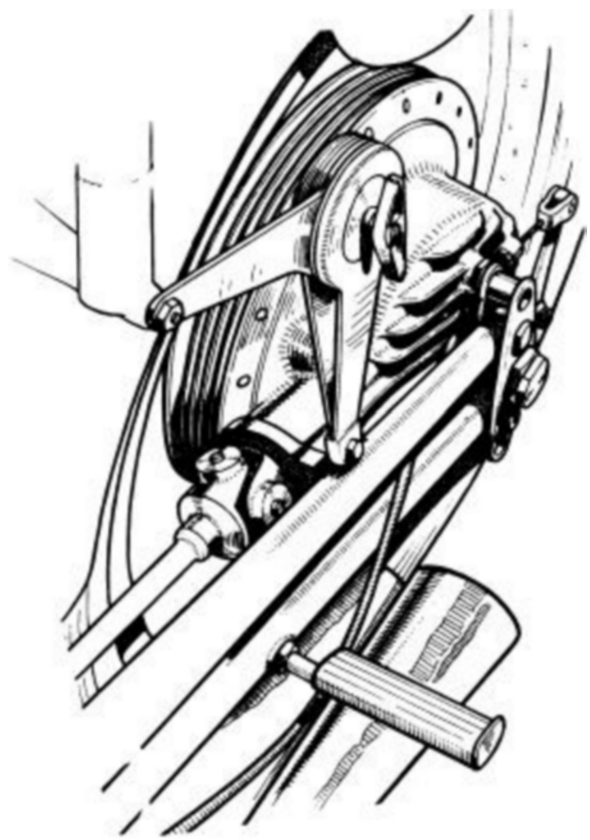
So is there anything that designers can do to improve the situation? Fortunately there is, but apart from using longer swing-arms few manufacturers seem willing to add the extra complication. Back in the 50s. though, MV Agusta used two separate swing-arms, on their shaft drive GP racers, to support the crown wheel housing and wheel assembly. The two swing-arms were arranged as a parallelogram. This is a rather heavy way to achieve the desired result and fortunately there is a lighter and cheaper way to get the same effect. Fig. 9.5 shows how.



**This 1950 MV Agusta racer had shaft drive and a rather elaborate way of partially isolating driving torque from the rear suspension, using a parallel pair of pivoted forks and a floating crown-wheel housing.**

**This drawing of the MV rear end shows the Hooke's joint at the end of the drive shaft. There was another at the front end with provision for slight variations in effective shaft length.**

**Note the friction damper connected between the top swing-arm and the frame. Another feature occasionally used for structural simplicity is the footrest mounted on the lower swing-arm.**



The crown wheel housing is mounted in such a way that it is free to rotate on the axle. In this way the swing-arm is isolated from the torque of the housing. Thus the driving torque cannot be passed to the frame by the swing-arm, which can now only pass forces along its length. To prevent the housing from spinning backward we need to attach a torque reaction link as shown. So the swing-arm pushes forward on the main part of the bike while the torque link pulls back. The forward push must be greater than the pull back or the bike will be going nowhere. As the angular motion of the housing is no longer the same as that of the swing-arm, it becomes necessary to have a universal joint at the rear end of the drive shaft as well as the front, and usually it is necessary to make provision for changes in drive shaft length.

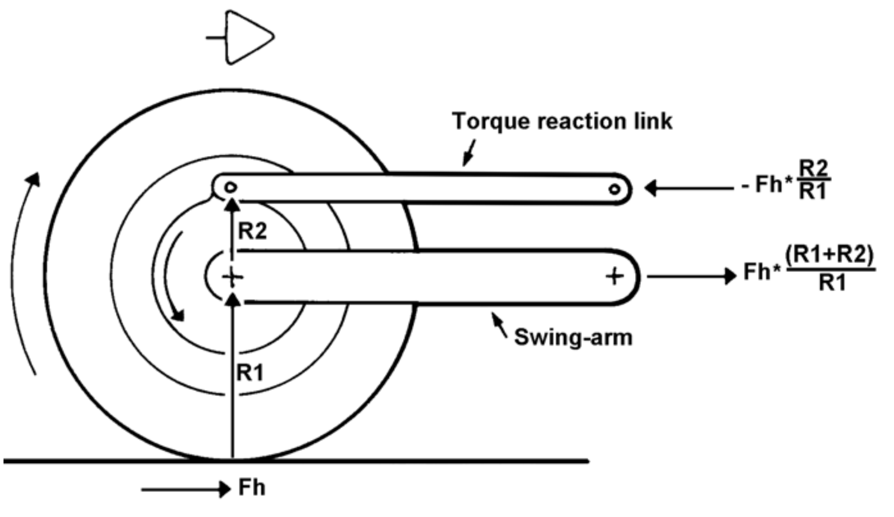



Fig. 9.5. **A floating crown wheel housing, needs an additional link to prevent it from spinning backwards under power. The swing-arm applies a forward force to the bike and the link pulls it back. As shown here with parallel and horizontal links the forward force exceeds the rearward one by an amount equal to the traction force at the tyre.**

Fig. 9.5 shows the torque link parallel to the swing-arm, but a parallelogram system like this has a severe disadvantage. The degree of anti-squat effect will change greatly over the range of suspension movement. If the arm and link point upward toward the front when lightly loaded then there will be some anti-squat, but when the arm is horizontal as shown in the sketch there will be zero anti-squat, just like the case of a normal shaft drive with the pivot at ground level. When the suspension is loaded some more such that the arm points downward the situation gets worse because then there will be a pro-squat tendency. This changing anti-squat characteristic has the same effect as a reduction in rear spring rate, and the more power that is applied the more will be this effective reduction. This would seem to be exactly the opposite to what we want at a time when more load is being transferred to the rear. As we shall see later a partial solution to this problem is to use non-parallel links converging towards the front.

Few manufacturers have adopted this system, obviously shunning the extra expense. Around the early 1980s. I built a chassis for a customer's MotoGuzzi LeMans which used the floating housing idea, but with slightly converging links. This machine was road tested by magazines and received very favourable reviews. More recently MotoGuzzi themselves have adopted a very similar system for their V11 Sport.

**To prevent driving torque from extending the rear suspension excessively on this Foale-framed shaft-drive Guzzi the crown-wheel housing was made free floating and was connected to the main frame by a pivoted link above the swing-arm. Note the second Hooke's joint at rear of drive shaft. Guzzi themselves later adopted a similar solution, along with a very similar chassis, for their production sports machines, the V11 sport.**

Fig. 9.6. shows the approach taken by BMW. The use of a single sided swing-arm would have made it difficult to float the housing about the axle as on the MotoGuzzi, and the rather clever solution to this problem was to float the housing at its forward end, pivoting off the end of a shortened swing-arm. The torque arm was mounted underneath the swing-arm, because with this layout supporting the static weight of the bike puts it in tension. Under hard driving that tension will be reduced, but increased under braking. Let's see how this design performs in comparison to the earlier models.

As in the sketch if we draw lines through the swing-arm and torque link, they meet toward the front. This junction and its location are very important. We can call it a "virtual pivot" or perhaps more correctly "instantaneous virtual pivot", because in any position of the suspension we can determine this point and the motion or arc of the housing will be identical to that which would occur with a non-floating housing and a long swing-arm physically pivoted at the virtual pivot point. (See appendix 6)

This virtual pivot can also be called the "instantaneous force centre" because we can consider this as a single point through which the two forces (in the swing-arm and torque link) act. This is a very conceptually special point as there will be no moments about it due to forces in the swing-arm and link. As the housing is constrained to move as if a long swing-arm was mounted at the virtual pivot we can analyze the anti-squat behaviour just as we did before, by drawing in a force line between the tyre contact patch and the virtual pivot. Comparing this line with line 2 in fig. 9.3, we can see that the slopes of both lines are almost identical. Therefore the newer BMWs have an anti-squat value close to that which was previously reasoned to be close to the ideal, much better than their earlier brethren.

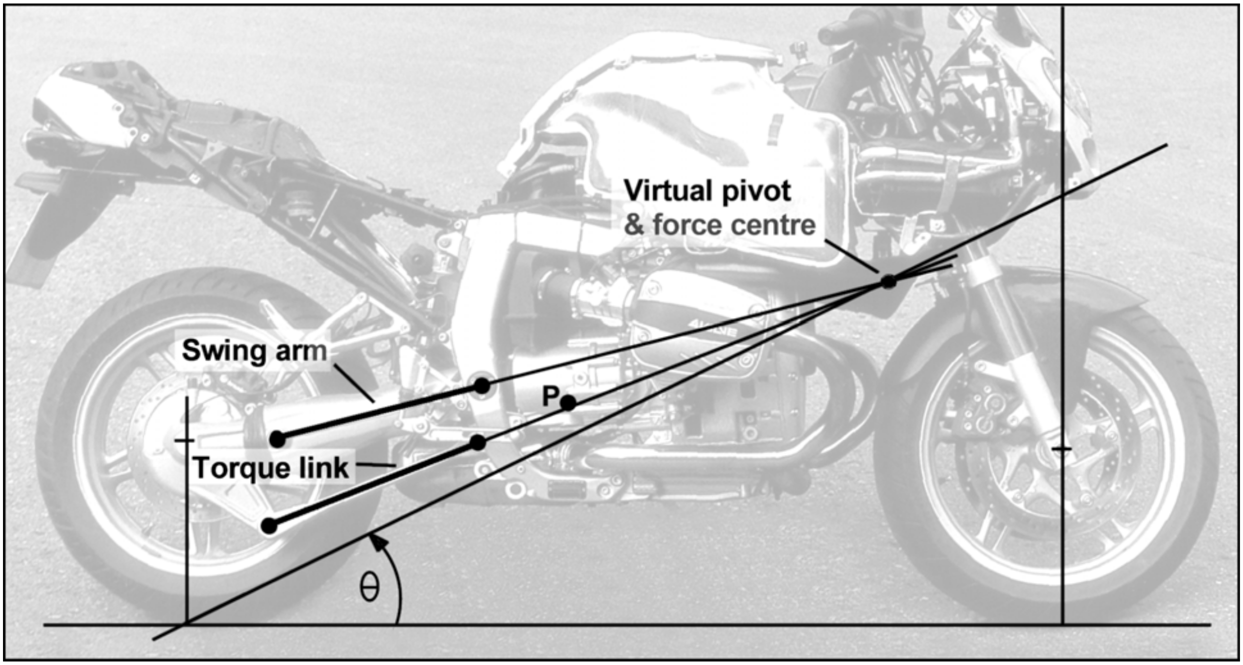



Fig. 9.6. **Current model BMWs (R1100S shown here) feature the "paralever" system. Which despite the name quite obviously does not use parallel links. Note that the torque link is under the swing-arm unlike the MotoGuzzi which places it above. Under static load the BMW link is in tension, which is reduced with driving and increased under braking. The anti-squat value would remain reasonably constant throughout the range of suspension movement if the forward link pivot was relocated close to point "P". The angle shown as Θ is often referred to as the anti-squat angle.**

We only have to look at fig. 9.6 to see that as the rear suspension compresses, the virtual pivot will move downward and give a reduced anti-squat. However, the variation will be much less than with a parallelogram system and will never get into the pro-squat situation.

It is possible to arrange the geometry of a floating housing design to give substantially constant anti-squat characteristics over the full range of suspension movement, a method for determining this geometry is shown later for chain drive systems. In this case if the frame mounted pivot for the torque link was close to point 'P' then the anti-squat tendency would remain close to constant throughout the full range of suspension movement.

## Chain drive

It might seem surprising, but we can analyze the driving squat behaviour of chain drive by using almost identical graphical methods to those that we used for shaft drive. Fig. 9.7 shows how the chain pulls backwards on the main part of the motorcycle but this is more than offset by the forward push from the swing-arm. As drawn, with both the swing-arm and chain line pointing upwards towards the front, the vertical component of the chain force is downward and acts as a pro-squat tendency, the vertical component of the swing-arm force is greater and acts upward thus giving an overall anti-squat effect.

As we'll see later when the suspension compresses past the horizontal position both the chain and swing-arm will point downwards at the front, when this happens it will be the chain pull that causes an anti-squat effect with an opposing squat tendency from the swing-arm force. We can determine whether the overall effect is one of pro-squat or anti-squat by constructing a force line as shown for shaft drives in fig. 9.6 and for chain drive in fig. 9.8.

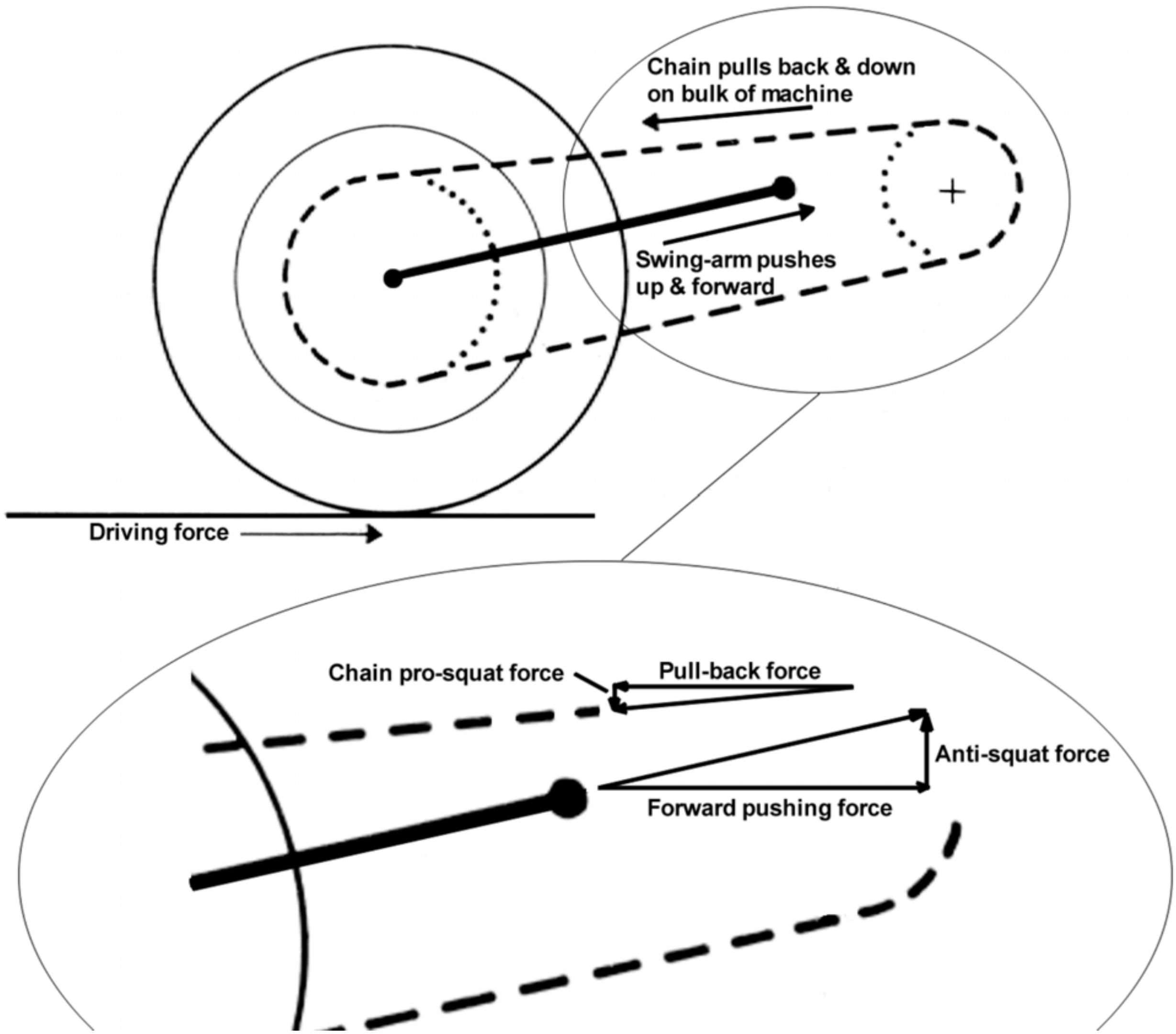



Fig. 9.7. **Compare this diagram of chain and swing-arm forces with those in figs. 9.2 & 9.5 for shaft drive. The chain pulls back and down on the main part of the bike whilst the swing arm forces work in the opposite direction. The horizontal component of the swing-arm pushing force must be greater than that of the chain pull-back by the amount equal to the overall driving force. Note how with this layout the anti-squat force from the swing-arm is considerably higher than the pro-squat force generated by the chain pull, giving an overall anti-squat effect.**

In the case of chain drive we can determine an 'Instantaneous Force Centre' by drawing lines through the chain and swing-arm. The meeting point of these lines can be considered as the point through which the combined chain and swing-arm forces act, this force must be balanced by equal and opposite forces at the tyre contact patch. (Neglecting the rear wheel mass as before.)

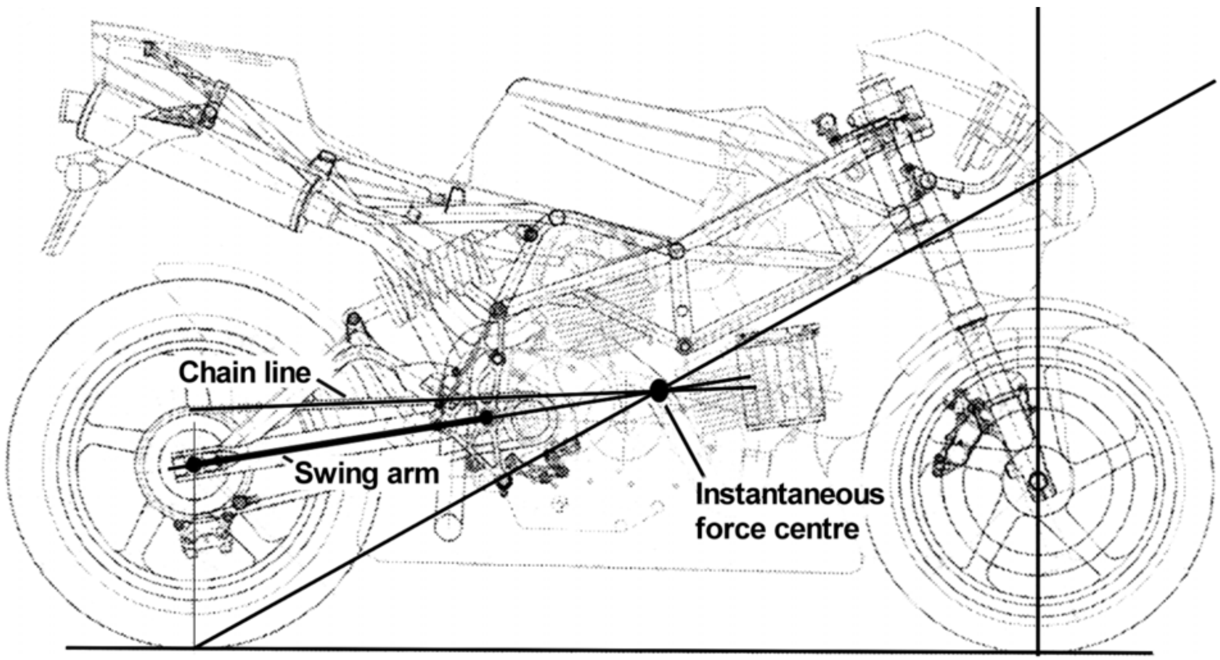



Fig. 9.8 **Using a similar geometric construction to that used to analyze the BMW paralever system in fig. 9.6 we can draw a force line through the tyre contact patch and the 'Instantaneous Force Centre' determined by the junction of lines through the swing-arm and chain line. We can't really talk about virtual arms & pivots in this case of chain drive with a single swing-arm because the motion of the wheel etc. is determined purely by the location of the 'real' swing-arm pivot.**

Comparing figs. 9.6 and 9.8 we can see that the slopes of the force lines in each case are very similar to each other and also close to the level needed to ensure a minimum of coupling between rear suspension movement and acceleration effects, as shown by line 2 in fig. 9.3. However, the situation can change drastically over the full range of suspension movement. As the rear suspension compresses for whatever reason the anti-squat effect will in general decrease and in some cases may become pro-squat, as explained above this reduces the effective rear spring rate.

Referring back to fig. 9.3, we saw that line 2 represented a degree of anti-squat which exactly balanced the squat tendencies of acceleration induced load transfer, hence effectively freeing the rear suspension from the effects of the driving forces. I would suggest that we refer to this degree of anti-squat as being 100%. We also saw that if the force line was at ground level then we had zero anti-squat and the residual suspension compression was that due to load transfer alone, and so this logically must be 0%. A force line with a slope equal to one half of the slope for 100% anti-squat will produce one half of the rear lifting force and so will be considered as having 50% anti-squat. Using this percentage scale gives a simple measure of anti-squat performance that we can use to compare different machines and suspension positions. Fig. 9.9 shows the locations of the force line for different values of anti-squat percentage. A line with a negative slope shows the case when the chain and swing-arm reactions add to the suspension compression instead of opposing it – pro-squat.

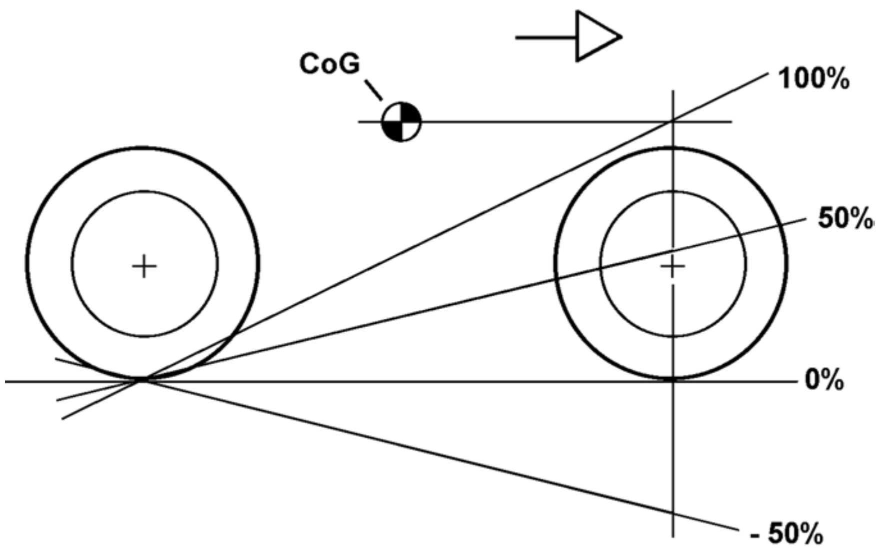



Fig. 9.9 **Four different force lines are shown with accompanying anti-squat percentages.**

**100% is when the force line passes over the front axle at the same height as the CoG. This effectively separates driving forces from the suspension response.**

**0% means no anti-squat tendency is produced and this occurs when the force line is horizontal.**

Fig. 9.10 shows how the anti-squat percentage varies with different layouts of swing-arm and sprocket positioning. Example ‘A’ shows how, when the swing-arm and chain line diverge towards the front, the instantaneous force centre can actually be behind the wheel. This has no special significance because it is the line through this force centre and the rear tyre contact patch which is important. However, when the force centre is behind the wheel then if it is above ground level we’ll have pro-squat performance and anti-squat when it drops below the ground. This is opposite to the situation when the force centre is in front of the wheel.

Cases ‘B’ and ‘C’ represent the same machine with the rear suspension at the two extremes of suspension movement. ‘B’ shows a high degree of anti-squat with extended suspension, as the suspension is compressed for whatever reason (bumps, cornering loads or higher payload), the anti-squat is usually reduced as shown by ‘C’. Therefore, the suspension appears to be softer with power applied.

Let's examine this in more detail. When accelerating on a smooth road the suspension will adopt a certain compression or extension depending on the degree of anti-squat, now if we hit a sizable bump the wheel will rise but we’ve seen that this will also reduce the anti-squat, and so the suspension will be compressed even more and squat enhanced. Thus under power, a bump will result in a greater wheel deflection than if the same bump were hit at the same speed, in the absence of any driving force. This means that the effective spring rate is reduced as the power is applied, the more power the softer will be the effective rate. As mentioned before, this is hardly likely to be a desirable characteristic at a time when more load is being transferred to the rear.

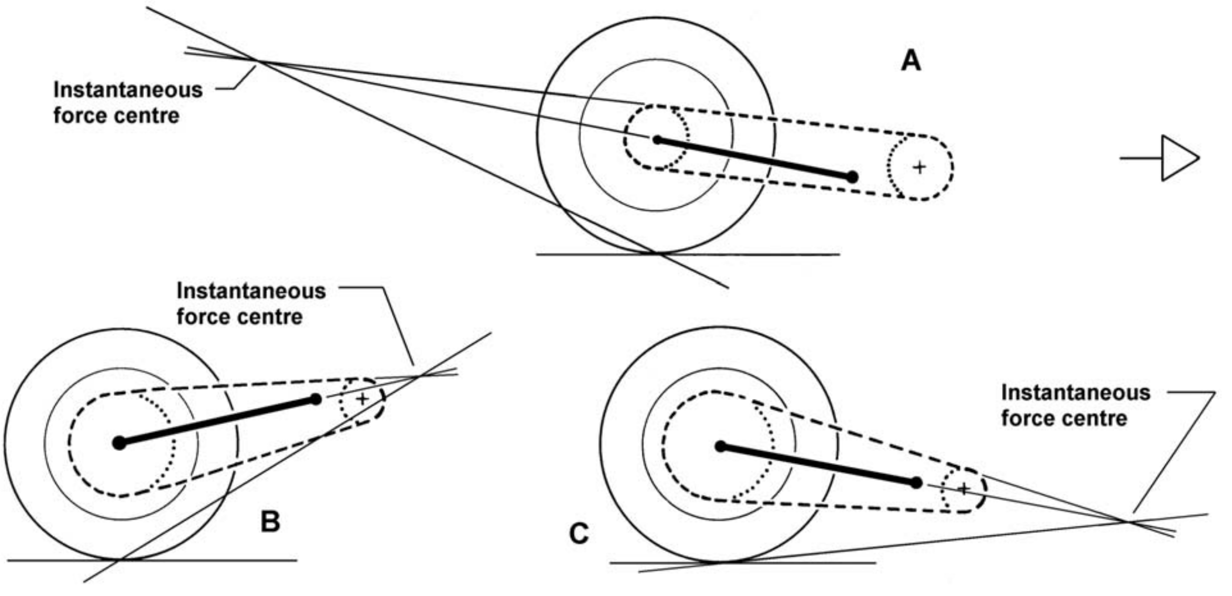



Fig. 9.10 **Sketch 'A' is drawn exaggerated to show the real possibility that the instantaneous force centre can be behind the wheel and not always in front, in this case the anti-squat effect is negative, or pro-squat. Cases 'B' and 'C' show the differences in anti-squat that can occur on the same machine throughout the range of suspension movement. 'B' shows the suspension in the extended state and the slope of the force line shows a high degree of anti-squat. 'C' on the other hand is with the suspension in the compressed state and has a greatly reduced anti-squat tendency.**

When comparing the two situations depicted by 'B' and 'C' we have to take into account the fact that in 'C' the CoG is vertically closer to the rear contact patch (the origin of the driving force) and hence the squatting moment from load transfer is also reduced in this case. Thus the anti-squat percentage is not reduced by such a large margin that a casual look at fig. 9.10 might indicate. However, the reduction is still quite considerable as shown in fig. 9.11. This diagram shows 'B' and 'C' superimposed in order to determine the anti-squat percentages. It is clear how the base for determining the 100% value is different for the two suspension positions, due to the different CoG heights relative to the rear wheel. The amount of wheel movement drawn is probably somewhat excessive for road or racing bikes and so the estimated change in anti-squat is exaggerated, but underestimates the effect on a dirt bike which would have greater suspension travel. For the motorcycle as drawn we see that with extended suspension we have an anti-squat value of 133%. Anything over 100% means that the resultant forces and moments are tending to extend the suspension farther, and so will tend to keep the suspension topped-out. Case 'C' with compressed suspension shows an anti-squat of just 30%, which means that 70% of the extra rear wheel load due to load transfer will be acting to further compress the suspension. This represents a serious enough extra suspension load when travelling in a straight line, but when cornering hard we already have the suspension loaded by up to 40 or 50% more than the static level.

This represents a range of anti-squat values of 4.4 to 1 from rebound to full bump. 'B' and 'C' are of course the extreme cases and the suspension will normally be somewhere between these limits, and so the application of power will generally have a smaller effect than indicated. Although not drawn in fig. 9.11 for clarity, the anti-squat when in the mid suspension position is equal to 100%.

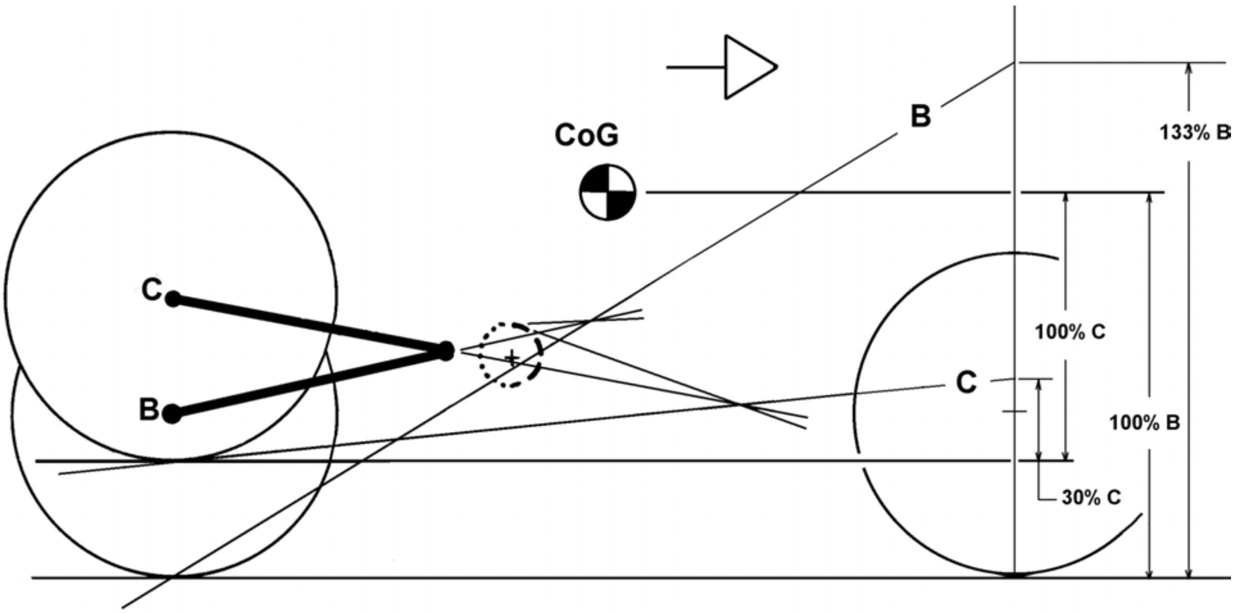
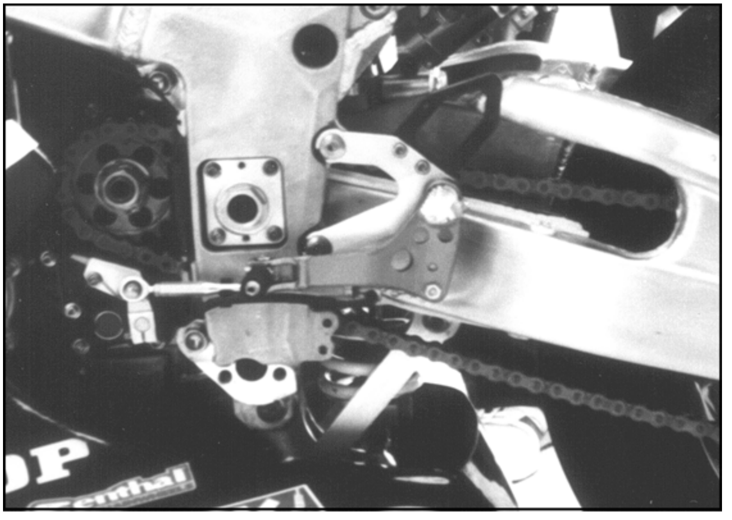



Fig. 9.11 **Cases 'B' and 'C' from fig. 9.10 are superimposed to determine the anti-squat percentage in each case. The detail of the chain lines and rear sprockets are not shown for clarity. It is clear how the base for determining the 100% value is different for the two suspension positions. Scaling from the sketches gives 30% anti-squat when the suspension is in the compressed state as against 133% when extended. Although not drawn for simplicity, the anti-squat when in mid compression is equal to 100%.**

However, we can see just how serious is the interaction between driving forces and suspension characteristics. The previous explanations have concentrated on the acceleration situation but opposite reactions occur when the throttle is closed, or changing down through the gears. That is, there will be a rise tendency due to load transfer and so we need the balance of forces to produce an anti-rise effect. These forces are smaller but will still widen the range of variation in the bike's attitude. When decelerating with the engine it is the lower chain run that is under load and so the instantaneous force centres must be determined using this run of the chain, and hence the anti-rise percentage will, in general, not be the same as the anti-squat for the same suspension position.

In the past squat and anti-squat was a largely ignored design feature and was certainly less important when power levels were lower. However, current power levels and/or large suspension movements have demanded that attention be paid to this subject and it is a very important feature of race and moto-X bike preparation and in fact some racing frames are now built with the possibility of adjusting the height of the swing-arm pivot to vary the anti-squat characteristics to better suit particular circuits and the styles of different riders.




**Note the swing-arm pivot mounting on this WSB 750 Kawasaki. The rectangular blocks can be changed for others with the pivot hole drilled in different positions to give track-side adjustment of the anti-squat properties.**

Such adjustability though doesn't really solve the problem of such a wide range of anti-squat variation with suspension movement. It just allows us to adjust the anti-squat to more desirable levels in the most important area of the suspension range. For example, keeping with fig. 9.11, if the swing-arm pivot is lowered by a little over 25 mm. then at full extension the anti-squat will be reduced to a more desirable 100% from the original 133%. A heavy price has to be paid at the other end of the suspension range though, here the anti-squat will change from 30% to about – 34%. A strong pro-squat tendency in other words. The adjustment has helped at one extreme but made the situation much worse at the other limit of suspension movement.

Sprocket size also has an effect on anti-squat behaviour as fig. 9.12 shows.

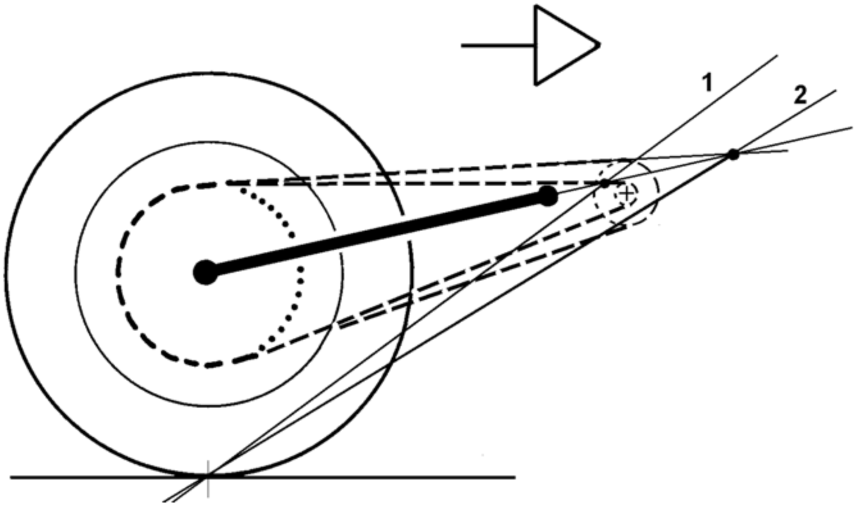



Fig. 9.12 **Effect of sprocket size on anti-squat. Line 1 is the force line with a small front sprocket and clearly gives a higher anti-squat than with a larger sprocket represented by line 2. A larger rear sprocket has a similar (but not exactly equal) effect to that of a smaller front one.**

It appears that more drastic measures are needed if we are to have proper control over squat characteristics, and it is hardly surprising that it has been the dirt bike field, with their large suspension movements and soft springing, which has seen the greatest amount of activity in this regard. Solutions tried have included; doubled up swing-arms in parallelogram and non-parallelogram form, drive sprockets concentric with the swing arm pivot and various ideas with additional idler sprockets such as the A-Trak system.

Before looking at whether any of these alternatives have merit or not, let's investigate the theoretical possibility of achieving constant anti-squat characteristics, and then see how close some of the other ideas come to accomplishing it in practice. Fig. 9.13 maintains the same basic dimensions used in fig. 9.11 but let's work backwards and start from the force lines drawn to give 100% anti-squat for the full bump and full rebound positions, we know from the earlier discussion that the instantaneous force centres must lie along these lines. These force centres must also lie along the lines drawn through the swing-arm and so their intersections with the force lines must define the actual force centres. These are points 1 and 2 in the sketch, 1 is for the bump case and 2 is for rebound.

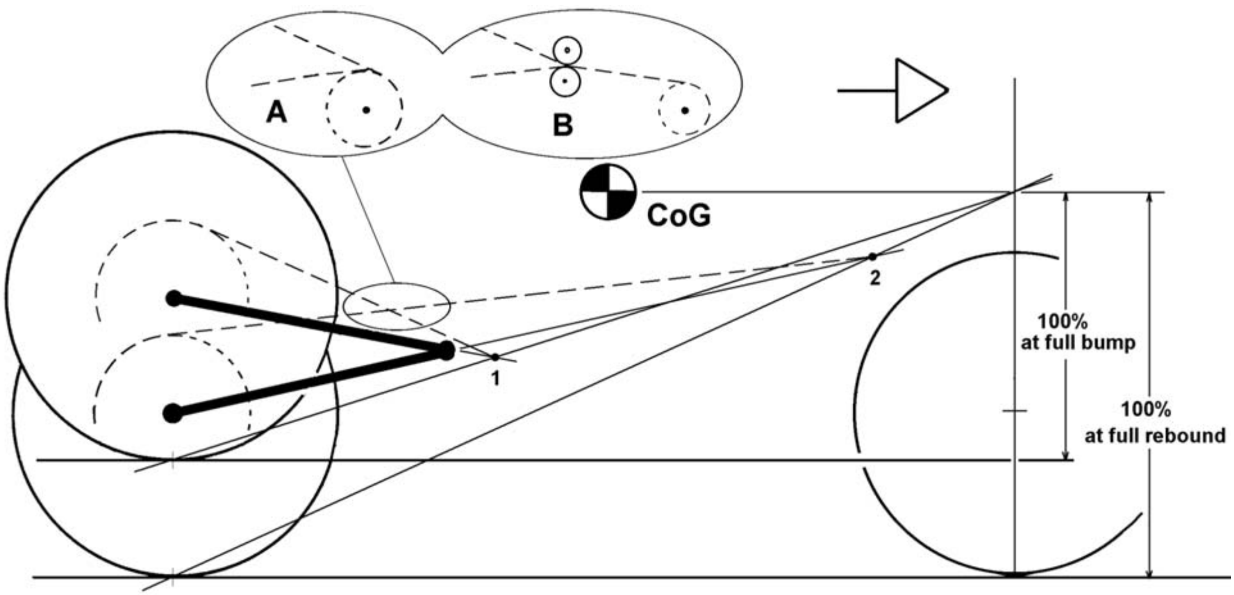



Fig. 9.13 **Showing how to determine the layout requirements for constant anti-squat behaviour, 100% in this case. Firstly draw in the force lines, for 100%, from the tyre contact patches to the intersection of the CoG height and the vertical through the front wheel spindle. Next draw in the forward extensions of the swing-arm inclination, the instantaneous force centres are defined where these lines meet, points 1 and 2. The required chain directions are then found by drawing the chain lines through the force centres tangent to the rear sprocket. The insets show two different methods by which the desired chain run may be achieved in practice.**

We have seen too, that the chain line must also pass through these force centres and so if we draw in the chain lines from the force centres tangent to the rear sprocket, then the path of the chain, needed for each suspension position, will be defined. From the drawing we see that the two chain lines actually cross each other somewhat to the rear of and above the swing-arm pivot. This point of intersection is the only common point on the chain runs to satisfy the requirement of 100% anti-squat for both extremes of suspension movement. So if we are to achieve our desired objective, we need to devise some system that forces the chain to always pass through this point, the insets ‘A’ and ‘B’ show two possibilities.

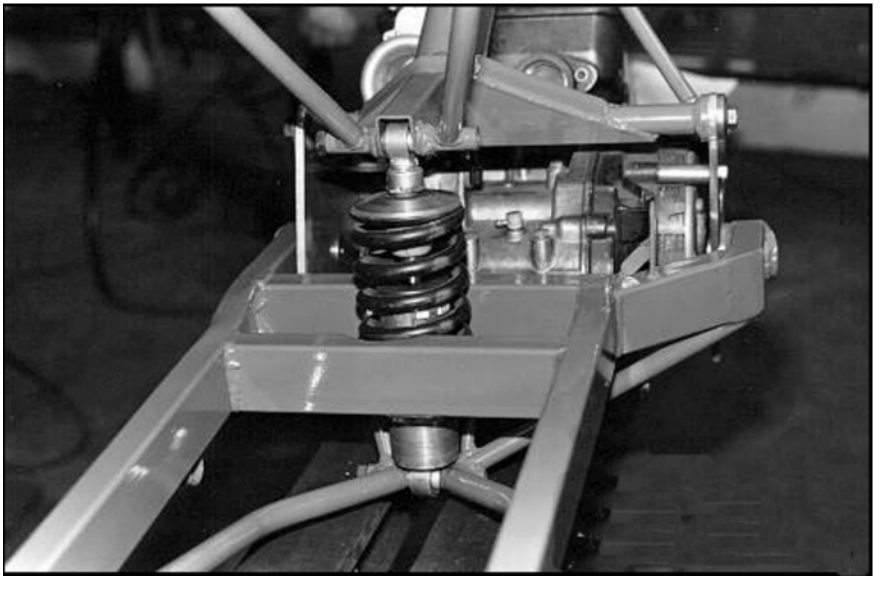
In ‘A’ the drive sprocket is mounted so that it is tangent to the two defined chain lines. In practice this might cause various problems, normally the drive sprocket is mounted on the gearbox which clearly presents packaging and weight distribution challenges in this case. These problems could be solved by the use of a jack shaft driven by a second chain from the gearbox sprocket in the normal location. Similar designs have been used in the past but to drive a sprocket mounted co-axially with the swing-arm pivot. Mechanically this would be an acceptable solution but the geometry of this layout would lead to excessive variation in chain slack with suspension movement.

What might be a better solution is suggested in ‘B’. Two idler sprockets or rollers are mounted above and below the chain run, fixed to the main chassis not the swing-arm, always confining the chain to the desired lines of action. The driving sprocket is still mounted on the gearbox in the normal fashion. Chain tension variation could still be a problem with this design but by fitting a similar roller arrangement on the bottom run of the chain we can solve that and at the same time gain separate control over the anti-rise on the overrun. Depending on the overall layout, amount of wheel movement and sprocket sizes in some cases it may only be necessary to use one of the two rollers and designs with only one roller or idler sprocket have appeared in the past.

Clearly, constant anti-squat characteristics are feasible from both a theoretical and practical standpoint. This was demonstrated only for the 100% anti-squat case but also holds true for other values.

Let’s consider some of the approaches that have been used in practice:

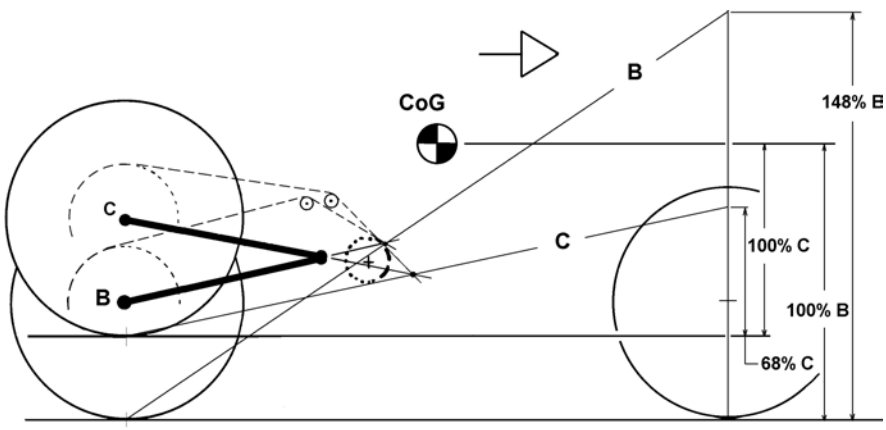
**A late 1970’s design by the author. Powered by a Yamaha TZ350 engine this racer featured a long swing-arm, widened at the front to pivot co-axially with the gearbox sprocket. Such an arrangement also eliminates chain slack variation.**

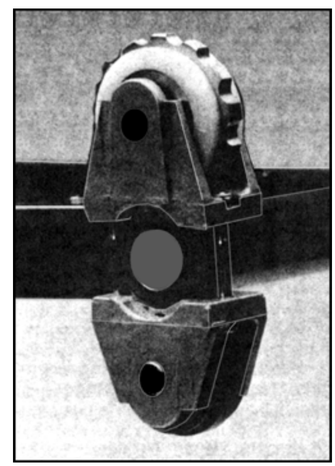


- **Mounting the drive sprocket coaxial with the swing-arm pivot.** This can be accomplished in one of two ways. By extending the swing-arm forward and putting the pivot in line with the gearbox sprocket, or by mounting a secondary pair of sprockets on the swing-arm pivot and using a second chain to drive it. The first approach is normally considered too difficult mechanically as it requires a very wide swing-arm at the pivot area. Bimota achieved it on one of their early models and the author also built a couple bikes this way. One a road bike based on a Kawasaki Z1R engine, and the other a Yamaha TZ350 engined racer. Using secondary sprockets is usually easier mechanically.

Referring back to fig. 9.11, if we just move the sprocket back so that it is concentric and mounted on the main frame then the anti-squat percentages change to 121% (133%) on rebound, 96% (100%) in mid stroke and 41% (30%) on full bump. The figures in brackets are the previous values with the sprocket positioned as in the diagram. The range of variation with the concentric sprockets is therefore 2.9 to 1 which compares very favourably with the 4.4 to 1 with the standard design.

- **A-Trak system.** This comprises of two additional sprockets or rollers mounted to the swing-arm above and below the pivot. These force the chain to approach the gearbox sprocket from a more upright angle as shown in fig. 9.14.



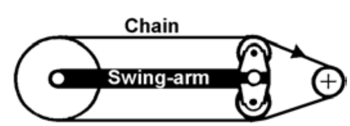



Fig. 9.14 **The A-Trak system features two idler sprockets mounted on the swing-arm, above and below the pivot. Because these sprockets are fixed to the swing-arm and not the main chassis, it is the chain line from the gearbox sprocket to the idler that is relevant for estimation of anti-squat performance. The idler sprockets are drawn unrealistically small above purely to aid clarity.**

Applying this system to the same machine as depicted in fig. 9.11, the anti-squat percentages change to 148% (133%) on rebound, 113% (100%) in mid stroke and 68% (30%) on full bump. The figures in brackets are from fig. 9.11. The range of variation with the A-Trak is therefore 2.2 to 1 which compares favourably with the 4.4 to 1 with the standard design and 2.9 with concentric sprockets. The variation is thus cut by 50%. To keep the diagram as easy as possible to read the A-Trak rollers are drawn with a very small diameter, however if they were drawn full size the variation would be slightly less. The actual values of anti-squat will vary with the details of gearbox sprocket size and location as well as wheel size, swing-arm length, wheelbase and CoG height etc. so the above figures don't represent any particular installation but just offer a comparison. On that basis it appears that the A-Trak system has the potential to reduce the variation considerably but not to eliminate it.

• **Double swing-arms.** This is the chain drive interpretation of the shaft drive MV racer pictured previously. Often the descriptions that accompany such designs indicate that even their designers don't fully understand the characteristics of such layouts. The claims often made would require the use of an alternative version of physics to have validity. However, a realistic evaluation of such designs is little more difficult than with a standard single arm. Basically from the two arms we can derive an equivalent virtual single swing-arm and virtual pivot point, using similar geometric constructions to those in fig. 9.6 for the BMW paralever system. For each suspension position though, the location of the virtual pivot may change. The anti-squat percentage can still be determined by drawing lines through the chain and the virtual swing-arm to find the instantaneous force centre, as before.

If the two swing-arms form a parallelogram, symmetrically disposed, and laid out as in fig. 9.15, which is the usual practice with this design, then all the effort is wasted. The virtual swing-arm will have its virtual pivot at infinity and be of infinite length, but the direction of the virtual arm will be identical to that of a single arm and so its intersection with the chain line will also be identical. Thus the anti-squat performance will not be changed from that of the single arm.

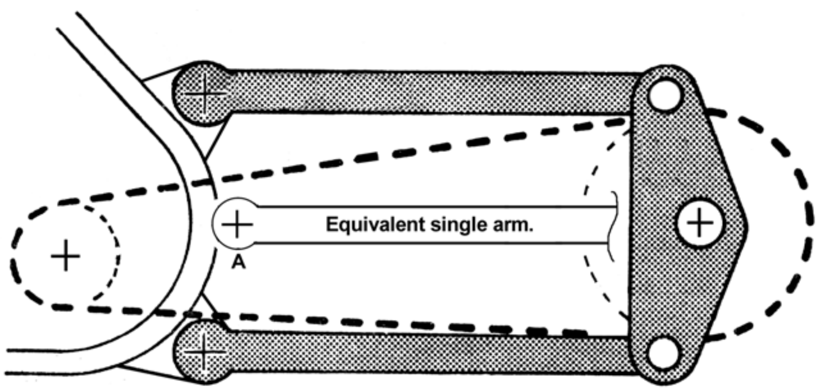



Fig. 9.15 **Double parallelogram swing-arms. The anti-squat and movement characteristics are identical to those obtained with a single conventional arm as shown, pivoted at point 'A'. Press descriptions of such dual-arm designs are very often highly flawed.**

If the two swing-arms are arranged in a non-equal length, non-parallel layout we have some degree of independent control over the characteristics. Rather than analysing various possible geometries, let's follow the example of fig. 9.13 and see if we can synthesize a layout that can provide constant 100% anti-squat throughout the full suspension range.

In this case we assume pre-defined chain runs and look for the existence of a common virtual swing-arm pivot position. Fig. 9.16 shows how. As in the method of fig. 9.13 draw in the force lines defining 100% anti-squat at the extremes of suspension movement, extend the respective chain lines until they meet the force lines. These intersections, 1 and 2, define the two instantaneous force centres. We know that the lines through the swing-arm must pass through these force centres and so we can draw in the swing-arm lines from the force centres back to the wheel axle. Where these two meet, at point P, is the only common point on each swing-arm line and so is the only pivot point to satisfy our initial requirement of 100% anti-squat at the extremes of suspension travel.

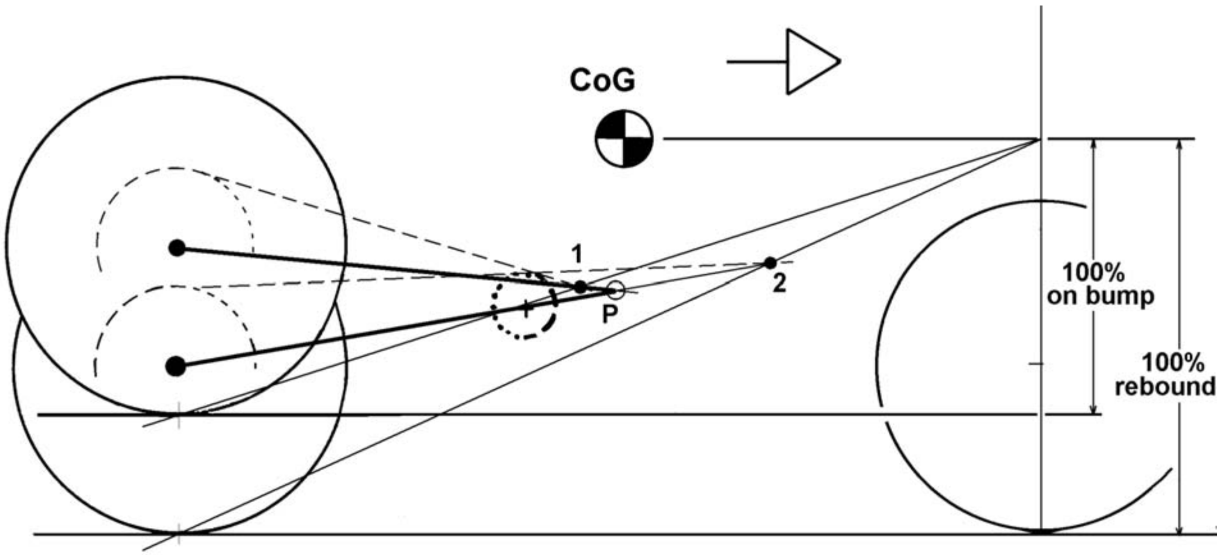



*Fig. 9.16* **Determining the swing-arm pivot position to achieve 100% anti-squat at both extremes of the suspension range. The intersection of the respective chains lines and the force lines for 100% define the instantaneous force centres (1 & 2). The swing-arm lines must pass through these points and where the two swing-arms meet, P, is the only common point and hence is the required pivot position.**

We see that this pivot point is in front of the gearbox sprocket which is likely to be impractical to implement as a conventional single arm. So instead of a real single arm we need to create a virtual swing-arm, with the same properties, but from a double arm design. It turns out that there are two possible solutions. One puts the forward pivot points of both arms coaxial with the required virtual pivot point. This is really just a glorified form of a single arm and is most unlikely to be any more practical than using a normal single arm. The other solution is actually a parallelogram with the two forward pivots above and below the required virtual pivot position. From fig. 9.15 we saw that a parallelogram acts just like a single arm, and so the trick is simply to design the layout to simulate a single arm pivoted about point P in fig. 9.16. It is unlikely that this will be anymore practically feasible than a using a simple single arm. Even without the extra weight and mechanical complication of the two arms and all that goes with it, it seems that there are no worthwhile advantages to the use of the double arm to offset the disadvantages, even though it is theoretically possible to achieve the desired characteristics.

- **Tracklever system.** This recently patented design (designer: Josef Lluis Belil) is being promoted as a method to prevent high side accidents. Now, whilst suspension response, to the varying loads involved when the rear tyre exceeds its maximum grip, can have important influences over the build up to a high-side, it is probably over ambitious to expect a complete solution to come from any form of anti-squat control. Fig. 9.17 shows the layout, the declared intention is to align the swing-arm and chain with the force line for 100% anti-squat in order to remove any residual squat or anti-squat moments. As we have seen, this is usually not possible over the full suspension range and this design does nothing to change that. If the swing-arm is aligned for 100% anti-squat at normal ride height (referred to as the neutral position) then the percentage will reduce on bump and increase on rebound, just like a conventional system. Using the same amount of wheel movement as in the previous examples, equally disposed about the neutral position, the anti-squat percentages change to 131% (133%) on rebound, 100% (100%) in mid stroke and 60% (30%) on full bump. The figures in brackets are from the conventional system shown in fig. 9.11. The range of variation with the Tracklever is therefore 2.2 to 1 which compares favourably with the 4.4 to 1 of the standard design.

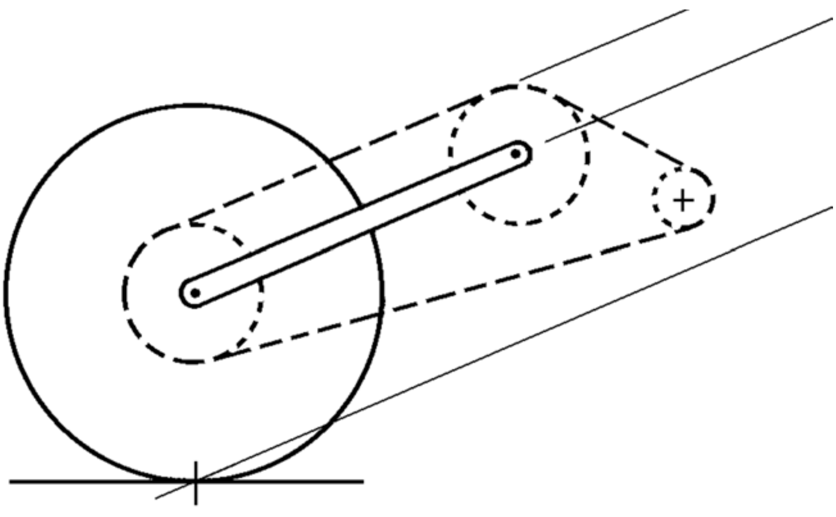


Fig. 9.17 **Tracklever system. An idler sprocket of the same size as the rear one is mounted to the frame, coaxial with the swing-arm pivot, to keep the chain line parallel to the swing-arm. In the normal ride position the swing-arm and chain are aligned with the force line appropriate for 100% anti-squat. The instantaneous force centres are always at infinity but the anti-squat percentage will still change with suspension movement.**

From these figures it can be seen that (in common with the A-Trak) most of the improvement occurs between normal ride height and full bump, this is likely to be the most important part of the suspension range, partly because the suspension becomes more compressed under hard cornering.

This design as illustrated appears to ignore the effects that occur on the overrun and in fact would have quite high pro-rise characteristics throughout the movement range. This is very likely to be detrimental to high-siding tendencies, because it is the rider's natural reaction to shut the throttle in times of trouble that often makes matters worse. This 'defect' would be quite easy to cure though. If the lower run of the chain was held against the bottom of the idler sprocket by a smaller idler mounted behind the gearbox sprocket, then the anti-rise properties would become similar to those of the anti-squat.

Unlike the A-Trak this design is not suitable for retro-fitting, it really needs to be designed into the bike from the beginning.

## Aerodynamic squat

So far we've looked at the squat characteristics as if all the driving force went into producing acceleration. As the speed rises more of the effort goes into overcoming aerodynamic drag. In fact at top speed virtually all of the traction force goes into parting the air. In this case the CoG position has no relevance and it is the height of the frontal aerodynamic centre of pressure that we must use to evaluate the anti-squat. Otherwise we can analyze the behaviour by exactly the same methods. In the chapter on aerodynamics we saw how the drag force causes a load transfer from the front wheel to the rear in like manner to the acceleration case. Fig. 9.18 shows how we can determine the direction of the force line to isolate the rear suspension from aerodynamic drag influences.

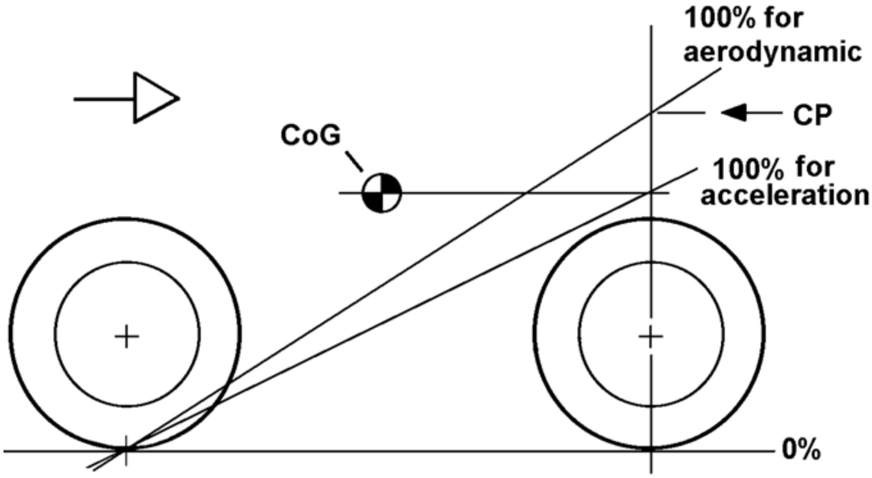



Fig.9.18 **The lower line gives 100% anti-squat under acceleration only, as in fig. 9.9. The upper line is similar but for the top speed case when all tyre force goes into overcoming drag. 'CP' is the frontal centre of pressure. If the CP and CoG have similar heights then there is no divergence of requirements to remove driving effects from suspension action.**

If the CP and CoG heights are the same then the amount of anti-squat needed to remove driving effects from the suspension will also be the same. This relationship will depend on the type of bike and positioning of the rider. A racer crouched over the tank behind a fairing may well have a CP height close to that of the CoG, but a normally seated rider on a big tourer will in all probability have a higher CP.

There is another effect that varies with speed and rider posture etc, and that is aerodynamic lift. Apart from the load transfer due to drag, the airflow may give rise to a nett vertical force, usually upward, which will also have an effect on suspension extension. Little wonder that most people only tend to consider acceleration in relation to squat or attitude characteristics.

## Braking reaction (rear)

Braking, whether from the front or rear, transfers load from the rear wheel to the front, but if the rear brake shoe plate or caliper is attached directly to the swing-arm in the normal way, then the rear braking torque is applied to the main frame through this component, trying to lower the swing-arm pivot, thereby tending to compress the suspension. Exactly the opposite effect to the shaft drive tendency of extending the suspension when driving. Under braking, some rear suspension compression (let's call it anti-rise) may be useful to counteract the change in the machine's attitude caused by forward weight transfer. But a serious problem can occur when the anti-rise is excessive, and that is rear-wheel hop or judder under heavy braking.

The sequence of events bringing this about is as follows: Sudden application of the brake applies a sharp torque to the swing-arm, tending to compress the suspension, and since the unsprung mass (the wheel etc.) is considerably less than that of the sprung mass, the wheel tends to leave the ground more quickly than the bulk of the machine can move downward, mainly due to gravity. As the wheel leaves the ground and the tyre therefore loses traction, the brake locks and the compressing moment vanishes. When the wheel returns to the ground, still locked, it may skid, and the shock load on the tyre may cause sufficient sudden braking torque to set the whole process in motion again – and again. This effect is likely to be more serious than the chattering previously described for the acceleration case. There are two main reasons for this, firstly the wheel is being momentarily lifted away from the ground rather than being pressed onto it. Secondly the largest portion of the braking effort comes from the front and so under heavy braking there is little load on the rear wheel to start with.

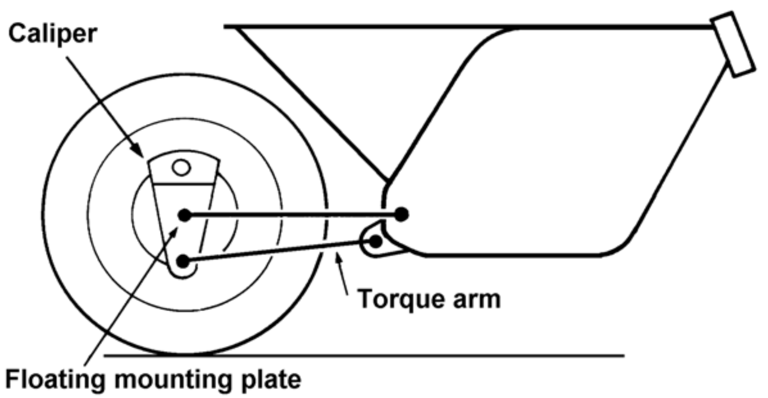



Fig 9.19 **To prevent brake torque from compressing the rear suspension excessively, the caliper plate can be made free to rotate on the wheel spindle and linked to the main frame by a pivoted torque arm.**

Riding style can determine whether this becomes a problem in practice, a rider who applies the brake in a gentle fashion may never experience trouble whereas someone with a less delicate touch may find it totally uncontrollable. (This is one of the reasons that different riders often report widely differing opinions on the handling characteristics of the same machine. Handling is affected by riding style) .

We can reduce this anti-rise effect in the same way that we reduced the driving-torque effect – that is, by arranging for the caliper plate to float on the axle and taking the torque reaction forward to the main frame via a pivoted linkage of approximately parallelogram shape, as shown in fig.9.19, but this is rare and it is more usual for manufacturers to tune the spring and damper rates to suppress it. Unfortunately, this introduces another undesirable compromise into the selection of suspension characteristics, which are best determined by considerations of roadholding and comfort.

Currently the only production road bike known that uses this linkage system to reduce anti-rise is the paralever BMW. This is a consequence of mounting the rear caliper on the crown wheel housing made floating to control anti-squat under power. The anti-rise is not separately adjustable from the anti-squat but is never-the-less a worthwhile improvement over the accepted norm. On the special Guzzi chassis built by the author (mentioned above), the disc was on the opposite side of the wheel to the drive housing and so it was possible to float both the caliper and drive housing and have independent control of the anti-rise and anti-squat. Riders with experience of a floating brake anchorage generally testify to its worth despite the slight complication.

Because rear-wheel hop is most likely on very rough surfaces, it is on motocross and enduro bikes that floating brake anchorages have been most often found – Husqvarna employed it for many years with others following suit later. In road racing, adoption of the scheme has been slower, but as with many other sound features, a precedent was set long ago by Ing. Giulio Carcano on the grand-prix Moto Guzzis (fig. 9.28). More recently it has been used on some single sided swing-arm works Hondas. However, the optimum setup for racing changes over time. Modern tyres ensure that there is little or no load on the rear under heavy braking, hence denying the rider the opportunity of using the rear brake. If the brake is mainly unused, it matters not, how we mount the caliper. Some riders just don't use the rear brake anyway.

Although we can evaluate the anti-rise characteristic in a similar way to the anti-squat, by constructing a force line through either the real or virtual pivot position, actually deciding on the most desirable degree of anti-rise is much harder than for anti-squat, because of the infinity of ways that front and rear braking can be combined. Fig.9.20 shows how the anti-rise depends on the swing arm length for the conventional system without a floating caliper mount (in this case we can use the actual swing-arm, there is no need for a virtual one) , a short arm may give more than 100% anti-rise, i.e. the geometry may cause the anti-rise to exceed the rise due to load transfer and the rear may actually sink down under the action of the rear brake alone. Any anti-rise such as this, relies on force and torque reactions which come from rear braking, therefore these effects will be completely absent when using only the front brake.

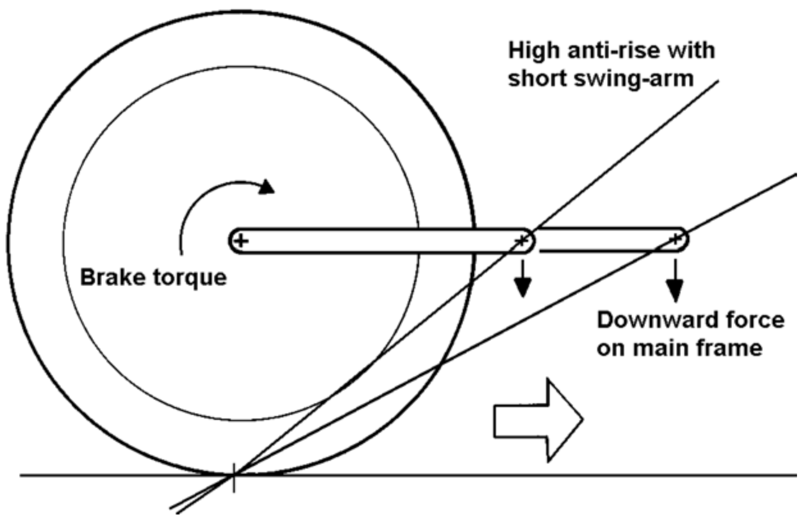



Fig. 9.20 **To resist braking torque the force at the swing-arm pivot is less with the longer arm, this means a lower anti-rise effect which is reflected in the lower slope of the force line. Compare this with the anti-squat case shown in fig. 9.4.**

The layout in Fig. 9.21. shows a floating brake mounting with two alternative locations for the torque stay pivot. With the torque stay mounted high up and well back, the virtual pivot for this arrangement is at the intersection of the swing-arm and the torque arm, we can see that this gives a very high anti-rise percentage.

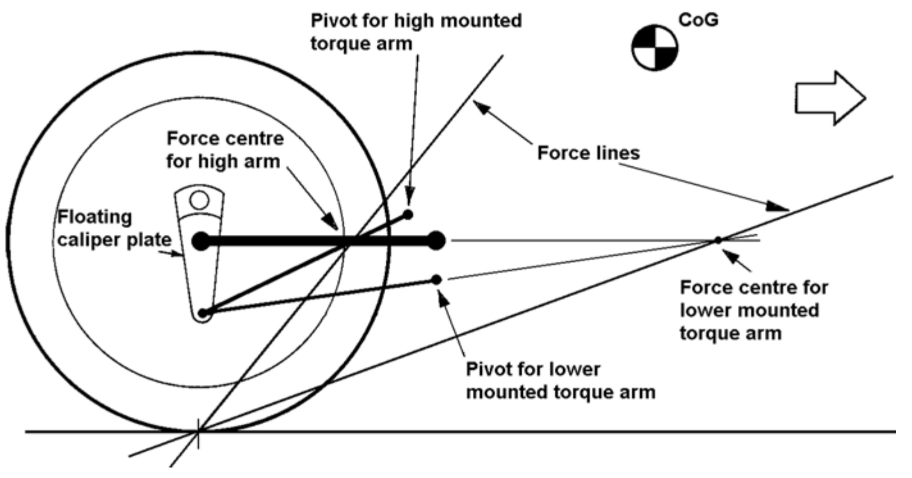



Fig. 9.21 **Anti-rise effects from two different positions of the torque stay pivot. The lower position is more desirable and would give around 100% anti-rise. The other high mounted arm would give an excessively high degree of anti-rise, possibly causing the suspension to bottom out. The lower arm has a virtual swing-arm longer than the 'real' one, whereas in the other case the virtual swing-arm is very short.**

The lower position of the pivot is situated to give about 100% anti-rise, thereby cancelling out any rear suspension reaction to rear wheel braking. As with the BMW paralever and chain drive constructions for anti-squat determination, we draw in a force line through the intersection of the torque arm and swing-arm (virtual pivot). Sometimes the high mounted torque stay is suggested as a way of keeping more pressure on the tyre during braking. This location will give a higher anti-rise effect than the conventional system with the caliper mounted to the swing-arm directly, and the possible disadvantages of that (mentioned above) will be worsened. The idea that the braking reaction will pull the bike down on to the road, was explained, in the first paragraph of this chapter, to be impossible under steady state conditions. The initial transient will in fact work toward reducing the dynamic tyre load.

If we adopt the floating caliper design then it would seem sensible to design for a reasonably constant anti-rise percentage throughout the range of suspension movement. This is simply a matter of determining a torque arm pivot position that satisfies the required anti-rise at each extreme of suspension position. Similar to the methods shown for constant anti-squat in figs. 9.13 and 9.16.

# Dive (front)

## Telescopic forks

There are two sources of dive associated with these forks - one is the obvious effect of load transfer, which is dependent on the CoG. height and the wheelbase, the other is a less obvious effect due to the rearward rake of the fork legs. This rake means that the braking force on the front tyre can be split into two components when fed into the forks, one in line with the sliders which tends to compress the springs (this force is approximately 42% of the braking force. at 25 deg. rake), the second component at right angles to the forks which tries to bend the fork legs (roughly 91% of the braking force). On a typical sport-bike the increased force in the fork springs due to load transfer is around 45% of the braking force. The calculations for a bike with 50:50 weight distribution and a CoG height of one half of the wheelbase are shown in fig. 9.22 from which we see that the dive tendency is nearly doubled by the additional effect of the angled sliders, over that due only to load transfer. The actual suspension load under maximum braking is therefore nearly tripled.

Fig. 9.22 **Under braking there are two separate sets of extra forces that work to compress the forks and cause dive. The first is due to load transfer (Ft), we can calculate a component (Fs1) inline with the suspension:**

**Fs1 = Ft.cosΘ**

**The second is due to the braking force (Fb), in this case the suspension component (Fs2) is:**

**Fs2 = Fb.sinΘ**

**The total extra suspension force(Fs) is therefore:**

**Fs = Fs1 + Fs2 = Ft.cosΘ + Fb.sinΘ**

**For a sport bike Ft will typically be about half of Fb and so if the rake angle Θ is 25 deg. this extra force becomes:**

**Fs = Fb(0.5x0.906 + 0.423) = 0.876.Fb**

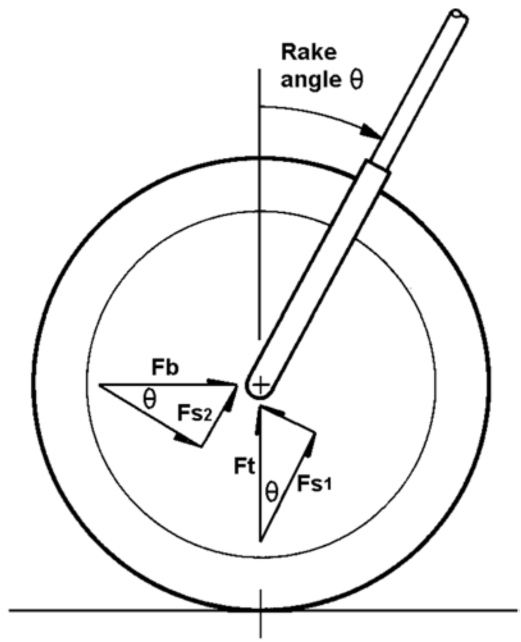
**The static suspension load = 0.5.Wt.cosΘ = 0.453.Wt**

**For 1g braking Fb = Wt**

**Then the total suspension load = 0.876 + 0.453 = 1.329**

**So the ratio of braking to static suspension load is:**

**1.329/0.453 = 2.93**




Little wonder then that telescopic forks dive so much. In the absence of any anti-dive system, there are two ways to accommodate this effect, use stiff fork springs to limit the movement or softer springs with larger movement. But these parameters should be selected on roadholding, handling or comfort grounds.

The disadvantages of hard suspension are self evident, and the large fork movement associated with softer springs, allows undue pitch changes and variations in steering geometry, to the detriment of comfort and stability.

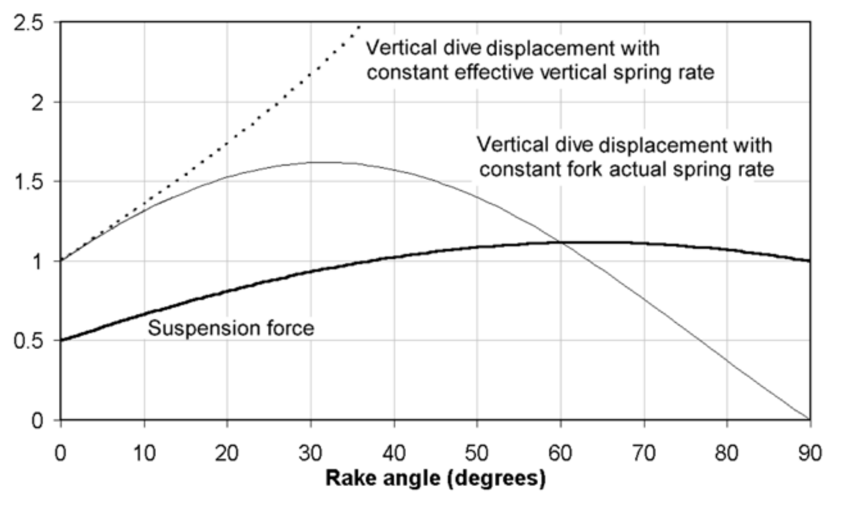
It would be interesting to see if we could find a value of rake angle that gave the best or worst front dive characteristics. Well, using the relationship derived in fig. 9.22 we can plot the additional suspension force, due to braking, against different rake or castor angles. Fig. 9.23 shows the suspension force in relation to the braking force for a machine with the CoG height at 50% of the wheelbase. The maximum is 1.12 at a rake of 63 degrees. That is, the increased suspension force could actually exceed the front wheel braking force by 12%, although at a most unlikely rake angle.

However, maximum dive does not occur at the rake angle for maximum suspension compression. Imagine the extreme case of 90 deg. castor. The horizontal braking force will act directly in line with the forks and cause some compression, but there will be no dive, only the wheelbase will change. The extra vertical force from load transfer will have no effect on the suspension, except for applying a bending load to the fork legs.

Dive motion is related to the fork compression by the cosine of the rake angle, taking this into account we can calculate that, if the fork spring rate is kept constant, the maximum dive will occur at about 32 deg.. However, to maintain a constant effective vertical spring rate, the actual fork spring rate must be reduced as the rake angle increases (by the inverse of the cosine of the rake angle squared). The graph shows that with constant effective vertical rate the amount of dive just increases as the rake angle gets larger. This is most valid case for comparison and shows that the dive problem just gets worse at greater rake angles.

These graphs are normalized such that the value of the displacement at a castor angle of zero is made equal to 1.0. They are drawn assuming single rate fork springs and ignoring the reduction in rake angle with braking, because this will depend on the actual spring rates used.

Fig. 9.23 **Showing the relationship between the additional suspension force due to braking and various castor angles. The other two curves show how the actual dive motion is dependent on the castor angle, depending on how we consider the spring rate. The case of constant effective vertical rate is the most valid for comparison, and we can see that the dive just continues to increase with rake angle.**




Surprising as it may seem, we can also evaluate the dive characteristics of telescopic forks by using the concepts of virtual swing-arms, pivots and percentage anti-dive (as shown in fig. 9.24) in a similar manner to that applied to the squat and rise problem of the rear suspension . The axle follows a straight path in line with the forks, as shown, if the fork legs are replaced by an infinitely long virtual swing-arm mounted at the wheel spindle and at right angles to the sliders, then the wheel motion will be unchanged, and the line through the contact patch and the virtual pivot will again define the anti-dive percentage. This line can be compared to the line required for 100% anti-dive, defined in similar fashion to the anti-squat percentage shown previously, except that this time it is the CoG height over the rear axle that is the base for determining what constitutes 100%.

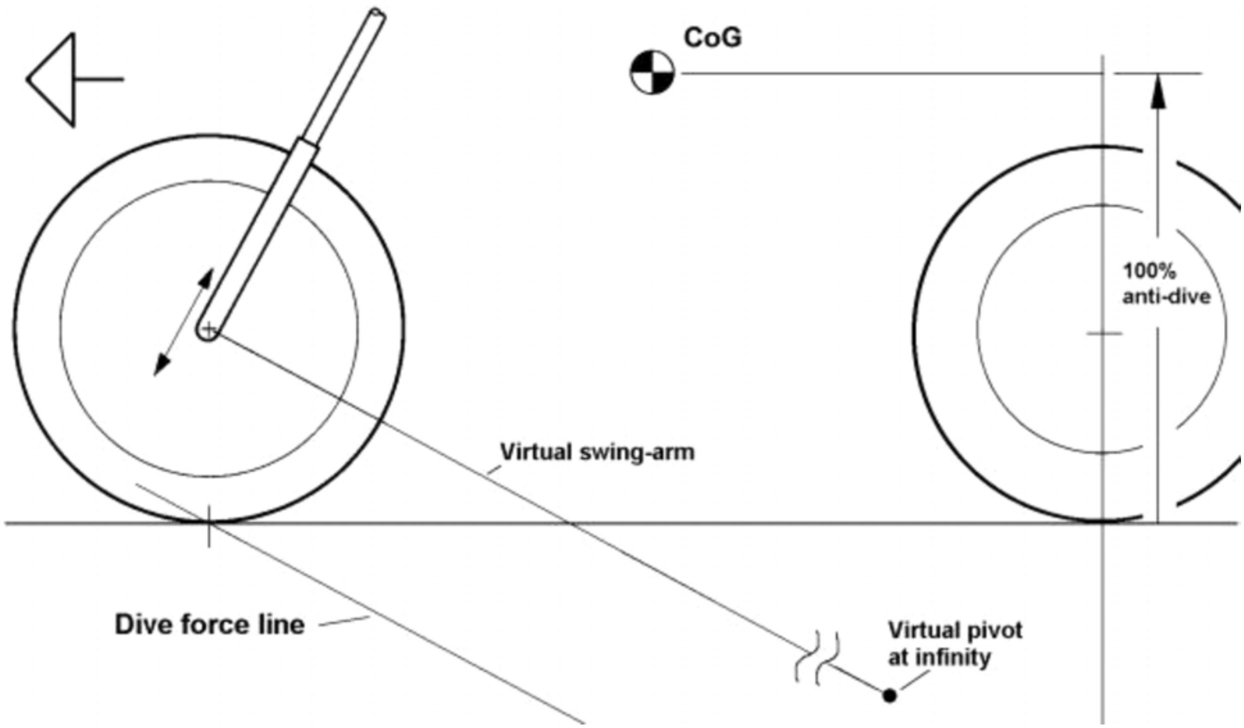
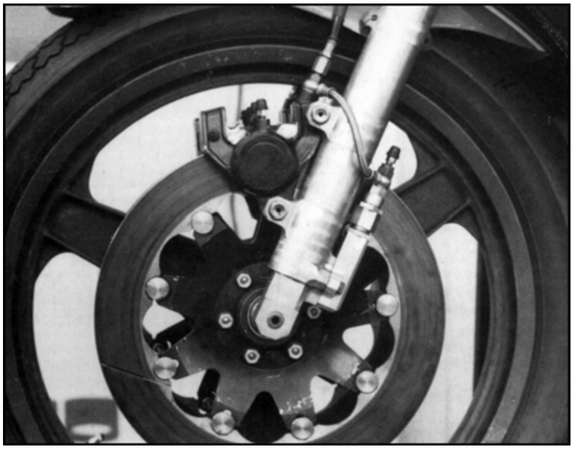



Fig. 9.24 **The forks move in a straight line and so could be replaced by an infinitely long swing-arm. The anti-dive force line can then be constructed between the tyre contact patch and the virtual swing-arm pivot and is therefore parallel to the virtual swing-arm. We can define 100% anti-dive by using similar reasoning to that for anti-squat.**

The sketch shows that this anti-dive force line has a large negative slope, indicating a large negative anti-dive or pro-dive behaviour, in accord with experience and the calculations of fig. 9.22. Normally we probably would not think of constructing anti-dive diagrams for telescopic forks in this way, but it is shown here because it is useful for evaluating some anti-dive mechanisms and other types of front suspension. The anti-dive percentage, using the parameters from fig. 9.22, is simply -200 x tangent of the rake angle (25°), which equals -93%, from which we can determine that the suspension force when braking at 1 g. is 293% of that under static conditions. This agrees with the value determined in fig. 9.22 hence demonstrating that this graphical method is just as valid for telescopic forks, as it is for link systems.

Because of the severe nose-diving inherent in this type of fork, various attempts have been made to counteract or control this response. One such system tried by some Japanese manufacturers, on both roadgoing and racing machines, is to interconnect the hydraulic braking and damping systems in such a way that fork damping is increased on brake application. This slows the rate of nose-diving but does not reduce the final amount of dive. This rate dive response is very important to the behaviour of the bike when the rider is setting the machine up for a corner, and is another factor in the always difficult compromise of suspension settings.



**Two very different approaches to the problem of controlling dive by opposing manufacturers. Above is a 1979 Suzuki RG500. Hydraulic pressure from the braking circuit is fed to valves on the forks which control the flow of damping fluid, slowing the rate of dive only. To the right is a Kawasaki GP racer, using a totally mechanical system. The calipers are attached to floating mounts which in turn are prevented from rotating by means of the approximately vertical links.**

Although seldom seen there is an effective anti-dive system which the mechanically adept can retro-fit to their own machines. At least, Garelli, Kawasaki and Honda have featured it on their G.P. racers at some time. The brake caliper is fitted to a floating bracket which pivots on a bush or bearing around the wheel spindle, the free end of this bracket is prevented from rotating by a pivoted rod usually fixed at its upper end to the bottom fork yoke. When the brake is applied the caliper bracket tries to rotate in the same direction as the wheel, but this is prevented by the rod which in turn pushes up under the fork yolk, thus acting in opposition to the diving tendency. The length and angle of the caliper bracket controls the degree of anti-dive, the longer the bracket the smaller will be the anti-dive effect. Indeed if the bracket is too short, then we may have the opposite problem, the front rising under braking.

This system has the added advantage of relieving the fork sliders of caliper forces, so reducing stiction and improving response to the road surface. Visually less tidy than simply bolting the caliper to the fork leg, this device is unlikely to be used by the major manufacturers for their road machines. As far as is known, its use has been confined to a small number of competition bikes. Technically, however, it is a superior solution to that previously described. It is difficult to see why this has not been more widely used as it can provide complete control over the anti-dive characteristics by means of simple geometric adjustments. Fig. 9.25 shows how we can use the idea of the anti-dive force line to assess and adjust its performance.

When discussing braking dive we must be aware that not all riders consider such motions to be a disadvantage. In racing, if braking continues during the initial part of the turn-in process then the reduction in trail may actually assist the rapidity of lean as explained elsewhere. This subject is very much one of the personal preference and past experience of individual riders. There are also important dynamic aspects which are covered later in this chapter.

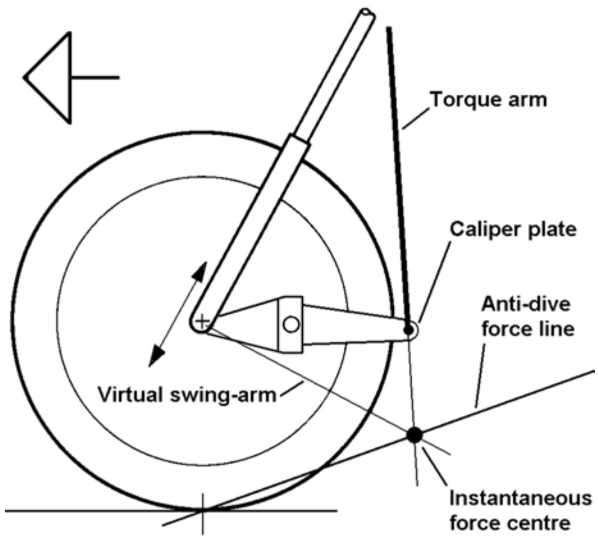



Fig. 9.25 **The caliper is mounted to a floating plate and torque reaction is resisted by compression in the torque arm. The instantaneous force centre is defined by the intersection of the extension of the torque arm and the virtual swing-arm. An anti-dive force line is formed by extending the line between the tyre contact patch and the force centre. In this case we can see that this line has a positive slope and therefore acts to produce an anti-dive effect, unlike the case with the caliper mounted directly to the forks as in fig. 9.23.**

## Alternatives to telescopic

Many of these alternatives are frequently described as having "natural anti-dive", but this is not really a helpful way of looking at their characteristics, because it ignores many of the subtleties and in many cases a small change to their geometry could in fact produce a pro-dive effect. However, the geometric layout often allows for a wide adjustment to their “anti-dive” properties.

## BMW Telelever

More details of the telelever form of construction are given in the chapter on front suspension. Whilst retaining some features of telescopic forks, the anti-dive properties of this fork are very different. Despite the apparent linear motion of the sliders on the inner tubes, the actual axle motion under suspension displacement is slightly forward and upward. The pivoting link above the wheel forces the fork to move like this, and so the pro-dive tendency of the rearward and upward motion of telescopic forks is absent. Fig. 9.26 shows how we can determine the anti-dive behaviour by constructing an anti-dive force line. Firstly we must construct the virtual swing-arm and virtual pivot, which is at the intersection of the rearward extension of the pivoting ‘A’ arm and a line draw at right angles to the direction of slider movement through the top steering joint. The reasoning behind this upper line may not be immediately apparent. It is probably easiest to consider that a very short link exists between the upper pivot and the fork tubes, this link would have to be at 90 degrees to the fork tubes to represent the same motion. So as before we can simply extend this imaginary link rearward to the virtual pivot.

From the sketch we can see that in the drawn attitude the anti-dive percentage is approximately 60%, but this will decrease with bump movement. Thus there is a residual diving force equal to 40% of the load transfer. In other words the front suspension load under maximum braking will be about 1.4 times the static load, which contrasts with a value of nearly 3 times for typical telescopic forks, shown above. In terms of suspension displacement this means that the telelever in the unloaded attitude would only dive about 20% of that for telescopics with 25° rake angle.

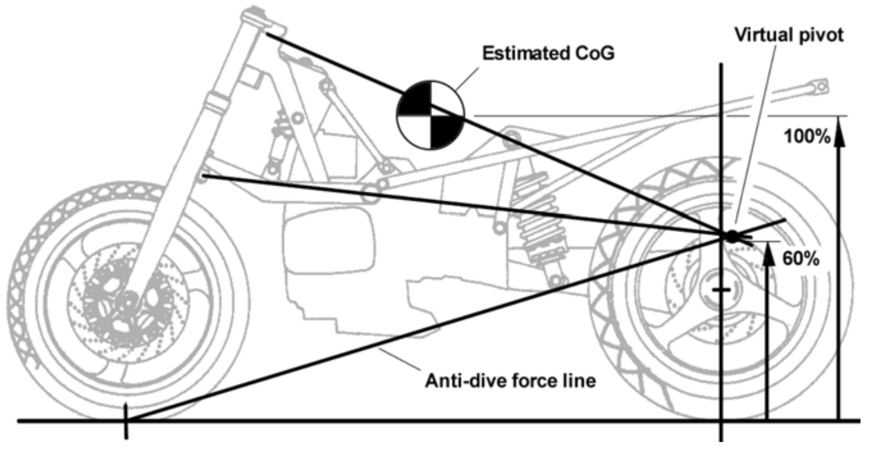
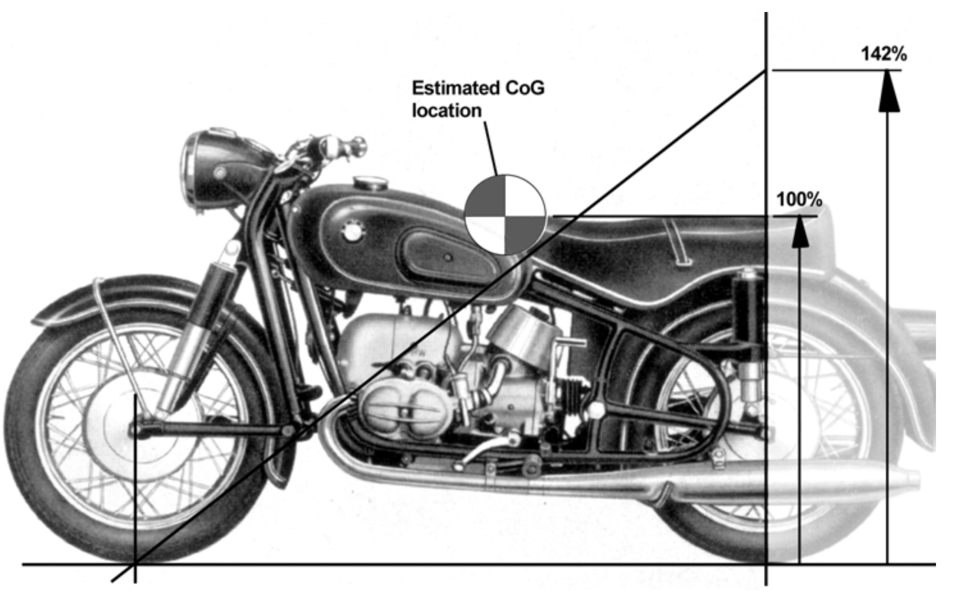



Fig. 9.26 **The BMW telelever has a virtual pivot point determined by the intersection formed by extending the pivoting link rearward and a line drawn through the top pivot at right angles to the fork movement. Depending on the accuracy of the sketch and estimated CoG position, the anti-dive percentage is approx 60%. This value will reduce with increased suspension compression.**

## Link forks

On old Earles-fork BMWs, where the brake shoe plate is directly clamped to the front pivoted fork, the action is similar to the effects of the shaft drive at the rear – i.e. brake torque tends to extend the suspension in opposition to the effect of forward weight transfer. Front braking alone would often cause the suspension to top-out losing sensitivity to road shocks.

Fig. 9.27 **This late 1960s BMW R69S featured Earles leading link forks. Based on the estimated CoG position the anti-dive percentage is about 142% which means that the suspension extension under braking greatly exceeds the compression due to load transfer. Which explains the observed behaviour of these bikes. The anti-dive force line is drawn through the tyre contact patch and the actual swing-arm pivot.**




Another problem here is loss of feel – some degree of dive seems necessary to aid the rider's judgement. On forks with shorter leading links, it is usual to incorporate a floating brake anchorage to prevent full fork extension, then any desired degree of anti-dive can be designed in by manipulating the link geometry. Some scooter forks have short trailing links to which the brake plate is directly anchored. In this case the torque reaction on the links reinforces the effect of forward load transfer to give more severe nose-diving.

As an example of short leading links with a floating brake plate fig. 9.28 shows a Moto-Guzzi racer. Ignoring perspective distortion in the photo, the anti-dive can be seen be around 64%, this is a much better value than the Earles forks above. Giving approximately 20% of the dive present with telescopics. Note how the links slope downward toward the rear, this gives a wheel axle movement over bumps similar to telescopics, that is rearward as well as upward. In the front suspension chapter we saw that this aids bump absorbsion and comfort. There is some variation in the anti-dive percentage, as the suspension loads the anti-dive reduces, but not by as much as many others. This can be a greater problem with short link forks in general, the shorter the link the greater is the range of angular movement for a given suspension displacement. The virtual pivot will move about more with this greater range of movement.

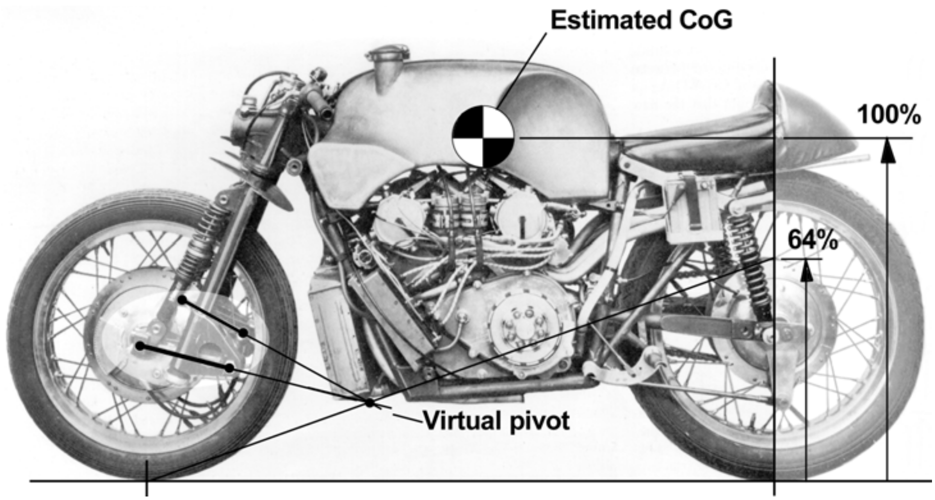



Fig. 9.28 **This 1956 500cc V8 GP Moto-Guzzi racer used short leading link forks as did all racing Guzzis of that period. The drum brake back-plate was made floating and restrained by a torque arm. The rearward extensions of this arm and the main link define a virtual pivot point, through which the anti-dive force line passes. This indicates an anti-dive of around 64%. Note that the rear brake is made floating also, restrained by a lower torque link.**

## Double-link, hub-centre and similar

The double-link system used in most of these designs is geometrically similar to that of leading-link forks with floating brake anchorages, in-so-far as anti-dive is concerned. Again, anti-dive can easily be built in, though other considerations may influence the layout.

As it is the most recent form of alternative front suspension from a major manufacturer (1993 – 1999), let's consider the Yamaha GTS in this final example of the anti-dive properties of various types of front end, fig. 9.29. Interestingly, this design appears to change from a strong pro-dive tendency when unloaded to a moderate anti-dive reaction with compressed suspension. Presumably this is to give a similar feel to that of telescopic forks during initial and light braking, but with reduced likelihood of bottoming out under maximum braking. In effect this is the same as having strongly progressive rate fork springs but only when braking.

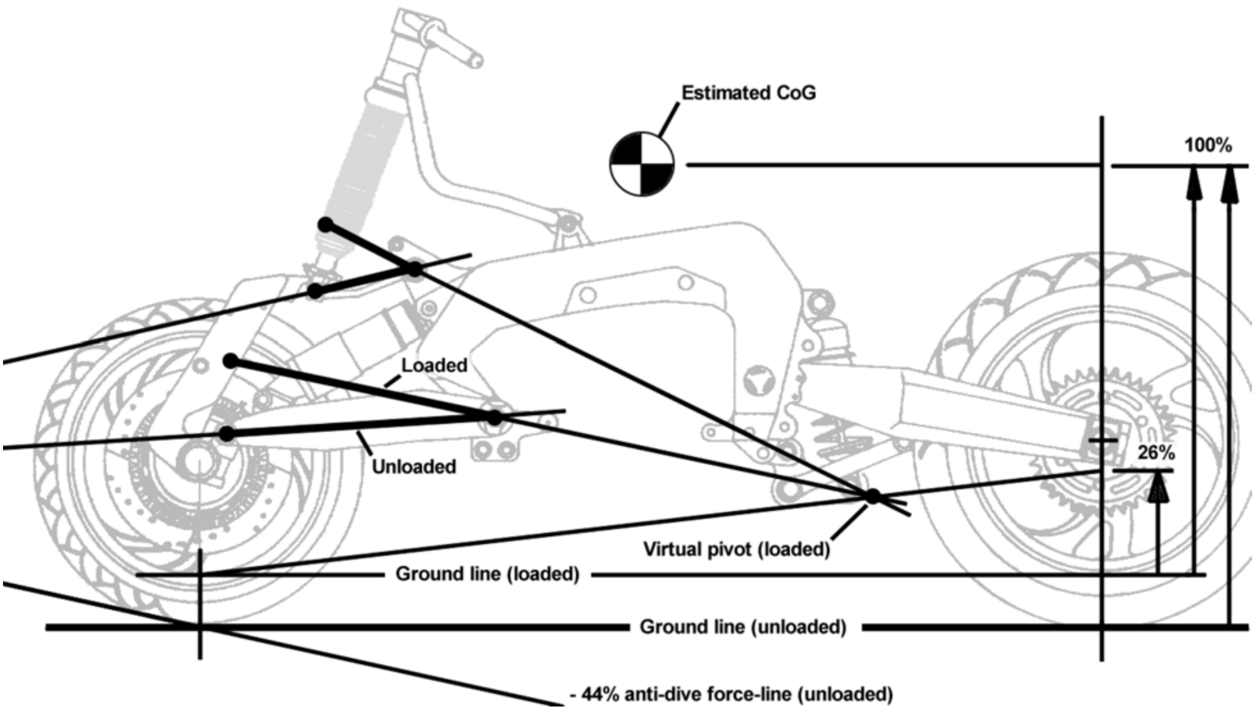



Fig. 9.29 **The GTS Yamaha, shown with the suspension links in loaded and unloaded positions. In the unloaded attitude the suspension links converge toward the front putting the virtual pivot ahead of the bike (not fully shown), as a result the unloaded anti-dive force-line has a negative slope giving approximately half of the pro-dive effect of telescopic forks (-44% compared to –93% at 25° rake). When loaded the relatively short top link angles more than the lower arm and the virtual pivot moves rearward giving an anti-dive of around 26%.**

Let's examine this in more detail. Suppose that when braking on a smooth road the pro-dive and load transfer cause some compression of the suspension, in this new position the pro-dive will have been reduced. Now, if we hit a sizable bump the wheel will rise but this will further reduce the pro-dive or increase the anti-dive forces. So as the suspension compresses there will be an increasing tendency to resist greater suspension deflection. This is the same action as we would get with progressive springing. In order to keep a constant effective spring rate (assuming that the suspension is not progressive, anyway) we need to have a constant anti-dive percentage throughout the full range of wheel travel, although unless the percentage is 100. there will still be some initial compression depending on the braking severity. Double link systems like this can be designed to have almost any characteristics that a designer practically requires. The relative length of the links and their angular disposition give a great deal of control over the reactions to braking.

## Dynamic effects

The preceding text has mainly considered the static aspects of the interaction between longitudinal accelerations (braking and driving) and suspension reactions, dive and squat. However, to evaluate such phenomenon as chatter and wheel hop, it is the dynamic aspects which are of the greatest importance, although the static squat and dive characteristics affect these dynamic responses. Traction and braking performance at the tyre/ground interface is controlled by the dynamic variation in vertical load at this point, which depends very much on the squat/anti-squat and dive/anti-dive reactions, as well as suspension and tyre characteristics in general.

This is a huge topic to cover in depth and here we'll just consider a few specific situations which highlight some important consequences of the degree of anti-dive and anti-squat chosen. These cases will look at the dynamic vertical tyre force variation and its significance to the following situations:

- The effect of different anti-dive values on front wheel braking, during the application of brakes and during a wheel lockup.
- The effect of different anti-squat values on rear wheel hop when accelerating hard.
- The effect of different anti-rise values on the application of rear wheel braking.
- The effect of different anti-rise values on the application of both wheels braking.

## Front braking

Telescopic forks, or any other type which has strong pro-dive characteristics, exhibit a strange and undesirable effect immediately after the application of the brake. The vertical load on the front wheel can actually decrease for a short period, which increases the chances of inadvertently locking the wheel. Conversely, when the braking force is removed or reduced, as for example when the wheel locks, this type of front suspension will cause a momentary increase in vertical tyre force which may help restore full braking. This may seem to be in direct contradiction to the known phenomenon of forward load transfer, but is quite easily explainable, consider figs. 9.22 and 9.30.

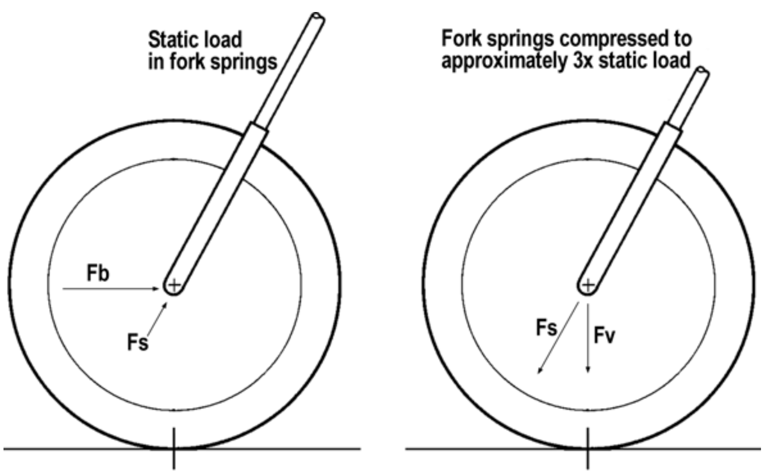



Fig. 9.30 **At the left is the application of the brake. The braking force Fb is suddenly applied which creates a component Fs acting on the springs in an upward direction, this force tends to lift the wheel reducing the tyre to road force.**

**The right side sketch shows the high spring force Fs due to the fork compression, if the braking force is suddenly released, this force creates an additional vertical component Fv tending to increase the tyre to road force.**

Load transfer onto the front tyre is not instantaneous, it is passed to the tyre through the fork springs and damping, and so it can only build up as the fork compresses. At the instant of applying the front brake a rearward horizontal force (**Fb**) will be created, and due to the rearward rake angle, this will produce a component of this force in the direction of the fork springs (**Fs**). Therefore until sufficient load transfer has built up, this additional force (**Fs**) will attempt to compress the springs from the bottom end, tending to lift the wheel thus reducing the tyre road contact force. A suspension system with 100% anti-dive completely eliminates any suspension displacement, passing the load transfer directly to the wheel without the delay of the spring compression, and so there is a quicker loading of the tyre to road interface. However, this loading is still not instantaneous because of the tyre spring rate, it takes some time for the tyre to compress to its equilibrium position.

A reverse action occurs when the braking effort is reduced or eliminated. This is unimportant when we deliberately release the brake, but has important consequences when the braking force is reduced due to exceeding the maximum tyre friction available which often results in locking the wheel. The right side sketch in fig. 9.30 shows the high spring force in the compressed fork, we saw in fig.9.22 that this force can be approximately three times the static level, under heavy braking. On removal of the braking force the spring compression takes time to return to its static length, and whilst doing so the excess spring force (which is no longer reacted by the in-line component of the braking force) creates a vertical component which increases the vertical load on the tyre. In the case of a slight wheel lock this increased tyre force may help to regain full braking effort, in other words we can consider pro-diving front suspension as having a limited degree of automatic ABS braking. Suspensions with high amounts of anti-dive will unload the tyre much quicker and tend to reinforce the loss of adhesion, thereby increasing the tendency for the wheel to fully lock. *(The idea of this "auto ABS" effect of telescopic forks was first proposed during conversations with Keith Duckworth – designer of the famous Cosworth F1 engines.)*

Fig. 9.31 shows the results of a computer simulation of firstly applying the brake to get a 1g. deceleration and then, after 1 second, reducing the coefficient of friction to 0.8 as representative of that which might occur when the maximum tyre friction available is exceeded. This simulation assumes zero structural compliance of the suspension systems in both cases.

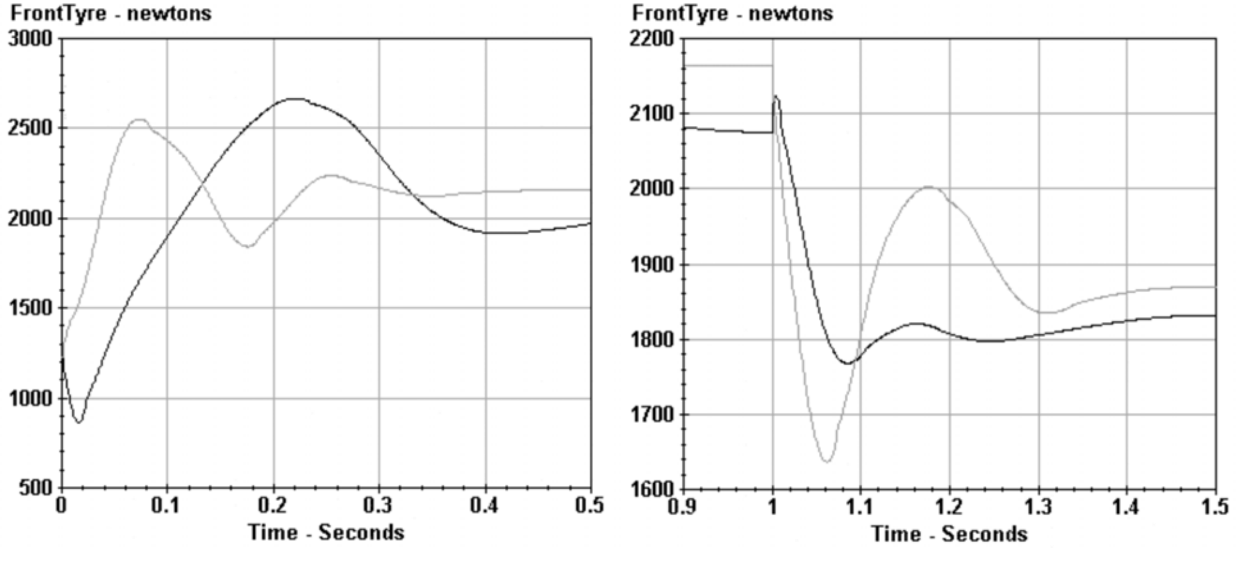



Fig. 9.31 **Computer simulation showing the vertical tyre to road loading during the application of hard braking and then at 1 second the sudden reduction of the coefficient of friction to 0.8 to represent a loss of adhesion. Note that to show the effects more clearly the vertical scale is expanded on the second graph. The dark line is representative of normal telescopic forks and the lighter line shows a system with 100% anti-dive.**

We can see that on application of the brake, the tyre load in the telescopic case is initially unloaded from 1200 N down to about 860 N and takes longer to reach its final steady value than a system with a lot of anti-dive. Thus we have to be more careful with the initial application of braking when using telescopic forks.

However, the second part of the curves, after the loss of adhesion, clearly shows the “auto ABS” load peak and slower loss of tyre to road load, which heavily favours telescopic forks. Note that the initial steady state tyre to road load is higher in the case of the anti-dive system, this is simply because the CoG height is maintained at a higher level, giving an increased load transfer. Also worthy of note is that the anti-dive system shows a greater oscillatory behaviour, this type of system only experiences a minimal suspension movement and so only a minimum of damping comes from the suspension dampers, the tyres are left to supply most. On the other hand, the large damper displacements of the telescopic forks smooth out these transients better. Not shown here but additional simulations showed that the braking transients of telescopic forks were hardly affected by different values of tyre damping, whereas this changed the oscillatory behaviour of the anti-dive system markedly. This oscillatory motion is due to the wheel bouncing on the tyre, often called wheel hop. In extreme cases this can become severe enough for the tyre to bounce off the ground, particularly if the wheel hop frequency coincides with a structural resonance.

It must be remembered that these simulations assumed no bending in the fork legs and this would tend to counter the “auto ABS” feature in practice. When braking hard the forks will be bent backward to some extent, but when the braking effort is reduced this flexing will be reduced also and due to the rearward inclination of the legs this will tend to quickly reduce the tyre load. It would require more complex simulations, than those used for illustration here, to investigate this behaviour more fully. However, it is probably safe to assume that under conditions of hard braking, wheel locking is likely to be more sudden with front suspension systems having a high degree of anti-dive. Although such designs are less likely to lock up due to initial hard application of the brake.

The performance of full ABS braking systems will be affected by the differences in dynamic tyre loading between pro-dive and anti-dive suspension systems. ABS systems will cycle between brake application and brake release, and so the two different dive values will each have their advantages and disadvantages at opposite ends of each ABS cycle. Ultimately, it may be other considerations which determine whether, in any particular installation, a pro-dive or anti-dive suspension system gives the best ABS response. Pro-dive is usually represented by telescopic forks which are likely to have less longitudinal structural rigidity than a double link or hub centre steered design, usually associated with anti-dive. This is likely to reduce ABS performance in the case of telescopic forks, particularly if the cycling frequency is close to the resonance frequency of the fork and wheel assembly. On the other hand, we have seen that the double link system is more prone to wheel hop under hard braking and this too will likely detract from braking performance, especially if the wheel hop frequency is close to the ABS cycling frequency.

## Acceleration

Different values of anti-squat percentage under acceleration raise issues similar to those of braking as discussed above. However, as we have seen in the example used in fig. 9.11, there is usually a large variation in anti-squat over the range of suspension movement of a typical swing arm design. On full rebound the anti-squat was 133% but dropped to 30% at full bump. This compares to a much smaller variation with normal telescopic forks, for an example with a 27° static rake angle which reduces to 21° under hard braking, the pro-dive percentage only varies from 100% down to 76% respectively. Whilst the anti-squat properties change considerably due to suspension movement, the range of practical variation in these values between different motorcycles is more likely to be just between different values of anti-squat, unlike the swing between pro-dive and anti-dive usual between different designs of front suspension. The largest differences at the rear are between chain drive and simple shaft drive designs, such as shown in fig. 9.4. Fig. 9.32 shows the results of some simulations of acceleration with different values of anti-squat.

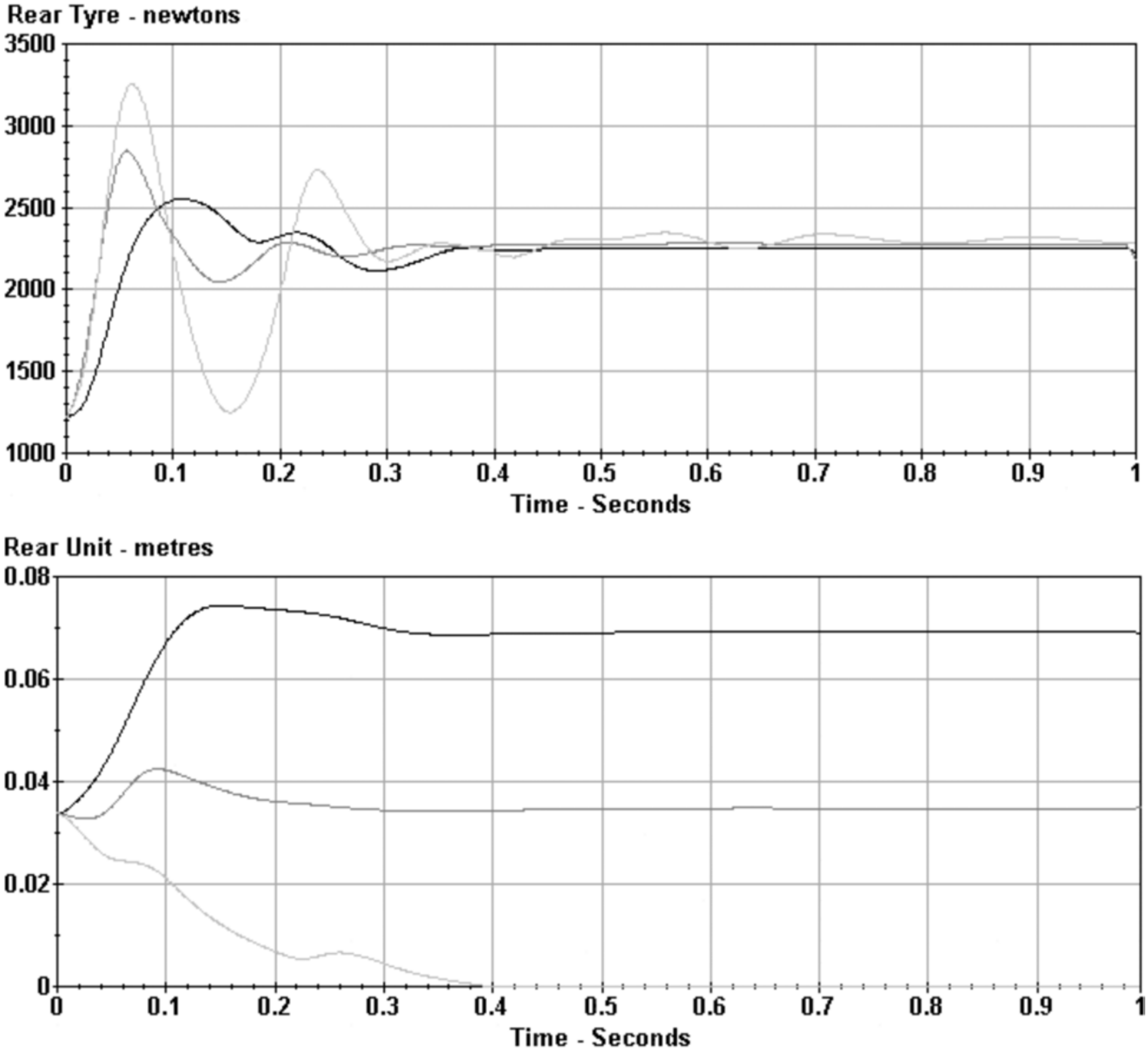



Fig. 9.32 **These plots show the effect of a step input of driving power on tyre force (upper) and suspension compression (lower), at different values of anti-squat percentage. The dark curves are for 0%, the lightest for 200% and the remaining curves show the results for 100%.**

The curves of suspension compression clearly show the quite different responses dependent on the anti-squat percentage. At 0% the suspension is compressed due to load transfer, at 100% the displacement is minimal, whereas the 200% case (representative of a simple shaft drive) shows the suspension being forced to extend to the rebound stops, a well known problem with such shaft drive machines. The tyre contact forces can be seen to initially build up quicker with the higher values of anti-squat, but (just as in the case of high anti-dive) this is later replaced by an increased oscillatory wheel hop behaviour. This varying tyre force generally is detrimental to traction and hence ultimate possible acceleration. Tyre damping has a large influence over this behaviour.

In a racing context, the most significant aspect of the above simply boils down to the question of what speed we can reach in a given time. The left hand curves in fig.9.33 show the speeds obtained using the three values of anti-squat from the previous example. Only the first 0.3 second is shown as this is were the tyre load variation was greatest.

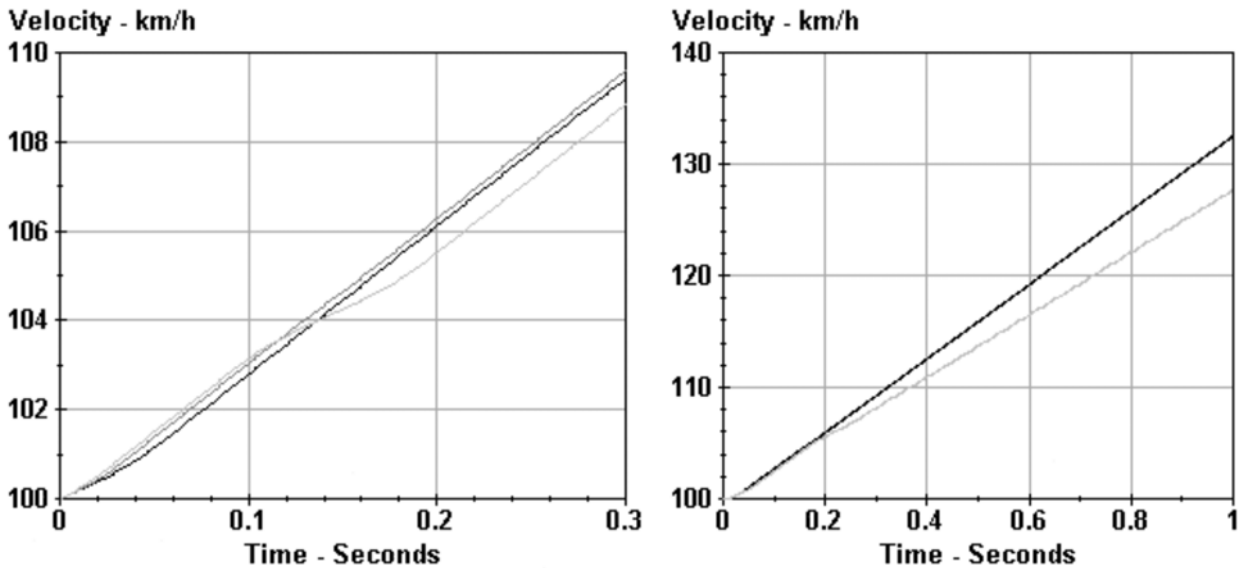



Fig. 9.33 **The plot to the left shows the velocity history for the three values of anti-squat used in fig. 9.32. starting from an initial velocity of 100km/h. The darkest line is for 0% anti-squat, the lightest for 200% and the other for 100%.**

**The right side graph shows the case of 100% anti-squat, but the dark line is on a perfectly smooth simulated road and the lighter line shows the acceleration on a road with continuous small undulations. These were sine-wave in nature with a wave length of 1 m. and a peak to peak height of only 0.01 m. or 10 mm.**

Despite a good start due to the rapid increase in tyre load, the 200% anti-squat case loses traction effort during the tyre un-loading between 0.1 and 0.2 seconds as can be seen from fig. 9.32. This is reflected in the speed and as shown in fig. 9.33 this configuration loses out to the other cases from about 0.14 seconds onwards. The right hand plot in fig. 9.33 is interesting because it takes the 100% anti-squat case and compares its performance over an ideal smooth flat road to that obtained over a series of small undulations. The continual variation in tyre load has a detrimental effect on the average value of traction obtained. In this case, suspensions settings, spring rates and dampers, could be adjusted to improve or degrade the acceleration. Although in practice, suspension settings are a compromise between all the conditions to be met, and optimising for one may well cause problems for another.

## Rear braking only

Unlike the previous two control requirements, front braking and acceleration, braking on the rear wheel causes a load transfer which unloads the rear suspension and braking tyre, i.e. causes rear rise. This greatly increases the chances that wheel hopping becomes bad enough for the tyre to intermittently leave the ground. We've seen in figs. 9.19 to 9.21 how we can alter the anti-rise percentage, by changing swing-arm length and/or fitting a swinging brake mount with a separate torque link to the chassis.

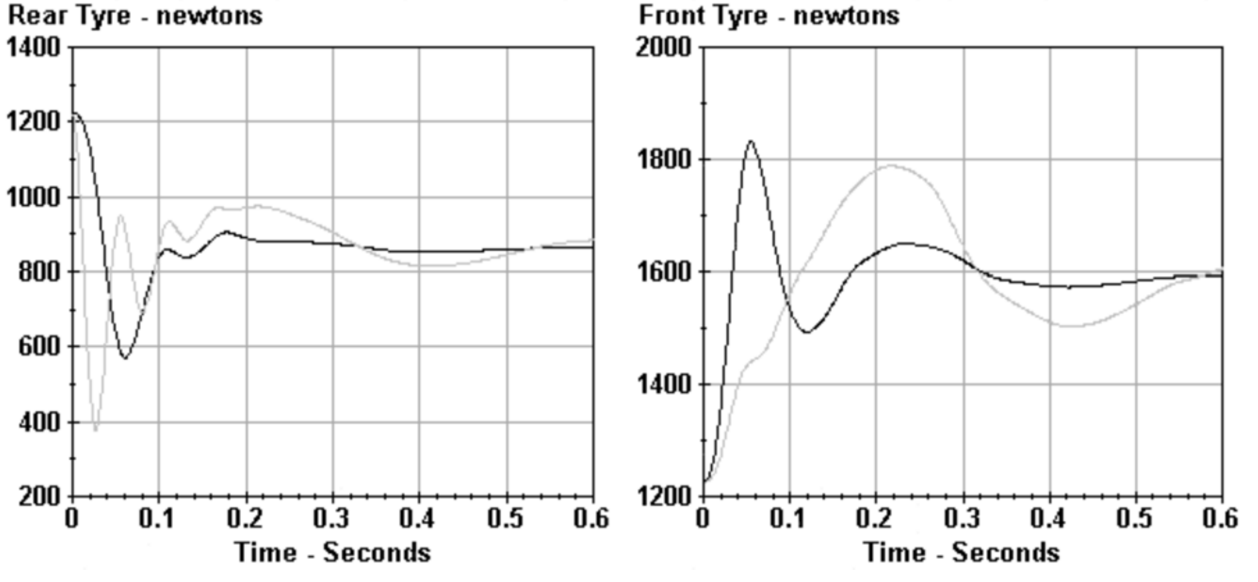



Fig. 9.34 **Tyre loading for rear wheel braking only. The dark line is for 0% anti-rise, typical of a floating brake design with parallelogram linkage in mid-position. The lighter line shows 200% which is typical of a fixed brake anchorage with a short swing-arm. The right side sketch clearly demonstrates the coupled natured of motorcycle suspension. Due to load transfer and pitch motion the dynamic loading on the front varies with changes to the rear under rear only braking.**

Fig. 9.34 shows two cases of rear braking with different values of anti-rise. One for 200% anti-rise which is typical of a short swing-arm (see fig. 9.20) with the brake anchor connected to the swing arm, the other for 0% anti-rise which is representative of a floating brake mount with a parallelogram linkage in the horizontal position. As explained previously, the anti-rise system tries to compress the suspension spring which in turn tends to lift the wheel and pull down the sprung part of the bike. The left side plots show that the tyre load is reduced quicker in the 200% case than with 0%, due to this tendency to lift the wheel. Thus in this case we have to be more careful with the initial application of the brake.

The right sketch shows that the front is strongly coupled to the rear, even in the case of rear braking only. The loading history of the front depends very much on the anti-rise chosen for the rear. Suspension settings, springs and dampers, at either end will also have significant effects on the dynamic behaviour at the opposite end.

## Braking at both ends

These simulations assume that both brakes are dynamically operated to achieve the optimum braking possible, even though this is very difficult for a real rider to do in practice. Like the case of rear wheel only braking, two cases are considered with 0% and 200% anti-rise at the rear, but in each case the front remains fixed at 100% pro-dive – telescopic forks in other words. Fig. 9.35 shows the tyre loading on both front and rear tyres and fig. 9.36 shows the pitch angle and suspension compression.

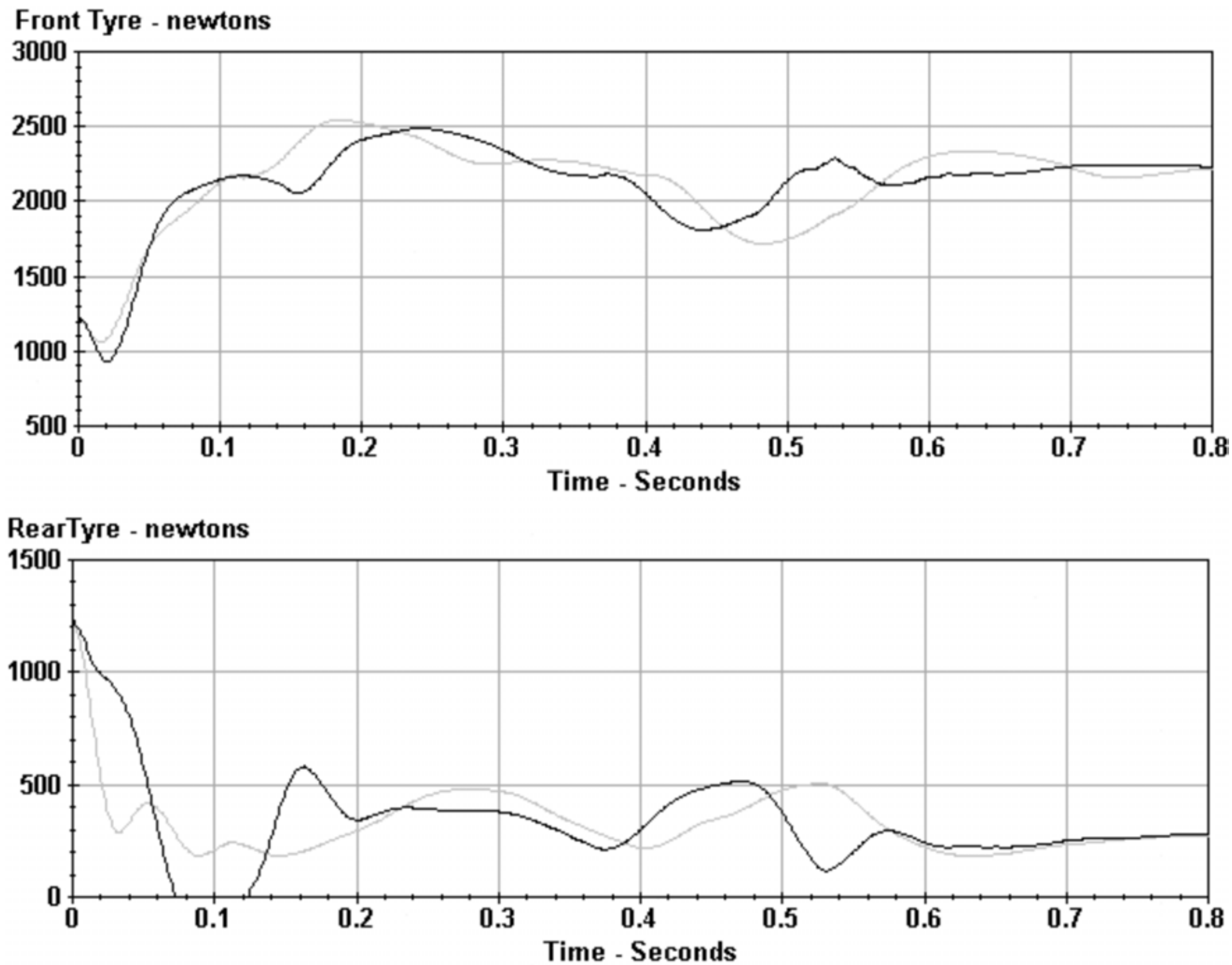



Fig. 9.35 **These two cases with both wheels braking are represented by the same line shading as in fig. 9.34 – dark represents 0% rear anti-rise and the lighter lines are for 200%. Note that the rear tyre leaves the ground around 0.1 seconds in the 0% case**

Despite the initial steeper removal of rear wheel load in the case of the 200% anti-rise (as we also saw in the case of rear braking only), it is in the 0% case that the load is removed completely for a short time as the wheel leaves the ground at around 0.1 seconds. This seems to go against the previous discussions, but just shows that when considering dynamic effects there are many things to account for. Study of the pitch motion in fig. 9.36 shows that in the 200% case the tendency to compress the rear suspension has slowed the build up of pitch velocity and also helped to lower the CoG height, thereby further reducing the load transfer to the front. The higher pitch velocity in the initial stages has been enough to continue on and raise the rear wheel, in the case of 0% anti-rise.

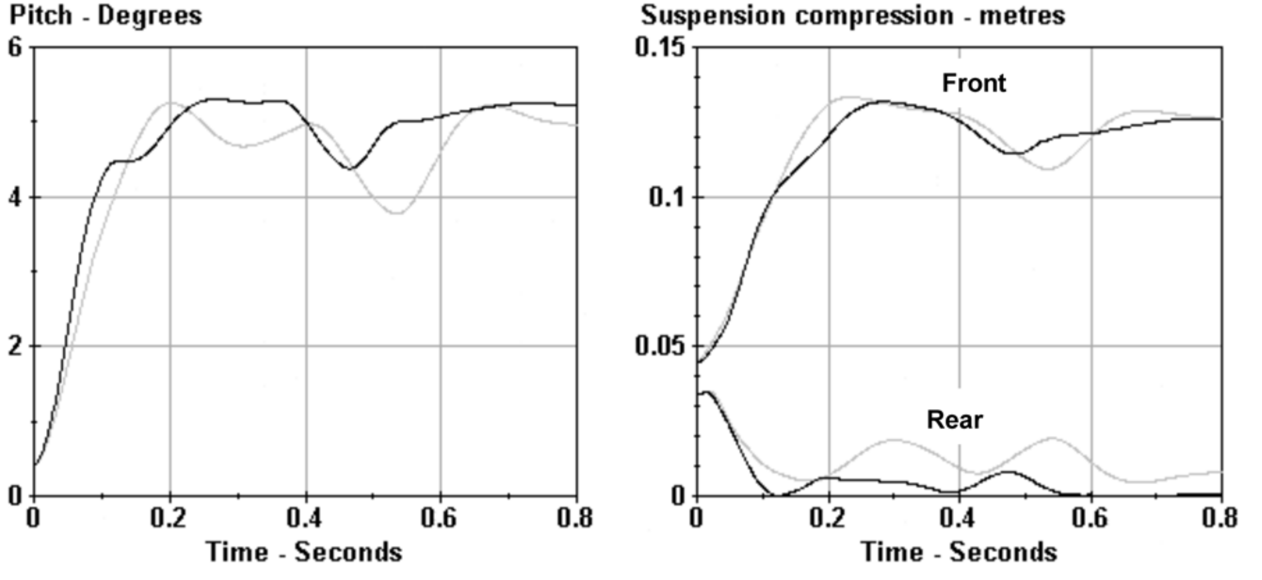



Fig. 9.36 **Showing the pitch angle to the front and the suspension compression for the same two cases used in fig. 9.35. Note the large difference in the initial slopes of the pitch curves around 0.1 seconds, the darker line shows that the 0% anti-rise case builds up a higher pitch velocity which in turn helps explain the rear wheel lifting off the ground in this case.**

## Some effects of preload

We have seen that due to the effects of load transfer, and various dynamic responses, during braking and acceleration, either wheel can be relieved of load. Commonly called wheelies and stoppies, these actions cause the forks and rear suspension units to react against the top-out springs or rebound stops. Accordingly we should expect that the degree of spring preload will have a significant effect on the transient responses of the suspension during these manoeuvres.

There are two principal aspects of greater preload in these circumstances;

- The static loaded position of the suspension is closer to the rebound stops, and so there is less suspension movement available before topping-out. So that occurs more often.
- The preload force means that when the suspension does top-out, it does so in a harder fashion.

Figs. 9.37 and 9.38 show some simulations of front wheel braking only and acceleration for two different amounts of spring preload. The first example shows the rear wheel motion caused by front wheel only braking, one curve with zero preload and the other with the spring adjustment wound up by 25 mm. In both cases we see that the wheel rises and unloads the tyre in an identical manner up to about 0.06 seconds, after which the preloaded case tops-out and the forward pitch momentum continues to lift the wheel off the ground, resulting in a sequence of increasing wheel hops.

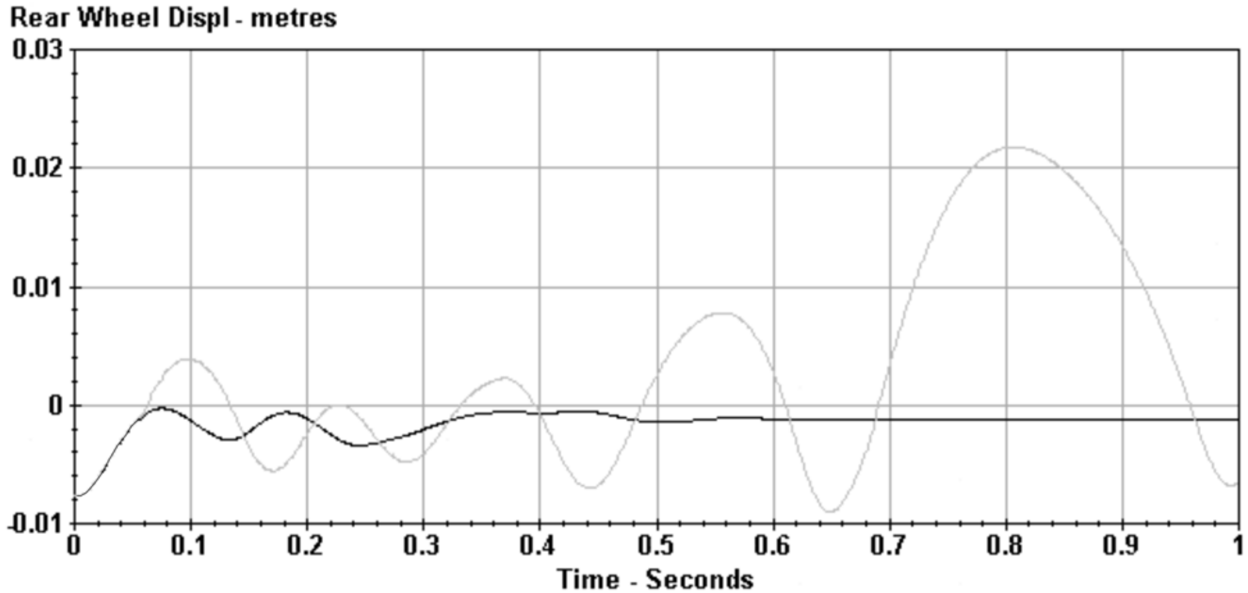



Fig. 9.37 **Rear wheel vertical displacement due to the action of front wheel only braking. Negative values indicate tyre compression and positive values show when the tyre has left the ground. The dark curve shows the motion with no rear spring preload, in this case the rear wheel stays on the ground. The light curve is when the same rear spring is preloaded by 25mm. This is enough to cause the rear wheel to start hopping with an increasing magnitude, which would probably lead to the rider easing off on the brake.**

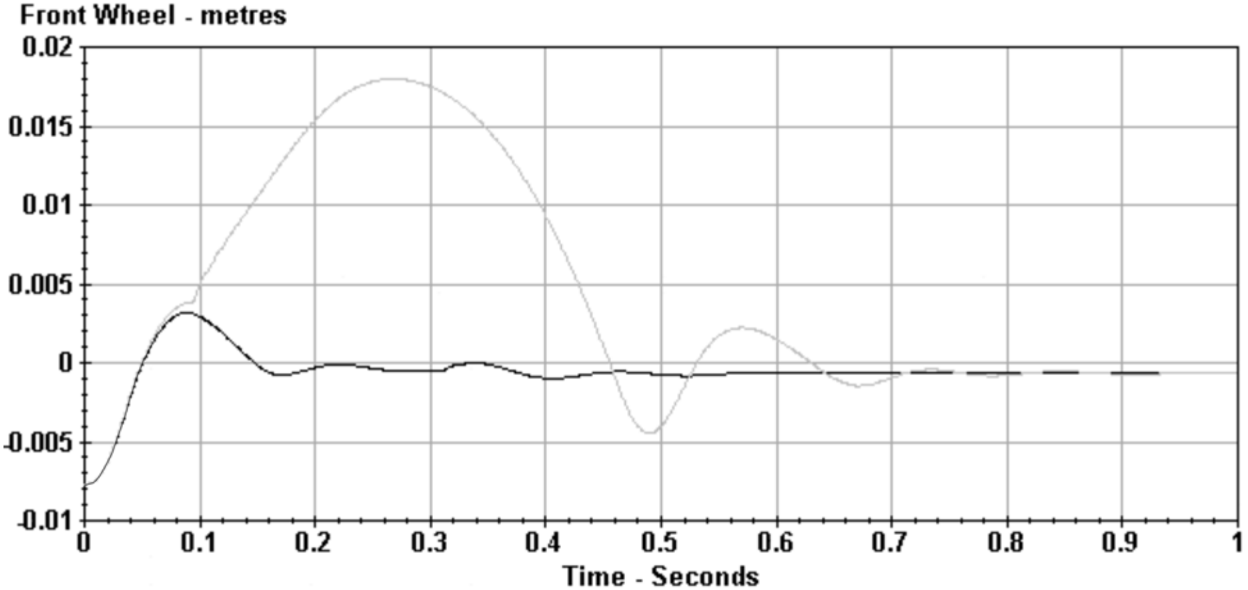



Fig. 9.38 **Front wheel motion under acceleration. The dark curve has no front preload and the lighter plot is with 25 mm. The acceleration is just below the level needed for a steady state wheelie, but in both of these cases there are small initial wheelies due to dynamic effects, as often seen when accelerating.**

The second set of curves show the front wheel response to acceleration, as in the previous braking situation the initial motion, up to nearly 0.1 seconds, is very similar for the two preload conditions and in each case the wheel leaves the ground by a few mm. After that the rearwardly pitching machine hits against the rebound stops and gives a sudden kick to lift the wheel higher up to about 18 mm. This is clearly indicated by a sudden change in the slope of the motion graph at that time. Unlike the braking situation, this wheel hopping reduces quite quickly. These are quite minor wheelies and are typical of those often seen when a performance bike accelerates hard. The point to note is that the preload and other suspension settings can affect this action.

***Note about these simulations.*** *In common with all computer simulation of dynamic systems, various simplifications have been made, and some factors ignored to produce the results shown in all the simulations above. For example, the assumption that power and braking effort can be applied instantaneously. Consequently, none of these values are meant to be totally accurate representations of any specific settings, they are just indications of the effects in general of changing some squat or dive parameters. In all cases the detail of the responses will be affected greatly by other suspension parameters such as spring rates and damper settings as well.*

# Summary

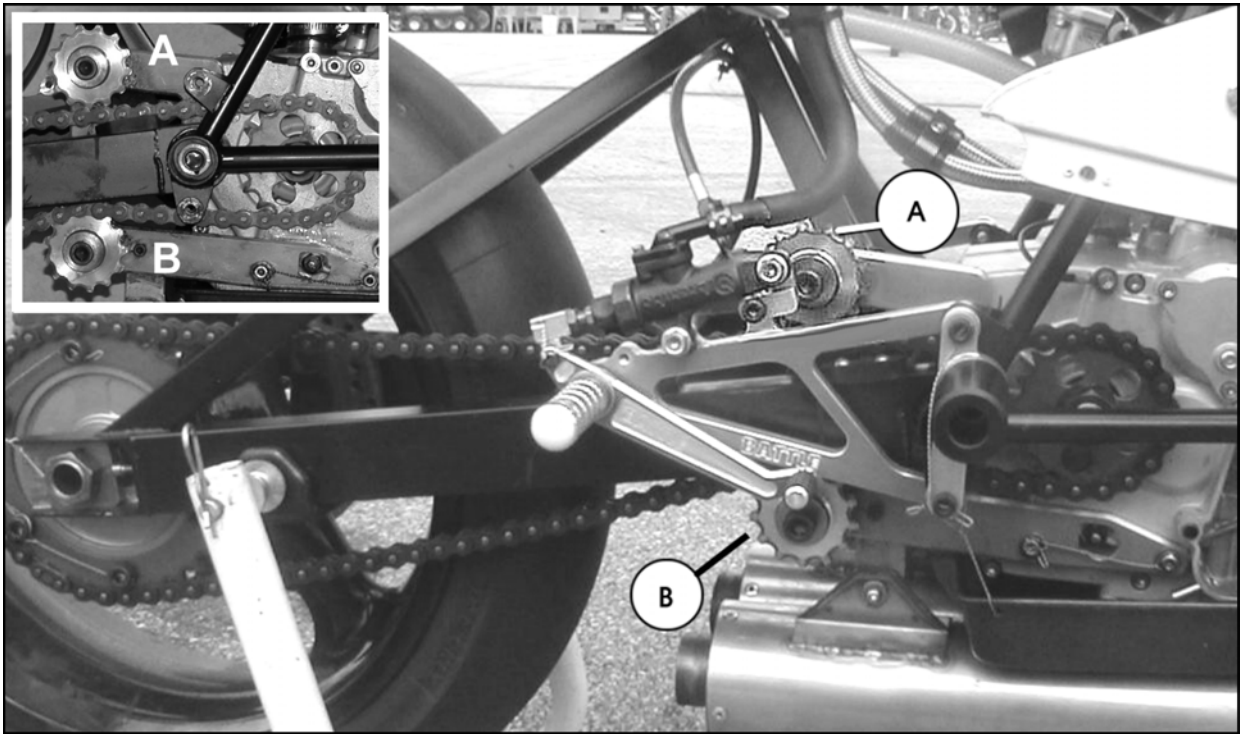
Load transfer from both driving and braking accelerations give rise to attitude change tendencies, front dive and rear rise for braking and rear squat and front rise when driving. The horizontal forces from driving and braking react through the geometric features of a particular bike to produce internal forces and moments that oppose or reinforce the squat and dive suspension movements. These reactions are known as pro-/anti-squat, pro-/anti-rise and pro-/anti-dive. Except for momentary transients and second order effects from raising or lowering the CoG these reactions do not affect the load experienced by either tyre.

The examples used, further help to show just how suspension parameters interact in very complex ways. There are an infinity of suspension settings possible with any individual motorcycle, but there is no such thing as the optimum setup. Something that works well in one set of conditions may lead to problems in others. The best one can hope for, is to achieve a compromise that works reasonably well under most of the expected situations to be faced by any machine. In racing the conditions would seem to be much more limited in their variance than on the road, and so it might be thought that setting up a racing machine would be easier. One reason that this is not true is simply because in racing the demands are for a much higher level of traction and braking performance. Even with the benefit of modern data recording systems, which can be analysed after practice sessions, it is sometimes very difficult to determine just what adjustments need to be made. We've seen how settings at one end can affect the response at the other, therefore there are times when it is difficult even to decide at which end adjustments should be made.

## Squat & Rise

We've seen how the degree of anti-squat is usually excessive with conventional shaft-drive bikes, and in like manner, the normal means of brake caliper mounting causes a degree of braking anti-rise that in extreme cases may lead to rear wheel bouncing. Improvements to both aspects can be made by removing torque reaction from the swing-arm and replacing it with forces in a separate torque link. The geometry of the swing-arm and link define an instantaneous centre, the location of which controls these anti-squat and anti-rise parameters. Currently (2002) only BMW and Moto Guzzi appear to regard the extra complication to be worth while, although some other manufacturers have longer swing-arms than possible with either of these two makes, which reduces the anti-squat and anti-rise by a smaller but useful amount. Parallelogram layouts, although sometimes used in the past, are undesirable because these characteristics vary widely throughout the range of suspension movement. In racing many riders don't use the rear brake, partly because there is so much load transfer to the front that there is little potential gain. So depending on the application it just may not be worth considering anything other than the simplest mounting for the rear brake.

Chain-drive too, introduces some anti-squat, but with long wheel movement bikes this sometimes may turn to pro-squat when the suspension is compressed. In any event, without the use of some extra mechanical features the anti-squat effect will vary throughout the range of suspension movement. A graphical method was shown to determine either a swing-arm pivot or front sprocket location to give substantially constant anti-squat characteristics. Unfortunately, the required positions are unlikely to be practical in most cases but a method making use of idler sprockets to achieve the same end was suggested.




**An example of the use of idler sprockets to control the anti-squat behaviour as suggested in the text. This single cylinder racer was built by Chris Cosentino in the USA, and also features a Hossack style front suspension with a very steep steering axis of just a few degrees. The sprocket at (A) controls squat under power and (B) controls the overrun. The inset shows more detail of the sprocket layout.**

Various designs have been marketed or proposed which change the anti-squat response, both the A-Trak and Tracklever were considered and found to reduce considerably, but not eliminate, the variation in anti-squat. The double swing-arm designs (parallelogram type) seem to offer nothing that can't more easily be achieved with a conventional single arm.

Over the past decade or so, increasing emphasis has been placed on anti-squat values as an important setup parameter for race bikes. Some have vertical adjustment at the swing-arm pivot to allow track-side variation whilst the others have to content themselves with rear ride height adjustment, but this changes many other things and so is not an independent nor desirable way of achieving the required end. To prevent changes in overall gearing from exerting an influence some racing teams now prefer to change internal gear ratios rather than sprocket sizes.

This aspect of machine setup is not always properly understood and it is often thought that the swing-arm angle is the critical dimension but this ignores the influence of sprocket size and location, others consider the anti-squat angle (as shown in fig. 9.6) as more important, this is much better but still ignores the influence of the CoG height over load transfer and so still gives an incomplete view of the problem. With fig. 9.9 the concept of anti-squat percentage was introduced. This parameter takes into account all the factors involved in the squat / anti-squat process and so is much better for comparison purposes. A value of 100% anti-squat means that driving forces will have no influence over suspension loads and so will give a very neutral feel. As the anti-squat percentage varies so much with suspension compression it is impossible to specify an optimum value suitable for all machines, like suspension and geometry settings it has to be done to suit particular tracks and riders. Whilst the anti-squat percentage will partially determine the bike's attitude, for a given driving force, and so affect handling, it was shown that the dynamic transient effects can be most important. As an extreme example, if we have a machine with pro-squat then on opening the throttle there will be a tendency to momentarily lift the rear wheel. In a straight line this might only cause a small loss of traction and maybe chattering but whilst cornering could be the precursor of more serious problems.

## Dive

The ubiquitous telescopic fork is well known for its propensity to dive. Within the normal range of rake angles between 20° and 30° they have a pro-dive percentage of about 75 to 115, and under maximum braking can load the suspension by up to about three times that of the static value. At various times there have been attempts to either reduce the degree of dive or to slow its rate. At first sight it might seem that such high amounts of dive are completely detrimental, but such is the complicated interaction with other handling parameters that the issue is by no means clear cut, particularly at the highest levels of racing.

The principal secondary effects of dive are :- Reduction in available suspension movement, reduction in rake and trail and a lowering of the CoG.

Using up suspension displacement means that there is less available for dealing with any surface roughness when braking, this can only be a disadvantage.

Reducing rake and trail can reduce stability but in the right hands this can be turned to advantage in racing. Many racers now continue the braking phase well into the initial part of the turn-in in order to keep the front tyre loaded at this critical moment to help lean-in, the trail reduction can be useful because it lightens up the steering a bit to counter the effects of the increased load, and helps turn-in.

There is a certain school of thought which reasons that dive is desirable because it reduces the CoG height and weight transfer along with it, hence allowing the rear brake to take on a bigger share of the stopping operation, and/or that it allows more front braking before lifting the rear wheel. However, this forgets that if dive is eliminated or reduced then suspension travel can probably be reduced also, which in turn means that we can lower the whole bike somewhat, i.e. the CoG. can be lower anyway.

Perhaps the most important criterion as far as the desirability or otherwise of having a large degree of dive is simply rider preference. Most riders grow up with bikes that dive and hence generally feel more at home with this reaction to braking and rightly or wrongly feel that it is a necessary part of the sensory feedback indicating the level of braking. There are other riders who feel that as the degree of dive varies from bike to bike, depending on front spring rate and various geometric parameters, that it is not a reliable indicator of how close the tyre is to reaching its maximum grip. After all, the drivers of F1 racing cars experience very little dive but seem to have appropriate feedback for the task.


**A perfect demonstration of both dive and load transference off the rear wheel as this bike is set up for a turn. Refer to fig. 9.37.** (SM30)

For those riders who prefer less dive we have seen that there is a way to control this with telescopic forks and that many of the alternative forms of front suspension can be designed with differing degrees of anti-dive. In practice, it seems that most of these alternatives have anti-dive characteristics which give a reducing degree of anti-dive with increased suspension compression. This mode of response gives the undesirable effect of a reducing spring-rate as the suspension is loaded, but only during braking. A notable exception to this is the Yamaha GTS which despite starting out with a strong pro-dive tendency (approx. half that of telescopics) changes to moderate anti-dive on full bump. This is equivalent to having stiffer suspension but only under braking. The GTS characteristics, intentional or not, may be a clever compromise, on the one hand giving the rider a feel like the familiar telescopics during initial or light braking, but on the other hand preventing the excesses of dive at full braking.

# 10 Structural considerations

Earlier chapters have detailed the effects of the various geometric parameters involved in chassis design and noted that these must be maintained by the structures concerned under all the load conditions to be expected in use. Moreover, the structures must be as light as possible consistent with an acceptable life span. To consider this in more detail firstly we have to define stiffness and strength. Stiffness is concerned with the ***temporary*** deformation of a structure when loaded and unloaded and is measured in terms of the linear or angular flexure compared with the force or torque applied. Strength is a measure of the loading that can be applied before structural failure occurs. This failure may be either breakage of some part or ***permanent*** deformation which remains after the load is removed.

## Fatigue

Failure rarely results from the static application of normal operating loads. Rather it is due to either excessive loading (such as a crash, which may result in breakage and/or permanent deformation) or fatigue, which ultimately leads to breakage. Metal fatigue is a very important and complex subject, beyond the scope of this book. Suffice it to say that, if a motorcycle chassis is subjected only to normal operating loads, fatigue will be the most likely cause of failure. The essence of good design in this respect is to ensure that fatigue failure would only occur long after the expected life of the machine.

Fatigue results from continual stress reversal, an extreme example is the fracture of a piece of metal by bending it back and forth several times. In practice, the stress levels in a structure will be such that many millions of reversals are required to cause a breakage. Fatigue characteristics vary from metal to metal. Some metals, such as steel for example, have a stress level (called endurance limit) below which they will not fail no matter how many reversals they sustain. Other metals, such as aluminium and its alloys, on the other hand, will eventually fail as a result of stress reversals, no matter how small the stress, although at low stress levels the number of reversals required to produce failure will be extremely large. Consequently, great care is essential when contemplating a frame design in aluminium, since failure is almost inevitable if it is used long enough. Porsche, indeed, for their famous 917 and other sports racing cars – with tubular space frames in aluminium and magnesium – pursued a policy of building new cars for every important event. Stress reversals in a motorcycle chassis can be caused both by road irregularities and by engine vibration, which can give rise to very large forces.

## Structural efficiency

If the components of a chassis are designed to be sufficiently rigid then, provided sound practice is applied to the details, strength will not usually be a problem. Hence, a good guide to the efficiency of a structure is its stiffness/weight ratio. In mass production, however, with common materials, cost is closely related to weight and so a major manufacturer might rather measure structural efficiency by the ratio of stiffness to cost. There are two basic routes to structural efficiency (as defined by stiffness/weight). One is to use many small-diameter straight tubes in a triangulated frame, the other is to use few large-section tubes and rely on their inherent torsional and bending stiffness. (A fine example of triangulation in nature, where efficiency is essential to survival, is the bone structure of some birds' wings.)

## Triangulation

To visualize the effect of triangulation, we need only consider the two simple structures illustrated in figure 10.1. If their bases are fixed while a force is applied as shown, then the four-sided frame may distort to a lozenge shape, with complete collapse prevented only by the tubes' resistance to bending at the corners. In contrast, the triangular frame can distort only by a change in length of any or all three sides.

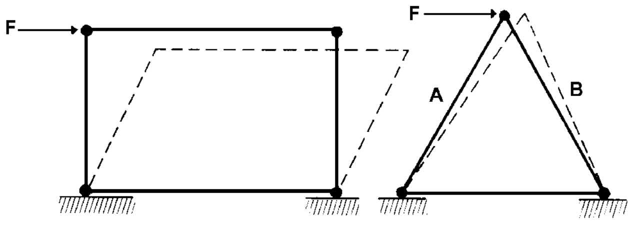
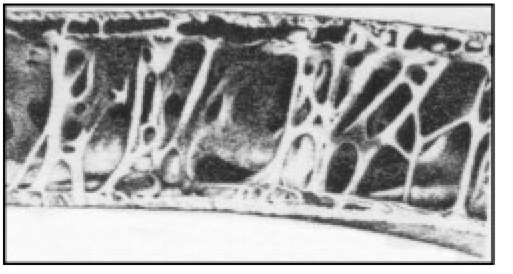


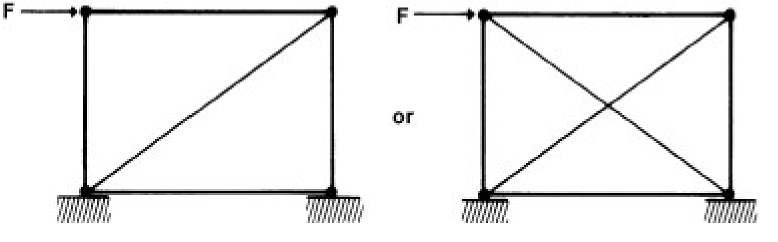

Fig 10.1 **Under force F, the four-sided structure easily distorts to a lozenge shape, relying on corner stiffness alone to prevent complete collapse. For the triangular structure to distort as shown would require side A to lengthen and side B to shorten.**



**Structural efficiency in nature. This section through a bone from a bird's wing shows a mixture of triangulation and honeycomb techniques to achieve a high level of stiffness to weight.**

A few experiments with a piece of wire or welding rod will show how easy it is to bend but how difficult it is to change in length, thus proving how much more rigid the triangular arrangement can be. A practical structure such as a motorcycle frame may comprise several such triangles, and, if designed correctly, should be very efficient. To test whether a design is fully triangulated or not, imagine that all the connections are by pin-joints – i.e. the joints have no resistance to bending. If the structure remains intact when loaded, then it is fully triangulated and may be regarded as a complete structure. If it collapses, however (as would our four-sided figure), then it is structurally unsound and is called a mechanism.

Fig 10.2 **In the pin-jointed structure on the left, the diagonal bracing strut provides effective triangulation. Two diagonal struts (right), however thin, stiffen the structure in both directions since one is always in tension.**




Of course, the four-sided frame may be stiffened tremendously by adding one or two diagonals (known as bracing struts), so converting it to two or four triangles. If only one bracing strut is added, it must be of sufficient diameter to resist compression loads if the direction of the applied force can be reversed. But if two diagonals are added they can be in very thin material, even wire, because one or the other will be subject to tension under any type of loading, and this one will complete the structure. The classic example of this technique is the wire bracing between the upper and lower wings of early biplanes.

Referring back to our pin-jointed structure, bending moments nor torsion cannot be fed into the individual members, which are thus subject only to tension or compression. If we know the magnitudes and directions of the applied loads, it is a straightforward matter (albeit tedious) to calculate the cross-sectional area of the metal needed to with-stand the individual member loads. If the load in a member is only tension, then the cross-sectional shape and size are irrelevant. But under compression a member may tend to buckle, particularly if it is long (note the difference between pulling and pushing a piece of string!). In that case buckling is best prevented by using a round tube of large diameter and thin gauge.

Motorcycle frames comprising bolted triangular structures have been built and they resemble our theoretical pin-jointed example. Although this approach has some advantages, such as simplified repair, there are disadvantages, too. Movement may develop in the joints – as a result of wear, corrosion or even manufacturing clearances – and this will negate the stiffness benefits of triangulation. Also, the nuts and bolts at the joints may weigh more than welding. Indeed, it is more usual to weld the joints, though this too can cause problems. In practice, it is rarely possible to ensure that all the loads in the individual tubes are either compressive or tensile. The very thickness of the tubes introduces physical offsets from the perfect triangular form, which in turn creates bending moments that are sometimes impossible to calculate. To complicate the issue further, the welding process itself will leave indeterminate residual stresses in the chassis, while the joints themselves may cause stress concentrations (see later). All these considerations mean that appropriate safety factors have to be applied to the calculated stresses if fatigue life is to be adequate.

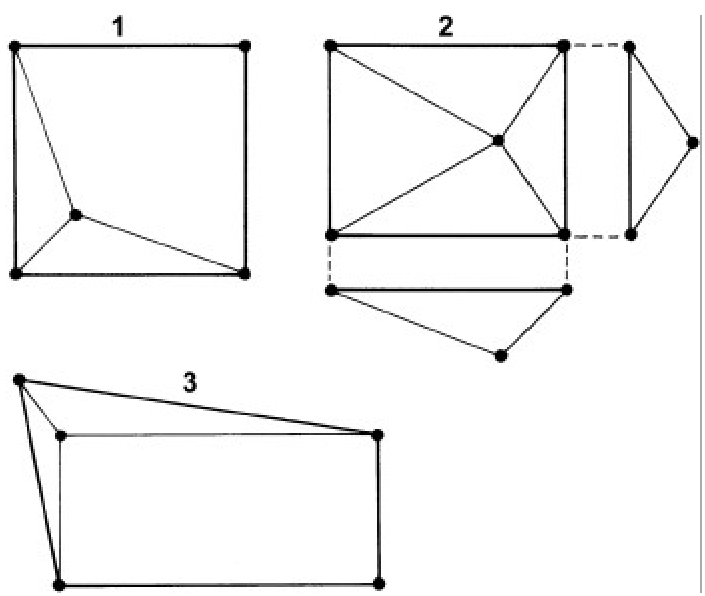
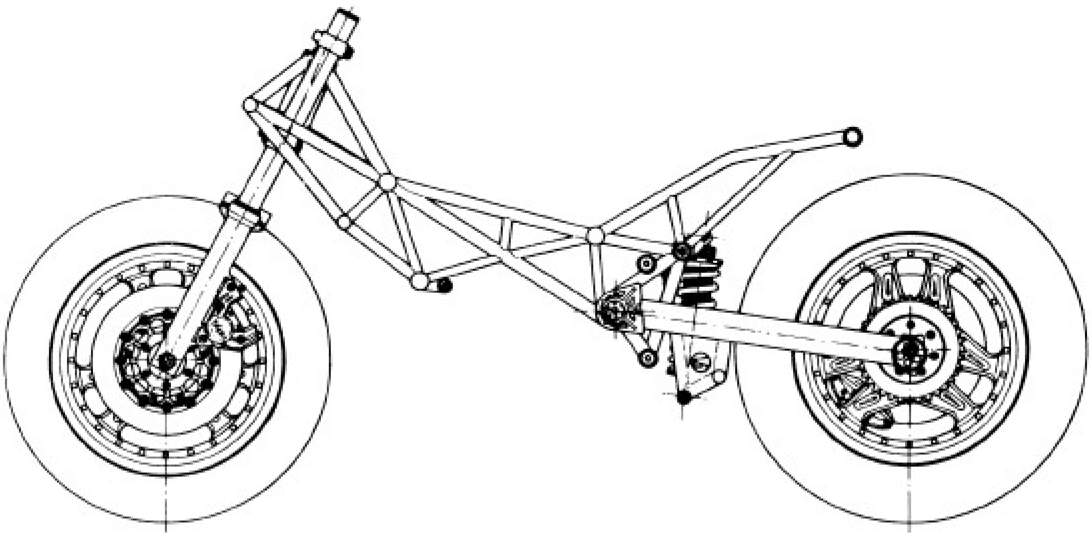



Fig 10.3 **Where there is no room for a direct bracing strut, a four-sided structure may be stiffened by any of these forms of triangulation.**

In laying out a triangulated frame, simple considerations of space and shape may present difficulties, since some engines just don't lend themselves to that type of construction. Nevertheless there are one or two ways of getting out of trouble. For example, three additional solutions to the problem of stiffening a four-sided structure where there is no room for a direct bracing strut are shown in figure 10.3. The second and third methods are examples of what might be called external triangulation – and the second is the basis of the Vincent-type of pivoted rear fork. An example of the use of external triangulation is the Bimota KB2 frame, in which the steering head is supported by several tubes. As a complex chassis, involving many tubes and much welding, this would not be a viable proposition for a large manufacturer, but it is acceptable in the context of Bimota's specialist output. Indeed, such is its visual impact that it may have marketing as well as technical merits.



**This sketch of the Bimota KB2 chassis shows the considerable use of external triangulation and the multi-tube support for the steering head. The large amount of welding involved restricts such a layout to small-scale production.**

For a machine with a conventional steering head it is possible to design a simple triangulated structure connecting the steering head to the rear-fork pivot, as shown in figure 10.4. However, this is seldom practicable because of the modifications necessary to accommodate the engine, seat and other components. Nevertheless the author managed this with frame for a 250cc Rotax-powered racer which was built on these lines.

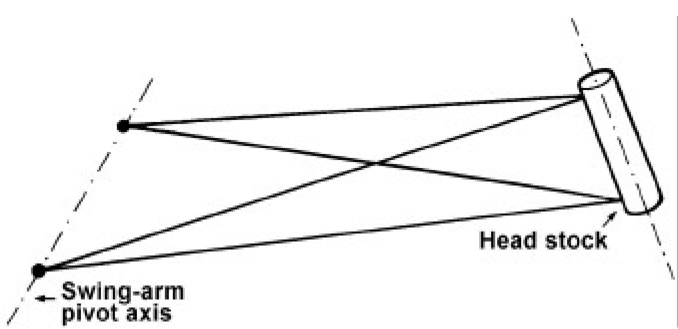



Fig 10.4 **This simple frame, triangulated in both horizontal and vertical planes, is seldom practicable because of the need to build around an engine and other parts**

**This Foale frame for a 250 cc Rotax-powered racer is of a simple triangulated type, as shown in fig. 10.4. The narrowness of the engine allowed this frame design, which is normally hard to implement with other engines.**

## Beam frames

This heading covers several different types (e.g. tubular backbone, pressed-steel beam, monocoque and the twin spar) that use large-section members for their inherent rigidity under torsional and bending loads. Beams can also be combined with triangulation to produce a practical layout. To illustrate the relationship between tube size and stiffness, let us consider two equal-length (L) segments of round tubing which are also equal in weight but of different diameters. Clearly, since the weights are the same, the larger tube has a proportionally thinner wall.

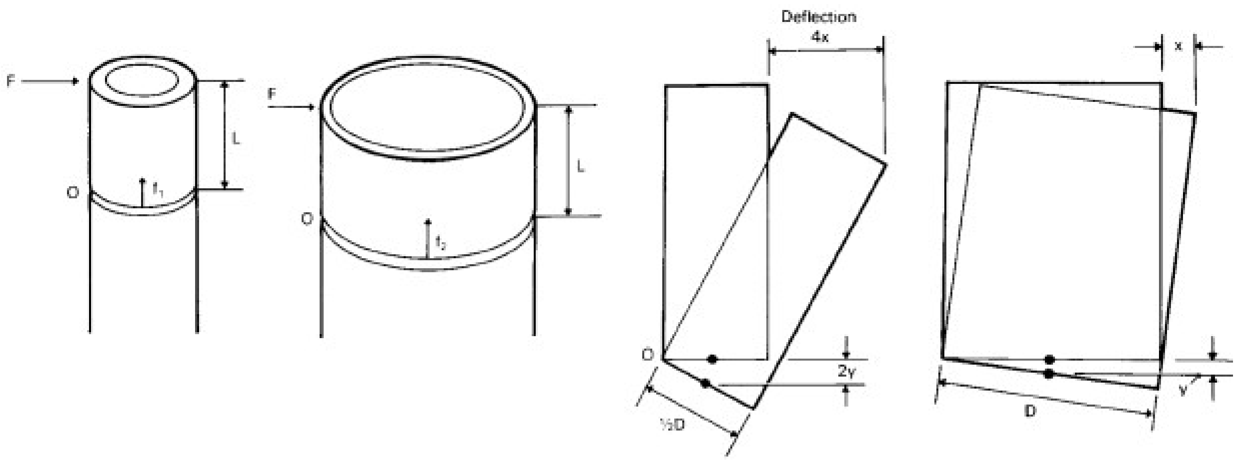



Fig 10.5 **Relative stiffness of equal-weight tubes of different diameters.**

If these tubes are subjected to a force (F) as shown, tending to bend them, the bending moment at the base of the segments is **F** x L. Imagine the bases to be pivoted at point O and the bending moment resisted by a force at the centre ($f_1$ and $f_2$). To balance the applied moment, $f_1$ and $f_2$ are inversely proportional to the tube diameters – i.e. if the larger tube is twice the diameter of the other, then $f_2$ will have half the value of $f_1$. See figure 10.5. Now let us assume that the deflection at the centre is proportional to this force – i.e. double for the smaller tube. But since this double deflection is at only half the radius, the net effect is that the *angular* deflection of the smaller tube is four times that of the larger tube. This angular deflection translates to a lateral movement at the top of the member, thus, for the same loading, the deflection about 0 of the top segment of the double-diameter tube is only a quarter of the deflection of the other. The same reasoning applies to all the segments of equal-length members, therefore the resistance to an applied lateral load is four times as great when tube diameter is doubled. Although the above is a simplified method of considering the problem, a rigid mathematical analysis gives the same results: ***the lateral stiffness of equal-length, equal-weight, thin-wall tubes of the same material is proportional to the square of the diameter.*** Torsional stiffness follows the same rule. And, although we have used round tubes to illustrate the principle, it holds for all other cross-sectional shapes.

For readers with more mathematical knowledge, a tube's stiffness depends on what is often called the moment of inertia (but should more correctly be called the second moment of area) of the cross-section about the bending axis (known as the neutral axis). If, in the pursuit of structural efficiency, we were to follow this large-tube route to its logical conclusion, we should arrive at very large sections with walls little thicker than foil. Such ultra-thin tubes would buckle and collapse under load – and so, in practice, we have to compromise between wall thickness and tube diameter. Modern large aircraft are an example of the extreme use of the large-tube principle, with an outer shell of thin aluminium sheeting as the main structural member. But in their case buckling is prevented by bulkheads and many stiffening ribs supporting the shell. In a beam-type frame, *local* buckling is another pitfall that must be avoided by paying great attention to the detail design of the method of feeding in the loads. It is possible to achieve adequate overall rigidity, while still having local weakness that can lead to fatigue failure. Before turning to the practical application of these principles, we must consider two more important structural concepts – the neutral axis and bending stress.

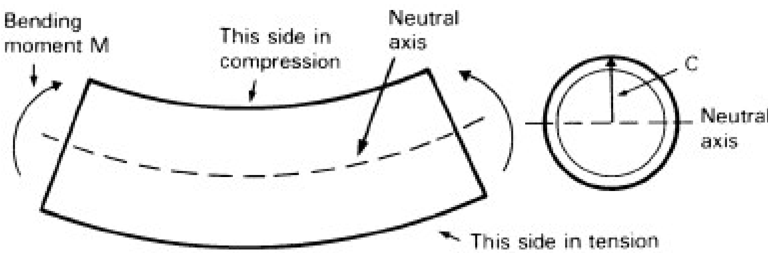



Fig 10.6 **In a member subject to a bending load, the inner surface is compressed and the outer surface stretched. At the neutral axis (between the two) length is unaltered by bending.**

Figure 10.6 shows a side view of a structural member subject to a bending load, this shortens the top surface (subject to compression) and stretches the lower surface (subject to tension). Somewhere between the two surfaces is a position of zero length change and this is known as the neutral axis. The bending produces no tension or compression stresses on this axis – unlike the outer surfaces, which are subject to stress that can be calculated by the following formula:

$$s_{\max} = \frac{M.c}{I}$$

where:

$\sigma_{max}$ is the maximum fibre stress

M is the applied bending moment

c is the distance from the neutral axis to the farthest fibre

I is the second moment of area about the neutral axis

From this it follows that, to keep surface stresses to a minimum, we must keep c as small as possible consistent with a large value for I. This requirement highlights a danger inherent in the use of simple sheet-metal gussets to stiffen a tubular chassis, especially where the tubes are of large diameter.

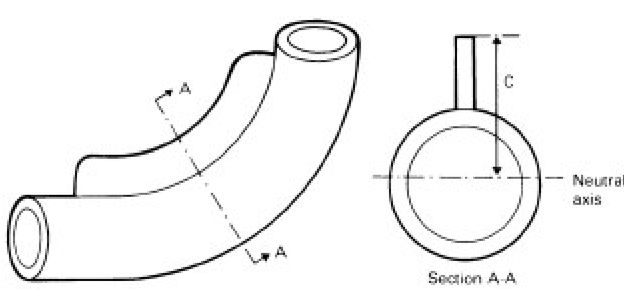



Fig 10.7 **Thin ribs or gussets may increase stresses. Unless I (second moment of area) is increased in proportion to c, the maximum fibre stress in the rib will be higher than in a plain tube.**

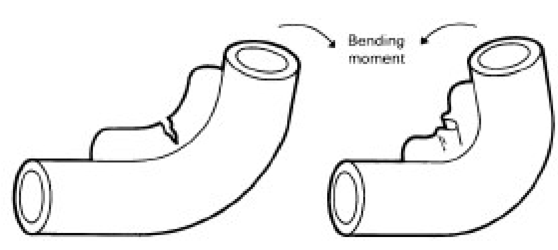



Fig 10.8 **Two types of gusset failure: tearing under tensile stress (left) and buckling under compression.**

Imagine that the frame section illustrated in fig. 10.7, with a welded-in gusset, is subject to a bending moment. If the gusset is relatively thin, then the increase in I is small and the neutral axis is moved only slightly, hence c is relatively large, this increases the maximum stress, compared with that in a plain tube, and thereby increases the chance of failure. (NB: this higher stress is only on the outer edge of the rib, the tube stress is not raised.) If the bending moment puts the gusset in compression, the failure will probably be buckling of the gusset, but if the gusset is in tension, then the failure will constitute tearing or cracking, which may spread into the tube itself and cause total failure.

Unless 'strengthening' gussets or ribs are sized correctly, therefore, there is a risk that they will actually weaken the structure. The same applies to ribs on castings. Another danger in the use of ribs, brackets and other causes of sharp changes in section is stress concentration. To explain this, figure 10.9 shows a solid bar subject to tensile stress, which may be represented by a series of equally spaced 'force lines'. When these force lines meet a sudden change in section, such as a notch, they bunch up in the vicinity to raise the local stress to two or three (or even more) times the average across the section.

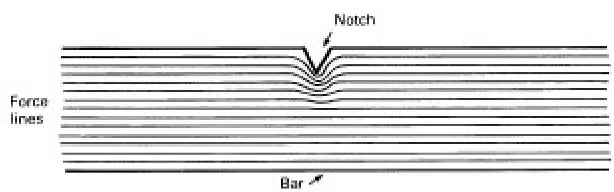
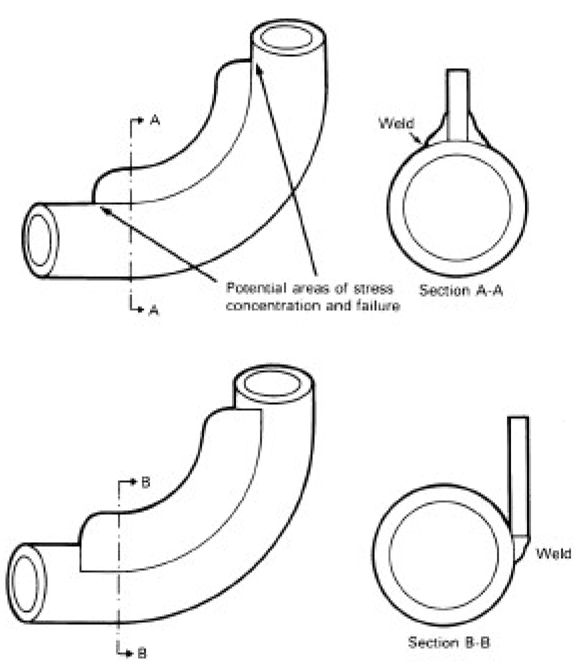



Fig 10.9 **The abrupt change in section at the notch causes the force lines to bunch up and increase local stress considerably, so shortening fatigue life**



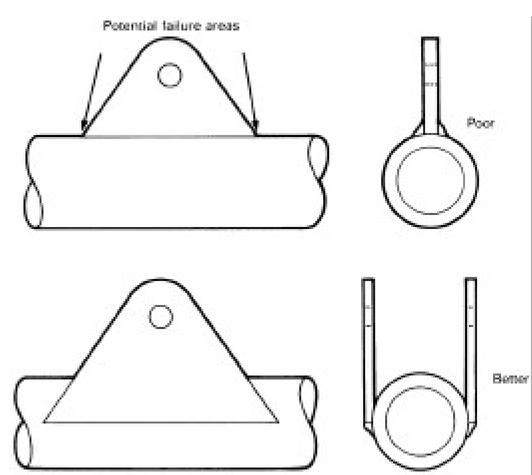




Fig 10.10 **Wrong (top) and better (lower) ways to weld a gusset on to a bent tube.**

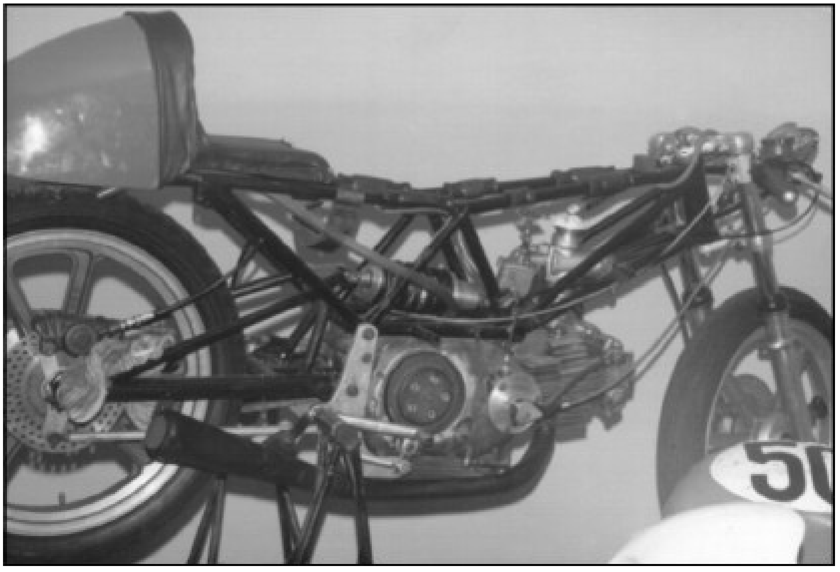
**The same principle applies to attachment brackets welded to structural tubes.**

Any change in section can cause stress concentration and increase the failure risk, the degree of which depends on the sharpness of the change in section and the ductility of the material, but *any* sudden change increases the risk of failure and this is very important to the fatigue life of a chassis. Typical areas of stress concentration are any attachment points such as the ends of gussets and brackets, the edges of welded joints and any places where a relatively flexible part of the structure is joined to a more rigid part. This explains the popular fallacy that a stiff frame is more inclined to break than a flimsy one. A gradual change in flexibility is required. To avoid problems arising from stress concentration, the designer should as far as possible position brackets, gussets and so forth at areas of low basic stress. This can often be done by welding them to the neutral axis of the supporting tube. The sort of techniques illustrated in fig 10.10 can prolong fatigue life greatly.

The preceding description of various structural effects provides an essential basis for a sound understanding of practical chassis design. Of necessity, the descriptions are brief and simple. Any reader wanting to study the subject in greater depth is recommended to read any of the many books on Strength of Materials. Now let us consider the more practical aspects of various basic frame designs.

## Triangulated frames

Although these can have extremely high structural efficiency, they have found few adherents among the major manufacturers. Probably this is because the shape and size of the most popular engine types require a wide and complicated (hence expensive) structure. Frames of this sort have found most favour with the Italians, having been used on several Moto Guzzi works racers in the 1950s and subsequently on the 500 cc Linto twins. In both cases, the engines had horizontal cylinders and so presented no great accommodation difficulties.

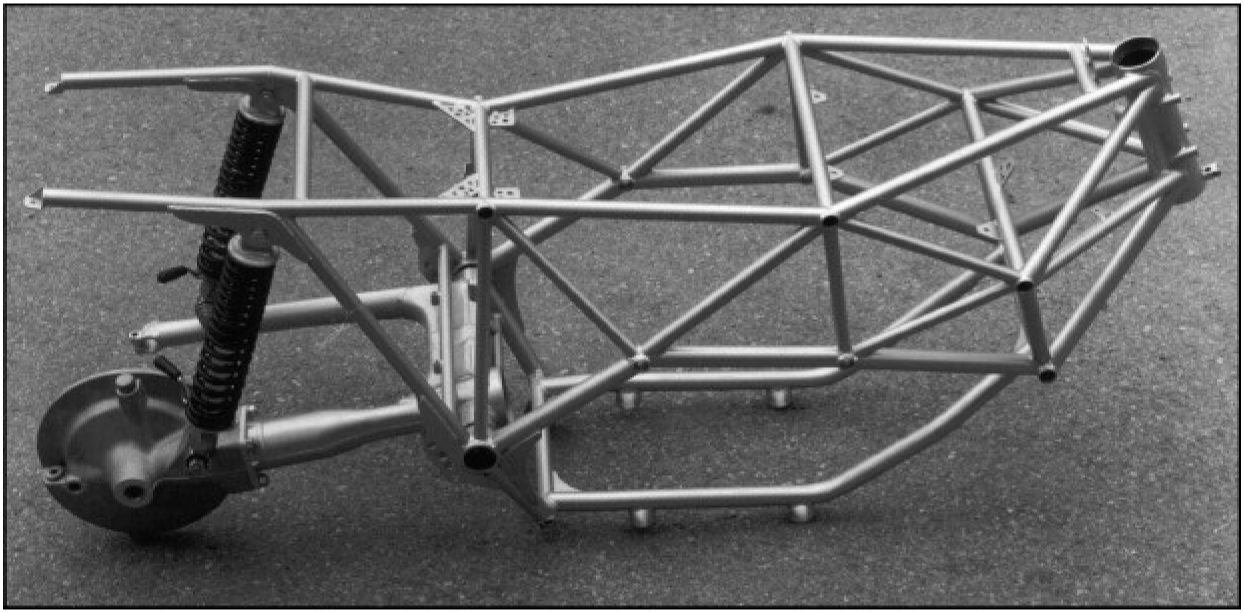


**Tony Foale's own racing Aermacchi from the early 1970s. Features a triangulated frame which also uses the engine as a structural member. The alloy plates at the rear of the engine support the rear fork pivot. The rear fork is triangulated with a single suspension unit above the engine.**

**Legendary designer Taglioni showing off a triangulated Ducati frame. Current versions of such triangulated frames made in steel tubing have proved very successful in SuperBike racing against the generally more fashionable aluminium twin spar machinery. Such frames have a high structural efficiency. This particular frame is structurally incomplete without the engine.** (Cathcart)

Other examples in the ranks of racing machines and low-production specialist roadsters include a few Norton racers in the early 1970s, the Krauser BMW roadster and Bimota KB2 chassis. A current example that has had much success both on the race track and street is the Ducati V-twin. Several recent world championships in the SuperBike class have demonstrated that the currently popular twin spar frame in aluminium is not the only worthwhile solution for performance.

A problem to watch out for with long tubes of small diameter is engine-excited resonance – that is, severe vibration in the tubes caused by unbalanced engine inertia forces at a critical frequency. The solution is to raise the tube's natural frequency, either by shortening it or increasing its diameter. This phenomenon is not unique to triangulated frames, it can occur in any design using long, thin members.



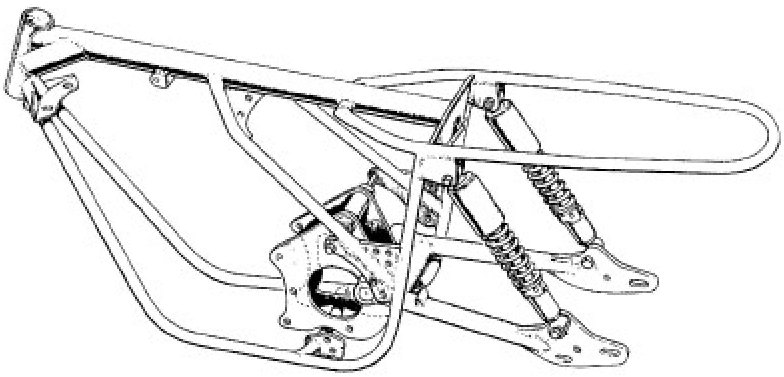
**Triangulation *en extremis:* a Krauser frame for a road going BMW transverse flat twin. Its high price reflected the large amount of labour involved in its construction.**

## Tubular backbone

This type of frame too has failed to achieve the wide acceptance its structural efficiency merits (provided the backbone has a large enough diameter). Again there may be difficulty in accommodating bulky engines. Ideally, the tube should be straight and connect the steering head directly to the rear-suspension pivot, but in practice this is seldom possible. With flat or medium-size engines, however, it is usually feasible to bring the backbone to within a few centimetres of the pivot and bridge the gap with a welded-up box section. Another way to gain the necessary engine clearance is to bend the tube. Even where these methods are unsuitable, an efficient frame can still be produced by using a high-level straight backbone and linking it to the engine and rear-suspension pivot by means of an arrangement of smaller diameter straight tubes – provided that these are properly triangulated (in both fore-and-aft and sideways planes). The Norton Commando frame was of this type but sacrificed some of the potential stiffness because the engine-gearbox assembly on which the rear fork was pivoted was itself rubber mounted to the frame.

For simplicity and low manufacturing costs, many mopeds have a curved backbone of smallish diameter. A typical example is the MZ Simson 50, where the subframe supporting the seat is bolted on, so facilitating both factory assembly and accident repairs.

Some designers have combined a curved backbone with use of the engine as a structural member. An example was the 125 cc Honda twin that spearheaded the Japanese invasion of the TT in 1959 – the small-diameter back-bone being curved through nearly 90 degrees and the engine bolted in to stiffen the assembly.



**The Norton Commando frame used a tubular back-bone and some triangulation but sacrificed some of its potential stiffness through the rubber mounting of assembly carrying the rear-fork pivot.** (MCW)

**Inexpensive Simson 50 cc moped features a small-diameter curved back-bone frame to keep production cost down. The engine was cantilevered from rear crankcase rubber mountings.**

## Structural comparison

We have seen that there are two basic structural approaches, some form of beam frame or a triangulated structure. Most real frames use either of these methods or some combination of both. Accordingly it is interesting to compare the stiffness characteristics of elementary versions of each type.

Fig. 10.11 shows two such examples. The beam frame is a simple circular backbone similar to the frame shown in the upper photograph on page 1-10, and the triangulated example is as shown in fig. 10.4. These represent the simplest and purest forms of both approaches to achieving structural efficiency, even though they are seldom practical in real situations without modification.

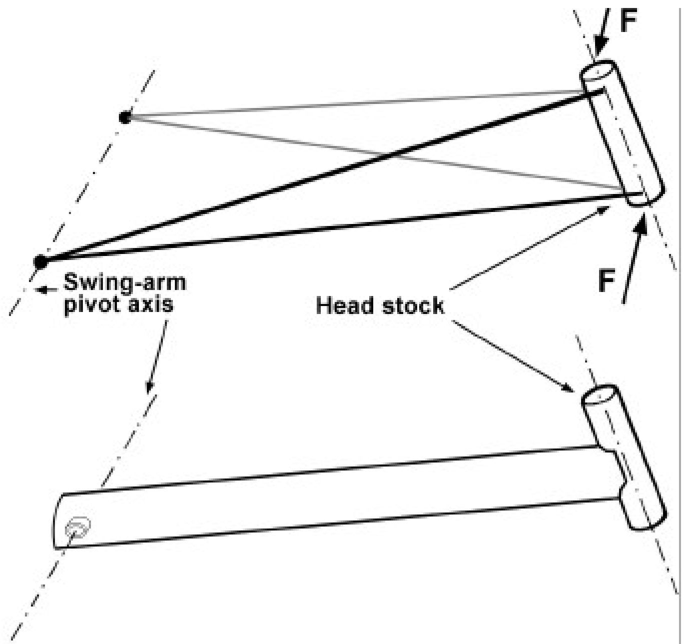



Fig. 10.11 **Elementary forms of beam and triangulated frames.**

**Each was subject to FEA (Finite Element Analysis) to determine the lateral and torsional stiffness characteristics of equal weight examples. For this analysis the swingarm pivots were regarded as fixed and equal loads (shown as F in the upper sketch) were applied to each end of the head stock. When these force are applied in opposite directions, as shown, the frame is loaded in torsion, but when both loads are applied in the same direction the loading is lateral bending. Both loading schemes were used.**

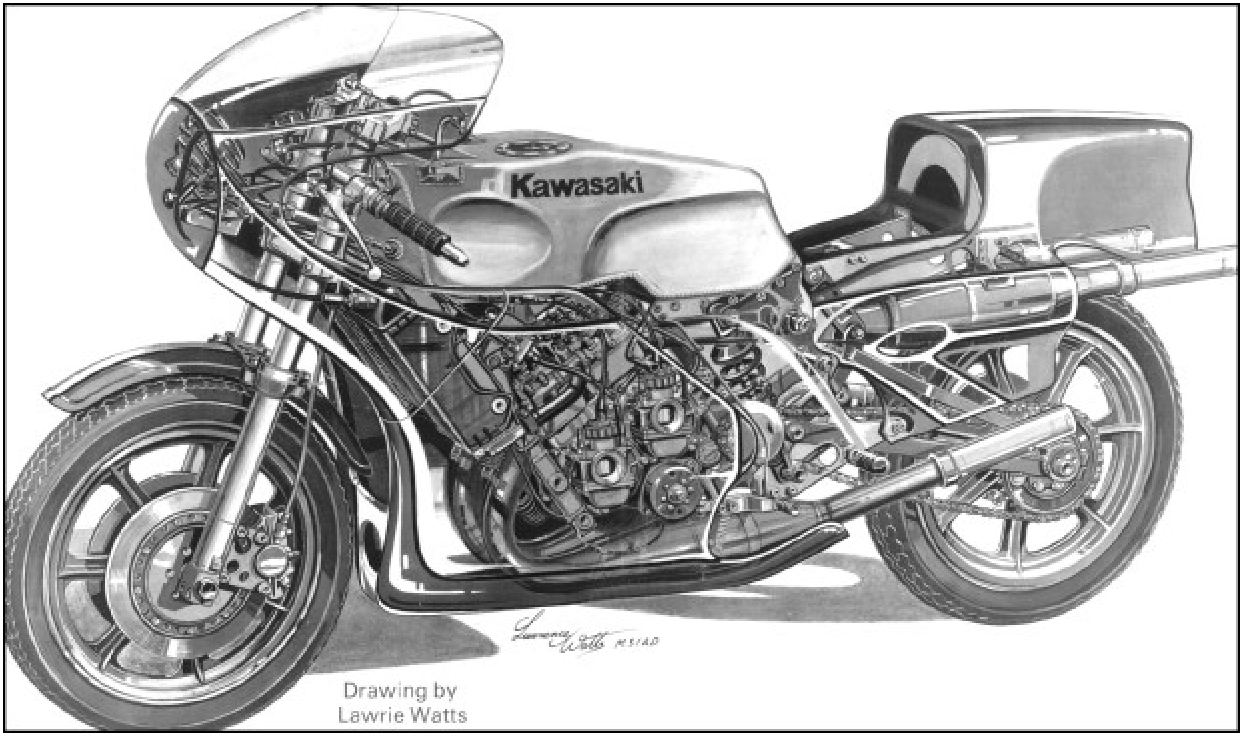
Both types of frame were analysed for torsional and lateral loading. In it's purest form the triangulated frame can be considered as pin jointed, and as such it is quite simple to calculate the stiffness properties manually, as it is with the simple backbone frame. However, we must remember that a real world triangulated frame is welded and not pin jointed. In this case it is easier to do a simple Finite Element Analysis and in the interests of uniformity this was used for all cases. The results in the following table are shown in a normalized form, comparing the properties to those of a tubular backbone of 100 mm. OD. and 1.0 mm. wall thickness. The wall thicknesses of the other examples were adjusted to give equal weight.

| **Type of structure and size of tube** | **Normalized lateral stiffness %** | **Normalized torsional stiffness %** |
|---|---|---|
| Circular backbone – 100 mm. diam. x 1.0 mm. wall. | 100 | 100 |
| Circular backbone – 75 mm. diam. x 1.34 mm. wall. | 56 | 56 |
| Triangulated – 18.75 mm. x 1.5 mm.-- pinned. | 411 | 36 |
| Triangulated – 18.75 mm. x 1.5 mm. -- welded. | 413 | 39 |
| Triangulated – 28 mm. x 0.91 mm. -- pinned. | 393 | 37 |
| Triangulated – 28 mm. x 0.91 mm. -- welded. | 399 | 42 |

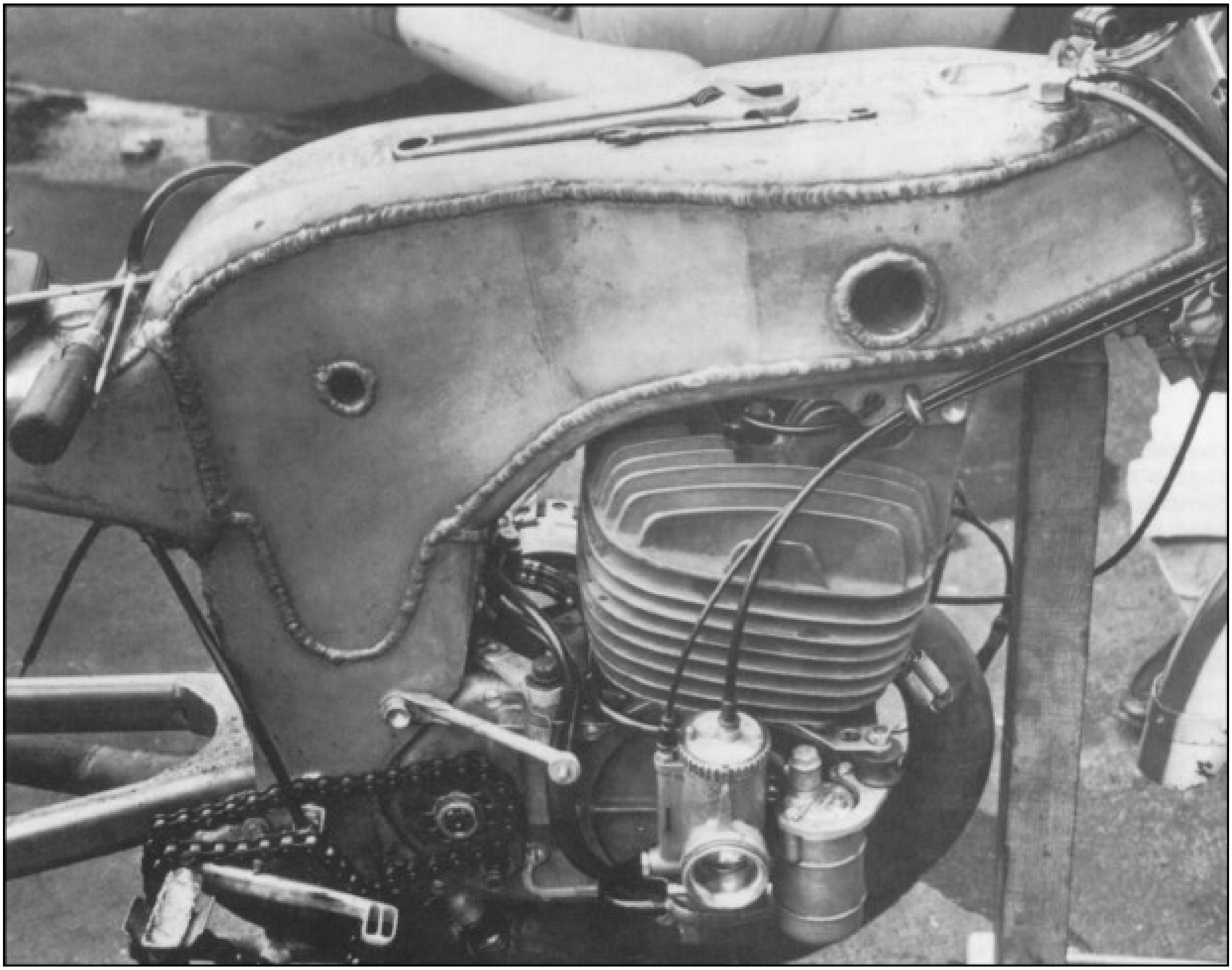
These values show some very interesting differences between the two structural types. The circular backbone is very sensitive to diameter (square law for equal weight cases), whereas we can see that the triangulated frame is not. There are large differences in the lateral bending characteristics, and the triangulated structure generally scores well, in the above examples this stiffness is 4 times greater than the larger backbone and 7 times more than the smaller one. The torsional case favours the backbone for structural efficiency, the larger one being around 2.5 times stiffer with the smaller example being 1.4 times that of the triangulated type. Selectively combining beams and triangulation gives a means to "tune" the ratio of lateral to torsional stiffness, which has implications for lateral suspension effects, see chapter 6. Note that with the triangulated examples, there is little difference between the pinned and welded cases, especially in bending. The difference is a little greater in the torsional cases, especially with the larger tubes where the inherent torsional stiffness of the tube starts to become more significant.

## Fabricated backbone

Quite a range of designs is possible here – the most popular being a T-shape structure comprising left and right steel pressings united by spot or electric-resistance welding, as in the Ariel Leader and the once highly popular Yamaha FS1E. This construction makes for rigidity and low production cost, though the high initial tooling outlay rules it out for small production runs and specials. Also, the end product is heavier than an equally rigid tubular backbone because of the inevitable excess metal in areas of low stress.




**The fabricated light-alloy back-bone on Kork Ballington's Kawasaki KR500 GP machine comprised a 32 litre fuel tank with the steering head incorporated at the front. Aluminium side plates at the rear supported the engine and swing-arm pivot.** (MCW)

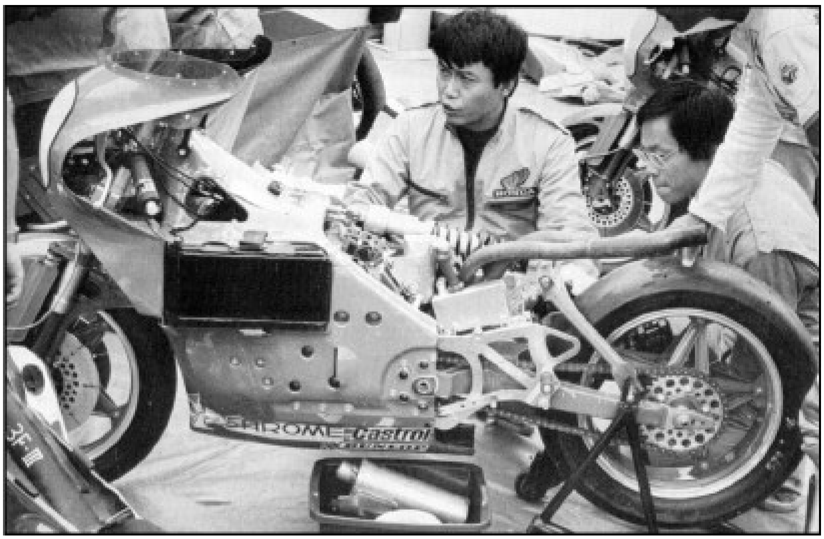


**On Santiago Herrero's highly successful 1960s 250 cc Ossa GP racer the welded aluminium chassis doubled as the fuel tank. This Spanish machine came within a hair's width of toppling the might of Honda for the world championship.** (MCW)

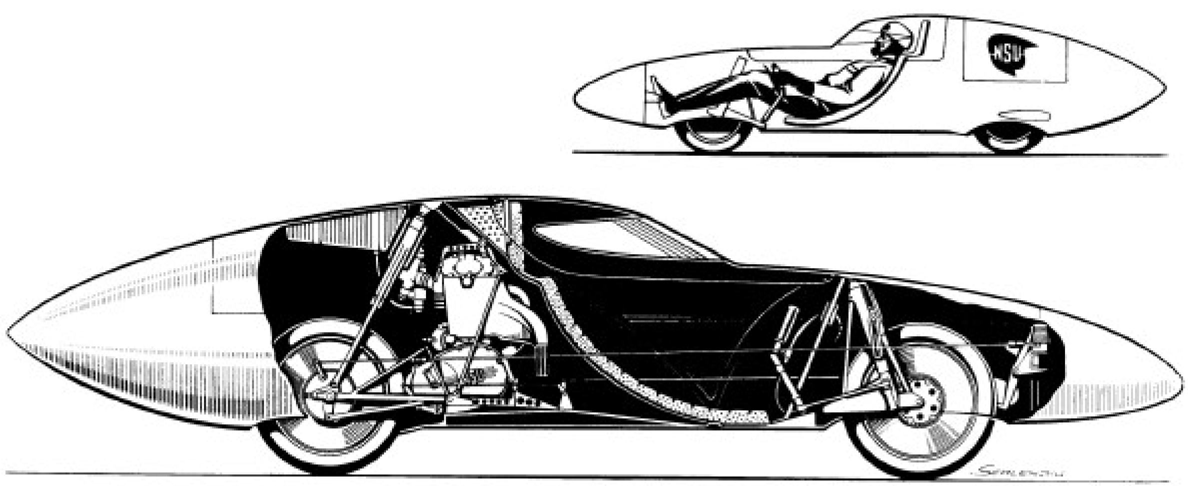
Other notable variations on the fabrication theme include the 250 cc Ossa grand-prix single of the late 1960s and the later Kawasaki KR500 square four. The Ossa had a complete welded-aluminium chassis that doubled as a fuel tank, while the Kawasaki had a 32-litre fuel tank as a backbone, with the steering head housed in the front and two inner and two outer aluminium side plates attached to the rear to support the fork pivot and engine.

## Monocoque

This term, often misused when applied to motorcycles, was originally coined to describe aircraft that used a skin of sheet aluminium as both the structure and smooth outer shape, it was later applied to cars employing a similar technique. A motorcycle, however, is much less amenable to this form of construction, because of its irregular shape (even with a fairing) and the need for several cut-outs. Many machines described as monocoques should more properly be said to have fabricated backbones. The original Honda NR500 racer was an exception, with the fairing an integral part of the bike. An unfortunate result, considering the frequent attention required in racing, was the fact that extensive dismantling was necessary for many routine maintenance tasks. (It is worth noting that the British Maxton concern was later commissioned to produce a more conventional frame for this machine.)



**Monocoque construction was originally used on the ill-fated NR500 V-four GP oval pistoned racer. One of the problems was inaccessibility for maintenance and a tubular frame from Maxton was substituted.**



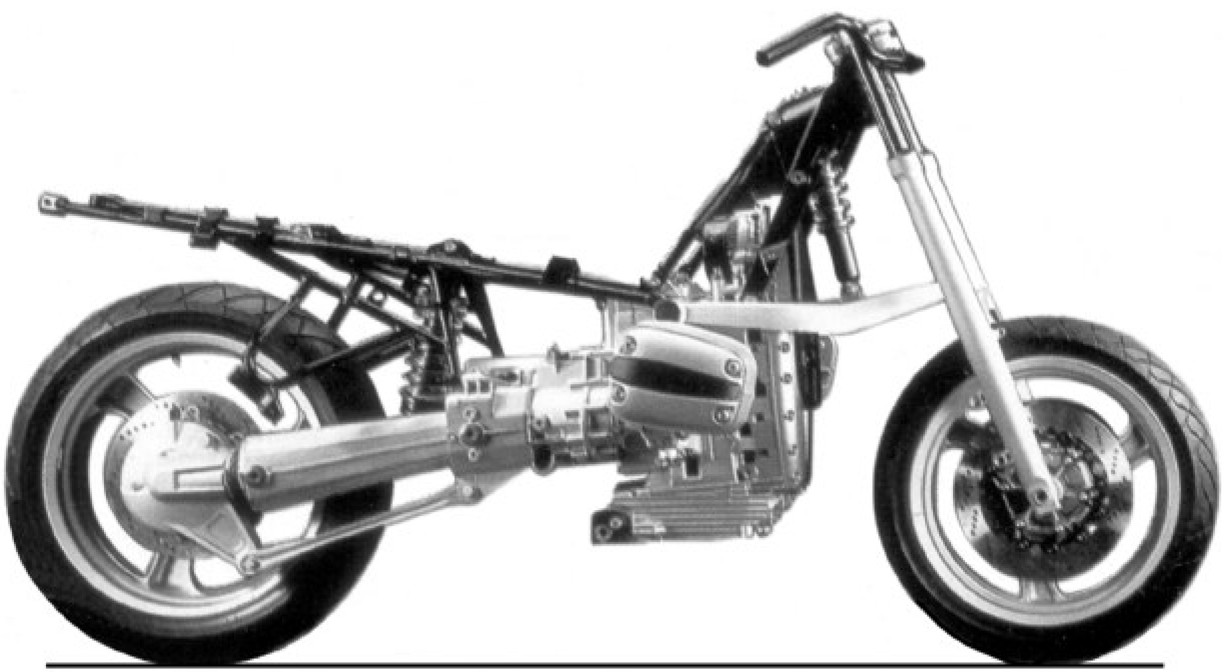
**The highly successful NSU "flying-hammock" record breaker of the mid-1950s had a true monocoque construction. The shell was in 1 mm thick aluminium alloy sheet, reinforced by bulkheads. Very similar to aircraft construction techniques.**

A true candidate for the term monocoque was the NSU 'flying hammock' record breaker of the early 1950s. In this, the primary structural strength was provided by the fully streamlined shell, hand-beaten in

1 mm-thick high-tensile aluminium alloy and reinforced, to take care of the points of application of chassis loads, by transverse channel-section members riveted in place.

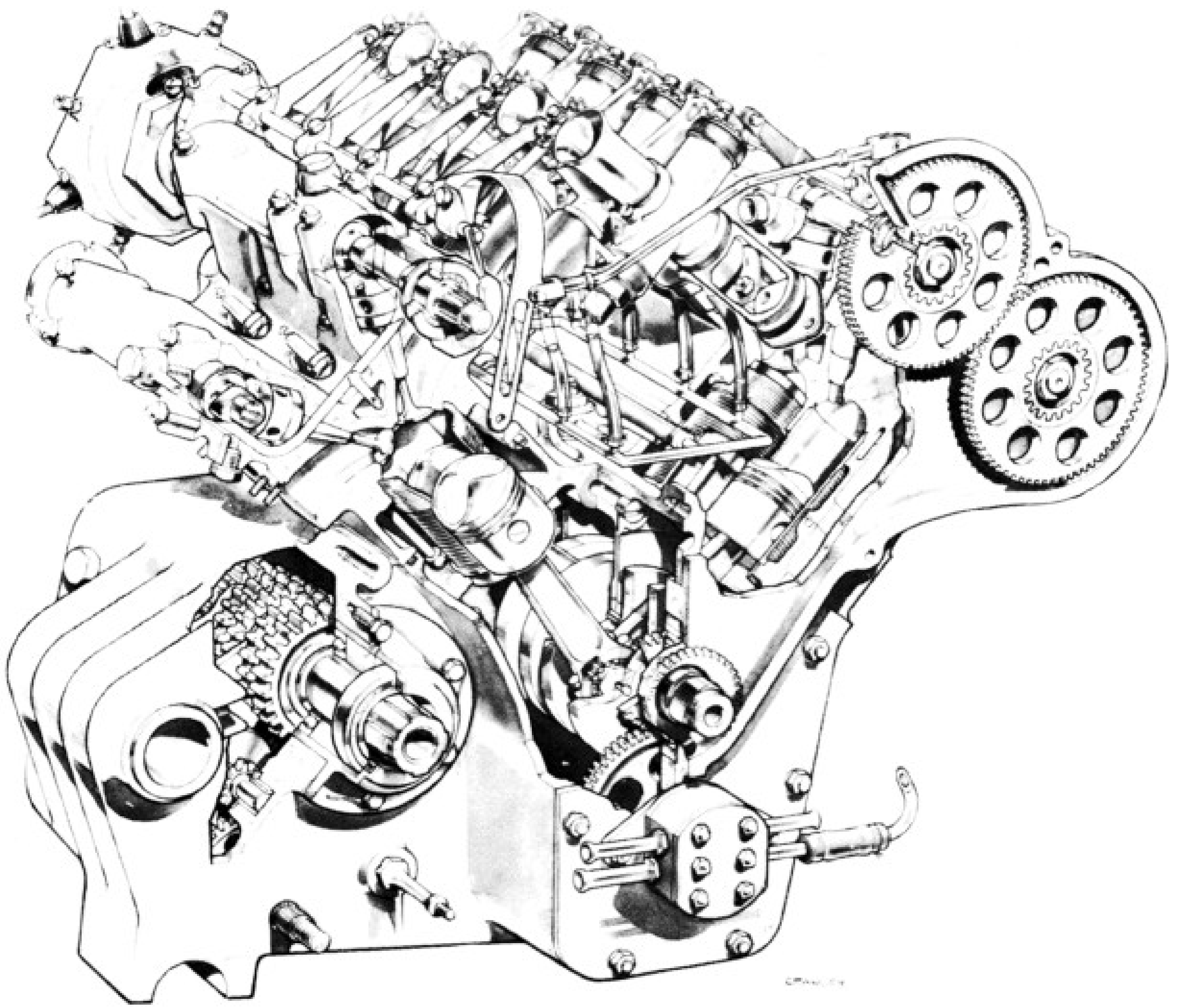
## Structural engine

Potentially, this is the most efficient way to build a bike with a large engine – and the post-war Vincent V-twin was an outstanding example. The principle here is to use the inherent stiffness of the engine-gearbox unit to provide the major support between the steering head and the rear-suspension pivot. If that pivot is incorporated in the rear of the gearbox casting, then a simple lightweight structure will usually suffice to join the steering head to the top of the engine. On the Vincent the rear-fork pivot was clamped between aluminium-alloy plates at the back of the gearbox, while both cylinder heads were attached to a fabricated backbone that doubled as an oil tank and incorporated the bolted-on steering head. More recently, the Norton-Cosworth parallel-twin racer demonstrated the potential of this method. The rear-fork pivot was cast in the back of the gearbox while the steering head was supported by a triangulated frame bolted to the cylinder head and a simple structure reached rearward to support the seat. Unfortunately, development was halted by the Norton company collapse.

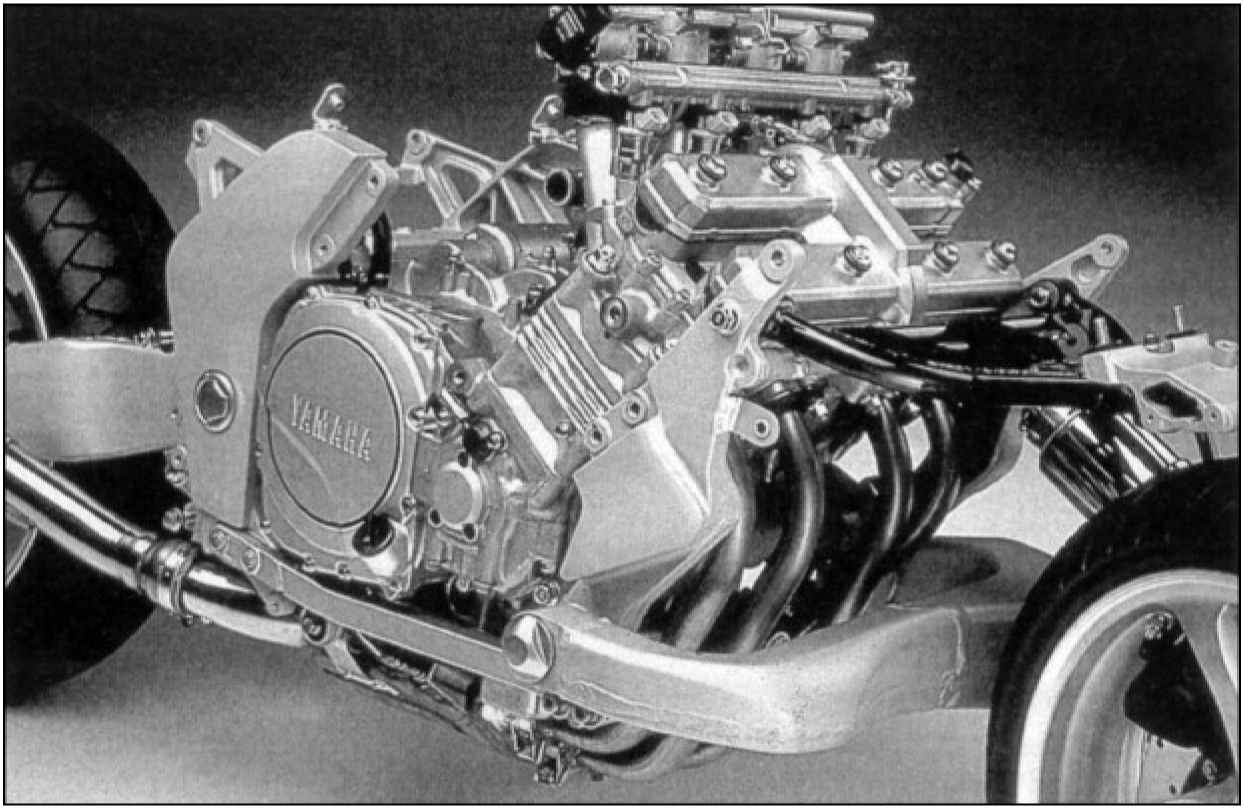


**Stripped down to the bare chassis this BMW boxer shows the simplicity that can be achieved by using the engine as the main structural member. The rear sub-frame to support the seat and rear suspension unit is the most complex part of the whole. The engine and gearbox unit form a very stiff and strong assembly quite capable of handling chassis loads.**

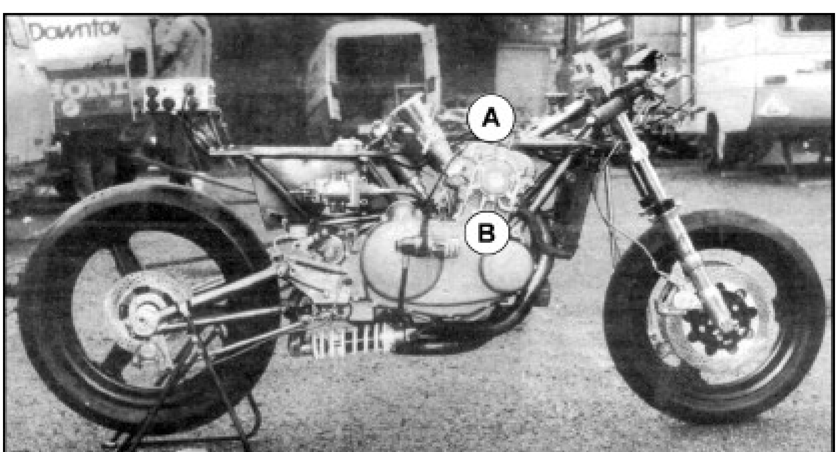
The current model BMW boxers are a good example of what can be achieved when the power train is designed from the ground up to act as the main structure also. The engine and gearbox form a rigid assembly largely bridging the gap between the wheels. At the rear the swing-arm mounts are completely integral with the gearbox castings, a construction method also used by a few past and present manufacturers of chain driven machines. At the front there is a simple structure to hold the dummy head-stock and the suspension "A" arm is attached directly to the engine. In some ways the use of the telelever front-end has made such a construction much easier as the loads fed back from the front suspension are spread over a wider area with lower local loads. Surprisingly, the most complicated part of the external structure is the subframe for mounting the seat and rear suspension unit.



**Stiffness was enhanced in the Moto Guzzi V8 racer and record breaker by incorporating the rear swing-arm pivot lug in the back of the gearbox casting.** (MCW)



**The Parker inspired front suspension necessitates a different type of frame structure for the Yamaha GTS. It is hard to classify this frame in any of the normal categories, the engine is structural but is not the principal structure. Aluminium castings at the front and rear support the suspension arms at each end. These castings are supported by a bolted on casting (not shown in this photo) that runs over the gearbox and to the side of the cylinders. A simple tubular frame extends from these casting up to support the top end of the steering column. Refer to chapter 7 for more details.**

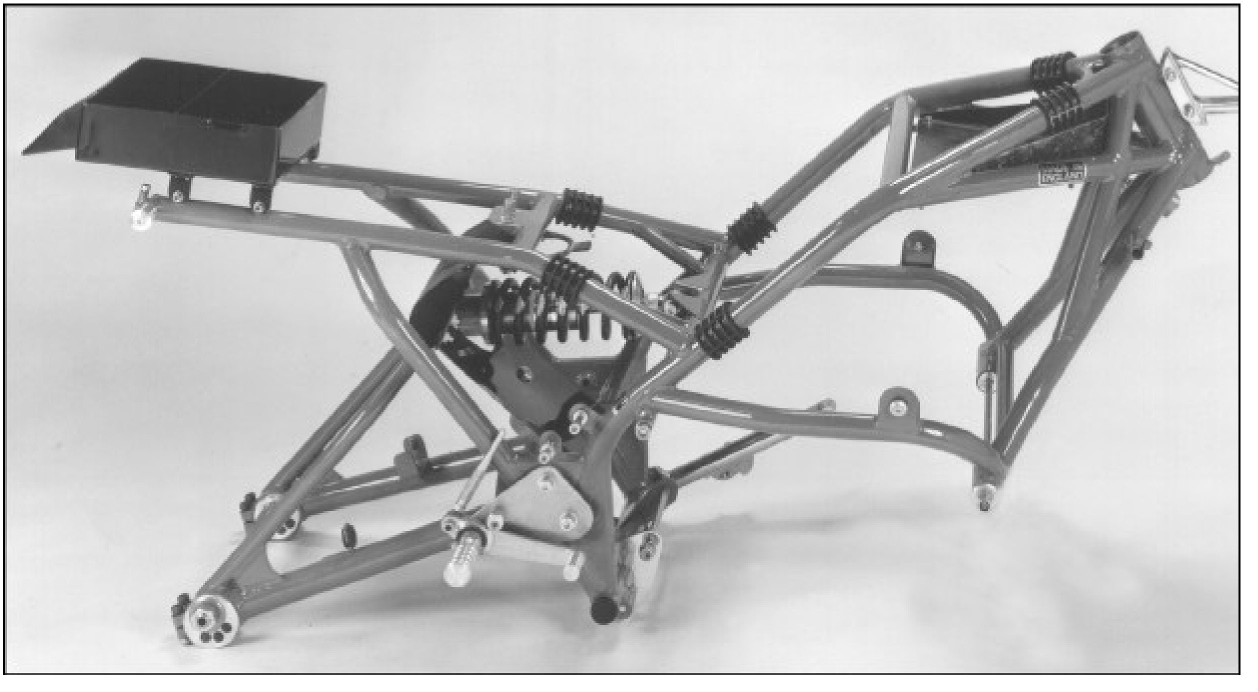



**The Norton Cosworth engine was designed from the beginning to be the main structural member and the remainder of the frame is very minimal. The head stock is supported by a simple triangulated structure bolted to the engine at points A & B. The swingarm pivots in the rear of the gearbox, this is shown more clearly in chapter 11.**

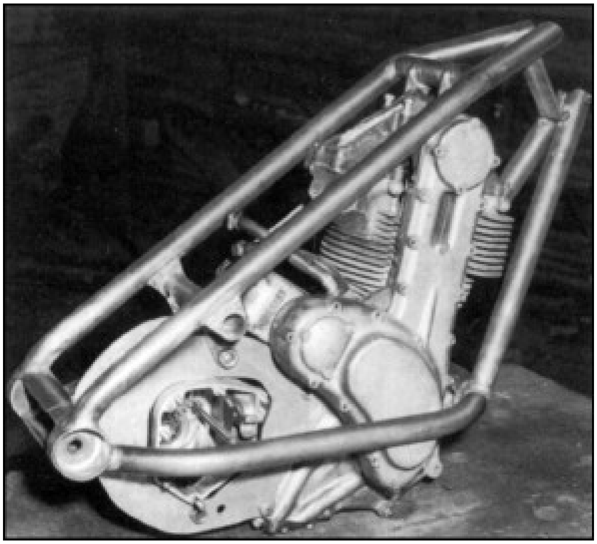
## Conventional multi-tubular

Although these were the most common prior to the advent of the twin beam type, these frames are potentially poor in terms of structural efficiency, their layout being determined primarily by availability of space and fashion considerations. They comprise medium-size tubes bent around the engine to connect the steering head to the rear-suspension pivot. The tube diameter is too small to gain much from the bending and torsional stiffness of the sections themselves, as in the case of a back-bone frame. Moreover, the layout is rarely such as to provide any significant triangulation. Indeed, many of these frames are relatively flexible and acceptable road behaviour is obtained only through the structural effect of bolting in a rigid engine. Despite these shortcomings, many machines with this type of chassis have achieved excellent handling (the featherbed Manx Norton being the most renowned), though only at the expense of weight, for a heavier frame is needed to give the required rigidity. In this context, it must be borne in mind that frame stiffness is not the only factor influencing handling. For example, when Norton substituted a racing parallel-twin Dominator engine for the single-cylinder Manx unit in 1961 the handling suffered slightly as a result of the changes to the distribution of mass.

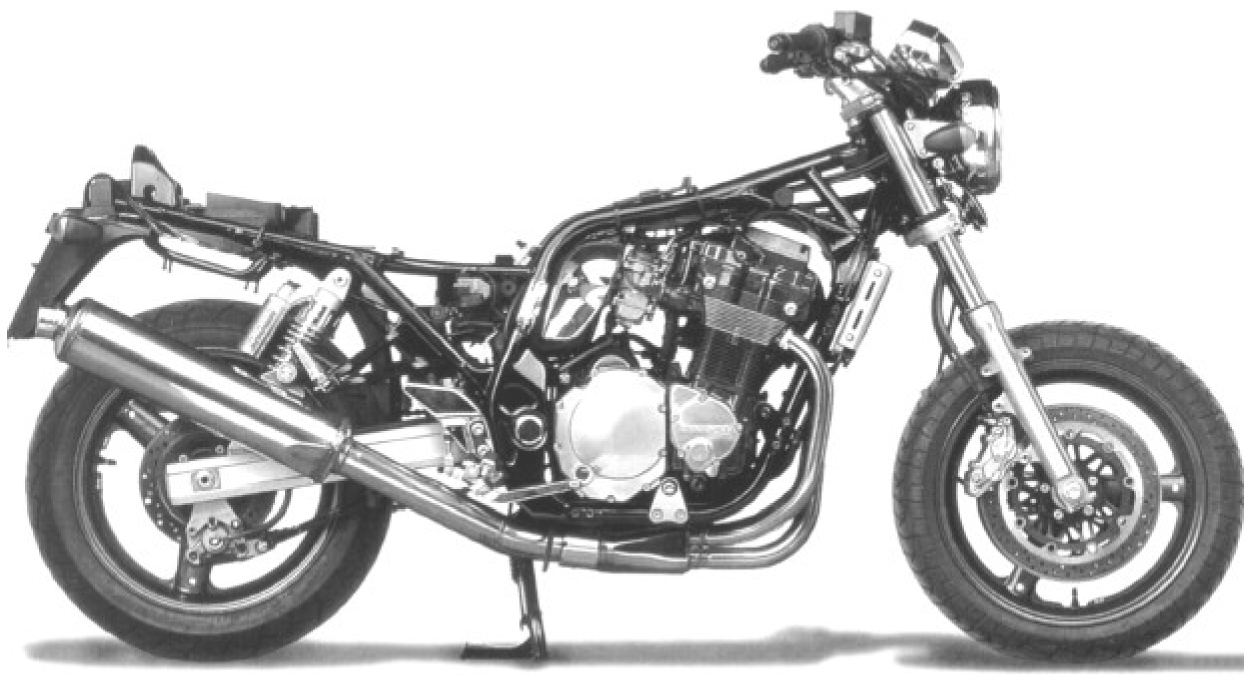
This frame type was traditionally made in welded steel tube and most current examples still are, but there was a period in the 1980s and early 1990s when some manufacturers used aluminium often in square or rectangular section, until the lower stressed twin spar became more common. This is not the best way to use aluminium tubing and this form of construction seemed to be marketing led design.



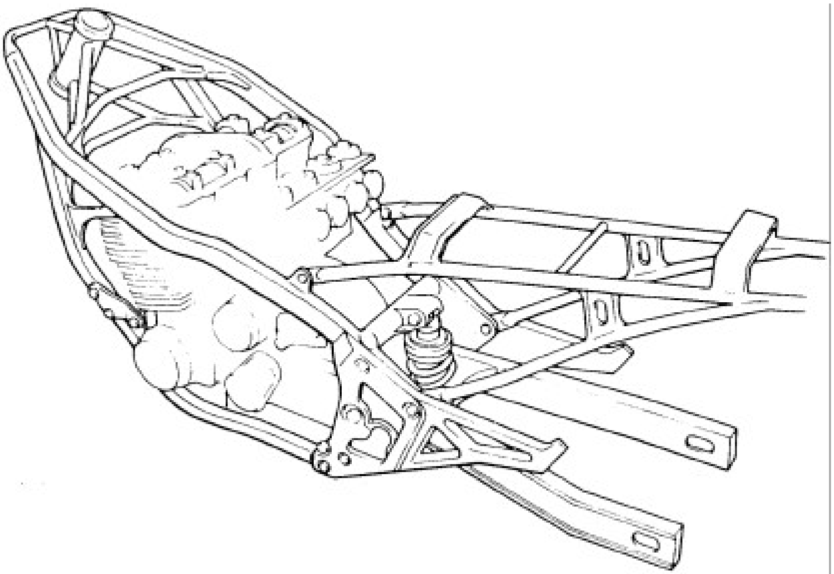
**With the engine installed, this Harris Magnum frame has a high degree of stiffness, helped by triangulation at the steering head, which is supported by eight tubes. The structure was formed with bronze welded 531 tubing.** (Parker)



**An interesting multi-tubular variant designed by the late Bob McIntyre for an AJS 7R and later used in the GPs by Jack Findley with a Matchless G50 engine. Engine and gearbox are low slung and well supported. This photo was taken before the head stock was welded in place. (MCW)**

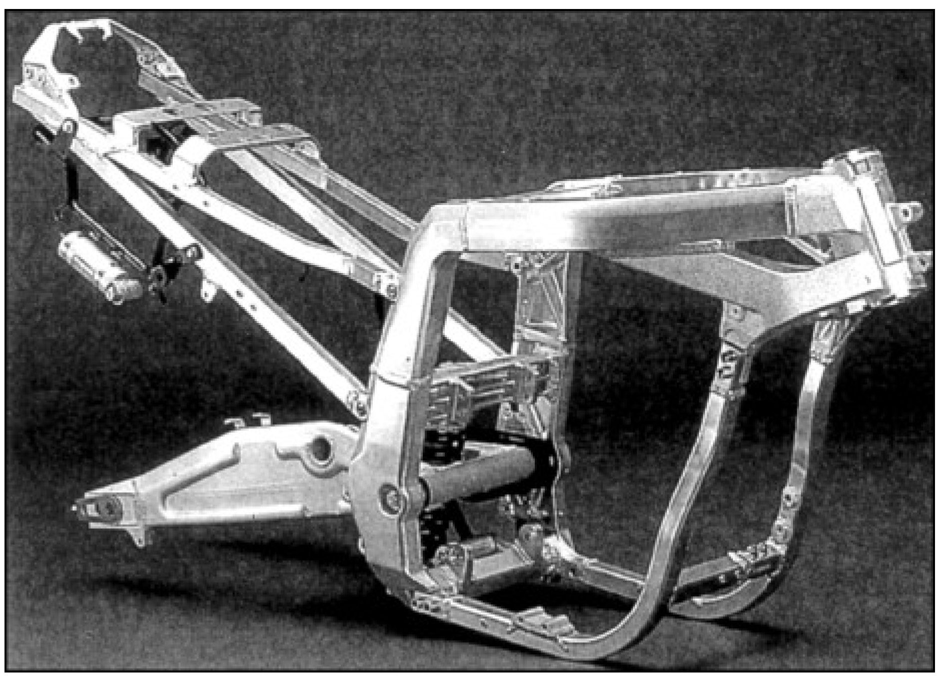


**Comparing this GSX1200 multi-tubular frame from 1999, with simple swing-arm and two rear suspension units, to the Hayabusa (pictured later) of the same period may lead one to believe that the designers at Suzuki were somewhat schizophrenic. In reality it just shows that frame designers are under fashion and image constraints as much as technical ones. In overall design this frame is little changed from those of twenty years before.**



**A 1984 Yamaha FJ1100 displays an unusual variant of the multi-tubular frame. It was called the "Lateral Frame Concept". The tubes surrounding the engine are kept wide at the front and the head-stock is supported in-side with multiple tubes. Some-what after the style set by the Bimota KB2 pictured earlier.**

**This Suzuki GSX-R 1100 from 1992 is typical of the application of aluminium to a relatively conventional frame. This is not the best way to use this material and appears to have been a stop gap until the twin spar took over. Note that the rear sections that hold the swing-arm pivot and just above are open section castings. The rear sub frame is bolted on as are the engine support tubes that pass under and in front of the engine. This enables engine removal.**

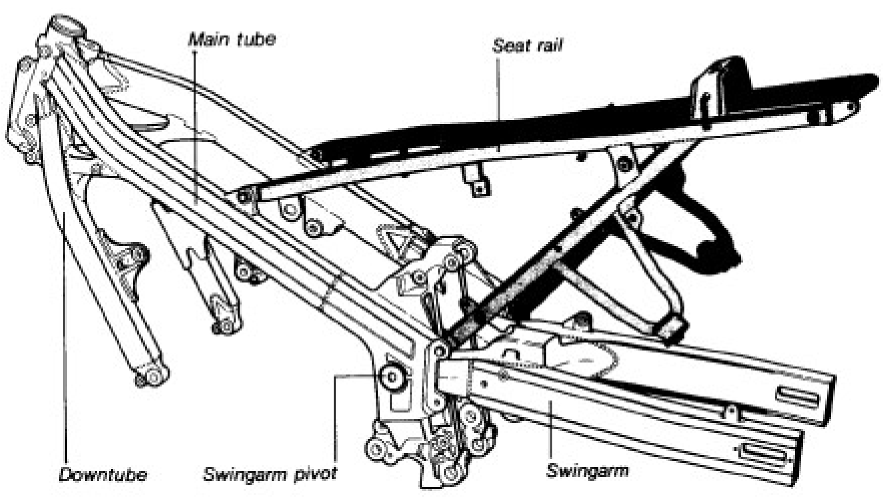


## Twin-spar

Spanish frame builder Antonio Cobas sparked a revolution when he first used what's become known as the twin spar frame, pictured in chapter 13. Virtually now universal for top of the range sports bikes and racers of all sizes, even many trials and other off-road bikes now use this form of construction. Mostly made in aluminium, it comprises of two beams running each side of the engine/gearbox unit, joining the head stock to the swing-arm pivot mountings. These side beams have generally got bigger with time to provide increased stiffness. Three different construction methods have been employed for the side members.

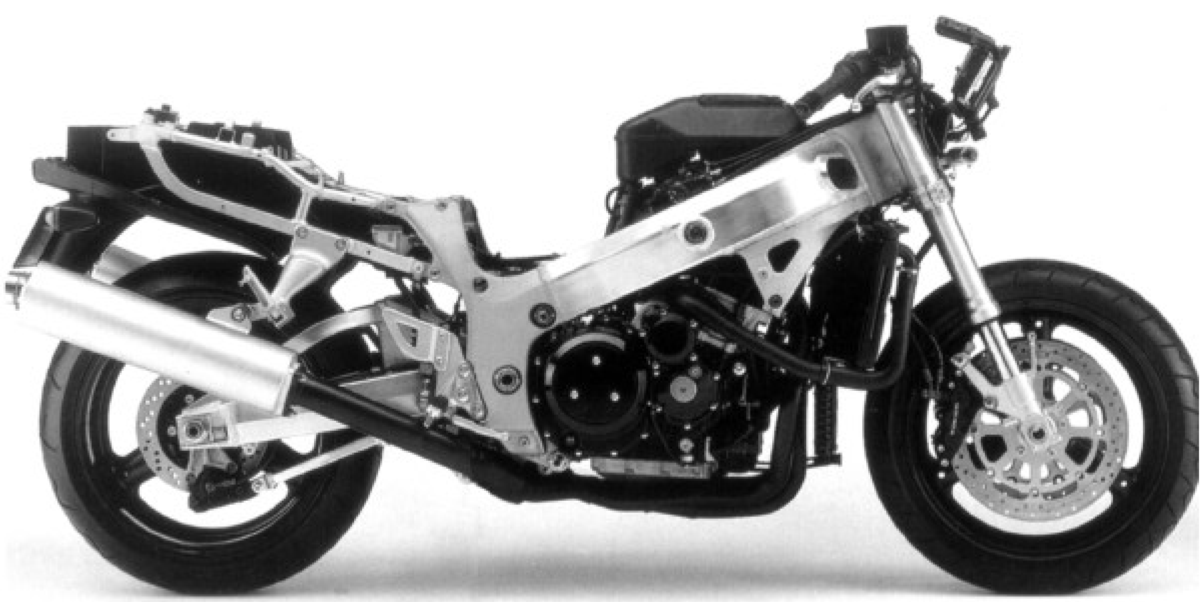
- Extruded tube, often with internal ribbing.
- Fabricated from sheet.
- Castings.

In all cases castings have often been used for the head stock area and the swing-arm and rear engine mounting plates. These castings are welded to the side spars. It is quite usual but not universal to have bolted on seat frames, which strangely have sometimes been made of steel tube.



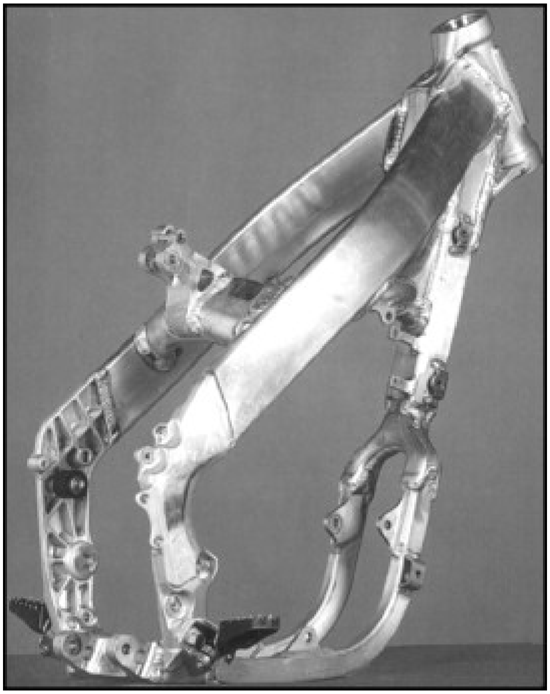

**An early example of a twin-spar frame is this Honda VFR from 1985. The main beams seem hardly sufficient to provide the required stiffness and no doubt the engine played an important role in enhancing this aspect. Note the bolted on seat sub-frame.**

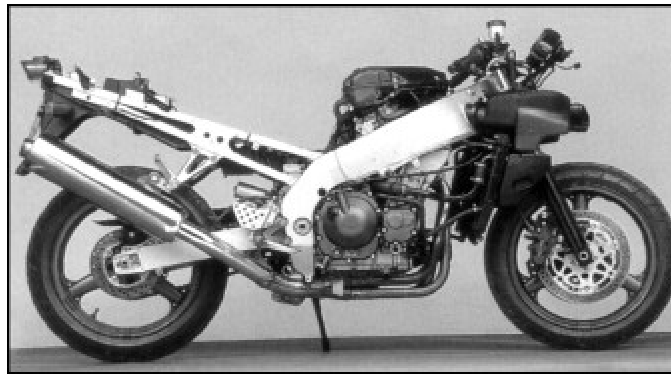
From a structural viewpoint the twin-spar frame could be considered as a bifurcated backbone or a boxed in version of the simple triangulated structure shown in fig. 10.4. Experience shows that given enough metal it can achieve sufficient stiffness, even for racing duty, but from a structural efficiency basis (stiffness to weight ratio) this type of frame is not particularly good. Even made in aluminium it is not a construction method that gives a particularly light frame, if made in steel as most other frames are, it would seem positively obese. However, historically very few production motorcycle chassis seem to have been designed with structural efficiency high on the list of priorities, and other considerations often take precedence. Success on the race track and in the showroom is of prime interest to manufacturers and from that aspect the twin-spar frame must be considered as successful as the multi-tube type was a few decades ago.



**Contemporary variations on the same theme. Above is the R1 from Yamaha and below is the Suzuki Hayabusa.**

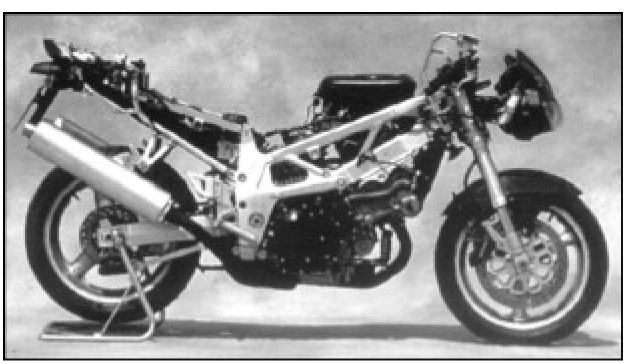
Modern large engines and their voluminous air-boxes present a huge challenge to the frame designer and the twin-spar offers considerable packaging advantages. Especially in the racing context this type of frame allows much easier access to work on the engine, in particular, removal and replacement of carburettors and access to spark plugs. The elimination of down-tubes and the lower cradle also frees up valuable space in the area needed by exhaust and cooling systems.



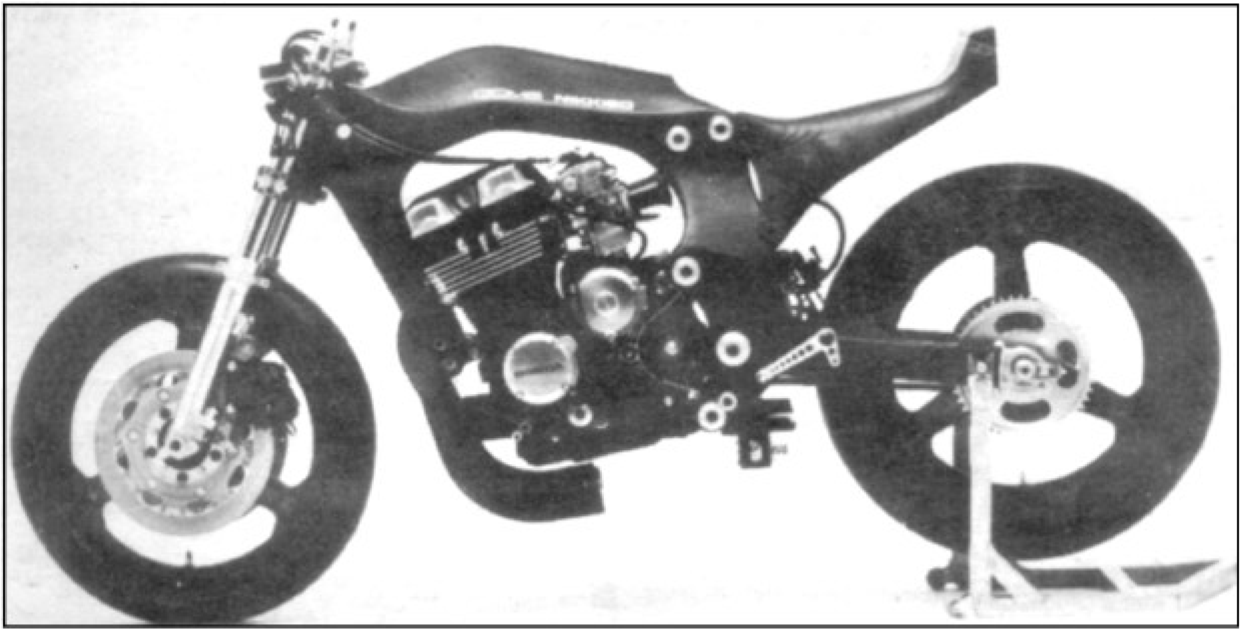


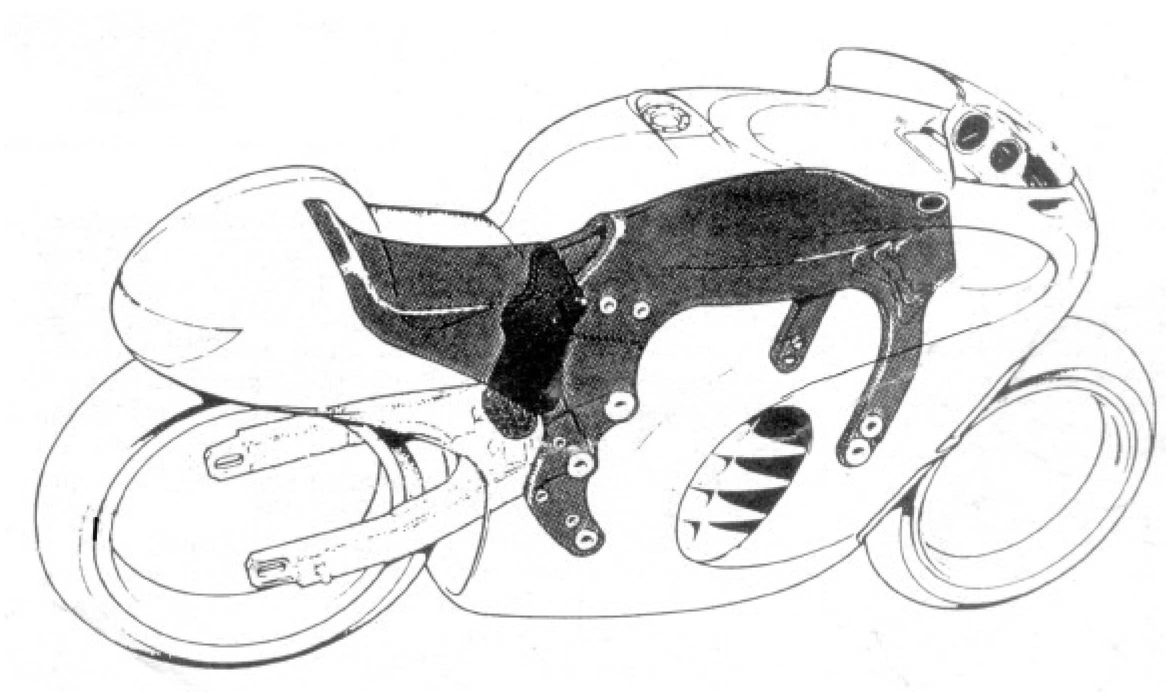
Above: **The 1998 Kawasaki Ninja used a twin-spar frame very little different from its main competitors shown above. Note that in common with the R1 and Hayabusa the front engine mounting is quite high up, rendering traditional down-tubes and bottom cradle unnecessary, thereby freeing up valuable space.**

Left: **The twin-spar is no longer reserved solely for pavement bikes as this shot of the frame for a Honda CR250 moto-X machine shows. Note the ribbed castings used for the swing-arm and rear engine mounting. The head stock is also cast and is joined to the rear by bent extruded rectangular tube sections.**



**Both the V-twin engined Suzuki TL1000s (left) and the Bimota DB4 have frames with a beam down each side of the engine and as such have similarities to the normal twin-spar. The principal difference being that the spars are formed as triangulated structures rather than box sections.**

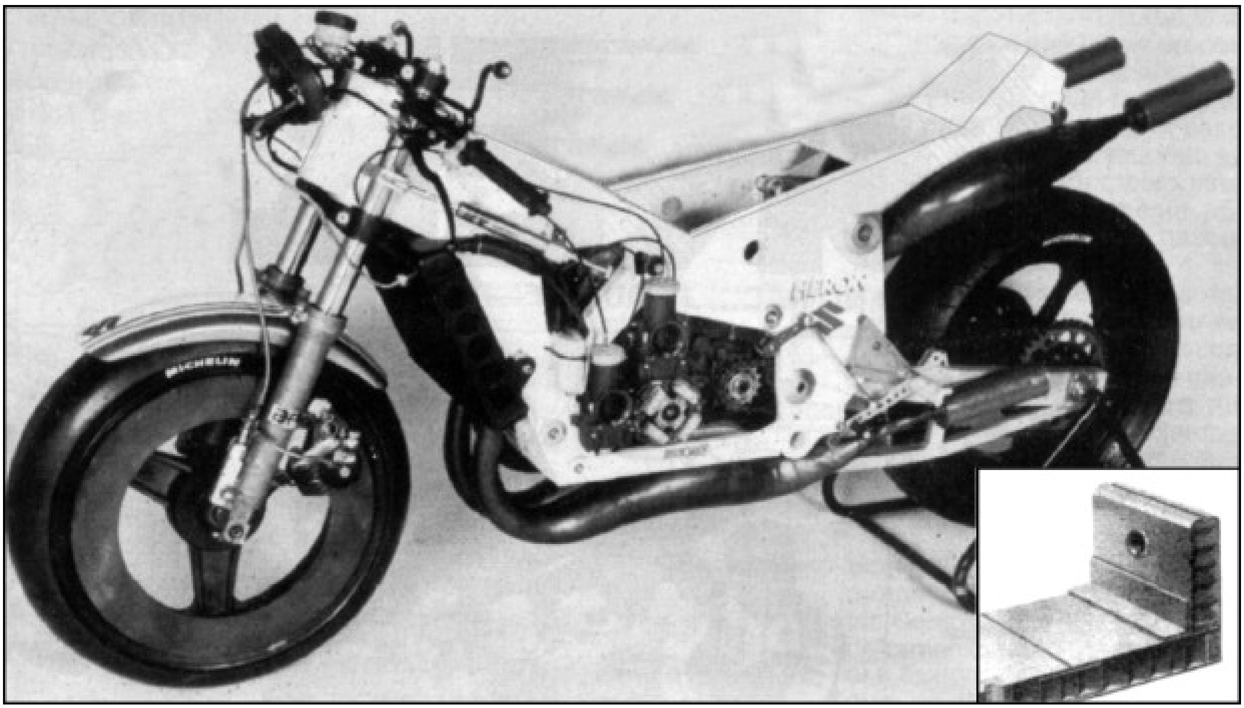




**Unusual carbon fibre frame made in Japan for the CBX750 Honda engine for endurance racing. The frame was reported to weigh less than 6 kgf. Called the Black Buffalo after the shape and colour of the frame. The backbone inspired design is very high above the engine and it would be very difficult to fit an airbox of the volume commonly used today, but perhaps the frame could double up as a substitute for that function.**

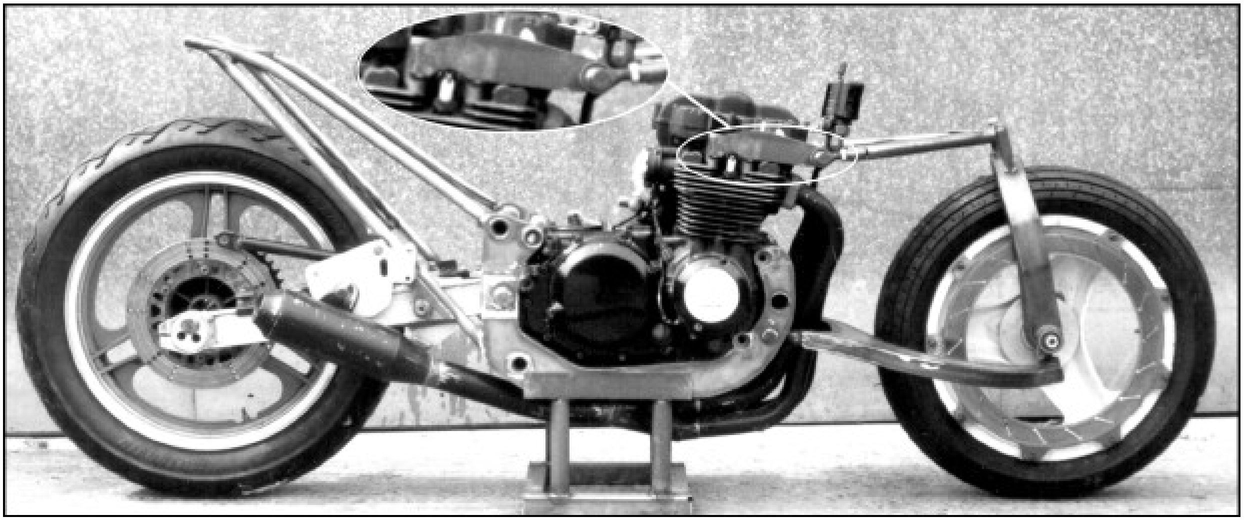
## Other types

Some frames don't fit into the descriptions above for a variety of reasons. The material used for construction dictates how the frame needs to be designed. This would be the case for example with any attempt to make a chassis from composite material such as carbon fibre. To use such material purely as a substitute for steel or aluminium in a similar design would be ill advised, each type of material needs a structural design to suit its own special properties. The Heron Suzuki pictured below is a good example. This was made from honeycomb material, as described in chapter 13, this material is not really suitable for bending, and so when this is necessary the inner sheet is cut away so that the bend only occurs on the outer sheet. To restore the strength of the inner sheet a folded strip is glued over the cut. This form of construction leads to the sharp edged boxy appearance shown.




**The frame for this GP racing 500cc. Heron Suzuki from the late 1970s is difficult to fit into any category. Sometimes called monocoque but it is hardly that. The boxy appearance is a result of being constructed from aluminium sided honeycomb material like that shown in fig. 13.3. The inset shows details of how this material is bent. Reportedly this chassis was very rigid and reduced lap times, but for whatever reason, development was not continued for very long. The front brake is of a rim type, but unlike the stainless steel example illustrated in chapter 12, this disk was made of carbon.**

As we've seen, with the telelever BMW and the Yamaha GTS, the use of alternative front suspension systems allows or demands that a different frame design be employed. The author did this with his range of Q2 machines, the Kawasaki 750 engined version below is one example. The double link suspension has completely different structural mounting requirements to that of a normal head-stock frame. This chassis is extremely simple and features two "U" shaped box section fabrications to house the engine and the lower suspension link fixing as well as the rear swingarm pivot. Three cross tubes are welded between the two U pieces and the assembly is quite stiff, which is further enhanced with the engine installed. The upper front suspension link is relatively lightly loaded and is satisfactorily located by a pair of connected brackets attached to the head (see chapter 11 for details on a similar attachment). A simple tubular super-structure over the engine (not shown) is all that is necessary to hold the steering mechanism and suspension unit mounting, thus completing the chassis.



**This chassis for a Q2 was designed by the author to handle the totally different structural requirements of a machine fitted with a form of double link front suspension. The enlargement shows more detail of the upper suspension link mounting.**

## Summary

The optimum chassis design depends on the size and shape of the engine and the intended purpose of the machine. While a small tubular or pressed-steel backbone suits a moped for both structural and economic reasons, large-capacity machines may best be designed with the engine as the main structural member. Despite potential cost and structural benefits this form of construction has not found many adherents amongst the main manufacturers. This method is rarely open to the specialist or low-volume producer either, as the engine should really be specially designed for such use. Where the engine is flat or compact, triangulation offers good structural efficiency, as does the simpler back-bone frame. Both of these approaches lend themselves to specials builders.

Tradition has probably ensured the survival of the multi-tubular frame for some time yet, perhaps with an increased use of steel pressings. In its favour is its ready adaptability to various engine sizes and styles. But it is difficult to mass-produce cheaply and that factor, rather than any technical one, may eventually woo manufacturers away from it to the structural-engine concept. It is difficult to visualize much application for genuine monocoque structures.

With very few exceptions the current frame of choice for racing and sports machines, over the past decade or two, has been the aluminium twin spar in one form or another. This form gives a lot of freedom for installation of the large air-boxes now common, and gives good access to the top of across the frame engines. Many examples do not provide sufficient stiffness on their own and rely heavily on the engine to make up the shortfall.

# 11 Engine mounting

Simple though it may seem, the method of mounting the engine in the chassis calls for careful consideration if annoying, or even destructive, vibration is to be avoided. The basic requirement is to support the engine in such a way that all its loads are adequately dealt with. In many cases the engine gearbox unit is a vital component in providing overall chassis stiffness and due attention must be paid to this aspect. The main loads to be handled are of three sorts:

- Engine weight and inertia loads due to road shocks.
- Unbalanced reciprocating forces in the engine.
- Final-drive chain tension, which can be extremely high in a large-capacity machine in a low gear. On shaft drive machines the transmission loads are manifest as a torque trying to twist the engine unit in roll.

In the traditional old singles and 360-degree vertical twins (pistons in step) vibration was a serious problem, exacerbated in many cases by relatively flexible frames that could very well vibrate in sympathy with the engine. If a frame resonance coincided with a particular engine speed, then the vibration transmitted to both rider and structure would be magnified. A common method of reducing this vibration (which, in a slowly rotating big single, was aggravated by pulsating torque reaction) was to fit a cylinder-head steady (i.e. a stay or bracket bracing the head to the frame). This worked by using the bulk of the engine to stiffen the frame, so raising its resonant frequency and reducing the magnitude of the vibration.

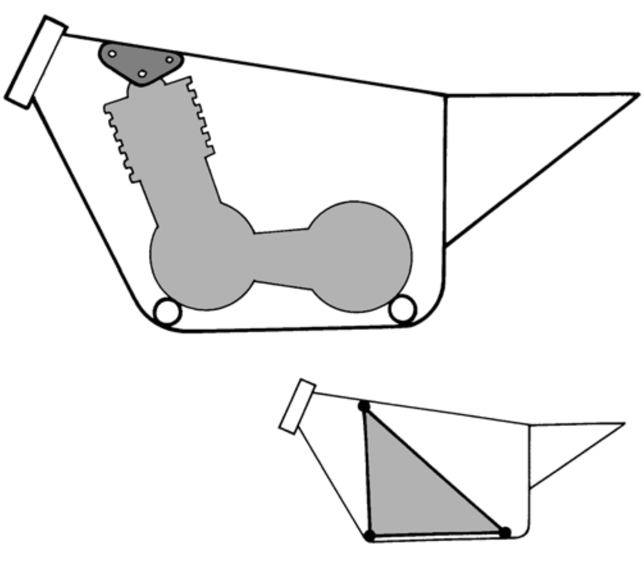


Fig. 11.1 **A cylinder-head steady as shown (top) effectively triangulates and stiffens the frame (bottom) raising its natural frequency.**

In some cases, however, fitting a head steady would make the problem worse, this would be due to feeding vibration from the engine into an area of the frame that was not capable of accepting it.

Another important influence on the level of vibration, especially in a big single or parallel twin, is the engine's balance factor – i.e. the proportion of the reciprocating mass that is counterbalanced in the flywheel assembly.

This factor determines both the direction and magnitude of the *primary* out-of-balance forces. With a zero balance factor these high forces act in line with the cylinder axis. By using a 100 per cent balance factor, the direction of the forces is turned through 90 degrees (i.e. fore and aft for a vertical cylinder). For intermediate balance factors, the forces can be reduced in magnitude while their direction varies between the two extremes. At first sight, it seems best to use a balance factor of 50 per cent and so minimize the out-of-balance force, in whichever plane – as, for example, did AJS and Matchless in their 500 cc parallel twins.

In practice, however, frames differ in their stiffness in various directions and the best balance factor can be determined only by experiment. At a time when proprietary engines (such as Blackburne, JAP and Python) were installed in different makers' frames, it was normal to alter the factor to suit each particular frame. Indeed, adapting any make of engine to a frame for which it was not intended became a specialized art. For various reasons, the problem of vibration has diminished with time. In the first place, large singles nowadays have contra-rotating balance shafts to oppose the out-of-balance forces anyway, other single-cylinder engines are mainly confined to small-capacity machines and are predominantly of the two-stroke type, which have lighter reciprocating masses. Even some of these small engines now have balance shafts.

Secondly, some parallel-twin four-strokes adopted 180-degree spacing of the crankpins (as in a two-stroke) so cancelling the primary inertia forces, albeit at the cost of a rocking couple, others kept the pistons in step (360 degree cranks) while incorporating balance shafts. Finally, engines with more than two cylinders are inherently smoother anyway. Nevertheless, some high-revving fours, particularly two-strokes, still generate enough high-frequency vibration to fracture components. Rubber mounting the engine can help here, although this solution may be anything but straightforward. For example, the 125 cc. Suzuki grand-prix twins of the 1960s were said to have shattered the insulators of their sparking plugs as soon as the engines were rubber mounted.

Clearly, the simplest form of flexible mounting is to fit bonded-rubber bushes in the engine's attachment lugs. With chain final drive, however, there is the disadvantage that the large forces, in the chain, pull the engine back in the bushes, so reducing their effectiveness and increasing chain slack. On the other hand this builds some cushioning into the drive train. With their Isolastic system, Norton skirted round the chain-pull problem in the Commando by coupling the engine, gearbox and rear fork rigidly together and rubber mounting the whole assembly in the frame. Thus the rear suspension was flexibly joined to the front. To minimize lateral compliance, a shim adjustment was incorporated, and provided this was correctly set, handling was said to be satisfactory. However, the chore of periodically making the adjustment was often neglected to the detriment of handling.

A better solution was James Love's patented Vibratek system developed in the late 70s. and demonstrated with a Triumph Bonneville application. The idea was to reduce the engine's balance factor to zero (so that the primary inertia forces were confined to the vertical), then pivot the engine at a precisely calculated point well to the rear, allowing a tiny vertical movement between rubber stops at the front. Because the main unbalanced forces were vertical, little of these were transferred to the frame via the pivot, the engine unit tending only to rotate about this point. The location of the pivot point was crucial and depended chiefly on the relative positions of the engine's centre of gravity and its crankshaft.

This point is called the centre of gyration and is the point about which the engine assembly would tend to rotate if a sudden vertical force was applied at the crankshaft axis. No vertical force is transmitted to the centre of gyration and so is the perfect point to mount from.

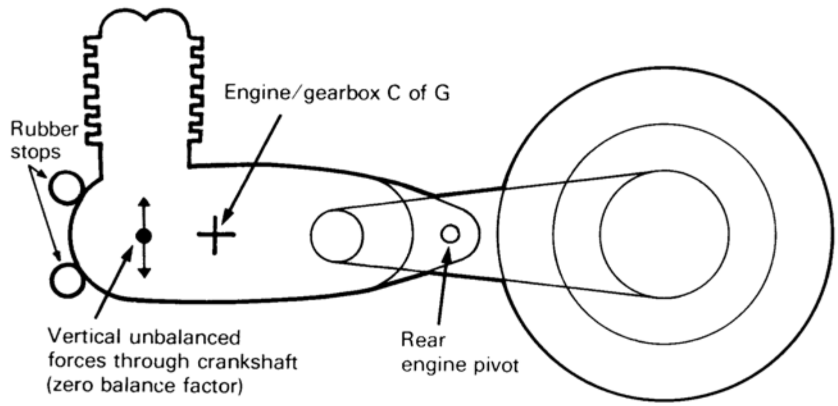



Fig 11.2 **Basic mounting principle behind the Vibratek system.**

Unfortunately, in the Triumph case the required pivot point proved to be within the rear-wheel area, which was clearly impractical. However, since the engine's angular movement about the pivot was extremely small, virtually the same movement was obtained by attaching the engine to the frame by short links top and bottom, so arranged that if they were extended rearward they would meet at the theoretical pivot point, sometimes called a virtual pivot. Tests on a rolling road indicated an enormous reduction in the vibration felt by the rider, as in fig. 11.4.

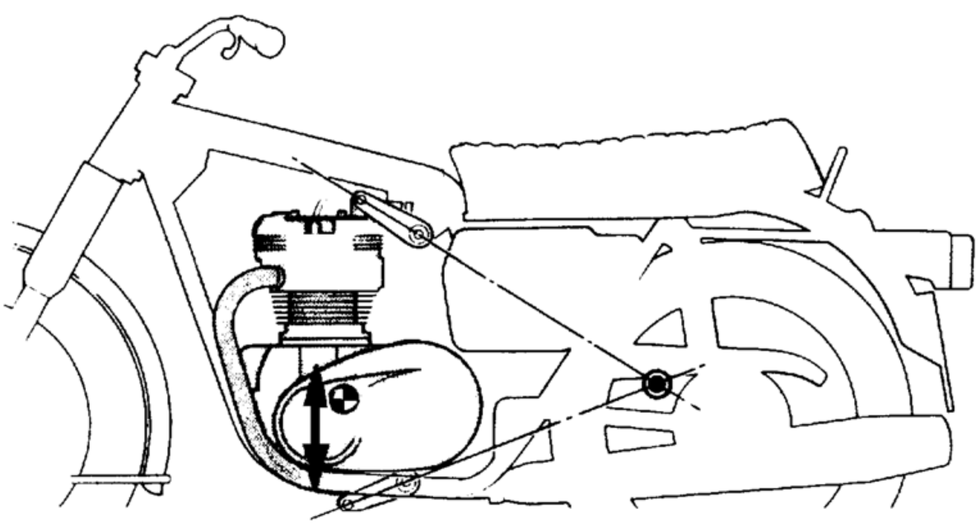


Fig 11.3 **This shows the actual method of execution.**

**Mounting by two links creates a virtual pivot at the centre of gyration, which in this case is within the rear wheel area.**

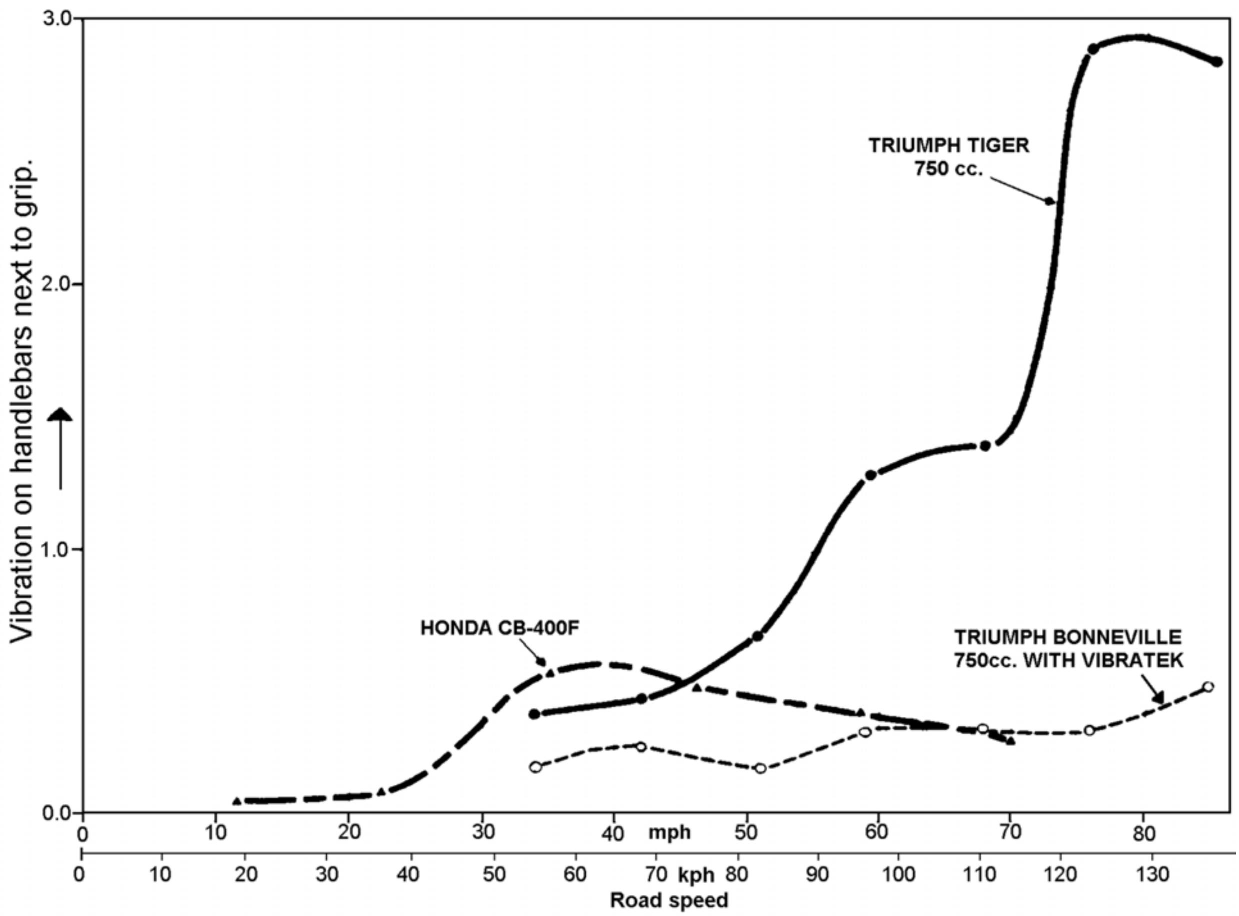



Fig 11.4 **Vibrations induced in the handlebars of a modified Bonneville are on average 25% of those in the handlebars of a Triumph Tiger of the same capacity, and over much of the range less than those of a 4 cylinder machine of just over half of the capacity.**

A somewhat similar system has been used for many years on 250 cc MZ roadsters and, though not perfect, worked quite well. As fig. 11.5 shows, the engine was pivoted on nylon bushes on the rear-fork spindle and rubber mounted at the back of the cylinder head. The engine's balance factor is not known to the author but the positioning of the top rubber mounting results in a fore-and-aft component of the inertia forces being fed into the frame through the pivot and rubber mounting bushes at lower engine speeds, although most of the loads at higher rpm would be reacted against the engine mass through its centre of gravity if the balance factor was zero.

Naturally, a rubber-mounted engine cannot be used in a structural way. In most multi-tubular frames a rigidly mounted engine stiffens the chassis considerably, hence the induced forces must be taken into account when designing the frame brackets and engine lugs involved. Where a chassis is sufficiently rigid on its own account (e.g. a good triangulated, backbone or twin spar design) it is best not to attach the engine at too many points. At the front, no more than one or two bolted-on tension struts may be needed to support the weight of the engine, while the chain pull may be taken through the rear engine mountings, preferably integrated with the rear-suspension pivot.

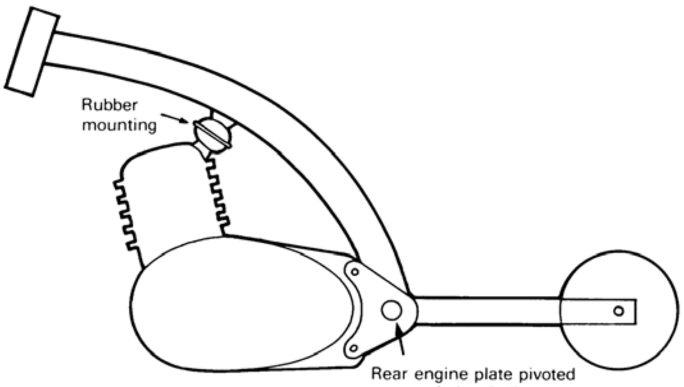



Fig 11.5 **A similar approach has been used by MZ, shown here are the main features. Not as sophisticated as the Vibratek system it proved quite adequate for a 250cc two stroke motor.**

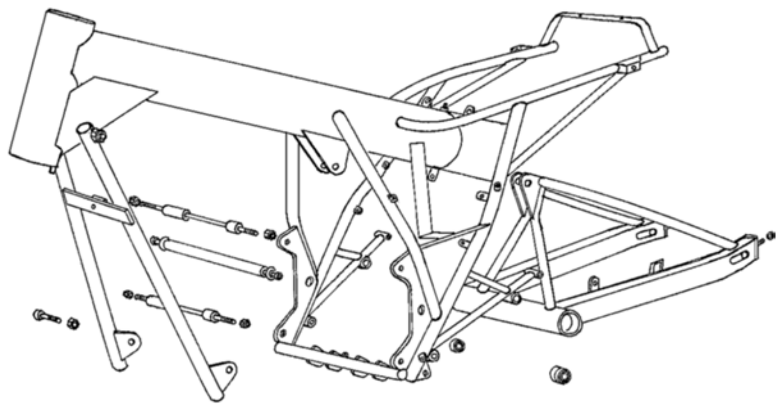


Fig 11.6 **Designed to accommodate a variety of tall transverse fours, this Foale backbone frame has two tension struts at the front to support engine weight. These members are bolted at the top to the main frame, and so can easily be swung out of the way during assembly and disassembly.**

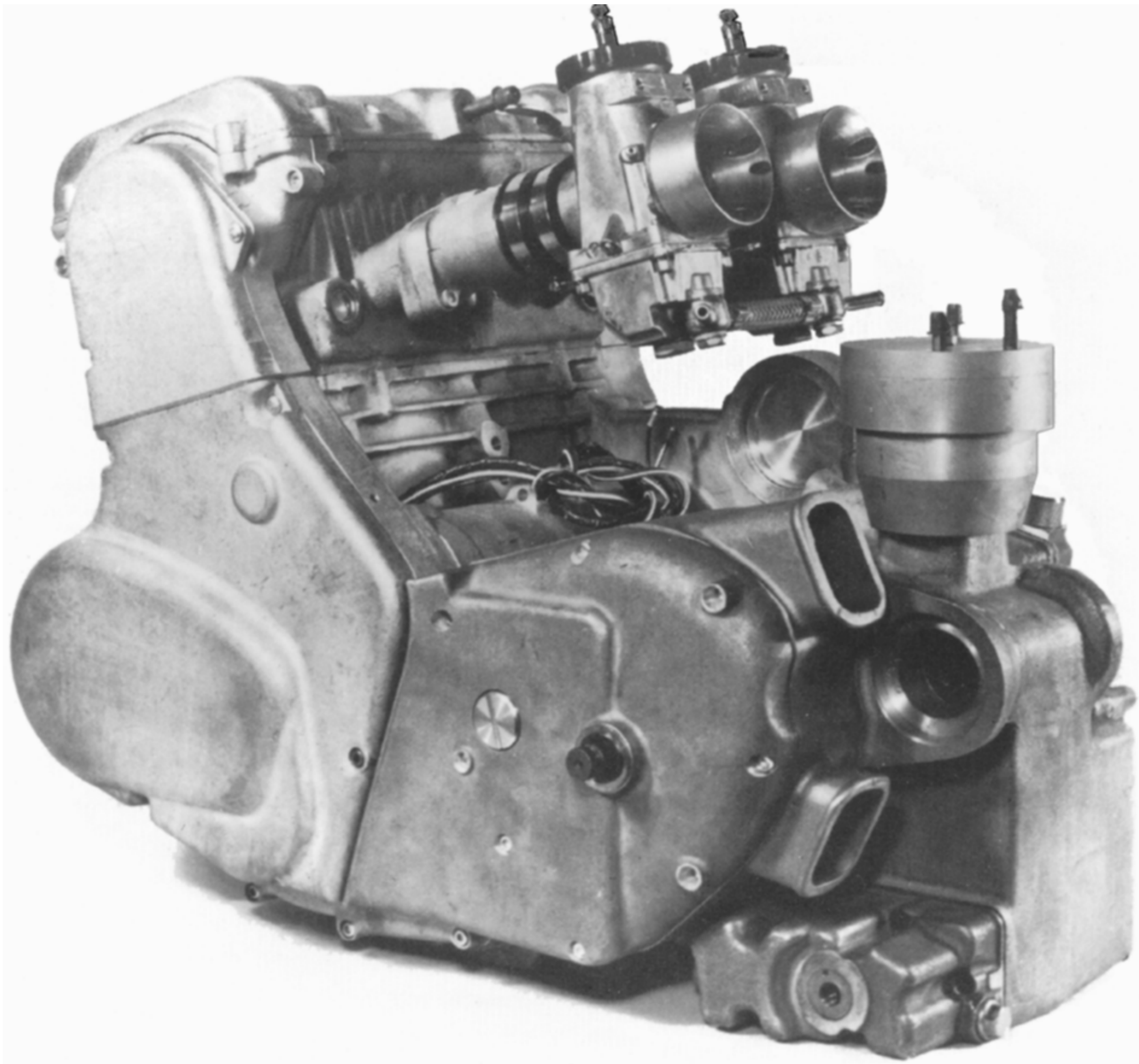
**The rear gusset plates provide mountings for the rear of the engine and the rear fork pivot.** (Sansum)

Although not commonly used, such detachable tension stays may have several benefits which include:

- easy installation and removal of the engine.
- elimination of bending moments at the upper end of the stays reduces the chance of fatigue failure.
- ease of accommodating dimensional tolerances in manufacture.

The author has used this form of front engine support quite successfully on a wide variety of designs. These include large capacity fours as shown above with solid engine mountings to small twin cylinder two strokes with rubber mounted motors.

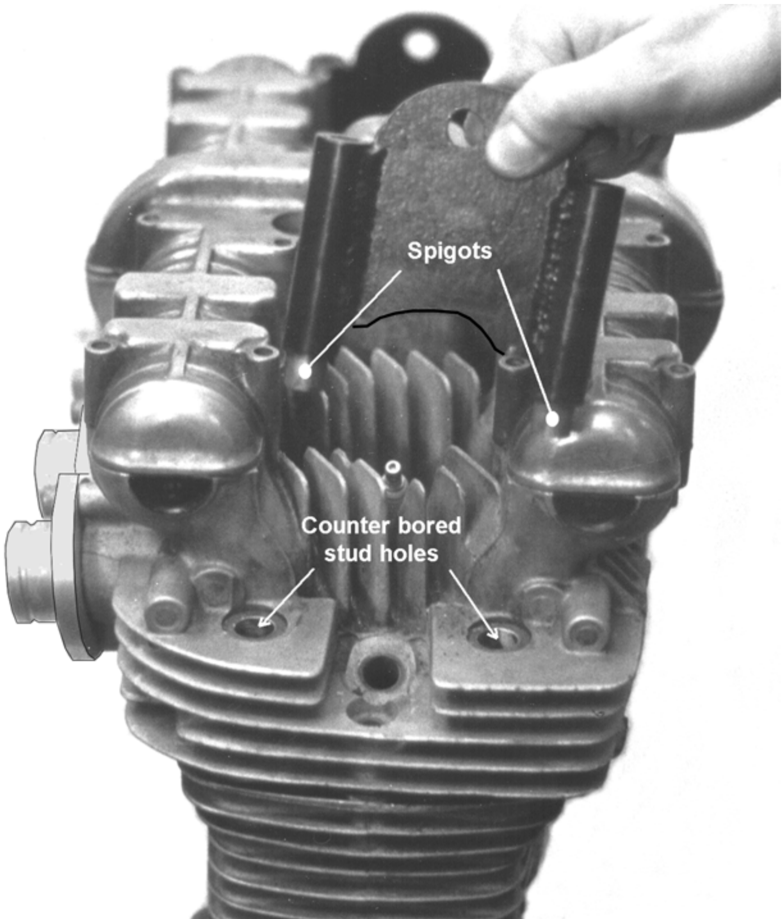
If the engine is to serve as the main structural member, the rear-suspension pivot is best incorporated in the crankcase/gearbox castings. It then remains only to build a simple structure at the front to support the steering head, leaving a minimum need for additional framework elsewhere to support the tank, seat and rear suspension. The Norton Cosworth Challenge was designed in this way.



**In the Norton Cosworth Challenge parallel twin the engine was the main structural chassis member, with the rear-fork pivot cast in the gearbox casing. A simple structure atop the motor to carry the steering head was all that was necessary to complete the main parts. See chapter 10 for an illustration of the frame.**

Naturally, this scheme is feasible only if the engine has been suitably designed in the first place. Many of the large multi-cylinder engines that are sufficiently substantial lack suitable mounting lugs. In these cases, the rear-suspension pivot may be housed in plates rigidly bolted to the rear engine-mounting lugs, while through bolts from crankcase to cylinder head may sometimes be lengthened to provide a suitable attachment for the front structure. Provided the engine is inherently fairly smooth (e.g. a good four or V-twin) vibration is no great additional problem when it is used as the main structure. Since the additional substructures are usually short and rigid, their resonant frequencies are often beyond the range of engine rpm.

Engines not specifically designed to be used as the main frame member can often be adapted for such duty. There is often good potential to provide a rigid structure and lighter all-up weight, but care must be taken with the details in order to ensure connections that don't work loose in use, hence negating any structural gain. During the 70s and 80s the Kawasaki Z900/1000 range was a popular donor of engines for those interested in a chassis change. The following picture shows how the author provided a strong and slop free frame mounting on the cylinder head.



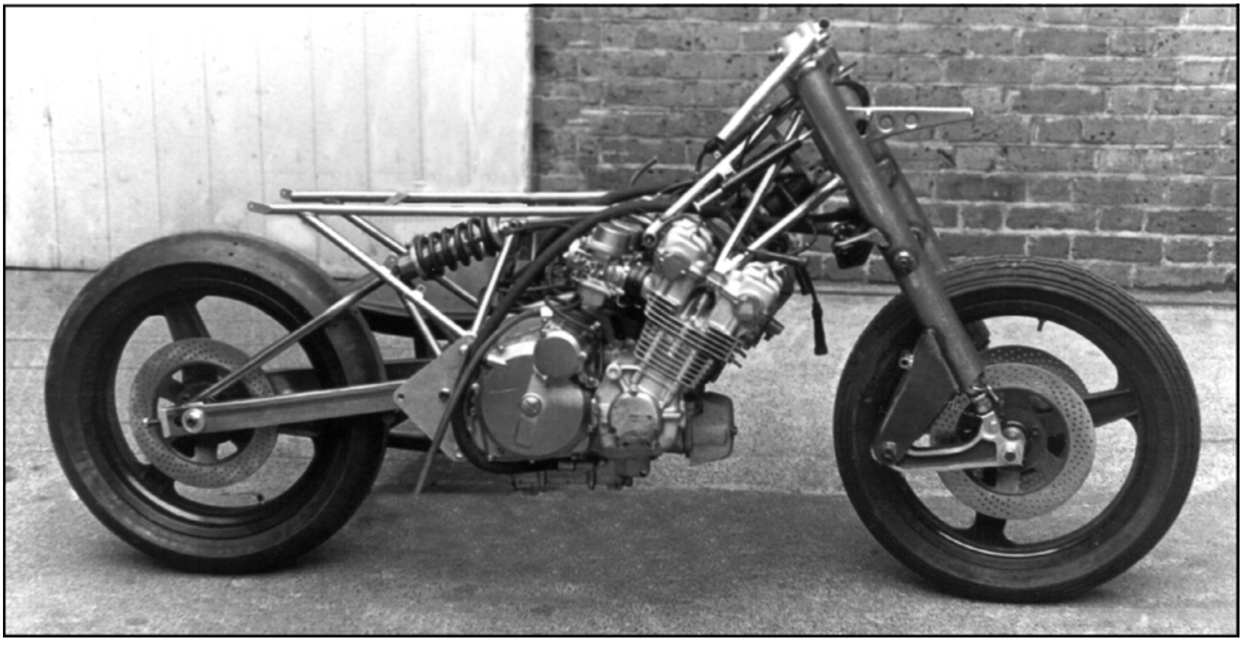

**To prepare the cylinder head of this Kawasaki Z1R for accepting the mounting a sub-frame, the normal stud holes were counter bored to accept the spigots on the mounting bracket. The spigots were a slight interference fit in the holes to ensure no fretting in use.**

**After being pressed on to the head, the brackets were considered to be part of the head and did not have to be removed for any service work.**

**Elongated head studs were made up to allow for the extra length needed to go through to the top of the two brackets.**

The new head mounting points and the normal front cast mounting lug on the crankcase provided a strong and widely spaced base on which to bolt the front subframe to support the head stock. At the rear, a long swing-arm was used which pivoted concentrically with the gearbox drive sprocket. On the right hand side a new clutch cover was cast incorporating a pivot bearing for the swing-arm. There was another special casting made for the right hand side to carry the other bearing. The result of this was a very light but extremely rigid chassis.

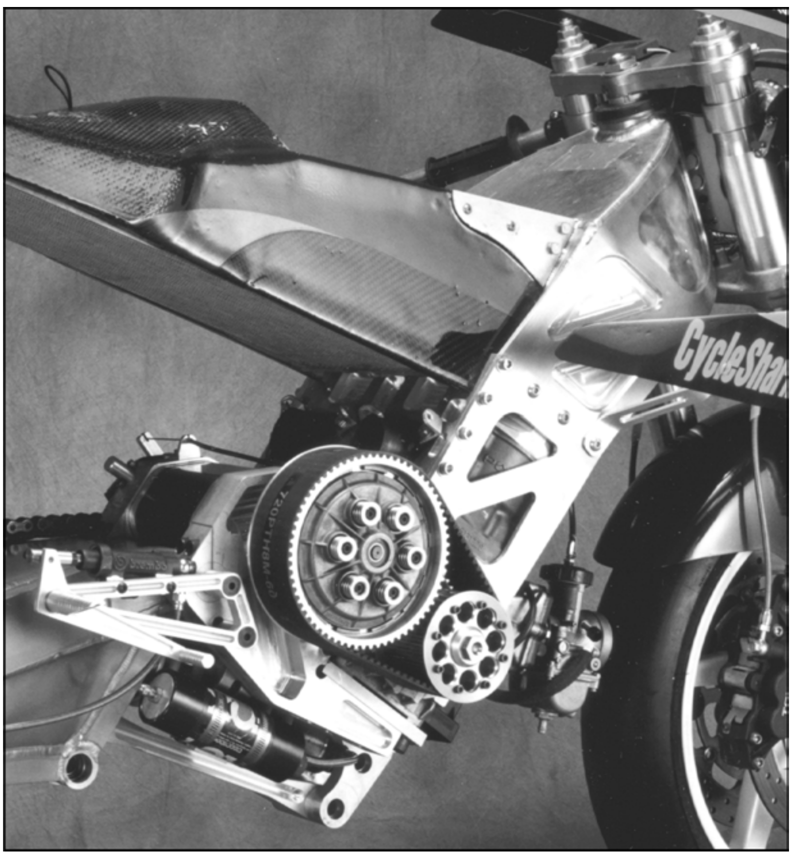

**A simple triangulated structure was bolted to the above brackets and the standard front mounting lugs on the crankcase. Specially made side castings replace the original parts to give support for the rear fork pivot bearings, which are coaxial with the gearbox sprocket.**



**This Honda CBX engine permitted a different approach to that employed for the Kawasaki above. Cylinder head lugs already existed that were suitable for mounting a similar front subframe.**

An interesting approach to engine mounting was taken by Dr. Robin Tuluie with the design of the Tul-aris. This machine uses the engine as a structural connection between front and back. The chassis and engine are integrated into one unit, the steering head uses an aluminium parallel beam construction to attach to the top of the engine, and the adjustable swing arm pivot attaches to the back of a CNC machined transmission case. The Polaris engine is a parallel twin without a balance shaft and large unbalanced forces were anticipated. The approach was to try and fix the front chassis to the engine so that there would be the absolute minimum of intermediate flexure, preventing addition vibration modes. In this case the frame mass effectively adds directly to the engine mass, reducing the vibration amplitude. The following illustration shows the large number of bolts connecting the frame to the engine.

The area just to the rear of the headstock was boxed-in and used as a secondary fuel tank. This added some damping to further reduce the vibration. In practice, this machine produced low amounts of vibration except at the handle-bars which vibrated badly in the first instance. Laboratory testing revealed that this was principally due to the handlebars themselves adopting a vibration mode. Subsequent work on tuning the vibration characteristics of the handlebars reduced this problem considerably.



**The Tul-aris uses a parallel twin Polaris engine with relatively large inherent shaking forces. The front chassis is designed to form a rigid extension to the engine, adding to its mass. Note the number of bolts fixing the front frame to the engine.**

**A small fuel tank is incorporated to provide some additional vibration damping.**

(Dr.Robin Tuluie)

# 12 Braking

## The basics

At the most fundamental level, vehicle braking is the conversion of kinetic energy into heat energy. Sometimes a small amount of the kinetic energy is initially converted into sound and light, but this ultimately turns to heat as well. Brake squeal is the release of the sound energy and when a disc becomes hot enough it glows red and so dissipates some energy as light.

Kinetic energy is the energy possessed by any moving object, it is dependent on the mass of the object and the square of its velocity. To slow from a high speed to a lower one we must remove the difference between the kinetic energy levels at each of these speeds. Fig. 12.1 shows the total amount of energy that must be dissipated for each 200 kg of laden mass to come to a halt from different initial velocities. This is drawn for a hypothetical machine similar to a typical 500 GP racer. Fortunately, at higher speeds the aerodynamic drag relieves the brakes from absorbing all of this.

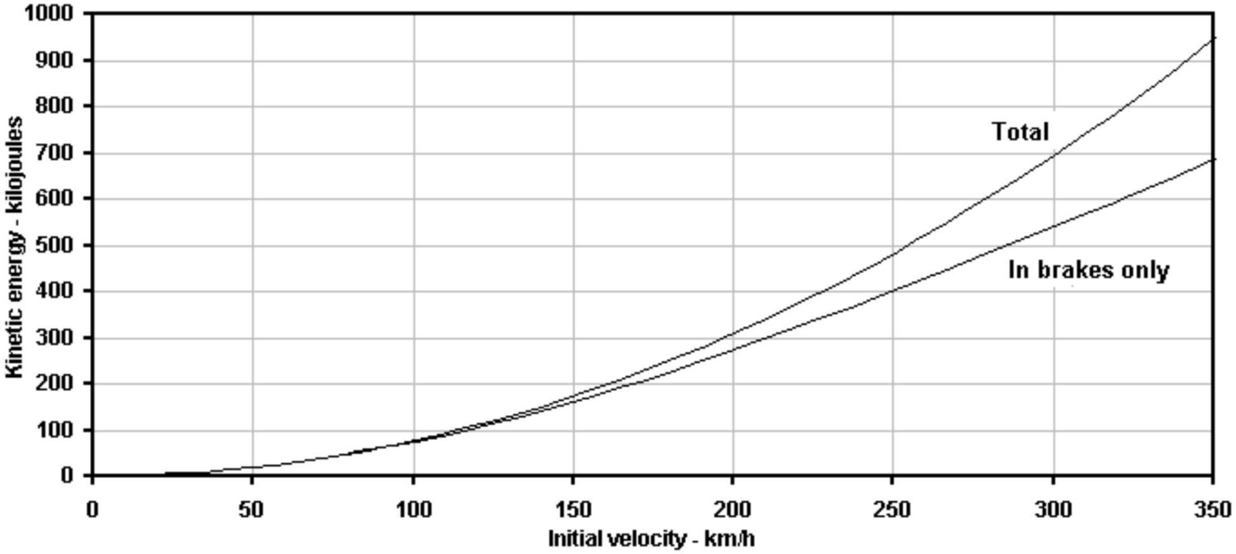



Fig. 12.1 **Showing the kinetic energy to be absorbed, to stop from different initial speeds. This example is drawn for a fictitious racing machine with 200 kg. of laden mass of the bike and rider and a value of 0.3 $m^2$ for CdA. The total energy must be dissipated between the brakes and aerodynamic drag, which becomes more important at the higher speeds. The lower curve is the energy to be dissipated in the brakes alone.**

The importance of aerodynamic effects on the degree of deceleration is shown clearly in fig. 12.2. Even at the relatively low speed of 125 km/h, the drag force adds 11% to the effect of the brakes alone. As the aerodynamic force acts directly on the machine, it does not use up any of the potential grip from the tyres, and so is additive to the deceleration provided by the brakes. In the rain the available braking through the tyres is reduced and so air drag becomes relatively even more important. Rolling resistance also adds to the total force slowing the bike but this has been ignored in these calculations because it is a smaller effect. Friction in the wheel bearings, chain and engine/gearbox has to act through the tyres to produce a retarding force, just like the brakes, and so it is implicitly assumed that this is part of the total force from the brakes. Engine braking is the most significant of these and in many cases is the principal braking mechanism at the rear.

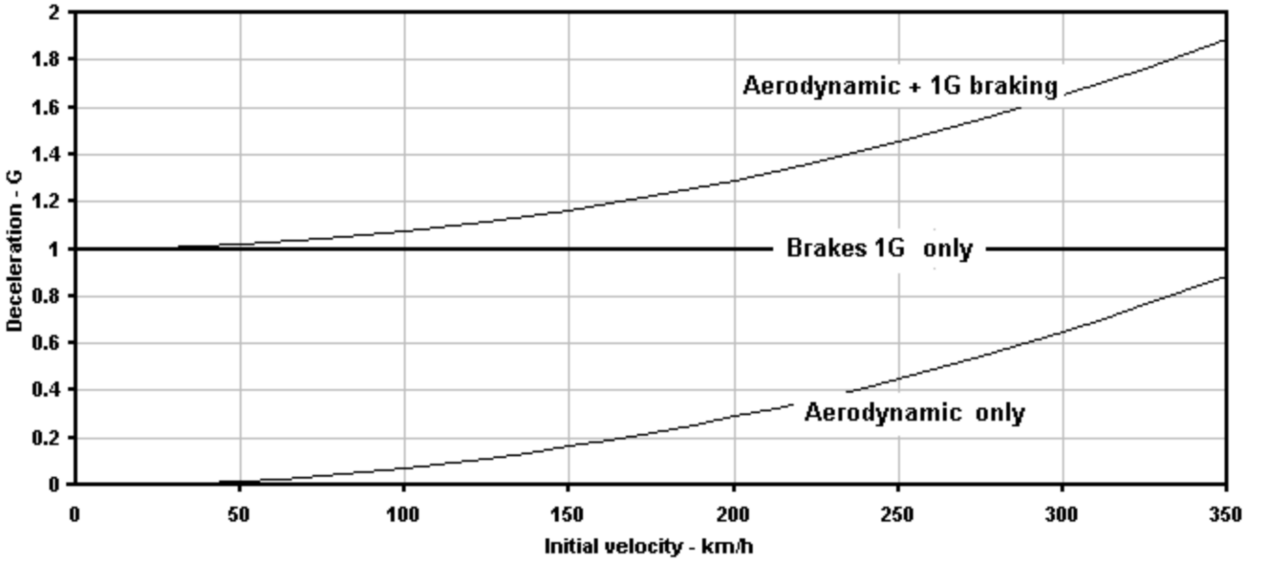
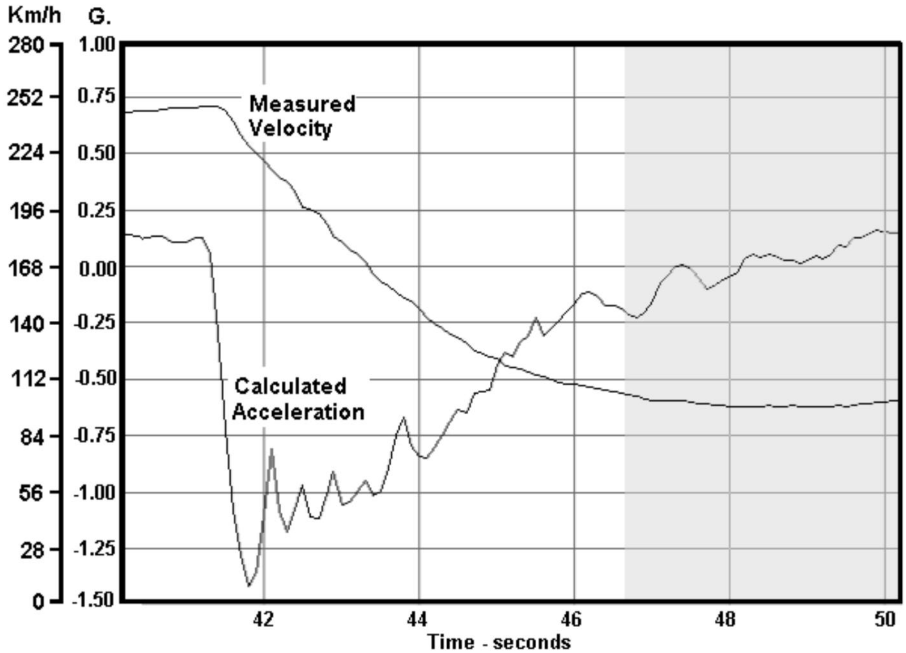
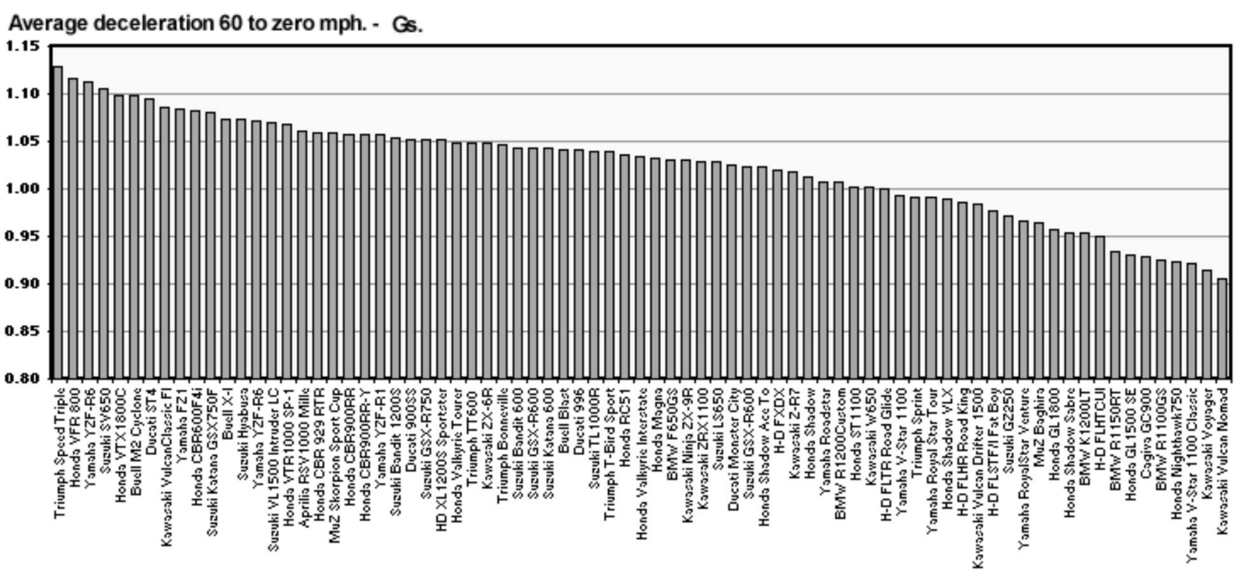



Fig. 12.2 **Aerodynamic drag adds significantly to the total braking potential particularly at high speeds. The brakes are assumed to give a constant tyre force equivalent to 1g. braking. At about 125 km/h the aerodynamic drag force is 11% of the braking force rising to 88% at 350 km/h. The total deceleration at 350 km/h can be as high as 1.88 G.**

The deceleration possible of a bike under various conditions is a good measure of the efficiency of the braking system and grip of the tyres, but in a racing context it is the time to slow from one speed to another that translates into slower or faster lap times. On the other hand, when riding on the road it is most often the distance required to stop from a given speed that is of most importance. This relates directly to collision avoidance and general safety. Fig. 12.3 shows stopping times and distances for the same bike used in the above examples. It is assumed that the effort from the brakes is kept constant to give a retarding force equal to the weight of the machine. Without aerodynamic effects that represents a constant braking deceleration of 1g. Surface conditions and tyre parameters determine the maximum braking force which may be higher or lower than this figure, 1 G is just representative of fairly strong braking.

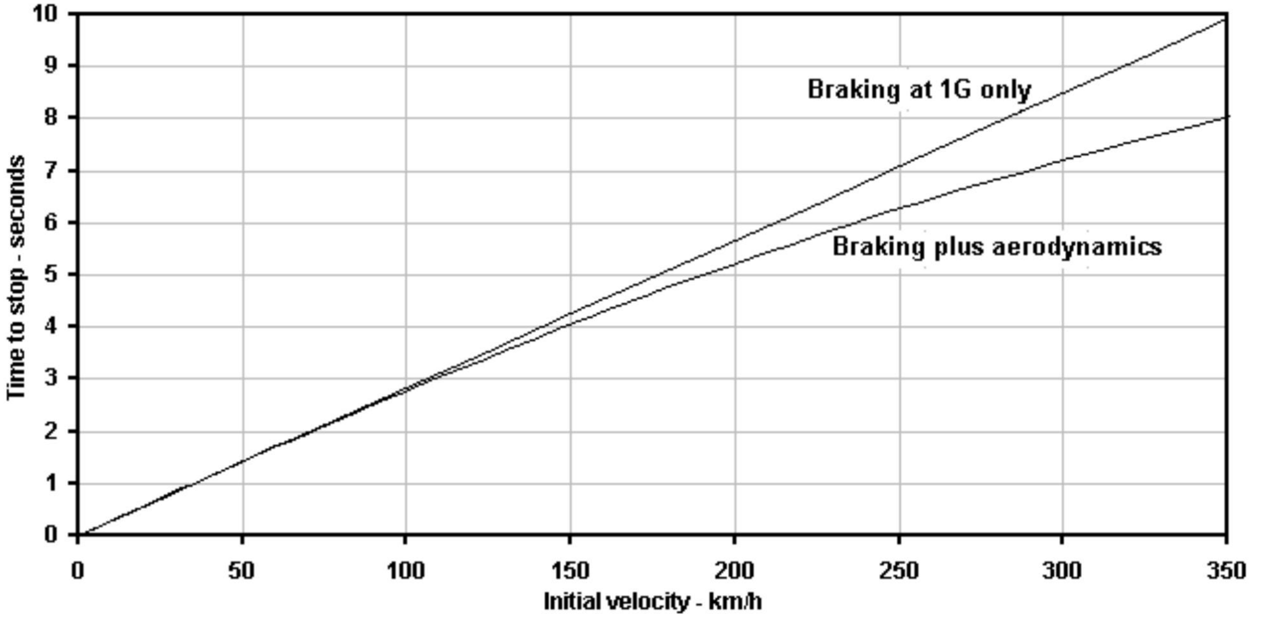



**Braking for the third bend at Misano. The shaded area represents when the bike is actually in the curve. The high initial deceleration is partly due to aerodynamic effects. Note how the deceleration then decreases as the speed reduces, some of which is due to the rider easing the brakes on approach to the corner. The velocity is measured at the front wheel and so is lower than the true velocity under hard braking due to tyre slip, and so the calculated deceleration will initially appear higher than the real deceleration.**




**Average measured braking decelerations of various road machines, expressed in Gs. All data refers to 60 mph to zero stops (96.6 km/h). At these speeds aerodynamic effects have minimal effect on stopping distances. These results were obtained with a sophisticated computer controlled radar system and probably have low error.**

(Data from Motorcycle Consumer News)



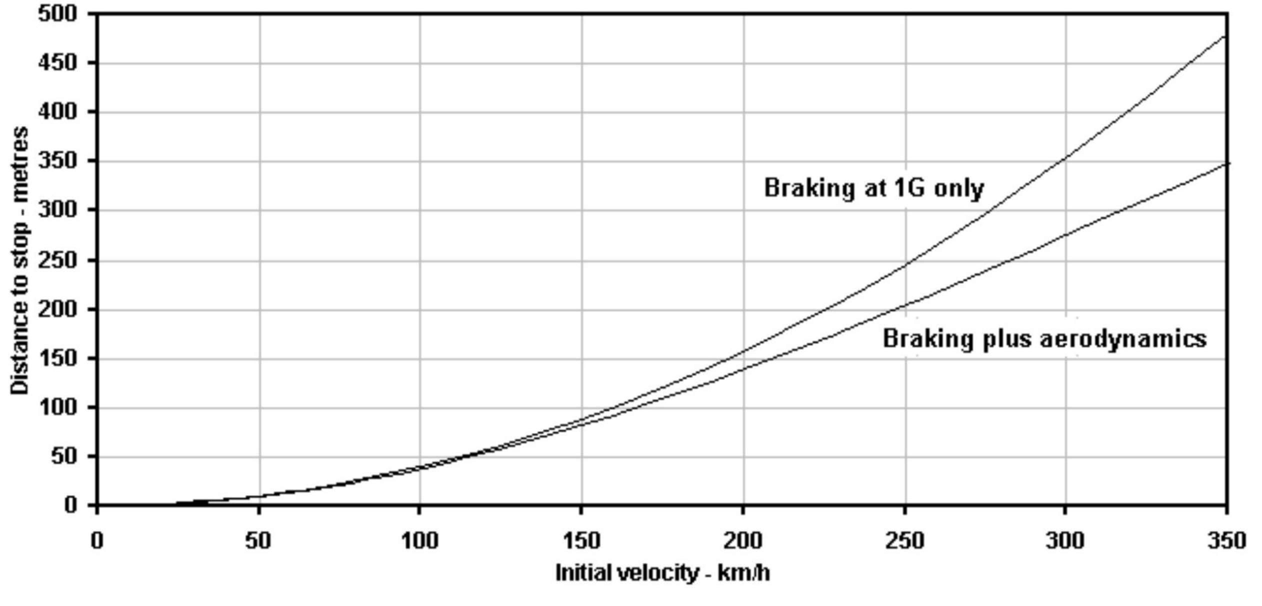




Fig. 12.3 **The upper pair of curves shows the time to stop from various speeds, with and without aerodynamic drag. From 280 km/h it would take 8 secs to stop instead of 7, without air drag. This is a lot in racing. The lower graphs demonstrate the distance required. Note how quickly this rises with speed. Again, at high speed aerodynamic effects are very important.**

Fig. 12.1 shows the total amount of energy to be dissipated by the brakes and engine friction etc.. This is the total amount of heat that must be got rid of. However, that gives no indication of how hot the brake system will get nor of how much cooling rate is necessary to maintain workable temperatures of these components. To calculate these things we must also consider how quickly we stop from a given speed. It is the time rate of dispersing this energy that is important and this is called “braking power”, fig. 12.4 shows this for the same case used earlier. Remember these figures are for a relatively light-weight racer, a heavier road bike would produce more power during emergency stops from the same speeds.

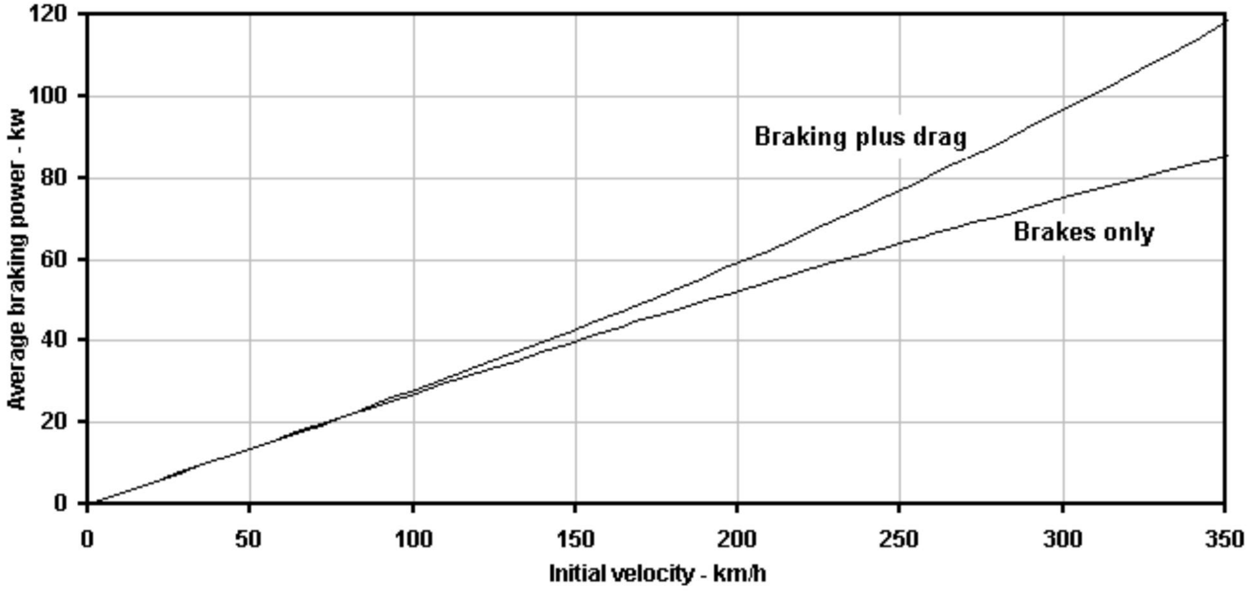



Fig. 12.4 **Average power required to stop. The lower line shows the power that must be handled by the brake discs and engine/power train friction. Fig. 12.1 slows the total energy that must be dissipated but it is the braking power that determines how quickly it must be dispersed and is the main parameter that dictates the degree of cooling needed to maintain acceptable temperatures.**

To put these figures into perspective, we can see that to slow this bike from 230 km/h we produce an average power in the brakes of about 60 kw., that is the same amount of heat to remove that we would get from one thousand 60 watt light bulbs. Most of this will be handled by the two front discs. Little wonder that brake discs often glow red. For those more used to thinking of power in terms of BHP, 60 kw. is the same as 80 BHP.

The actual temperature that the disc reaches depends on three factors.

- Braking power required.
- Degree of cooling.
- Heat capacity of the disc.

Brake cooling comes from several sources, but the principal one is from the flow of air carrying the heat away. Some heat travels through the mounting components and into the wheel, but most of this is ultimately taken away by air cooling also. Heat radiation removes some of the heat, although this is relatively more important a factor as the disc gets very hot. Radiation is the same process by which we receive heat from the sun. To maintain a steady temperature the rate of cooling at that temperature must be equal to the rate of heating or in this case the braking power. Cooling from all sources increases as the disc temperature rises, and so the temperature will continue to rise until this equilibrium condition is reached.

Heat capacity is the measure of how much heat we have to put into an object (brake disc) to raise its temperature, or in other words, the amount of heat stored in a disc at a certain temperature. For a given material the heat capacity depends on the mass of the disc. The heat capacity of a braking system doesn't affect the ultimate steady state temperature reached, but it does affect how quickly the disc reaches that temperature, and how quickly it cools down after the braking has ended. Fig. 12.5 shows this effect for two discs, one with double the mass of the other.

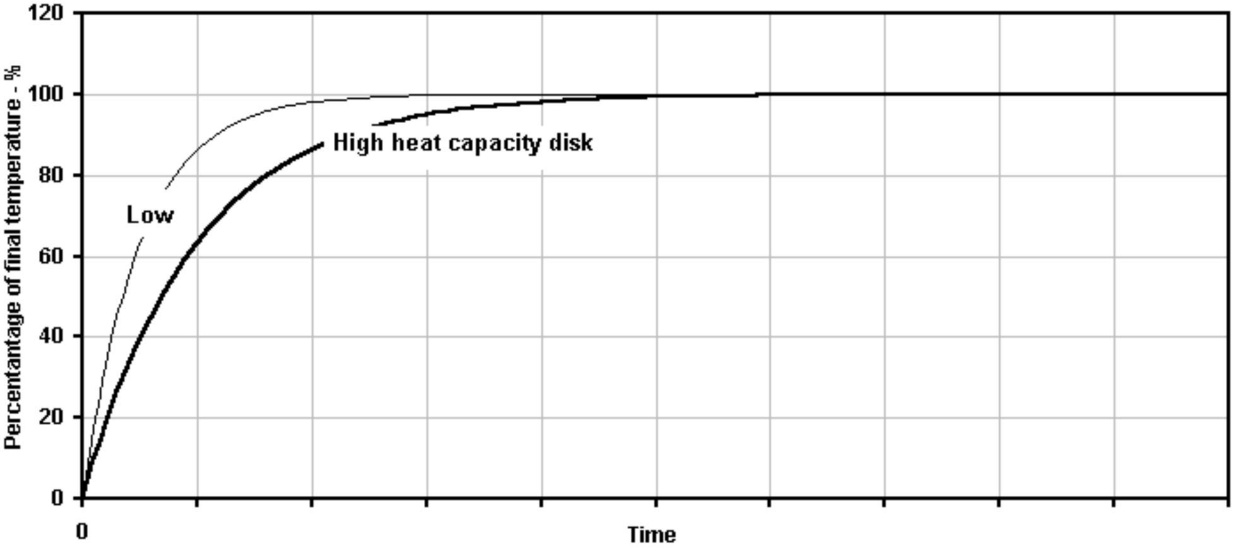



Fig. 12.5 **Illustrating in general terms, the effect of the disc heat capacity on the temperature build up. The high capacity disc has double the mass of the other. The final working temperatures will be the same in each case.**

In the past, the pad materials in use would lose some of their grip at high temperatures. Called brake fade this could be a serious problem when braking from high speed, and high heat capacity braking systems could be of considerable advantage by keeping the disc cooler for longer. It is obvious just from a casual glance that today's brake discs are much lighter than those from a couple of decades back, for both racing and sports bikes. Amongst the reasons for this are reduced gyroscopic effect for faster turn-in and less unsprung mass for improved roadholding and general suspension performance, but also to reduce the heat capacity. Modern high performance braking pads use materials that work better at elevated temperatures and so to get a consistent feel when first applying the brakes we need the discs to get up to temperature as quickly as possible.

## Effects of CoG height

In chapter 9 on *Squat and dive* we saw that load transfer occurs under braking and how it is related to the ratio of CoG height to wheelbase. The higher the CoG the higher will be the load transfer to the extent that racing and sports bikes can easily lift their rear wheels when braking hard at lower speeds. Depending on the grip of the tyres this will typically occur if the CoG is higher than about 45 – 55% of the wheelbase. Under these conditions the rear wheel is incapable of helping with the braking effort and the front must take the full load. However, if the CoG was lower then some vertical load would still be present on the rear tyre, at maximum braking, which could then be used to relieve the front of some of this duty. Fig. 12.6 shows how the maximum possible braking forces vary between the front and rear tyres depending on the CoG to wheel base ratio, between zero (CoG at ground level) and 60% of the wheel base. The example bike has a laden 50/50 weight distribution and under static loading is assumed to have identical friction coefficients of 1.0 at each tyre.

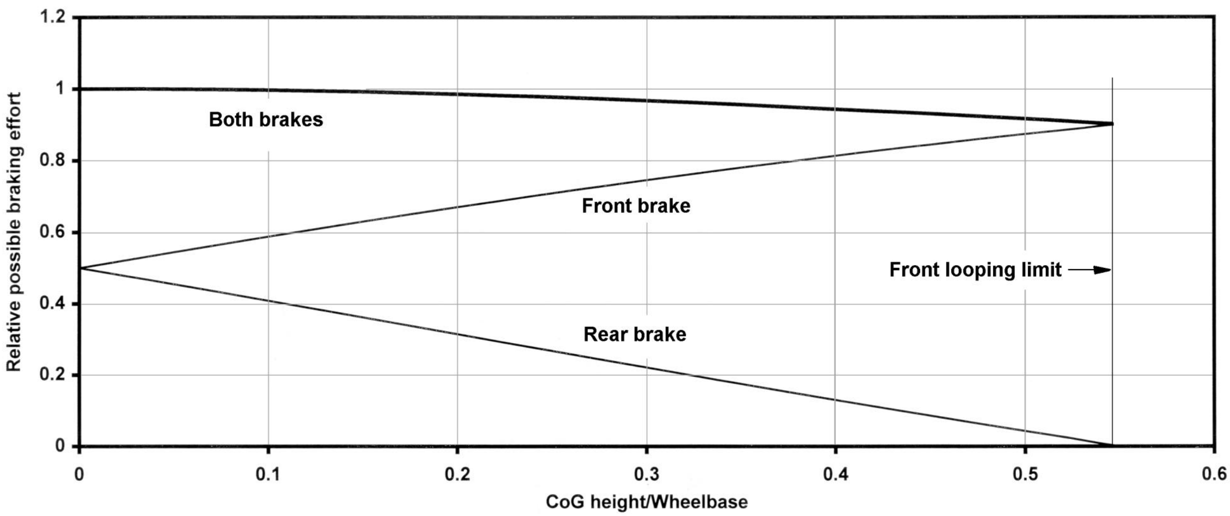



Fig. 12.6 **Maximum possible braking for various ratios of CoG height to wheelbase length. At the higher ratios less load on the rear tyre means that the front must provide most of the braking effort. When there is no vertical load on the rear tyre the bike will loop forward if subject to increased braking. This case is for speeds low enough for air drag to be minimal.**

In chapter 2 on *Tyres* we saw how the coefficient of friction between the road and tyre generally decreased as the vertical loading increased (here assumed to decrease by 10% with a doubling of load.), this is seen in the above figure as the CoG is raised resulting in increased load transfer, and the total possible braking effort is reduced slightly. The best technically possible braking would occur with a very low CoG to minimize load transfer. Then the braking is shared equally between both wheels, however, many riders find it difficult to coordinate the application of both brakes such that both are braking near an optimal fashion. In practice CoG heights are around 50% of the wheelbase and so under hard braking in the dry, most if not all of the braking is only possible at the front and so little is lost if the rear brake is left alone. On wet roads or those of poor grip for whatever reason, the maximum braking is naturally reduced and so too the load transfer, hence under these conditions rear wheel braking cannot be ignored if the best stopping is required.

In chapter 5 on Aerodynamics it was shown that air drag caused a load transfer off the front wheel onto the rear. At very high speeds this can be sufficient to almost unload the front wheel completely, and so the balance of vertical forces and hence available braking forces will be very different at low and high speeds. Fig. 12.7 shows this for the same example machine as above, except that it is assumed to be travelling at a speed with enough aerodynamic drag to just unload the front wheel completely. This is an extreme but not totally impossible case.

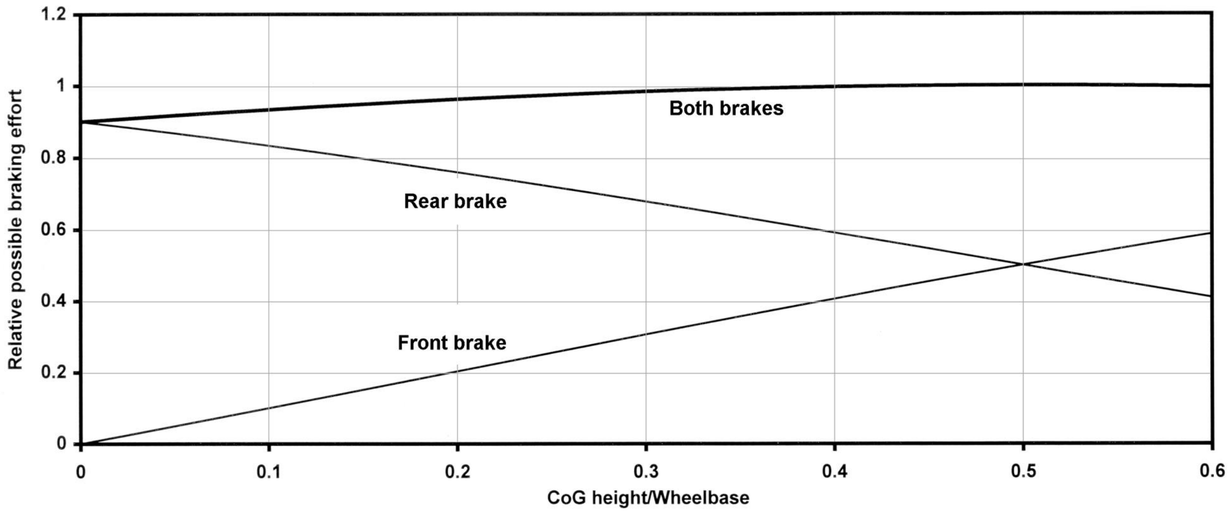



Fig. 12.7 **At very high speeds drag forces can unload the front wheel and load the rear, completely changing the required braking balance compared to the slow speed case. Optimal possible braking occurs with a CoG height to wheelbase ratio of around 0.5 or 50%.**

The curve showing the total effort from both brakes is now reversed from the slower speed case above, the maximum possible braking effort occurs when there is significant braking load transfer with a CoG height of 50% of the wheelbase. This is where the load transfer due to braking balances the load transfer due to drag. There are several other important features of this high speed graph. With a low CoG and zero braking load transfer, the front wheel is unloaded and so cannot support any braking load. The same situation occurs with a normal height CoG at the instant of applying the brakes from high speed, the front is not carrying any load and so any attempt to brake would just lock the front wheel. The rear wheel though is heavily loaded and capable of substantial braking. So the safest way of braking from high speeds, on a bike with nett aerodynamic lift at the front, is to apply the rear brake first and then as load transfers to the front, the front brake can be used with increasing force. The Honda linked braking system has a delay valve to the front brake. The figure shows that even with the CoG height at 50% (typical) of the wheelbase only half of the total braking effort can come from the front wheel. Therefore from very high speeds, ignoring the rear brake as many riders claim to do, will significantly reduce the total potential braking effort technically possible.

The ideal braking control from very high speeds would require, initial heavy use of the rear brake which must be gradually reduced, combined with an increasing dependence on the front brake as the speed reduces. Anything that can be done to reduce the height of the frontal CP (centre of pressure) will reduce this very wide variation in the required braking balance between slow and high speeds. The technically best layout for maximum braking would be a long and low machine to reduce the load transfer from both drag and braking, but to take advantage of this layout the riders would have to be capable of adjusting in an optimal manner the front to rear balance. Such a layout is similar to that used for F1 car racing, but here the driver has but one braking control to optimize, although front to rear brake balance is usually adjustable in-flight by means of a separate control.

## Generation of torque

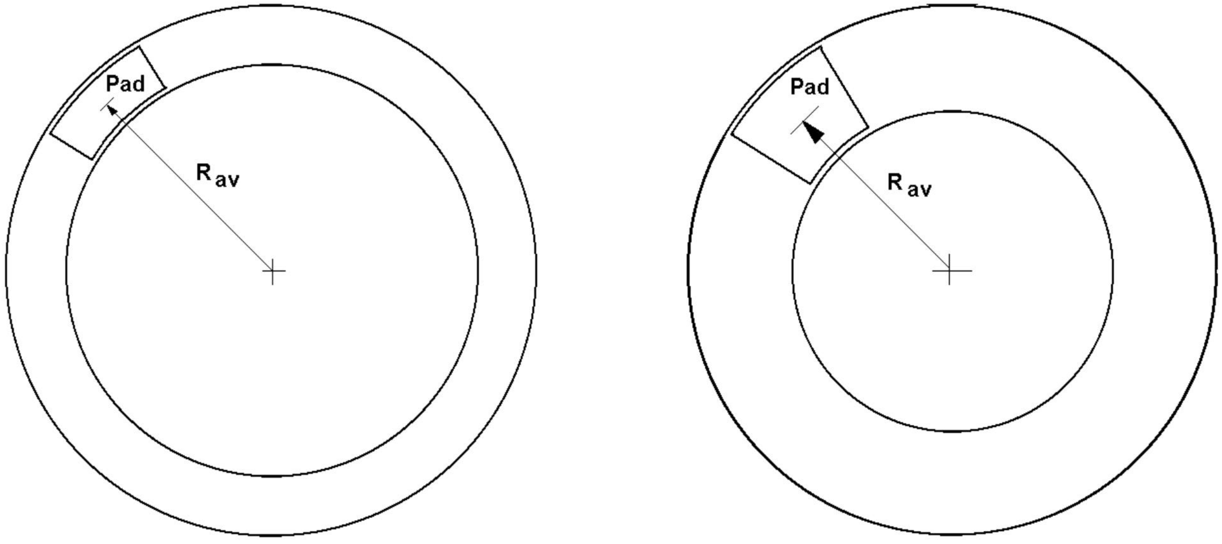



*Fig. 12.8* **The disc on the left has a smaller radial working depth than the one on the right and it can easily been seen that the effective or average radius of the area swept by the pads is greater. The disc on the left produces a higher torque than the other for a given line pressure and piston area. The heat capacity will be less also and so reach its operating temperature quicker.**

The actual production of braking torque is essentially dependent on four factors:

- Hydraulic line pressure.
- Total caliper piston area.
- Coefficient of friction between the pads and discs.
- Average working radius of the discs.

*Torque = 2 x pressure x piston area x coefficient of friction x average disc radius*

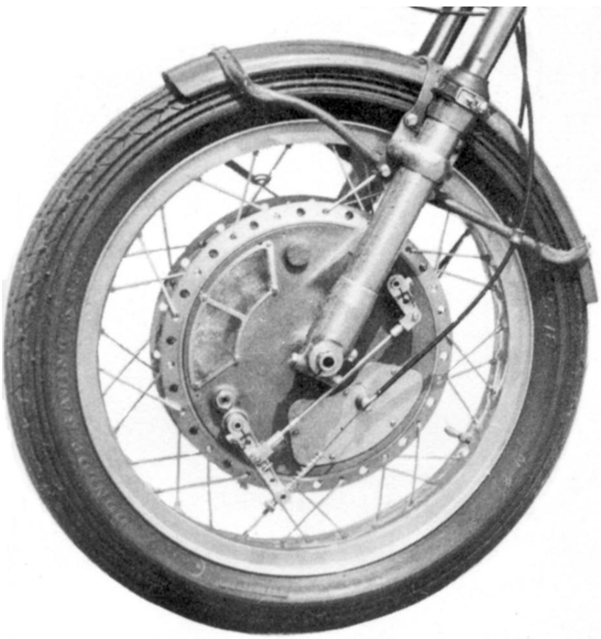
To get the braking force at the tyre we have to take the tyre radius into account as well. Then:

*Force = 2 x pressure x piston area x coefficient of friction x average disc radius / tyre radius*

The factor "2" in the above is because the same force acts on both sides of a disc. The line pressure is determined by the area of the master cylinder piston, the lever ratio of the operating mechanism and the effort applied by the rider. Fig. 12.8 shows the meaning of the average disc radius and shows how this varies depending on the radial depth of the disc swept by the pads. The modern trend toward discs of small radial depth increases the average working radius for discs of the same overall diameter.

## Hardware

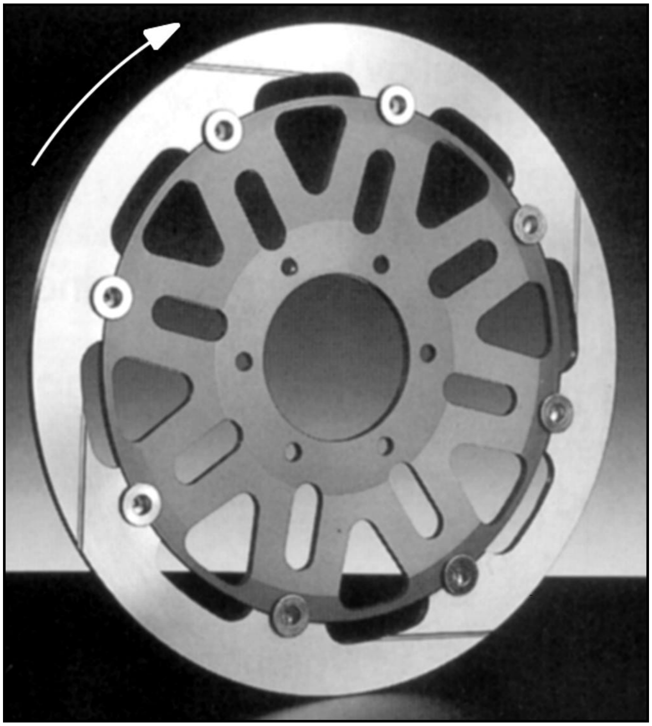
Disc brakes are now so universal on motorcycles, with only a few exceptions even at the moped end of the market, that it is easy to forget that it is really only from the 1970s. that discs have been in common use.

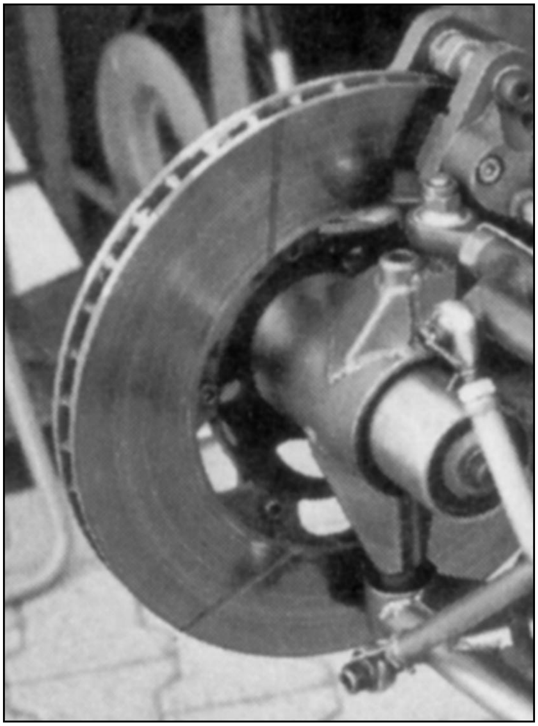


**The peak of development of the drum brake is well illustrated here on the left, fitted to the 1968 works 250/4 Yamaha GP racer. A brake drum on each side of the wheel and each one with twin leading shoes, operated by cable with a balance roller fitted near the hand control to equalize the effort applied to each side. Air ducts at the front and rear were to try and get sufficient cooling. Just over a decade later disc brakes were firmly established in racing and on other performance orientated bikes. From the same marque on the right is a 1979 500 GP machine. Note the relatively large radial depth of the swept area, and how the disc is firmly riveted to the disc carriers, both features in contrast to current design ideas.**

Hitherto, it was drum brakes in various stages of sophistication that were universal from mopeds to GP racers. For performance applications drum brakes always had severe limitations. With the use of duplex twin leading shoe brakes, it was generally possible to get sufficient stopping power, at least when cold, but the necessary servo effect of the leading shoes would often lead to a brake that was prone to grabbing when first applied. This characteristic has been responsible for many riders hitting the tarmac, including the author. On the other hand the servo effect of the leading shoes is more affected by fade and so there was a much greater fall off in performance as the brakes heated. Arguably one of the biggest problems with such brakes was expansion of the drum itself. Caused by both the physical force of the brake shoes pressing against the inside surface and also thermal expansion. The structural expansion could be minimized by rigid design but the thermal problem was never easy to solve, drum brakes are not the easiest to cool.

## Discs



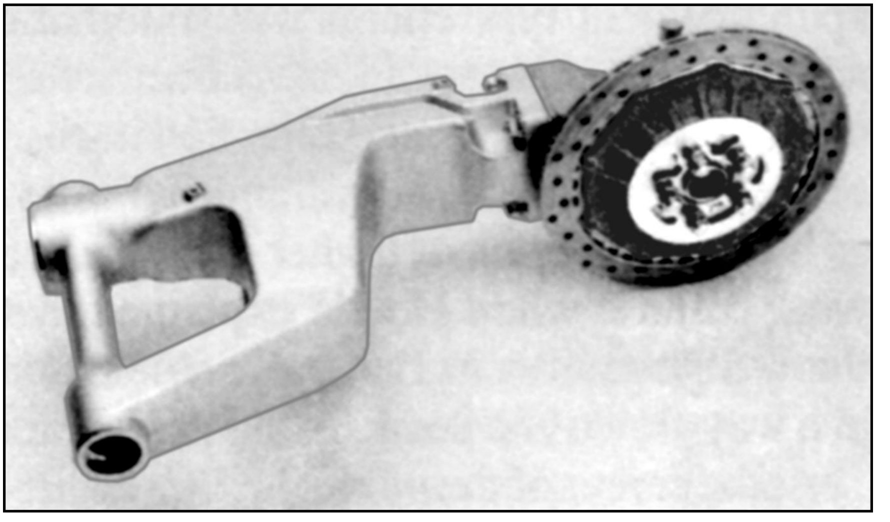


**On the left is a modern high performance brake disc from AP racing. The working disc is made of iron and has a small radial depth. The centre carrier is of aluminium alloy for lightness. The two are held together by means of 8 top-hat fixings, which allow the disc some lateral clearance but transmit the braking torque in a secure manner. The floating nature of this disc fixing is a precaution against warpage due to uneven temperature distribution. The surface slots are to permit dust from pad wear to be cleaned off the working surface, these slots are "handed" for each side of the bike, it is important that they rotate in the correct direction, the disc shown is for RHS mounting as shown by the arrow.**

**The ventilated disc to the right is fitted to a racing sidecar where air flow is somewhat more restricted, although tried on racing solos (Suzuki) it has never been accepted for general use, although used by Honda on the rear of the GL1800 Gold Wing.**

Disc brakes on motorcycles generally get sufficient cooling from the air stream on either side of the wheel, and it has been rare to see extra cooling air ducts fitted to solos. Racing sidecars, on the other hand have wide tyres and wheels which tend to shroud the brake like some cars, both ducting and ventilated discs have been employed to reduce disc temperatures. In the past some manufacturers experimented with ventilated discs on solo racers but this has never been accepted as the norm. Extra weight being the principal disadvantage with this feature. The moment of inertia of the discs is of great importance on a bike due to the effect on cornering lean-in performance, an increase in gyroscopic effects adversely affects this response. Disc mass is all unsprung and so suspension reaction also suffers as a result of heavier discs.

**Cast single sided swing-arm from a Honda GL1800 Gold Wing. The rear end of this machine is largely covered by luggage bodywork and so the brake disc is shrouded from a cooling air-flow. Hence the use of a radially ventilated rear disc. The front discs were not shrouded and are of the normal plain plate type.**



Although disc brakes had occasionally been used before, it was during the 1970s. that the real changeover began in earnest. Initially for racing by European and American riders. In the beginning design just followed car practice with the use of cast iron for the disc material. When the Japanese factories started to fit discs to road bikes they used stainless steel, although less suitable from a braking point of view it had the advantage that it didn't rust in damp climates nor if the bike was unused for a while. Poor wet weather braking was another feature of the early stainless steel brakes and it was usual to see slots and holes used to try and break up the surface water. Italian manufacturers continued to use cast iron even on road bikes. As mentioned earlier, it is desirable to keep disc weight to a minimum and to this end there were many experiments with alternative materials such as aluminium, but due to the relatively soft surface these needed some form of hard coating. Hard chrome plating, plasma sprayed ceramic coatings and hard anodizing have all been tried with varying degrees of success but none of these have stood the test of time. More recently, for top level competition, discs made exclusively of carbon have become firmly established in those racing classes that don't ban their use. Such bans are based on the high cost of such technology rather than on technical grounds. Carbon has many features on its side, importantly it can withstand high temperatures and is very light. It must be used with pads of the same material and early designs only gave good braking at high temperatures causing some problems with stopping during the initial cold application of the brakes. Different manufacturing methods can now produce carbon components that are much less sensitive to this problem.

**An interesting design from AP racing during the early 1980s. Although various rim discs had been tried by other private constructors before. Termed the ISO disc (In Side Out) it made use of the maximum possible radius. In place of the normal central carrier the disc was mounted on lugs cast on to rim of especially made wheels. The fixing system was patented and allowed some float of the disc both laterally and radially to allow for expansion. The special calipers were dual piston and proved adequate for the job. Shown here with a stainless steel disc fitted to a BMW front end it was also tried on works GP Suzuki 500s with carbon discs.**

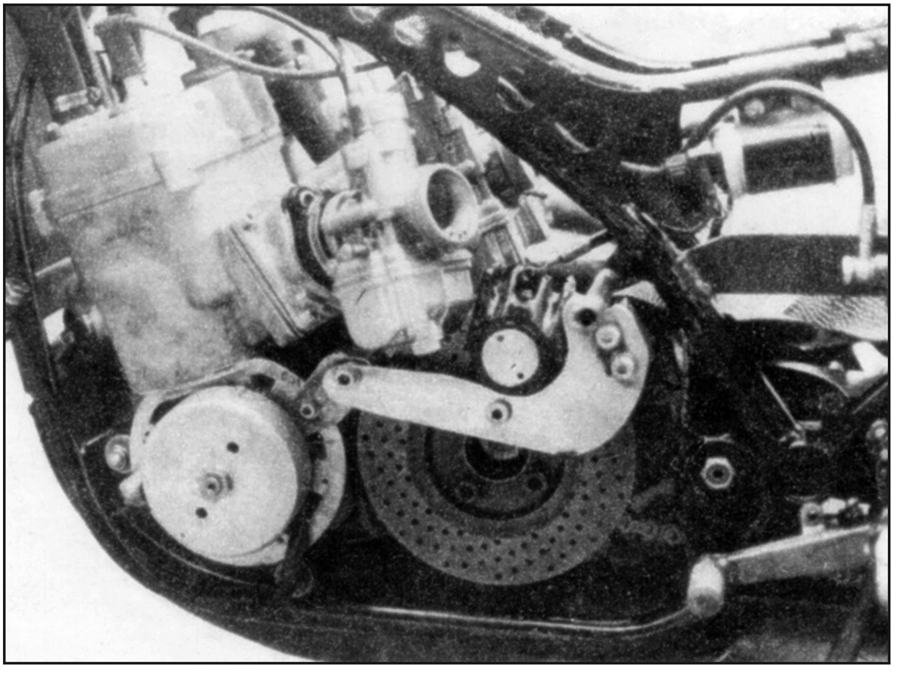
**Despite a general lack of interest at the time, this type of disc has recently been 'reinvented' by other manufacturers.**

**With telescopic forks there weren't any overwhelming advantages, but it was the ideal disc design for the author's dual link suspension as shown on page 7-24. The open centre area gave plenty of space for the suspension components.**

Aermacchi Harley-Davidson conducted an interesting experiment on their racing two-stroke twins in the mid-1970s. They mounted the rear brake disc on the gearbox sprocket so that the braking force was taken through the chain. Getting sufficient braking at the rear of a 250 was unlikely to be a great problem and so we must look elsewhere to find any potential benefits, which might include:

- less unsprung mass
- less torque on the swing-arm
- a smaller and lighter disc, since the secondary gearing increased its rotational speed.

Offsetting these advantages, the disc was more shielded from the cooling breeze and chain slack could cause brake judder. If the bottom chain run is slack when the brake is first applied, little load is initially applied to the disc from the chain, and so it may momentarily lock or be slowed considerably below the corresponding road speed. Then, when the lower chain run is tightened by further rear-wheel rotation, a shock load is applied to the wheel, so leading to a possible oscillation between disc and wheel, hence to judder. This may well be the reason that the idea was abandoned. The disc also proved to be no smaller than those seen on the rear of other similar sized machines.

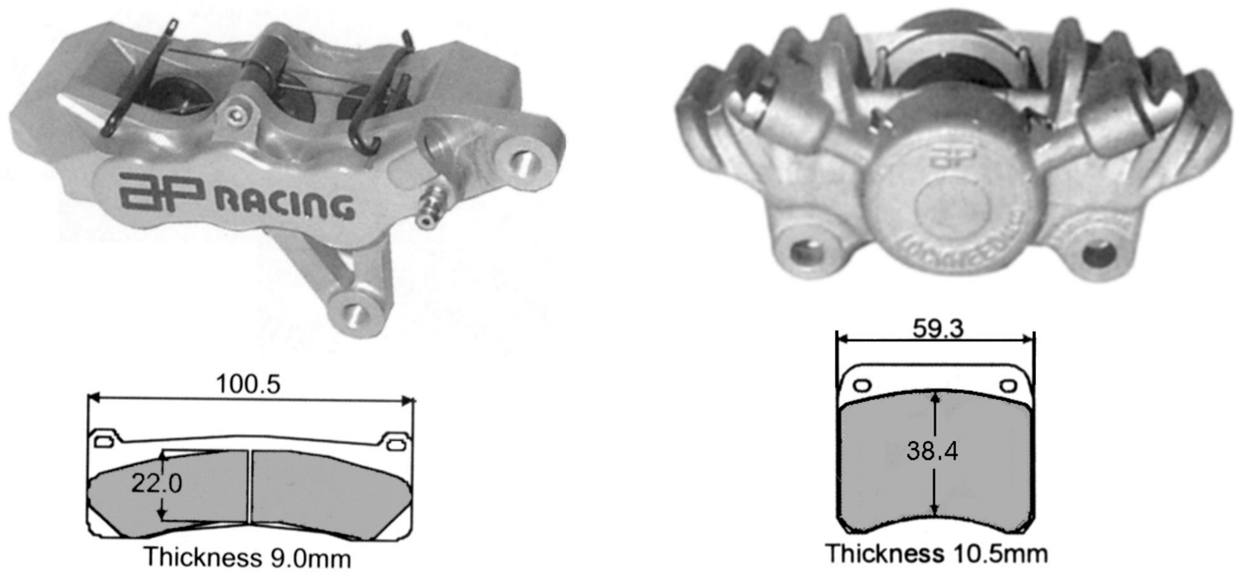


**Unusual gearbox sprocket mounted rear-brake disc on the Aermacchi two-stroke racing twins of the mid-1970s. The pros and cons are explained in the text.**

## Calipers

There are two main classes of caliper – single acting and double acting. The double acting have a piston or pistons on each side of the disc. The body of the caliper is fixed to the fork leg and only the pistons move to apply force to the pads. The single acting only has pistons on one side of the disc and to balance the pad force the caliper body has to be free to move laterally. This movement has been provided either by allowing the caliper to slide on two pins or by having an approximately vertical pivot giving the caliper a swinging motion, (early BMW and Honda). The pivoted design gives rise to uneven pad wear, the pad on the inside of the disc wearing more at the front and the outside pad wearing more toward the rear. Both movement methods have a serious deficiency in every-day road use, particularly in countries prone to wet weather, and that is corrosion of the moving surfaces. This causes the caliper body to seize on the sliding or pivoting pins, thus losing the ability to properly balance pad pressure with a consequent reduction in braking ability. The principal reason for the use of single acting calipers is simply one of cost, although sometimes there are space saving benefits. The inside half of the single acting caliper can be thinner, allowing the discs to be mounted closer to the wheel, thus reducing fork width, but this is more of theoretical benefit than a practical one.

Calipers can also be classified according to the number of pistons. Early racing calipers and many current road caliper use a single set of pistons, two for a double acting caliper and one for a single acting. As the radial depth of the working surface of the disc has be reduced with time, the length of pads has increased to compensate for the lack of area, and to ensure an even distribution of pressure it has become necessary to use multi-piston calipers. Four or six pistons are common with double acting units and two or three with single acting. Sometimes, the pistons of such calipers are of slightly different diameters to alter the distribution of pad pressure to even up on pad wear. Without such measures the pads often wear to a wedge shape.




**The old and the new. Nearly thirty years separate these two calipers. On the right is an old style twin piston type made from two separate aluminium dye castings, weight without pads was 900 grams. The advent of CNC machine tools has made it economically feasible to machine critical components from solid billet as typified by the modern racing caliper to the left. Weighing just 690 grams this has been made in one piece to give maximum rigidity, and six pistons are used to spread the load over the long thin pad. Different piston diameters are used to reduce the tendency of the pads to wear to a taper, four pistons are 28.6 mm and two are 25.4 mm in diameter. Total piston area of 35.8 cm$^2$ compared to 26.8 cm$^2$ of the older model. The different pad shapes are shown underneath, note the much smaller radial depth of the newer one which also has a smaller surface area of 19.8 cm$^2$ in place of 22.4 cm$^2$.**

## Pads

Brake pads were traditionally based on asbestos until health concerns forced a search for different materials. Various compounds are now in use, tailored to suit the application. Pad compounds are developed through high-tech chemistry and are continually evolving. At the highest levels of racing with carbon discs it is necessary to use carbon pads.

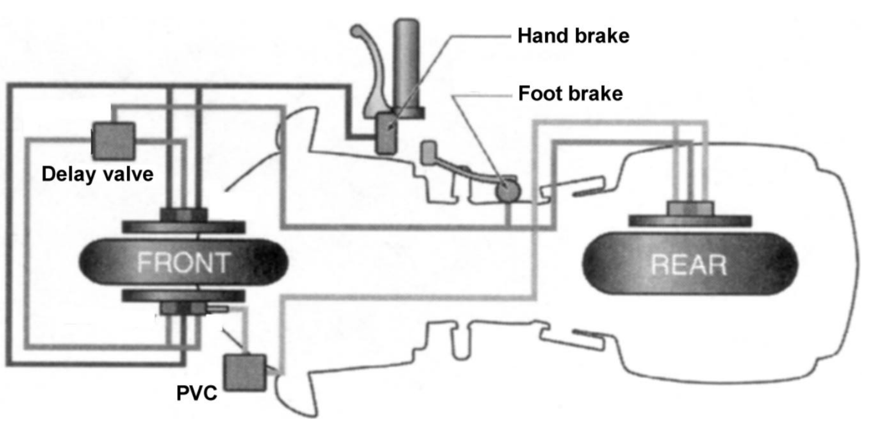
Even for use with metal discs, carbon compounds and carbon-metallic are finding use in high performance applications. Sintered metal has become the standard high performance pad, with low fade and quick cooling properties. Lower cost general purpose pads can be ceramic, which give relatively long life.

## Linked brakes

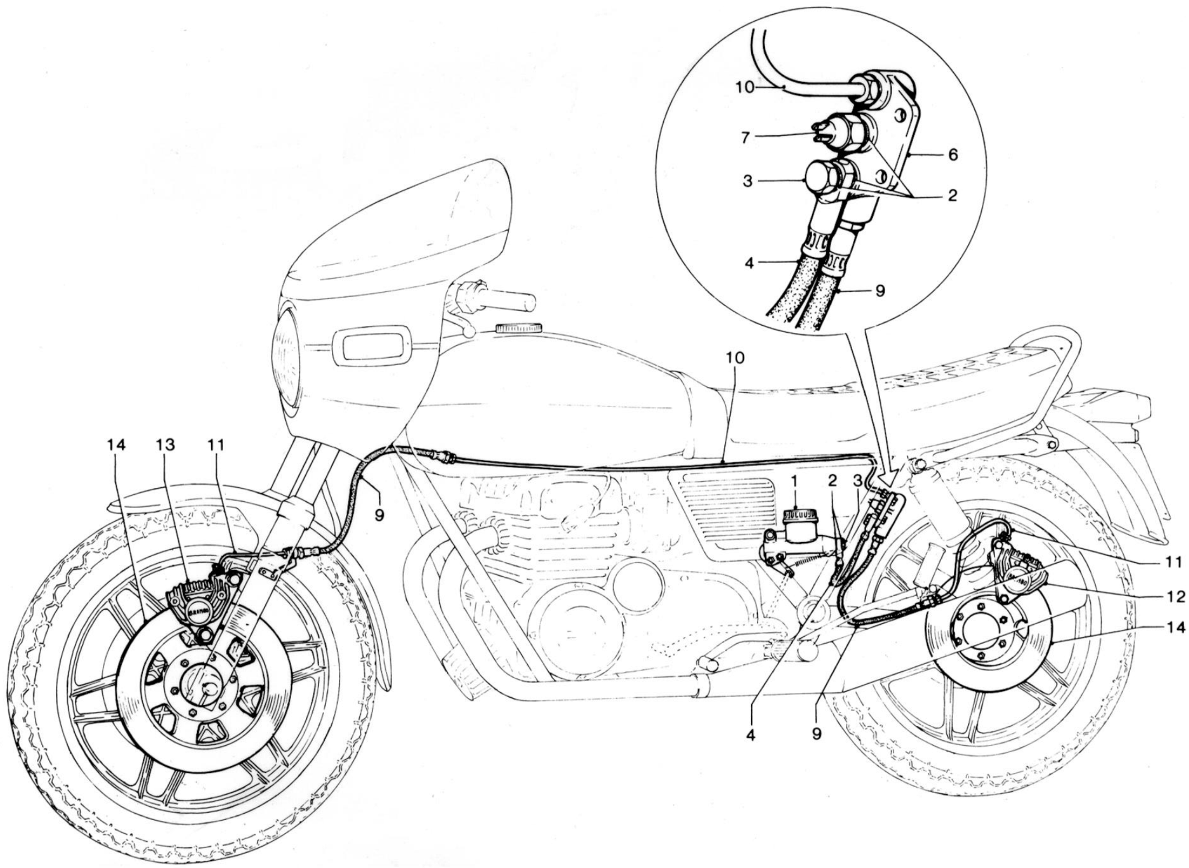
Whilst cars have traditionally had one control only for the working brakes, motorcycles have always had two, in fact racing and road regulations usually insist on it. We've seen that motorcycles experience a wide range of load transfer under braking and so it would be very difficult to provide a single control that achieved the optimum brake balance under all conditions. However, in practice very few riders are able to achieve a reasonable balance themselves and many, possibly the majority, don't even try, preferring instead to ignore one of the brakes. It is often the more useful front brake that is least used by the least experienced.

Recognising this fact there have been various attempts at producing linked brake systems, to try and protect these riders from themselves. Rudge in 1925 had a mechanically linked system, which used a very simple design to increase the front bias as braking increased. The front brake was activated by the foot pedal directly through a normal cable, but the rear brake was pulled through a spring. So the stronger the effort put into the pedal the more that was directed to the front. Probably the first since the era of hydraulic disc brakes began was the Brembo system used by Moto Guzzi, firstly around 1975. The rear brake and one of the two fronts were linked hydraulically with identical calipers at each end, but the front disc was greater in diameter giving a forward bias to the braking. The second front brake was operated by the conventional handlebar master cylinder. This second brake satisfied the legal demands but also gave the skilled rider the option to increase the front brake bias as and when required. The Guzzi was a relatively long and low bike with a low seat height and so was subject to less load transfer than many other bikes, making it a more suitable machine for this linked system. Later models have a swing-arm actuated proportioning valve to increase the front bias according to rear wheel load.

Yamaha and Honda have also used linked brakes on some models. The Honda Silver Wing scooter uses an interesting system with multi-piston calipers. As is common on scooters the two controls are handlebar mounted, one on the left grip and the other in the normal position to the right. The front caliper has three pistons. The right hand lever controls two pistons on the single front caliper, just like a conventional motorcycle, but the left lever controls the rear brake and the remaining front piston. There is a delay valve to the front brake to ensure that the rear is applied slightly earlier. This is a clever feature which will enhance the initial braking stability, particularly at high speed, see also chapter 14.




**Linked brakes on the Honda GL1800 Gold Wing. Complicated, but there's enough redundancy to ensure fail-safe operation. The hand brake operates two pistons on the right-hand front caliper and one on the left, through an isolating secondary master cylinder and proportioning valve it also powers two rear pistons. The foot brake directly feeds one rear piston and a total of three front pistons via a delay valve.**




| | | | | | |
|---|---|---|---|---|---|
| 1 | Master cylinder | 6 | Four way joint | 11 | Rigid pipe |
| 2 | Copper washer | 7 | Stop light switch | 12 | Rear brake caliper |
| 3 | Banjo bolt | 9 | Flexible pipe | 13 | Front brake caliper |
| 4 | Flexible pipe | 10 | Rigid pipe | 14 | Brake disc |

Fig. 12.9 **Diagrammatic layout of the original Brembo hydraulic coupled brakes as used by Moto Guzzi. The object is to eliminate reliance on a high degree of expertise in apportioning front and rear effort.** (Brembo)

## ABS

Anti-locking brakes were originally developed for aircraft, the Dunlop Maxerat system, and then applied to trucks, buses, cars and finally made their way onto motorcycles, but not until the late 1980s. As shown in chapter 14 on stability it is very important to avoid locking the wheels of a motorcycle, especially the front, and so, in many ways ABS is more important to have on a bike than other vehicles. Therefore it might seem surprisingly that manufacturers, with the notable exception of BMW, have been very slow to adopt a system with such a safety potential. Perhaps at least part of the reason is that highly skilled riders, under controlled conditions, without ABS have sometimes beaten the stopping distance of motorcycles equipped with early ABS designs. However, that is to deny the majority of less skilled riders an important aid, particularly under slippery and emergency conditions.

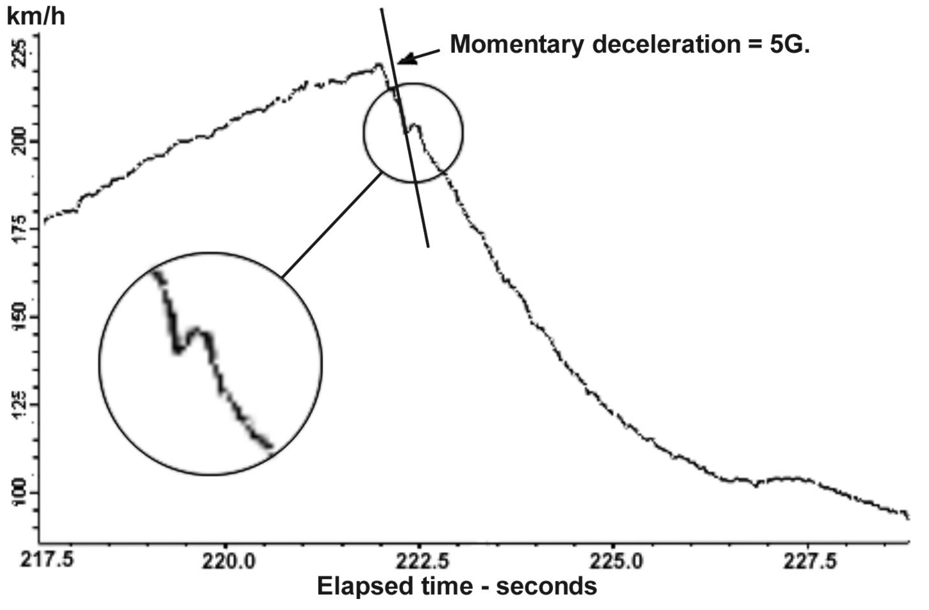



Fig. 12.10 **Wheel locking in practice. This is the front wheel velocity trace from a 250 GP bike. The enlargement clearly shows the sudden deceleration of the wheel and the relatively slow return to road speed. The deceleration can be determined by measuring the slope of a line drawn as shown. In this case the effective deceleration at the tyre was close to 5 G.**

**This was a very brief locking/unlocking sequence, lasting only about 0.25 secs.**

(Data courtesy of 2D.)

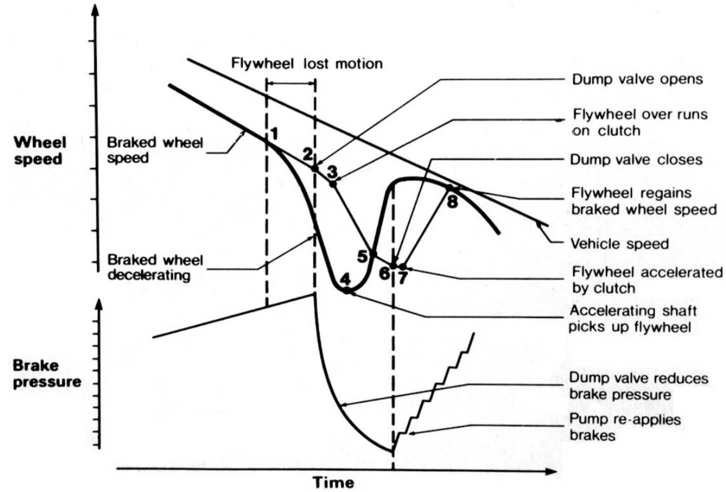
Although development of motorcycle ABS systems had been going on from the late 1960s it wasn't until 1987 that they became available on production models. Until then, design was heading in two separate directions; electronically controlled and mechanically actuated.

The operation of the two systems is basically similar, the elementary sequence of events for both is as follows:

- Detect the onset of wheel locking.
- Remove the braking effort.
- Allow wheel to regain a speed close to road speed.
- Reapply brake.

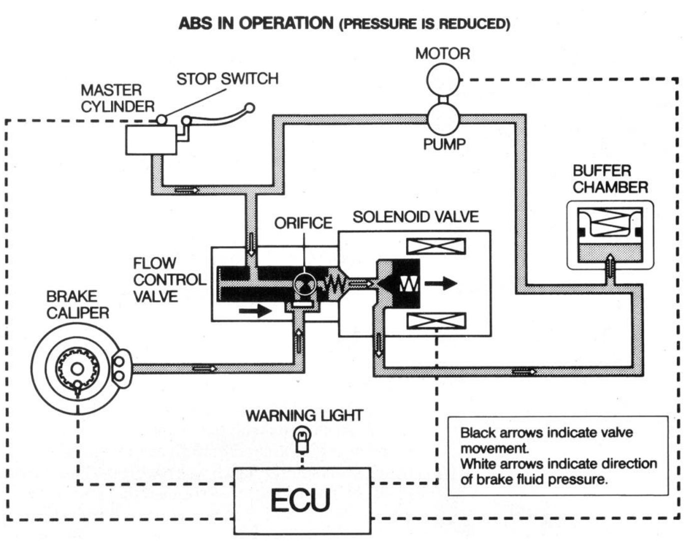
This may occur up to about ten times per second. It is in the detection phase that the main differences exist between the electronic and mechanical systems. In essence the mechanical systems, such as the early Dunlop Maxerat and later Lucas-Girling SCS, use a small flywheel, geared to spin faster than the wheel. When the wheel locks it decelerates rapidly but the flywheel wants to continue as before, this difference is detected by an overrun mechanism and used to operate valves to control the pressure fed to the brakes.

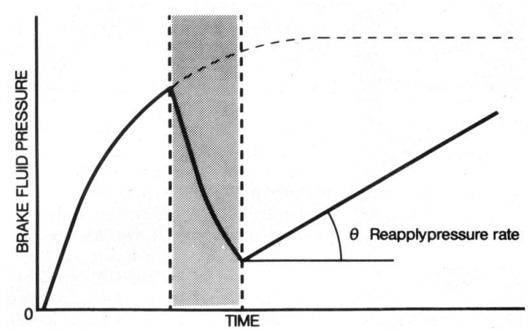
The electronic systems measure the speed of the road wheels by means of some form of toothed disc and electronic sensor. This velocity signal is fed to a micro-processor which calculates the wheel deceleration, when this is too great the micro-processor signals that the hydraulic pressure must be released.




**Lucas-Girling SCS (Stop Control System) mechanically actuated ABS. The SCS unit can be seen just ahead of the forks, a notched rubber belt, behind the brake disc, drives the internal flywheel at three times wheel speed. The graphs to the right follow a single lock-unlock cycle.**

Although the SCS system was still in development at least until 1987, as with many other products it could not withstand the onslaught of modern electronics and as far as is known no mechanical systems actually made it into motorcycle production.



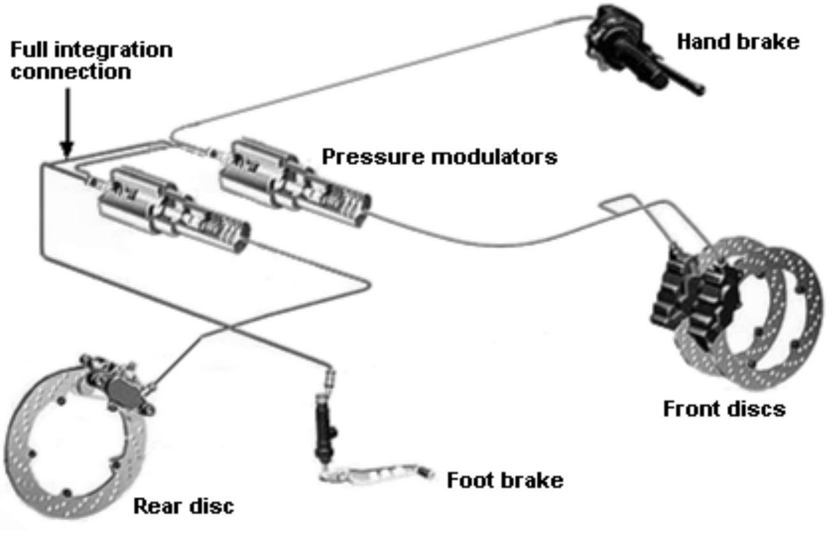




**As the ABS is ativated, the solenoid valve is opened releasing brake fluid pressure to the buffer chamber. The flow control valve moves due to the pressure difference across the orifice. Movement of the flow control valve cuts the passage to the caliper before the orifice. Further movement opens the passage after the orifice allowing the caliper to depressurize.**

**Yamaha ABS system. First used on the FJ1000A in 1991 and later miniaturized and silenced in 1998 for the 250 Majesty scooter.**

BMW were the first to offer machines for sale with ABS, followed by Honda and Yamaha, and remain the only manufacturer so far to have really committed themselves to it. Other manufacturers have only offered it on a few specific models, but BMW have offered it across their range, either as standard or optional equipment. Recognising, perhaps, that on average, scooters are often ridden by the least skilled, there is a growing trend for ABS to be offered on such models. A handicap to this process has been the high cost of ABS, but that is changing and prices are coming down, opening up the range of models to which it can be economically fitted.

The first generation BMW systems were very heavy and added significantly to the selling price. The current, third generation versions are better in both respects, and more sophisticated in concept and performance. The brakes are linked and both front and rear brakes have their ABS function controlled individually.




**Layout of the third generation BMW ABS system. This is the fully integrated system which reacts the same regards of whether the hand or foot brake is used. The partially integrated system is identical except for the removal of the connection between the foot circuit and the front modulator as shown.**

An interesting feature of BMW's linking system is that it features an "adaptive brake force distribution" system. This senses wheel speed and the amount of brake bias is also modulated according to the load condition of the bike. By sensing wheel-lock on either wheel it then transfers more braking effort to the opposite wheel. So each wheel will receive the maximum amount of braking torque that it can use. This system can handle up to 10 lock/unlock cycles a second.

# 13 Materials and properties

Before deciding which material is most suitable for any particular component, we clearly need to know something about material properties, and the main properties of concern to us are:

- Strength
- Stiffness
- Density (or specific gravity)
- Ductility
- Fatigue resistance
- Available joining methods
- Cost of material
- Cost of machining and working

The relative importance of these properties depends on the purpose for which the machine is intended. For example, low cost is much more important for a mass-produced moped than it is for a works grand-prix racer, where cost is secondary to low weight. In a previous chapter we defined the terms strength and stiffness in relation to a complete structure. The same concepts apply to a single piece of material and, when quantified, provide a yardstick for comparing different materials. The term stress – more particularly ultimate tensile stress (UTS) or Yield stress (also called the elastic limit) – is used as a measure of strength. Stress is akin to a measure of the force density in the material. Expressed as the force applied per unit of cross-sectional area, e.g. if we apply a load of 100 kgf. to a piece of material of 1 mm.– square section, then the stress in the material is 100 kgf per square mm. (kgf/mm$^2$).

The UTS is the stress under which the material breaks completely. In some cases, other stress criteria may be more valid for comparison. Yield stress, for example, is the stress where permanent deformation begins and is useful when comparing ductile materials. Under load, all materials deflect to some extent. This deflection is called strain and simply expresses the proportional change in dimensions; e.g. if we pull a 1000 mm. ( 1 metre.) length of material until it stretches by 1 mm., then the strain in the material is

$$\frac{1}{1000} = 0.001$$

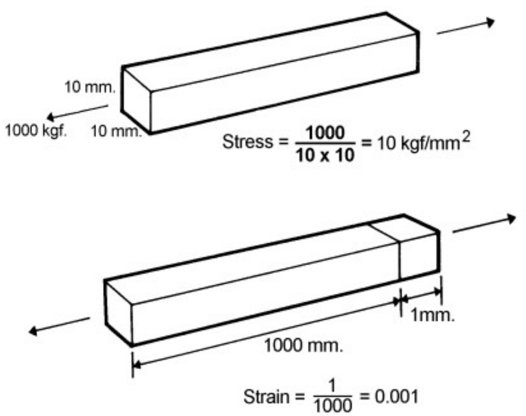
As we have seen, stiffness is the ratio of the applied load to the deflection it causes. This is defined by Young's Modulus, which is simply the applied stress divided by the resulting strain; e.g. if a stress of 10 kgf/mm$^2$ is needed to produce a strain of 0.001, then Young's Modulus for the material under stress is

$$\frac{10kgf/mm^2}{0.001} = 10{,}000kgf/mm^2$$

The stress strain curve in fig. 13.1 shows the typical performance of a non-brittle material such as the grades of steel used in frame construction. If we subject a piece of the material to a stress below the yield limit then a certain degree of strain occurs, as explained above, but this is elastic strain and when we remove that stress then the material returns to its original unstrained shape and size.

This is called elastic deformation because it behaves like a spring. However, if we attempt to apply a higher stress than the yield point, then the material *gives* and deforms permanently. When the stress is removed the object does not return to its original condition. This is known as plastic deformation.

When we continue to apply sufficient load beyond the yield point we reach the point of ultimate failure and the material actually breaks. The amount of strain that occurs between the yield point and the failure point is a measure of the materials ductility.



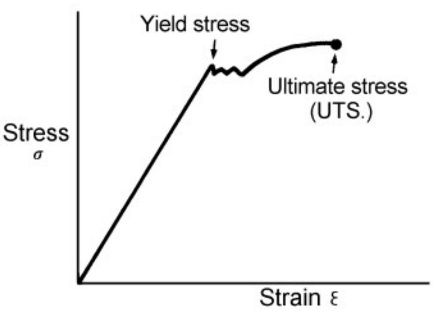




Fig 13.1 **Top left shows the meaning of the term stress. The significance of strain is shown below that. A typical stress-strain curve for steel is on the right. The initial slope of the line is the Young's Modulus.**

Ductility determines the type of failure exhibited by a material. If it undergoes much permanent deformation before final fracture it is said to be ductile. But if it fails suddenly, with little prior distortion, then it is brittle. Generally speaking, a ductile material is preferable because it can withstand a certain amount of overloading without total failure. For example, if a cast wheel is made from a ductile material it may be only dented or buckled on hitting a kerb or sharp pot-hole, but if it is brittle a section of the rim may break off or the spokes may fracture.

Unfortunately, as we increase the strength of materials we usually increase their brittleness too. So, the selection of a suitable grade of metal and its heat treatment is a compromise that can be determined only in the light of the component's duty and applied loads. As a rule, cast metals are less ductile than their forged or wrought counterparts. As mentioned in a preceding chapter, fatigue characteristics vary with the type and condition of a metal. An additional point here is that a brittle metal is more likely to suffer a fatigue fracture than is a ductile one – the non-brittle material will distort in such a way as to reduce the stress concentrations and so lessen the possibility of ultimate failure.

Ductility is also strongly influenced by the type of loading. An extremely ductile material can fail in a brittle manner if subjected to triaxial stress – i.e. to loading that results in a component of stress on each of three mutually perpendicular axes. Fig 13.2 demonstrates this pictorially. When we apply strain to a piece of material in one direction it tends to contract in the two side directions. This ratio between the longitudinal strain and this lateral strain is known as Poisson's ratio and for most metals is about 0.33. It is easy to calculate that this ratio must be between 0 and 0.5, and in fact rubber is around 0.5.

When we apply an outward stress on all three axis we prevent this lateral strain and the material has no choice but to fail in a brittle manner.

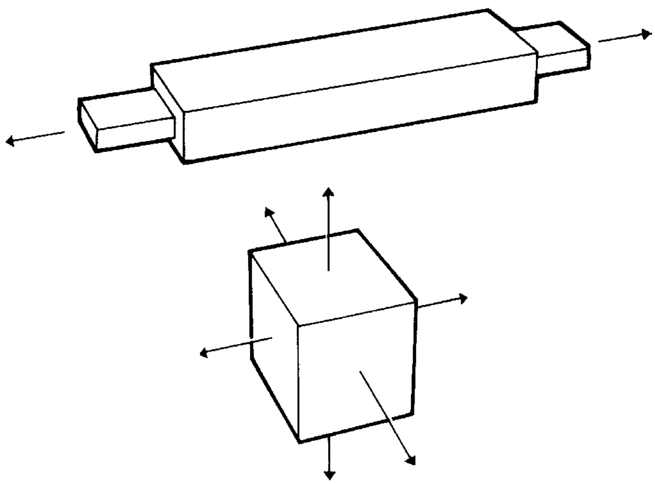


Fig 13.2 **When a bar is stretched along one axis (above), the longitudinal extension is accompanied by a lateral contraction.**

**Triaxial stress (below) prevents such contraction and the material will act in a brittle manner.**

## Typical properties of some common materials

| Material | UTS Kgf/mm$^2$ | Specific gravity | Young's Modulus x $10^4$ Kgf/mm$^2$ | Relative stiffness |
|---|---|---|---|---|
| Grey cast iron | 16 - 22 | 7.3 | 1.27 | 0.65 |
| Structural/Stainless steel | 47 - 63 | 7.8 | 2.1 | 1.00 |
| Reynolds 531/4130 tube | 78 | 7.8 | 2.1 | 1.00 |
| Aluminium alloys | 17 - 63 | 2.7 | 0.7 | 0.96 |
| Magnesium alloys | 19 - 32 | 1.7 | 0.44 | 0.96 |
| Titanium alloys | 47 - 126 | 4.4 | 1.2 | 1.01 |
| Nylon | 8 | 1.1 | 0.14 – 0.28 | 0.47 – 0.95 |
| PVC | 6 | 1.4 | 0.025 - 0.042 | 0.07 – 0.11 |
| PTFE (Teflon) | 1.3 | 2.2 | 0.035 - 0.042 | 0.06- 0.07 |
| GRP (Glass Reinforced Plastic) | 16 – 35 | 1.7 | 0.07 – 0.2 | 0.15 – 0.44 |
| Carbon fibre (In direction of fibres. Fibres only, properties reduced in resin matrix.) | 140 | 1.6 | 2.2 | 5.11 |

Density is a measure of mass per unit volume; hence, size for size, it compares the masses of different materials. We get the same comparison from specific gravity, since that is just the density of any material compared with that of water under standard conditions. The above properties are a rough guide only, as the tensile strength may vary considerably, depending on the metal composition or alloy and its state of heat treatment and working. In particular the tubing used for frame construction will lose some strength after welding, and composites vary considerably.

The specific gravity and Young's Modulus do ***not*** vary in this way. In the above table the term "relative stiffness" means the ratio of Young's Modulus to specific gravity referenced to that of steel, which is a measure of stiffness per unit weight. For the metals in our table, except for cast iron, this ratio is almost constant at 0.96 – 1.01. Thus, while aluminium alloys are approximately only one-third as dense as steel, their stiffness too (size for size) is only one-third that of steel. To put it another way, weight for weight, aluminium has approximately the same tensile stiffness as steel, magnesium and titanium. However, this applies only to tension and compression stresses. The attraction of carbon fibre is easily explained with a value of 5, its stiffness is similar to steel but is much less dense, and so lighter components can be made with the same stiffness.

Stiffness in bending and torsion depends not only on the modulus and cross-sectional area (proportional to weight) but also on the second moment of area (often called the moment of inertia) – and this provides a clue to the efficient use of lightweight materials. Consider a solid round bar subject to a bending load. To determine the bar diameters that will give the same stiffness in both steel and aluminium, we need the product of "E" (Young's Modulus) and "I" (second moment of area) to remain constant. Now "I" depends on the fourth power of bar diameter, while bar weight depends on diameter squared and material density. Calculations show that – since aluminium has one-third the density and one-third the modulus of steel – the diameter of the aluminium bar needs to be larger by 32 per cent, at which its weight will be only 58 per cent of that of the steel bar.

In a motorcycle chassis, of course, we are seldom concerned with solid bars but mostly with round tubes. To maintain the same bending and torsional stiffness with a less dense tube material, we can either keep the diameter the same and increase the tube thickness or (more efficiently) keep the thickness the same and increase the diameter. Between the two extremes is a wide choice of proportions. If we increase the tube diameter and use the same wall thickness, we find that an aluminium tube needs approximately twice the diameter of a steel one but would weigh only 70 per cent as much.

Thus the most efficient way to use light-weight materials is to make the sections as large as possible consistent with maintaining a practical wall thickness. But, in maintaining similar structural characteristics to those of steel, our light alloy tube will weigh more than a simple comparison of density indicates. The density of aluminium is 33 per cent that of steel but the structural weights of our bar and tube in the foregoing examples are 58 and 70 per cent respectively.

The terms chrome-moly, T45, 4130 and 531 are frequently bandied about as though they have some magical significance, implying extra stiffness and lightness, to such steels. In fact, these terms are simply standards, or commercial references and refer to steels with alloying elements calculated to enhance strength, particularly strength after welding. Their Young's modulus, hence stiffness, is no different from that of other steel alloys, nor is their density. Hence, if they are substituted for lower-strength steels and the same size tubing is used, then the weight and stiffness of the frame will be unchanged. Where they score is in the ultimate load that the frame will take before breaking. If stiffness is no problem with a particular structure or member, then the use of high-strength tubing will permit thinner walls, hence reduced weight. But if stiffness is vital, then the best way to use this tubing is to reduce wall thickness and increase diameter, but in a smaller proportion. In this way only can stiffness be improved and weight reduced.

A Reynolds trade name, 531 is often referred to as chrome-moly, whereas it is actually a manganese-molybdenum steel, which Reynolds claim has superior properties to those of a chrome-molybdenum steel. For many years this tube type was a favourite amongst British specialist frame builders.

The main alloying elements in Reynolds 531 are as follows:

| | |
|---|---|
| Carbon | 0.23 to 0.29 per cent |
| Silicon | 0.15 to 0.35 per cent |
| Manganese | 1.25 to 1.45 per cent |
| Molybdenum | 0.15 to 0.25 per cent |
| Sulphur | 0.45 per cent maximum |
| Phosphorous | 0.45 per cent maximum |

Its minimum strength properties are:

| | **Yield stress** | **Ultimate stress** |
|---|---|---|
| As drawn | 71 kgf/mm$^2$. | 79 kgf/mm$^2$. |
| After brazing | 63 kgf/mm$^2$. | 71 kgf/mm$^2$. |

These figures indicate the excellent retention of strength after brazing, which is a great boon. The use of this and other high-strength tubing is normally confined to competition machines, for roadsters the extra cost is not usually warranted.

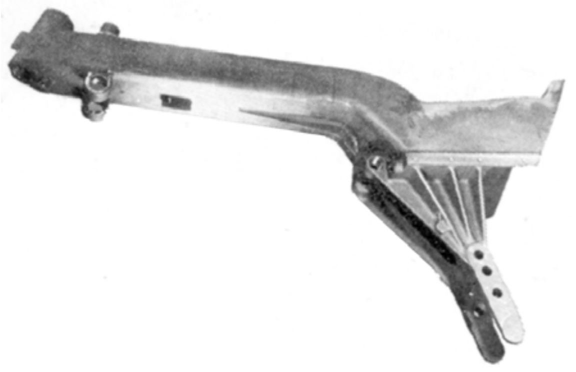
Now that we have dealt with the principles underlying the selection of materials, let us consider the choices open for various components.

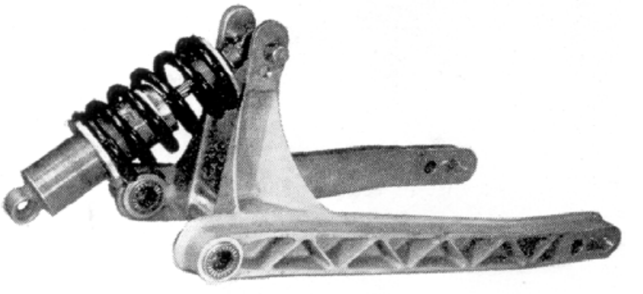
## Frame

Steel is easily the most common material here, either as tube or sheet, depending on design. There are several reasons for its choice, viz:

- Raw material cost is relatively low.
- Well developed manipulating and joining techniques are available.
- Young's Modulus is high, so the required stiffness can be obtained with small tube sizes.

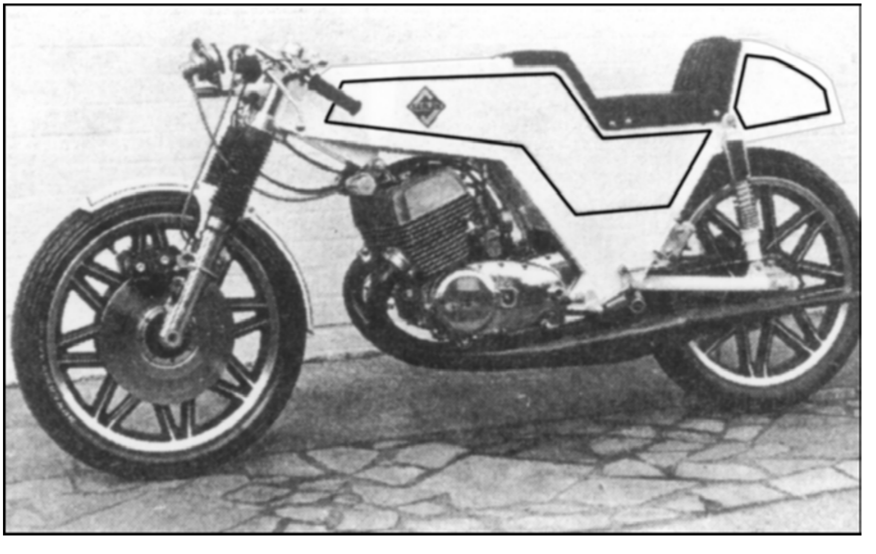
Aluminium has often been used for specials and racing machines in the form of monocoques and large-section backbones such as the fabricated Ossa and Kawasaki mentioned earlier. Cast-aluminium backbones have been tried by Eric Offenstadt in France and Terry Shepherd in England. However, components such as complete frames are rarely cast because the minimum material thickness needed for the casting process usually results in excessively heavy components.





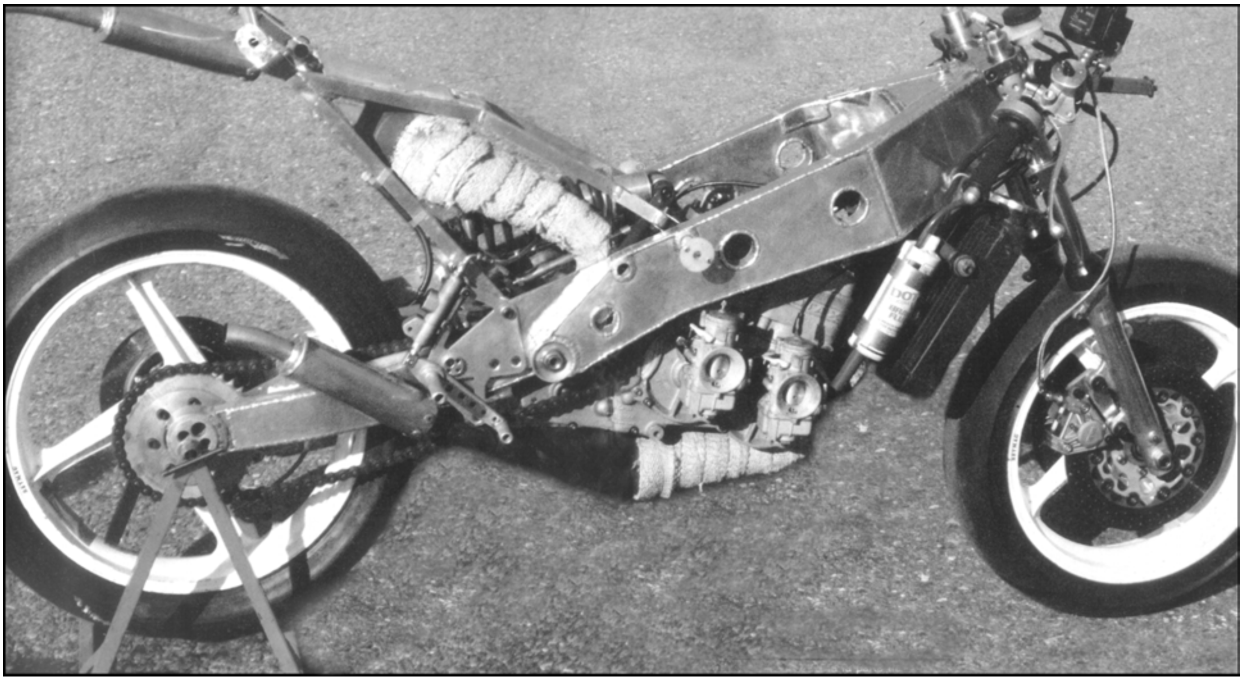
**Examples of cast chassis components. On the left is a cast frame from Eric Offenstadt, the backbone design is hollow in the main section. To the right is the swing-arm for the same frame. To avoid the increased pattern making and casting costs inherent with hollow sections the swing-arm uses a ribbed design. This is easier to make but results in a less structurally efficient component.**

Then tubular aluminium frames started to appear on works racers, with Yamaha taking the initiative. This trend started cautiously, when just the pivoted rear fork was made in light alloy, before spreading to the complete chassis. In the development of aluminium frames, however, it is interesting to note that tube sizes increased rapidly to compensate for the low Young's Modulus, as explained earlier. A great help in this context would be the spread of proper triangulation. In GP racing now, the use of aluminium alloy fabricated twin-spar frames is almost universal, and is also widely featured on expensive sports models for the street. It must be remembered that the fatigue characteristics of aluminium are such that failure is inevitable eventually in components subjected to alternating stress, hence limited life must be accepted. In the case of works racers, their natural rapid obsolescence makes this less of a serious problem.



**Using racing car technology of the time, this 1970s. Colin Seeley framed racer was fitted with a Suzuki twin cylinder 500 cc. two-stroke. Basically The frame consisted of flat aluminium alloy sheet, folded at the edges and glued and riveted together. If due attention is paid to the detail design of the mounting points there is no doubt that this form of construction can produce a very rigid frame. The swing-arm design seems spindly by comparison.**

For touring machines, where long life dictates lower stress levels, it may well be that aluminium's weight advantage over steel is lost.



**This trend setting twin-spar GP Kobas from the early 1980s. housed a 250 Rotax engine. Note that the swingarm and main frame are constructed of flat aluminium alloy welded at the corners. Today's frames may be neater in appearance due to different manufacturing methods but this early Kobas had structurally stiffer sections, and a more direct connection between head-stock and swing-arm pivot than many that have followed.**

Tubular frames have also been made of titanium – as, for instance, on the BSAs for the world motocross championships in the 1960s. The likely drawback here was excessive flexure, because the tube sizes appeared to be no larger than in the team's successful steel frames, whereas titanium has approximately half the modulus of steel. This does not mean that titanium is unsuitable for frames, but rather that design must take account of the properties of the materials used. With its low weight and high strength, titanium is probably used to best advantage in a triangulated design. Its chief disadvantages are high cost and the sophisticated welding techniques required, but its corrosion resistance is excellent.

Magnesium, both cast and fabricated, has been used for backbone-type frames. Besides high cost and welding difficulties, however, it has the added disadvantage of limited life as a result of both fatigue and corrosion. Porsche accepted this limitation in their 917 sports racing cars, the chassis of which comprised welded-up triangulated structures in magnesium tubing – so the technique might be worth trying on racing motorcycles, given ample financial resources.

The use of composite materials, such as carbon fibre and Kevlar reinforced plastics, is extensively used in Formula One car racing. The monocoque style of chassis construction there lends itself to this approach much more than does a motorcycle; but clever design and future development in materials may alter the picture, though at present construction costs are very high.

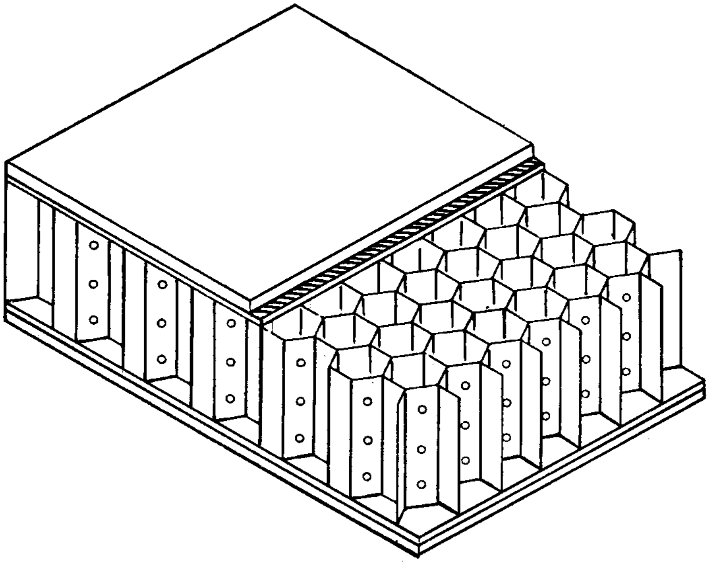


Fig 13.3 **Typical composition of honeycomb composite. Two sheets of material are spaced apart by a light weight centre formed as a honeycomb shape. Adhesive is used to bond the separate components. Slabs made in this way weigh little more than the outer two sheets because the spacing structure is often of very thin material. However, the depth given to the slab gives it much higher bending and torsion rigidity than that of an equal weight plain sheet. Whilst this material has been used to make some frames, the designs have to be composed of flat surfaces glued together. This means that it is only suitable for a limited range of engine types and frames layouts.**

The John Britten designed V twins had very successful chassis largely built of carbon reinforced plastic, but to date there have probably been more failures than successes with this material.

Another form of composite material that has been tried is aluminium honeycomb, although similar material is available made with carbon fibre reinforced plastic in place of the aluminium.

## Wheels

For most of motorcycle history, the traditional wheel was a composite of hub, spokes and rim. Hubs have been made in steel, cast iron, aluminium and magnesium. In the days of drum brakes, the light-alloy hubs usually had a cast-iron brake drum. Although some people experimented with various forms of plating or other hard surfacing direct on the drum surface to improve heat dissipation and save weight.

Spokes were of steel, sometimes titanium, usually with brass nipples, though these were sometimes in aluminium for racing. Rims have mostly been of steel, except that aluminium took over for sports and competition machines and some roadsters.

Since the late 1960s, however, cast wheels have become increasingly popular, first for racing (where magnesium predominates) then on roadsters, where cost and corrosion problems favour aluminium. Even cheap mopeds now use die-cast aluminium wheels.

In magnesium, a properly designed cast wheel may well be lighter than a steel-spoked wheel with an aluminium rim and magnesium hub; but cast-aluminium wheels usually have a weight penalty though in some cases they may be stiffer laterally and run more accurately.

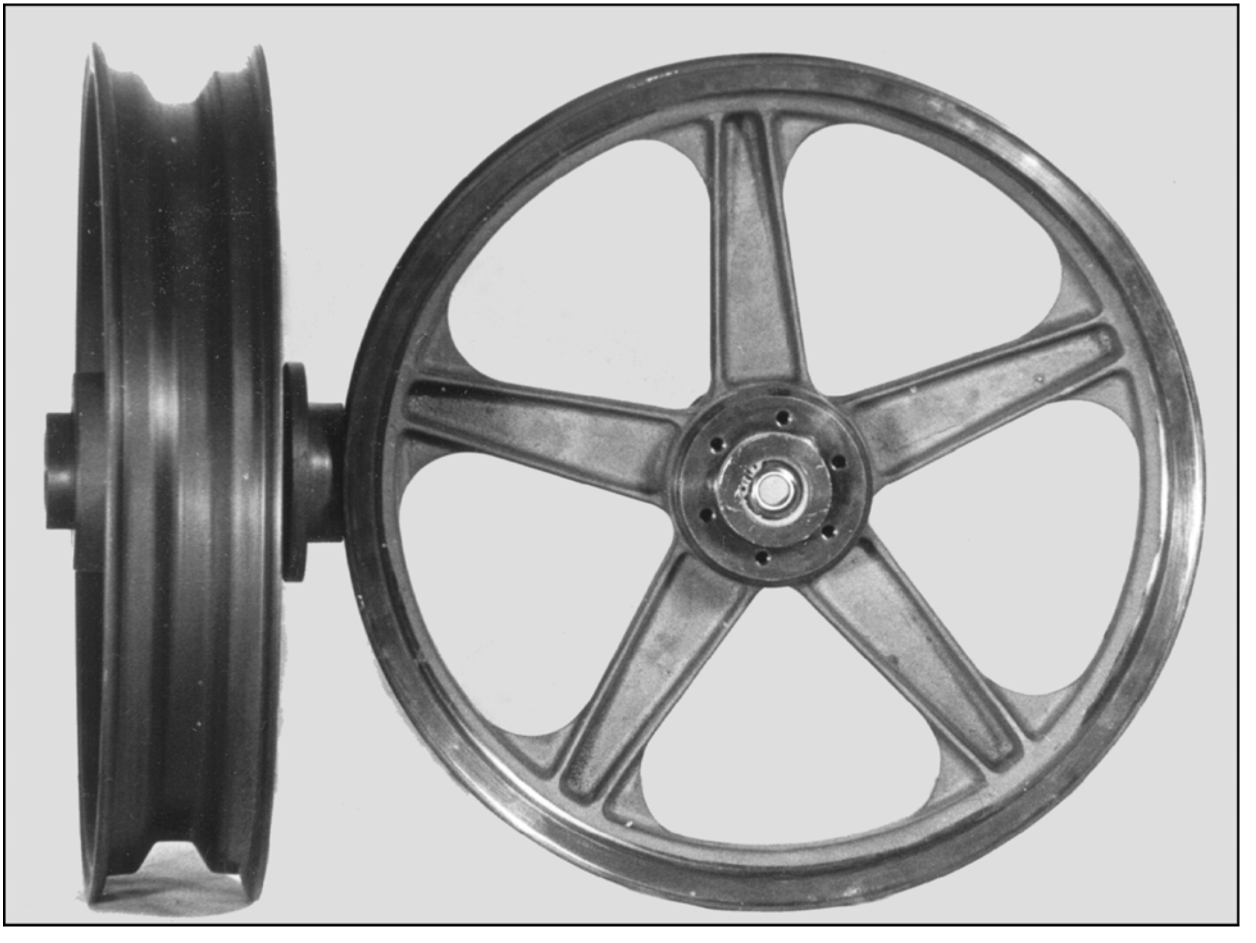

**Off-road bikes have generally stayed with wire spoked wheels as shown on this Cagiva. Despite attempts to use cast wheels, experience has shown that the wire wheel is on average more robust for this kind of duty.**

An ingenious sheet-aluminium design by Tony Dawson consists of left and right pressings riveted together at the rim and bolted to a cast hub. The higher strength of the wrought material used enabled these wheels to compete with cast magnesium for weight, while the greater ductility of the sheet material gives a high safety factor. It would be interesting to see this technique tried with magnesium sheet.

Honda developed a multi-part wheel called the *Comstar.* As with conventional wire spoked wheels this comprised separate rim, hub and spokes, but where it was different was in the spokes. In place of the normal wire spokes they used aluminium stampings and these were bolted to the hub and riveted to the rim. The rims for these wheels had a rib running around the inner circumference on which to fix the spokes.

A similar rim can also be used in another type of composite wheel. The hub and spokes are cast, like a complete cast wheel without the rim, and the end of the spokes are machined to fit the afore mentioned rib inside the rim. This form of construction is useful for low volume production “specials” because it is considerably cheaper to make patterns and have castings made and machined than for full cast wheels. It also has the advantage that various width rims can be tried without the expense of new castings. Compared to full cast wheels it has the disadvantage that the rim would not run so true, although it will likely spin truer than a conventional spoked wheel. Standard aluminium rims are available with this inner rib and the central spider could be cast in either aluminium or magnesium.

Where expense was of little concern, complete wheels have been machined from an Aluminium billet and this has become a more practical proposition for one-offs and show machines, with the spread of CNC machine tools.



**Peter Williams was a pioneer in the use of cast magnesium wheels for motorcycles, on the Tom Arter G50 Matchless. He later went into production with the wheels shown above. Nowadays 3 spokes are considered sufficient, but early wheels seldom had less than 5, some even having 9. An odd number of spokes is best as this reduces the chances of the casting cracking when cooling from the as cast state. Some manufacturers used an even number of spokes but avoided this problem by grouping them in pairs (3 pairs for a 6 spoke wheel) such that no two spokes were diametrically opposed.**

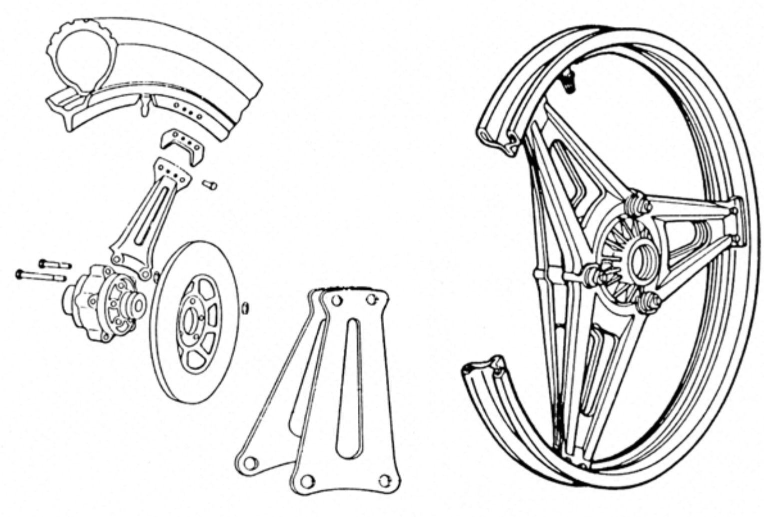
**An ingenious form of construction was employed by Tony Dawson to make the Astralite wheels.**

**Two identical pressing in aluminium alloy sheet are riveted together at the rim and bolted to a central cast hub, containing bearings and mountings for the brake disks.**

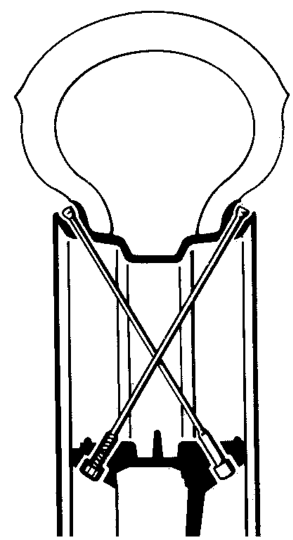
**A similar construction was later used by Bimota.**

Considering the speed of development in plastics technology, it may soon be a practical proposition to make wheels in some form of this material, thus considerably reducing unsprung mass. Such wheels are already well established on BMX cycle-type machines. Carbon fibre reinforced plastic was for a long time touted as the material of the future for wheels but some high profile and spectacular structural failures delayed further experimentation for quite a while. However, these earlier problems have now been overcome and carbon fibre wheels and rims are now a reality.

**These Dymag wheels are of composite construction. The rim is made in carbon fibre but the central spider and hub is cast in magnesium. This has the advantages that various centre sections can be installed in a given rim, and the weight reduction is in the rim which is the most important for a reduction of the rotating moment of inertia of the wheel, thus improving acceleration and corner lean-in. Such wheels have been approved for road use.**

**Honda Comstar wheel. The extruded and rolled rim has an inner flange to which the stamped spokes are riveted. The bearings and disk mountings are in a relatively normal cast hub which bolts to the inner end of the spokes.**



**A novel way by BMW to allow the use of tubeless tyres with a spoked wheel. The spoke holes in the rim are outside of the air space.**

## Fuel tank

Steel is the traditional material here for roadsters, aluminium for racers. To prevent cracking, care must be taken to isolate aluminium tanks from vibration. Plastic tanks – both glassfibre-reinforced and moulded thermoplastics (ABS or similar) – have been successfully used for competition duty (particularly on off-road machines) but the *Construction and Use regulations* forbid the use of non-metallic tanks on the public roads in Britain. As with many other components Carbon Fibre reinforced material is now making an appearance in fuel tanks also.

## Brake discs

Stainless steel is most commonly used on road machines because its freedom from corrosion preserves a smart appearance. But it has the disadvantage of poor wet-weather performance, though this can be improved by drilling or grooving the discs to break up the surface water film. Another improvement has been brought about by a change in brake-pad material. Cast iron is functionally superior to stainless steel, consequently it was widely used on racing machines and some Italian sportsters. Its chief drawback is its poor appearance in damp weather, due to rust. To reduce unsprung mass, racing machines have tried aluminium discs with a hard surface coating. This may take the form of hard anodizing, chromium plating or, more expensively, a plasma-sprayed material. Generally, such discs had a shorter life than the cast-iron variety. Because of the higher coefficient of expansion of aluminium alloys, care must be taken to prevent distortion due to temperature differentials between the disc operating surface and the hub. The best method seems to be to use a flat disc, attached in a semi-floating manner to a central carrier. To allow for the greater radial expansion of the aluminium disc there must be more clearance between its periphery and the caliper, otherwise the brake could lock when hot. High disc expansion can also cause a hard surface coating to crack if it is not of a compatible material.

These experiments with aluminium based disks have been superseded by the use of carbon disks. Carbon is strong, light and can stand very high temperatures, making it ideal for brake disks. Initially there were problems because the surface friction increased with temperature and they had to get hot before good braking performance was obtained. Different production methods for the basic carbon have largely resolved these problems. The pad material for carbon rotors also needs to be made in carbon. Apart from any other considerations, at the moment the very high cost of this material will ensure that we'll only see its use on well funded competition machinery.

## Bodywork

The use of steel or aluminium for seats, mudguards, fairings and suchlike has been largely superseded in racing by reinforced plastics. Initially this was GRP or Glass Reinforced Plastic, polyester being the plastic or resin most used. This has been over taken by the use of Carbon Fibre Reinforced Plastic, polyester has given way to the stronger and more stable epoxy resin. Carbon fibre has the advantage of having a very high Young's modulus, that is, it is very stiff. Some of this stiffness is given up when imbedded in the epoxy but the overall resulting composite material is still stiffer than most other forms of construction. This enables thin and hence light weight panels and shapes to be moulded, without undue flexibility in the finished component.

Like GRP, carbon fibre parts can be made at home or in a small workshop, but for the best results the work needs to be done with specialist facilities. The final setting or hardening of the material is done in autoclaves (ovens) and some form of pressure moulding (such as vacuum bagging) is best to ensure an even thickness and uniform matrix. It is important to expel any air trapped in the liquid resin. Working with the resins used in composite materials can be hazardous and appropriate precautions and protective clothing should always be worn.

On road machines, too, metal is being ousted for these components in order to save weight, but in this case thermoplastic mouldings are commonly used, some of which have greater flexibility, which reduces the chance of permanent damage in a minor accident. A disadvantage, however, is their tendency to look tatty in time as a result of scratching and other surface blemishes.

# 14 Stability & control

Stability of a motorcycle means different things to different people and so we need to be clear in our use of the term. There are precise mathematical definitions for the stability of various systems but these are beyond the scope of this text. Here we'll use a more qualitative version: Stability is the tendency to return to an equilibrium position (also known as a trim state) after being disturbed from that same equilibrium state. A vehicle can have more than one stable state of equilibrium, for example travelling in a straight line is such state but a motorcycle leant over at 27 degrees and in a steady turn of 0.5 G is also in a stable trim state. Control is closely related to stability because the purpose of control is generally to change from one stable trim state to another. For example, when travelling straight the rider exerts some control input to change into a stable cornering state. The greater the straight line stability the greater will need to be the control input to overcome it.

Motorcycle engineers have been somewhat slow in employing analytical techniques to stability and control investigations when compared to their aircraft and automobile counterparts. Almost from the beginnings of flight, designers studied in detail this aspect of their machines, car designers were slower off the mark but basic handling theories began to emerge in the 1930s. Largely due to the work of Maurice Olley. However, it has only been during the last couple of decades or so that much progress has been made in this direction with motorcycles.

Because of the wealth of knowledge that has been built up in the car world it is only sensible to look there for guidance on how to analyse motorcycle stability, but if we do this we find that whilst many car techniques are quite useful, there are several important differences. Directional stability is the most important aspect to consider with cars although in some circumstances roll-over stability can be important also. Motorcycles are principally concerned with directional and also balance stability, put simply; motorcycles can fall over. Under some conditions we might also be concerned with pitch stability, when a bike is in a wheelie state there is a pitch angle beyond which the machine will fall over backwards. A similar consideration occurs when performing a “stoppie”.

Above all, there are three very important differences between the two types of vehicle.

- The need for a bike to lean whilst cornering.
- The need for countersteering.
- The car can have differences in the side to side loading of its tyres. In a corner the outside wheels are more heavily loaded than the inside, this can be controlled by suspension characteristics and adds another dimension to the parameters available for tuning stability and control.

We sometimes talk about other forms of stability such as aerodynamic and braking, but strictly speaking aerodynamic or side-winds should more correctly be seen as a disturbing influence on balance and directional stability rather than a different type of stability. Braking can also be considered as creating disturbing moments that principally affect directional stability.

When subject to a side force, a motorcycle, through its trail, creates a steering effect but it is not always appreciated that the point of application of such a side force has a large influence over the magnitude and even direction of the steering torque. Fig. 5.17 shows a simple experiment with a bicycle which the reader is encouraged to repeat. These self steering characteristics are of great importance to directional stability in general and particularly to side winds, this aspect is elaborated upon in chapter 5.

## Under-/over-steer

These terms were briefly explained in chapter 2 from the tyre point of view. It is well known that both stability and control of cars is largely a function of their under-/over-steer characteristics. In general we can say that under-steer leads to stability and conversely over-steer indicates lack of stability. This generally applies to under-steer in cars in both the linear range during gentle cornering and also at the limit of adhesion. Under-steering changes with speed and the application of power, etc. Although as we shall see, there are complicating factors with bikes that can alter the relationship between under-steer and stability/control drastically.

In recent times these terms have become part of popular motorcycle jargon, but they are generally misused, particularly when considering the linear region. In chapter 2 on tyres, we saw that the rider perceives under-/over-steer basically through handlebar torque rather than steering wheel displacement, but we have also seen in various chapters that handle-bar torque is affected by many other factors. For example if a bike needs the application of considerable inward torque to keep it on line in a corner, then the rider is most likely to report that the machine under-steers, whereas that torque requirement is affected more by trail and front tyre width than it is by the slip angle differences between front and back. Therefore a rider is likely to perceive differences in feel and stability, from a change in trail, as changes in the under-steer characteristics. On the other hand, limit under-/over-steering is more likely to be correctly diagnosed but is more frequently referred to as push and loose respectively.

This is a book on motorcycles and not cars of course, but as the under-steer concept is so well developed for cars it is useful to compare the two different machine types. We'll look at two different situations and see how the vehicles cope with each. Firstly we'll look at crossing a gentle slope which will load the tyres in the linear range and secondly we'll consider changes to the turn radius when cornering near or at the limit.

### Crossing a slope (car)

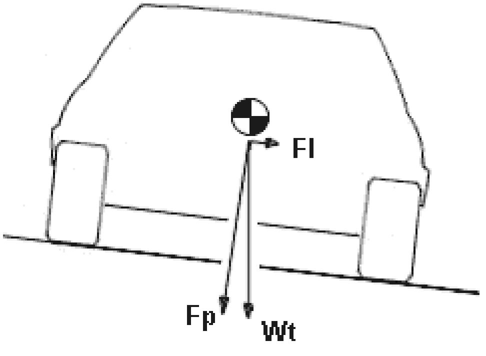



Fig. 14.1 **When a car crosses a slope it adopts the same angle as the road. The weight (Wt) can be resolved in a lateral component (Fl) and a component perpendicular to the car (Fp). The lateral component tends to make the car run down the slope and must be resisted by lateral tyre forces acting up the slope.**

Fig. 14.1 shows that the tyres must produce a lateral force to counter-balance the lateral component of the weight. We've seen in chapter 2 that in order to create a side force a car tyre must have some side slip velocity. In other words in a plan view the car must be set at an angle to the direction of travel, as shown in fig. 14.2 below, fig. 14.3 shows the different paths followed by the car, without driver control, depending on the under-/over-steering characteristics. The lateral force at the centre of gravity (Fl) causes some lateral load transfer, and the fore-and-aft distribution of this transfer is controlled by the relative roll stiffnesses at each end. This gives some control of the under-/over-steering. The paths followed as in fig. 14.3 have been used as the actual definition of under-/over-steer, in some cases. To maintain a course aligned with the road the driver must apply appropriate steering angles.

Whilst this is a simplified view of the phenomenon it is quite accurate for tyre loading in the linear range of characteristics and suitable for our comparison.

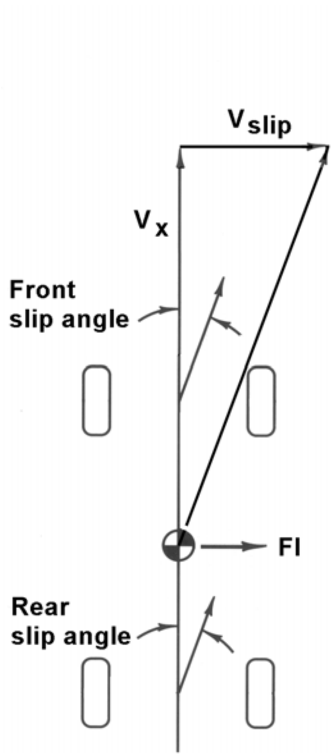
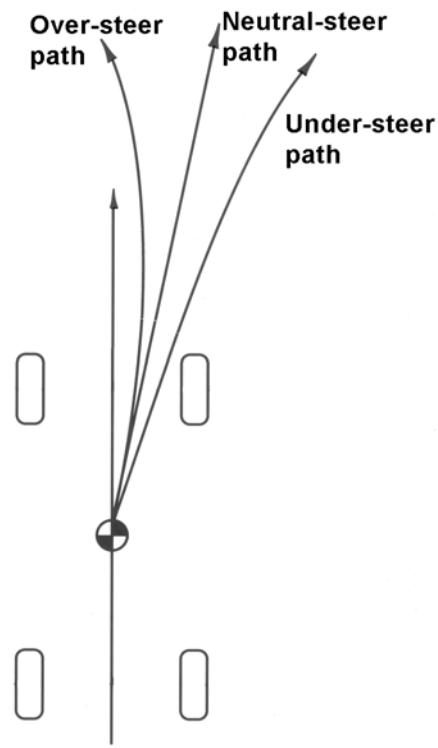



Fig. 14.2 Left.

**In order to generate a tyre force to balance the lateral force (Fl) the car must have some side slip velocity. I.e. the car must be set at an angle to the direction of travel.**

Fig. 14.3 Right.

**Without driver intervention (with the steering wheel fixed) the car will adopt different paths, as shown, in order to create the appropriate front and rear slip angles to balance the lateral force down the slope. The required slip angles depend on the under-/over-steer characteristics.**




## Crossing a slope (motorcycle)

Fig. 14.4 shows the case of a motorcycle crossing a slope and just as in the car case the tyres must produce a lateral force, but this is where the two vehicles can differ greatly. The motorcycle must keep a vertical attitude to retain balance, unlike the car which leans over to match the slope. In other words, the motorcycle adopts a camber angle relative to the road surface. Refer back to chapter 2 on tyre grip and in particular figs. 2.20 & 2.21. If both tyres have the characteristics as in fig. 2.20 with normalized camber stiffnesses equal to 1.0, then camber forces will exactly balance Fl, the force down the slope, without any slip angle being needed. Therefore there will be no slip velocity down the slope an hence the bike will just run straight on in its original direction. This lack of reliance on slip angles to produce the required tyre forces simply means that the car definition of under-/over-steering has no particular significance in this case. However, the under-/over-steer possibilities are more numerous in the case of motorcycles, depending on the relationship between the values of camber and steering stiffnesses at the two ends. In fact there are thirty three different combinations that we could consider. We have seen in this example of riding across a slope that a camber stiffness equal to 1.0 has a particular significance, therefore we need to consider three sets of values of camber stiffness, at each end. Viz. Less than 1.0, equal to 1.0 and greater than 1.0. Front and back tyres can have different values and within each possible combination we can have three different relationships between the front and rear steering stiffnesses: Front and rear equal, front greater than the rear and front less than the rear. Without the complicating effects of camber force these three combinations would define under-/over-steer, but the situation is not that simple.

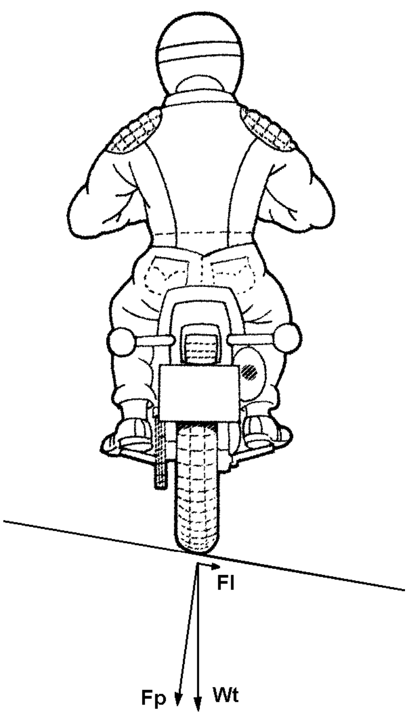



Fig. 14.4 **As in the car case there is a component of the weight (Wt) that can be resolved parallel to the road (Fl) and another perpendicular to the road (Fp). The force Fl must be reacted against by tyre forces. Unlike a car the bike remains vertical to retain balance and there is no over-turning moment created.**

It would be too tedious to consider each of the thirty three possible groups of combinations of relative camber and steer stiffnesses, suffice it to say that; if the handle-bars are held in the straight ahead position, and the camber stiffnesses are less than 1.0 then the path of the bike will depend on the under-/over-steer characteristics in the same manner as a car. If the camber stiffnesses are both greater than 1.0 then the effect of under-/over-steer will be reversed, i.e. the neutral and under-steering cases will have paths leading uphill and the over-steer case will be down the slope. Which is completely at odds with our normal conception of these terms.

## Free steering (crossing a slope)

In both the car and motorcycle cases above it was specified that the steering was held in alignment with the vehicle. This is not too far from the truth in the car case. There is usually a lot of friction in the steering system and there is a considerable gear ratio between the steering wheel and the steering angle at the road wheels, so without deliberate steering from the driver the wheels will remain aligned.

The situation with a bike is quite different, the steering is direct without any gearing and is usually with a minimum of friction. We have seen in other chapters that there is very little steering movement necessary and the principal rider input is steering torque and not displacement. Riding on a flat straight road the average steering torque can be taken as zero. In other words, the control effect on the bike is similar to riding *no-hands.* Let's compare a motorcycle and a car on a slope both with no steering friction and both driven *hands-off*. Fig. 14.5 shows a fundamental difference between the two vehicles. The motorcycle remains vertical but the car adopts the slope of the road, we shall see that this completely changes the *hands-off* steering reactions.

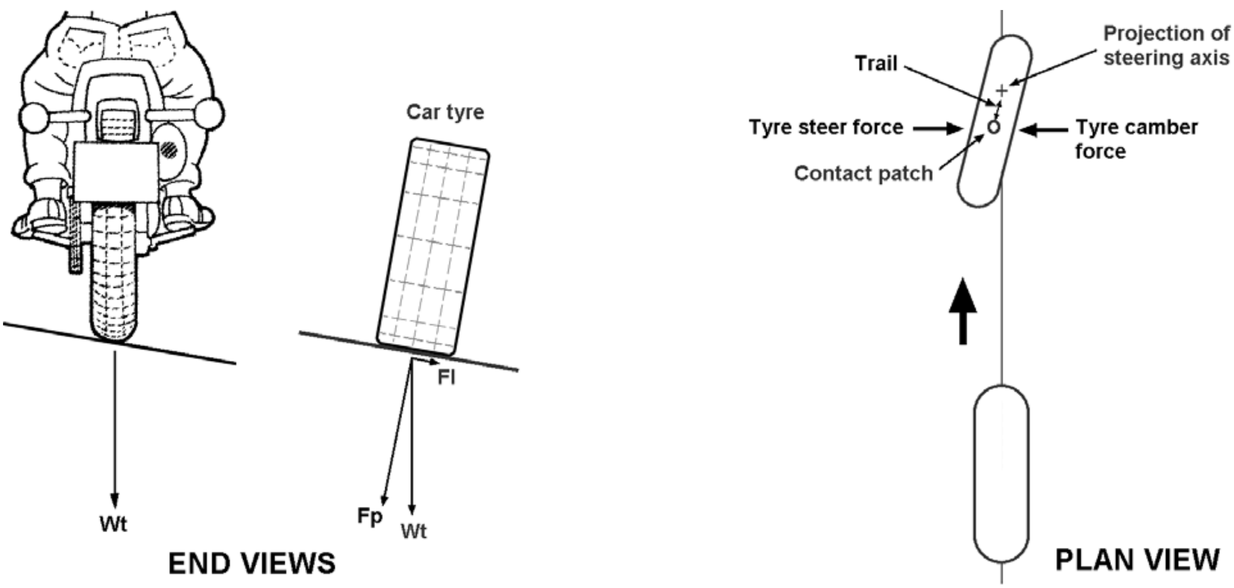



Fig. 14.5 **The end views show the difference between the upright attitude of the car and motorcycle tyres. The plan view on the right shows how the motorcycle tyre forces at the contact patch can cause a self steering effect about the steering axis.**

We can see that for the motorcycle to remain balanced it must remain vertical and hence there can be no resultant component of the weight force acting at right angles to the machine (although there is a force down the slope). The plan view shows the case of a motorcycle with a front camber stiffness greater than 1.0. Thus, the front camber force is in excess of that needed to balance the force down the slope and so the steering will be turned about the steering axis as shown. This will create a steering force to oppose the camber force. The steering will stabilize at an a angle where the total tyre force balances the weight force down the slope. If the camber stiffness was less than 1.0 the steering would automatically turn in the opposite direction and the steer force would augment the camber force until the total balanced the force down the slope. In other words the motorcycle has self stabilizing steering in this case. In reality there will be some steering friction and so the motorcycle may have a small tendency to steer either up or down the slope depending on its under-/over-steer characteristics.

On the other hand, the weight force has a residual component (Fl) that acts laterally on the car. In the absence of any driver supplied steering torque to oppose this, the wheel will steer down the slope until there is sufficient slip angle to generate enough steer force to balance this lateral force. However, unlike the bike case this slip angle will be in a direction to reinforce the tendency to turn down the slope. In practice the friction in the system will work against the free steering of the wheels. Additionally, a car normally has much less trail than a bike and so the steering torque will be reduced accordingly. The steering action of a side force on a vehicle is explained in more detail in the static stability section of chapter 5.

It is fairly clear that the behaviour of cars and motorcycles are so different in this case that any attempt to apply the accepted stability and control aspects of under-/over-steer would be of dubious value.

## Limit steering (car)

When cornering on the peak of the frictional properties of the tyres, our previous simplified definition of under-/over-steer based on the steering stiffness of the tyres is no longer valid. In this case let us consider these characteristics more in terms of the way that most people perceive them. If the front tyres are closer to their adhesion limit than the rear, then the car is considered to be under-steering and vice versa.

Imagine cornering in an under-steering car (push)at a fixed speed, with the front tyres at their limit. Then steer inwards to tighten the turn, we can see from fig. 2.15 (and surrounding text) that if we apply a greater slip angle the lateral tyre force will reduce. It follows that the front end of the car will not be able to maintain its previous course and will tend to slide out and travel on a larger circle, this in turn will reduce the lateral force needed and the car will settle into a stable turn of greater radius. As long as there is sufficient road this is a stable situation.

If we did the same with an over-steering car (loose), then the vehicle would start to turn on a tighter circle but this would increase the slip angle of the rear wheels which would then slide outwards, further increasing their slip angle. This is an unstable situation and the car would rapidly go into a spin.

## Limit steering (motorcycle)

As we might expect the situation is quite different with a motorcycle. Counter-steering and gravity see to that. As with the car above imagine cornering to the right in a limit under-steering situation at a fixed speed. If the rider steers right into the curve hoping to tighten the turn, like the car driver, then the front lateral force will likewise be reduced, but as we see from fig. 14.6 (and chapter 4) the outward moment holds the bike up against gravity and so any sudden reduction in this moment will result in a very quick crash. With the bike leant over at a large angle the gravity moment is high. However, we know that this is not the normal way to steer a motorcycle, we have to counter-steer. When we counter-steer at a slower cornering speed with some reserves of grip, the bike will start to lay down but additional lateral force will build quickly and the machine will settle down to turning a tighter curve with balance restored with a higher lean angle. When there are no more reserves of grip, as in the limit situation, there is no direct possibility of generating the additional lateral force needed to prevent the motorcycle from falling.

Both steering into or out of the limit curve results in an unstable situation, the bike will crash in each case. The limit under-steer stability criterion for cars is just not applicable to a motorcycle.

An additional interesting problem with limit under-steer is that it becomes very difficult to exit a turn from that condition.

On the other hand, the limit over-steer situation is very similar for both vehicles, but the motorcycle has the additional problem that when the rear slides out the reduction in total lateral tyre force will probably result in a fall also. In the case of the motorcycle both limit under- and over-steer are unstable situations. In general the over-steer situation is preferable as a rear slide is sometimes possible to control and recover from. Limit over-steering is often brought on by the application of engine power and so according to the friction circle concept introduced in chapter 2, there will be a reduction in available lateral tyre force, simply reducing the throttle opening with some steering correction is often sufficient to restore stability and control. Set against this possibility of recovering from a rear slide is the more dangerous possibility of a “high-side”.

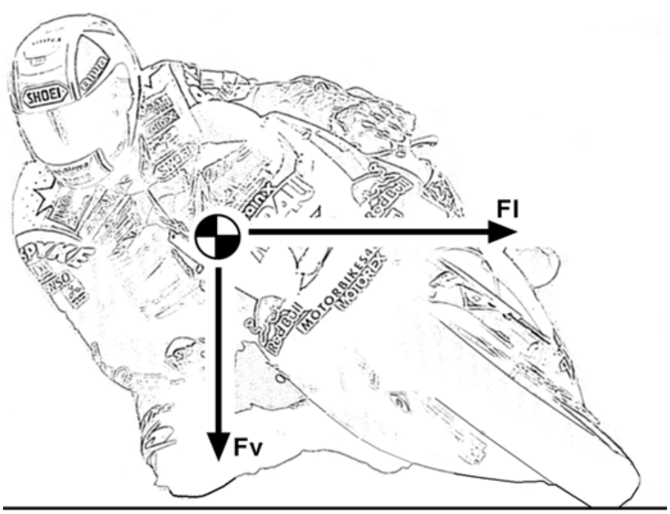



Fig. 14.6 **To keep a motorcycle balanced the moment due to the lateral cornering force (Fl) must balance exactly the vertical gravity force (Fv). If Fl is suddenly reduced then the gravity moment will pull the bike to the ground very quickly. Refer to fig. 4.3 for more detail on balancing these moments.**

## High-siding

A motorcycle is a very integrated system, i.e. changes in just one parameter usually cause effects throughout the whole machine, and so chain pull and suspension characteristics will exert some influence over phenomenon such as high-siding. There are two broad groups of high-side;

- the classic "slip and flip" which is relatively easy to understand.
- the second type promotes a series of violent weaves and this is affected by suspension settings much more than in the case of the classic "slip and flip" type of high-side. But this should really be seen as a separate type of accident, when a sudden loss of adhesion promotes a wild weave.

The classic "slip and flip" high-side is, more a tyre grip issue. The standard explanation that the tyre slips and then just grips again, throwing the bike and rider over the top, is near the mark but grossly over simplified, a high-side can occur even if the tyre continues to slide and never recovers full grip. Consider a bike exiting a corner under power close to the limit of rear wheel adhesion. Let's assume that 80% of the total possible rear tyre grip is supporting the cornering force (about 39 degrees lean for tyres with the coefficient of friction mu=1, or say 46 degrees with warm race tyres with mu=1.3) which means that about 60% of the total grip is available for delivering engine power to the ground. Refer to the friction circle to see how 80% and 60% add to 100%.

When the rider overdoes it, total tyre grip is reduced and the bike will start to slide, whereupon the rider is likely to shut the throttle, thus removing the need for traction force from the tyre, making available 100% of the reduced total tyre force (assuming tyre is still slipping) as a lateral force, let's assume that this reduced total force is say 90% of the pre-slide total force, in this case the available lateral force is now 90% whereas pre-slide it was just 80%. Therefore even without the tyre regaining its full grip it has 12.5% more lateral force available than in the pre-slide state, in many cases this would be enough to "trip" the bike up with familiar results. So the tyre didn't have to regain its full grip to cause the high-side, it is more a question that some of the potential grip originally being used for traction now becomes available for lateral grip. Of course, if full grip is re-established then the flip will likely be quicker.

If the rider fights the natural urge to shut the throttle it will most likely result in a low side crash, but then of course in many cases shutting the throttle will save the day, it is a question of knowing which. This scenario and the resulting numbers becomes even more probable as the rider straightens up and applies more power, because the proportions of the tyre grip needed for driving and cornering change, and these are just the conditions when most high-sides occur. When leaning right over using most of the grip for cornering with only a smaller amount for drive it is more likely that over-enthusiasm will result in a low side.

***** Note on tyre grip****.*

*It is often incorrectly thought that when a tyre passes the limit, then its available grip decreases to a small fraction of its pre-limit state. This is not the case, as we have seen in chapter 2, the available lateral force only decreases slowly with more lateral slip, a loss of 10% as used in the above example would actually be a lot. (The loss of adhesion against traction forces is more severe and decreases a bit more rapidly.) When we experience a slide, it occurs rapidly and it certainly feels like an almost total loss of grip, but this is a false perception, when we are truly at the limit then any reduction will allow a slide. A 10% drop will cause the backend to step out by a third of a metre in a fraction of a second."*

The second type of high-side occurs after a number of violent weaves. As we shall see later in this chapter, weave is a coupled yaw and roll oscillation that can reach high magnitudes. It is a fundamental mode of instability inherent in motorcycles and can be triggered by various external disturbances. The rapidly changing lateral tyre forces that occur when a slide occurs are sometimes sufficient to excite such a weave, which is characterized by both bike and rider being flung from side to side until the inevitable occurs.

## Stability under braking

The requirements for directional stability under heavy braking are quite involved and need careful thought. There can be basic lack of directional stability inherent in braking, and this must be countered by the rider if he is to maintain control.

Suppose a bike, while braking, is deflected from its true direction of travel, as in fig. 14.7. The inertia force ($F_I$) gives rise to an unsettling couple ($F_I.x_1$) about the front tyre contact patch. At the same time, the rear braking force ($F_R$) creates a correcting torque ($F_R.x_2$). For natural stability, the correcting torque must exceed the unbalancing one. This can occur only under low or moderate deceleration, when there is little load transfer and so the rear wheel can provide a relatively high proportion of the total braking. Heavy braking, on the other hand, involves considerable forward load transfer, so that the front wheel is required to do the lion's share, if not all, of the work, and the unsettling couple then exceeds the correcting torque – an unstable condition. If we are to reduce this tendency, we need to minimize the load transfer by means of a long wheelbase and low mass centre. A forward centre of gravity also helps by reducing the destabilizing moment by a greater amount than it reduces the rear wheel stabilizing effect. However, a forward CoG position limits the degree of braking before the bike spins forward.

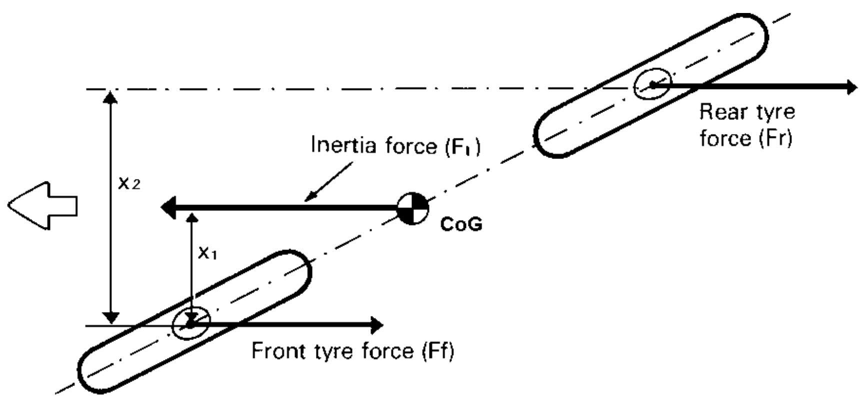



Fig. 14.7 **Yaw moments under braking. To balance the moments about the front wheel, the stabilizing moment from the rear brake force ($F_R.x_2$) must equal the destabilizing couple from the inertia force ($F_I.x_1$). i.e. $F_R.x_2 = F_I.x_1$**

To complete this analysis let us consider four separate braking conditions.

- ***Rear brake only, locked or not.*** The rear tyre force balances the inertia force and the resulting couple tends to stabilize the machine.
- ***Front brake only, locked or not.*** Here the above situation is reversed so that the couple tends to destabilize the machine. A completely locked front wheel also eliminates steering control and reduces balance stability, exacerbating the loss of directional stability.
- ***Both brakes applied, rear wheel locked.*** As soon as the wheel locks, its tyre loses some of its grip and this can have two effects: firstly, we lose some of the rear brake's stabilizing effect; secondly, because the total braking force is reduced there is less forward load transfer and, if we are already braking close to the limit, this may cause the front wheel to lock, too, with disastrous results.

- ***Both brakes applied, front wheel locked.*** Here we have strong directional stability from the still-braking rear wheel; also the reduction in load transfer restores some load to the rear so permitting increased braking at that end. Despite the increased directional stability the stationary front wheel loses steering control and balance stability as above.

All these cases refer only to directional stability while braking in a straight line. The situation changes drastically when negotiating even a mild bend while braking. As soon as either wheel locks then, it loses some of its ability to provide the sideways centripetal force to balance the load of cornering, and the wheel slides outward. Depending on the rider's skill (or luck) it is usually much easier to recover from a rear-wheel slide than a front one. As explained earlier, the balance of a single-track vehicle depends on the precessional and steering forces of rotating wheels, especially the front one – hence the desirability of not locking the wheels. The situation of a locked front wheel and a braking rear wheel, though directionally stable, is unstable so far as balance is concerned. Unlike cars, where all four brakes are controlled by a single pedal and the front/rear balance is pre-determined by the manufacturer, most motorcycles have separate controls for the front and rear brakes in the questionable belief that the rider himself is best able to judge the required balance in all circumstances. This may be so for a highly skilled rider but is unlikely for the less expert. Moto Guzzi, for one, introduced coupled braking in an attempt to reduce the degree of expertise required. In their system, the pedal controls the rear brake and one of the two front brakes, while the other front brake is controlled by the handlebar lever. Thus, additional front braking is available if required. Since the degree of forward load transfer depends on how hard the brakes are applied, it is clear that optimum braking calls for a greater front/rear bias in dry conditions than wet. Motorcycles are more critical in this regard than cars because of the larger degree of load transfer due to their higher CoG height to wheelbase ratio.

## Roll change when braking

It is often observed that a motorcycle tends to stand up in a corner when the brakes are first applied. Although this is a topic that seems to promote much discussion the explanation is really quite simple. When leant over, as we have seen, the tyre contact area moves sideways away from the steering axis toward the inside of the turn. When a braking force is applied through this offset contact area, it will generate a steering torque into the curve. Due to the counter-steering control of a motorcycle this will cause a roll reaction in the opposite direction, i.e. the bike roll angle will tend to decrease. This is much more pronounced when using the front brake but a greatly reduced effect is still in evidence when only braking at the rear. The rear yaw torque applies a lateral force at the steering axis which steers the front wheel into the turn also, but this is of minor importance.

The reduced speed after braking will also require a lesser lean angle, but this is a relatively slow acting requirement and is under the control of the rider, it is the sudden application of a steering torque when first applying the brakes that is mainly responsible for the tendency to stand up, prior to the rider applying a counter-balancing handle-bar torque.

# Instabilities

Over the past two or three decades there has been quite a bit of academic research aimed at discovering the reasons and possible cures for certain types of motorcycle instabilities. This is undoubtedly because it is easier to get research funding for issues that directly affect some easily identifiable aspect of safety. Manufacturers are always concerned about liability litigation and want to be seen to be involved in such work and government grants are more likely to be forth-coming for such projects rather than research into how to make motorcycles turn quicker. Although any work that leads to handling improvements can arguably be considered as likely to have a safety benefit also.

There are generally considered to be three basic modes of instability inherent in the motorcycle layout. These are fundamental to all motorcycles and whilst they may not ever be noticed in practice with any particular machine, the possibility is always there, but may be kept in check by various forms of damping. Sometimes it only needs some wear or lack of proper adjustment at critical components, like swing-arm pivots and steering head bearings, for dangerous instabilities to occur. These three modes are known as:

- Capsize.
- Wobble. An angular oscillation of the steered mass about the steering axis.
- Weave. A complex roll/yaw oscillation of the main part of the machine about the steering axis.

## Capsize

This ranges from the rather trivial and obvious case, that a stationary motorcycle will simply fall over unless held upright, to the slow speed case where the machine will start to turn and lean to one side in a gradually tightening curve until it falls over also. This can be observed by gently pushing and releasing a rider-less bicycle, as it slows down it will veer to one side and as the curve radius reduces it will eventually fall.

## Wobble

Trail is generally thought to provide a stabilizing influence to the steering, it is ironic then that it can itself be the cause an oscillating, or wobbling, type of instability. This happens if, when the front wheel is displaced by some road irregularity, the restoring torque created by the trail happens to be strong enough to over-correct for the initial disturbance. The wheel will then swing beyond the straight-ahead position and will be steering in the opposite direction. This in turn creates another but opposite restoring force, which repeats the whole process, and we have a side-to-side steering wobble caused by the trail. The above is a simplified explanation of a phenomenon known as harmonic motion; the essential elements are a mass free to move but restrained by a form of spring or other means which generates a force (dependent on the displacement or velocity) tending to restore the mass to its rest position or beyond. The best example of this that has some similarity to our steering problem is the pendulum. Consider figure 14.8.

As we move the pendulum to one side, gravity creates a force that starts to accelerate the mass back towards the centre; but, by the time it gets there, it is moving fast enough to carry on to the other side and the whole process continues, with the pendulum swinging from side to side. If we wait long enough, the extent of the swings will progressively diminish until the pendulum eventually comes to rest. This is due to damping, which may take many forms. In the pendulum's case, the damping comes mainly from air resistance and friction in the pivot. The amount of damping determines the number of swings before the pendulum comes to rest. It is possible, however, to have an oscillating system that doesn't come to rest in this way or which actually increases the magnitude of successive swings (sometimes until the system is destroyed). For this to occur we need negative damping, which may also be termed a forcing function. This simply means the application of some added impetus to the system at the appropriate time. In the case of a pendulum in a clock, the impetus comes from a spring (or a weight on a chain) which, through the escapement mechanism, gives the pendulum a kick on each swing to ensure that it does not come to rest. A child's garden swing is just a special case of pendulum. We have only to give it a gentle push each time it reaches a peak for the amplitude of the swing to increase very quickly; in extreme cases the swing may even be made to go over the top. If the gentle push is timed wrongly, however, it is surprising how quickly the swing can be brought to rest. (No apologies are made for this lengthy discussion of swings and pendulums because a basic knowledge of simple harmonic motion is essential to an understanding of the causes of several motorcycle problems and of the function of suspension systems.)

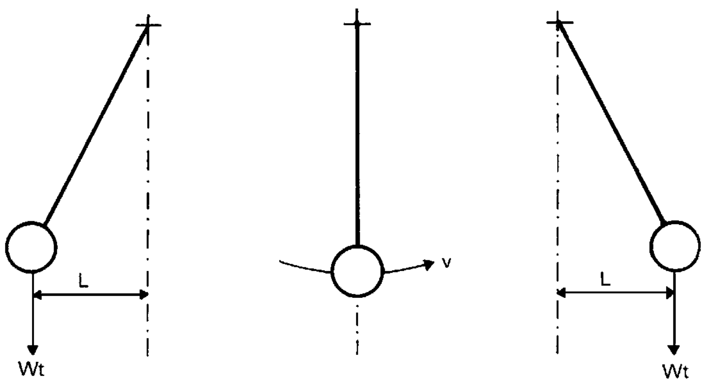



Fig. 14.8 **When the pendulum is displaced to the left by a distance L and released it will return to its central stable position, but by then it will have acquired some velocity V, which carries it past, until, in the absence of any damping, it reaches a distance to the right of L also. This process will repeat itself.**

Now let us revisit the concept of resonance, first mentioned in chapter 6 on suspension. Experiments with our pendulum or swing will soon show that (except for extreme angles of swing) the number of complete oscillations in a given time is almost constant, regardless of the amplitude of the movement. This is called the resonant frequency (or natural frequency) and is mainly determined by the length of the pendulum. As we have seen, if our push (forcing function) to the child's swing is correctly timed (i.e. in phase) and at the natural frequency it is easy to build up high amplitudes of oscillation. However, if the forcing function is either out of phase or of a different frequency, then the oscillation will be much less inclined to build up and may even be heavily suppressed.

If we apply the principles of simple harmonic motion to a motorcycle we find that our wobbling front wheel has a natural frequency determined mainly by:

- The moment of inertial of the front wheel and fork about the steering axis.
- The magnitude of the restoring torque due to a given angular displacement, this is determined by the rake, trail, tyre size and characteristics, and to a lesser extent the stiffness of the frame and fork.

The *higher* the steered moment of inertia, the *lower* the natural frequency, and the *higher* the restoring torque per degree of deflection, the *higher* the natural frequency. The forcing function trying to build up a wobble on a motorcycle results from unavoidable imperfections in the machine. A small unevenness in a tyre or a slight buckle in a rim will give a kick to the steering at each revolution of the wheel, as will any wheel unbalance that is offset from its centre plane. The frequency of these forcing functions is that of the rpm of the wheel (or a multiple of the rpm); and if our speed is such that this coincides with the natural wobble frequency, then the bike may develop a bad wobble at this speed.

Several factors contribute to the fact that not all bikes wobble in this way – and damping is an important one. Not only the deliberate damping provided by a steering damper but also damping inherent in the wobbling system, such as friction in the steering bearings, wiring and control cables, tyre friction and internal tyre damping (hysteresis); often the decisive factor is damping by the rider's body through his contact with the handlebar. Ultimately, the natural wobble frequency, road-induced forcing function and damping fight it out together and the result determines the wobble characteristics of the machine. Some machines are free from this type of wobble, yet on others it is difficult to eliminate.

It usually occurs between 40 and 65 km/h and is felt most strongly, sometimes violently, while slowing through this range with the hands removed from the bar. As an example, the old BMWs with the Earles-type pivoted front fork (pre-1970) could develop a pronounced wobble at about 55 km/h, especially when carrying a pillion passenger or heavy luggage at the rear. An hydraulic steering damper effected a cure without any adverse effect. A most important point to bear in mind is that the fundamental mechanism for causing this sort of wobble is inherent in the layout of a conventional motorcycle, the only way to prevent it is to damp or tune it out of the system and the following measures may be helpful:

- Increase lateral fork stiffness.
- Reduce the trail; there are limits to this approach as it may impair directional stability.
- Reduce the mass of the front wheel and fork, so reducing their moment of inertia about the steering axis. This cuts down the energy in the oscillating parts for a given magnitude and frequency of wobble, so that the inherent damping has a greater proportional effect. This reduction of inertia also raises the system's natural wobble frequency.
- Fit an hydraulic steering damper.

In many cases wobble is hardly noticed when the rider has his hands on the bars, and only becomes a problem when riding "hands-off". The rider provides a measure of damping when his hands are in place.

## Weave

As *wobble* is an oscillation of the front steered mass about the steering axis, *weave* is considered an oscillation of the rear steered mass (the main part of the bike and rider) about the steering axis. It is a much more complex form of instability, because there is a lot more cross coupling between the various degrees of freedom of possible motions.

There is some degree of cross coupling with *wobble,* for example the oscillation will create relatively high angular steering velocities which in turn through gyroscopic reactions will produce oscillating roll torques, but because the roll inertia is relatively high the response to these torques will be relatively small at the wobble frequency. Therefore the wobble instability is principally a question of steering wobble only. Such is not the case with weave. The weave frequency is lower which allows for a greater roll response, which in turn will cause greater gyroscopic steering torques on both wheels. The whole motion is a complex combination of yaw, roll, suspension and steering displacements. It is generally more dangerous than wobble because it occurs at higher speeds although the oscillating frequency is lower. It might be thought strange that the weave frequency is not tied to the rotational frequency of the wheels, but this is a result of the gyroscopic influence. Appendix 4 shows how gyroscopic reactions are tied to the rotational velocity of the wheels.

Weave can represent extremely violent manoeuvres and structural compliance can be a significant parameter affecting the onset and severity of the instability. Thankfully most riders never experience a full blown weave and some bikes are known to be much less susceptible to it than others, but on occasion some particular combinations of riding conditions and road disturbance will excite a violent reaction often resulting in a crash. As mentioned above about high-siding, the slip and grip of the tyres can be such a stimuli.

The tendency for a bike to weave is also closely tied to various setup parameters such as tyre sizes, rake, trail, weight distribution etc.. Figs. 6.28 – 6.31 show how an under-damped rear end can result in a weave initiated during corner entry. In that example the weave was gradually damped out, showing that not all such instabilities diverge into catastrophe.

Other oscillatory systems are possible on a motorcycle and their interactions affect the overall wobble and weave stability characteristics. With most heavyweight touring machines, for example, if we stand alongside and turn the steering rapidly from side to side, then the rear of the machine will develop a surprising sideways oscillation, usually more pronounced if panniers and luggage are fitted. Close examination will reveal that most of the movement takes place in lateral flexure of the rear tyre. This oscillation has its own natural frequency, affected by passenger and luggage weight and tyre properties. If this (or other) resonance coincides closely with the weave frequency, then the overall effect may be greatly intensified.

To look at these subjects adequately requires complex computer simulations and as with stability of various control systems in general it is difficult to discuss the finer points without mathematical techniques outside the scope of this book. Any reader anxious to learn more is recommended to review the academic papers published by organizations like the SAE. There has probably been more written on this aspect of motorcycle behaviour than on any other.

## Damping

As we saw in chapter 6 on suspension, damping can be used to control and reduce tendencies for systems to oscillate. The same applies to the instabilities mentioned above. The difference is that with suspension damping we have control over it through selection and adjustment of purpose built hydraulic dampers. Although adjustable hydraulic steering dampers are sometimes fitted there are many less obvious sources of damping that are important to the control of wobbles and weaves. The rider's body is one such source. Made mainly of soft tissue there will be some internal energy absorption as the rider is generally shaken about. Remember that the principal property of a damper is to remove energy from a system.

Another very important form of damping has its source in the characteristics of the tyres. In chapter 2 we looked at one form of tyre damping called hysteresis, but in the current context that is of less importance than "yaw damping". We have seen that tyres provide a lateral force through a mechanism of a lateral sliding velocity. In fact the lateral force is proportional to the sliding velocity over much of the range of characteristics and is in opposition to the direction of the sliding velocity. These are exactly the characteristics of a viscous damper, and hence a tyre behaves just as a damper against lateral or yaw motions. This is known as yaw damping and is very important to the general wobble, weave and directional stabilities. It is probably fair to say that without yaw damping, motorcycles would be harder to control and much more likely to develop serious instabilities. Tyre relaxation length is important in this context too.

# 15 Performance measurement

Since the earliest days engines have been subject to quantitative performance testing. We've long had dynamometers with which we can measure the power and torque produced under various operating conditions. Also cylinder pressures, flow benches for gas flow and exhaust gas analysers have long been used. This ability to measure and analyse performance has in no small way been a prime reason behind the incredible development and increase in power and drivability of the internal combustion engine. Unfortunately, until quite recently there has never been analogous measuring facilities available to access the performance of the overall chassis. This hasn't only been due to the lack of measuring devices, perhaps the main problem is defining just what should be considered as good chassis performance.

In the car world, the analysis of various aspects of handling have been evolving steadily since the 1930s and there are various standard tests and performance indexes. Probably because the job is harder and the market smaller, but similar analysis has been much longer in the coming for motorcycles. Initially, testing was primarily concerned with structural and fatigue aspects, although suspension dynamometers were used in the development of dampers. Manufacturers wanted to know if frames would break, much of this physical testing has now been replaced with computer based structural analysis using a technique known as finite element analysis (usually abbreviated to FEA). It is much cheaper to play with different designs on a computer than to make and test new ideas physically. Poor design can be improved or eliminated before a prototype needs to be constructed.

From around the early 1970s. considerable academic attention has been paid to some stability aspects of the overall handling equation, wobble and weave mainly. Probably the principal reason that much research has been so closely focussed is due to the safety implications. Funding has been available from government safety agencies and also from manufacturers increasingly worried about the escalating costs of large settlements in liability litigation. Funding for similar research into the performance aspects of handling has not been so readily available, even in the ultra competitive atmosphere of racing there has been little fundamental analysis of the features necessary to give good chassis performance.

That is not to deny that improvements have been made, but this is largely the result of empirical methods and much trial and even more error. Over the past decade such methods have been given a huge boost through the application of on the track data measurement and recording techniques. Various sensors measure some physical parameters and store this information electronically so that it can be downloaded into a computer for subsequent analysis, after a race or other test sequence. Sometimes instead of onboard data storage, the information is relayed directly to computers by means of a radio link. This is termed telemetry, a word often incorrectly used to refer to data recording in general. In racing these developments have proved to be an invaluable tool to aid the detail tuning of suspension and geometric parameters. Engine performance is monitored also but these aspects are not of direct interest to us in this text.

## Track side

There are many more parameters that can be measured than those that are actually routinely measured. For track side setting up the most important are probably:

- Both wheel speeds.
- Rpm.
- Front and rear suspension movements.
- Throttle position.

An essential addition to these is a lap marker. This puts a marker into the recorded data to give a reference point that is the same for each lap. In addition to its use as a reference for the data it is also used to provide on-board lap time information for the rider. Physically the lap marker consists of a bank of infra-red light sources placed on the pit wall or tripod and aimed across the track. As the bike passes, a sensor on board detects the beam as it passes, and stops and starts the lap timer and marks the recorded data. To prevent interference with other bikes, the light beams are pulsed at quite high frequency and each team uses a different pulse frequency, thus only the correct marker is used. Multiple light sources can be positioned at different locations around the track to provide split timing.

Wheel speed is measured by means of optical or magnetic sensors which get triggered a fixed number of times per wheel revolution. There are a variety of methods that can be used for this, for example inductive pickups can be triggered by the heads of steel bolts used to hold the brake disk to the wheel, hall effect magnetic pickups need a small magnet to pass and these can be inserted into the heads of these disk bolts also. The analysis software will measure the elapsed time between each pulse and will calculate the tyre surface speed depending on the specified tyre rolling radius. For example, if the rolling radius of the tyre is 300 mm. then at 150 km/h the wheel rotates 22.1 times per second. That is, each wheel revolution takes 45.2 msecs. So if we get two pulses per wheel revolution then 150 km/h is represented by 22.6 msecs. between pulses, and if we use six pulses per rev. then the elapsed time will be 7.5 msecs.. Normally the times between several pulses are averaged to provide a more reliable indication of speed.

Rear wheel speed can be compared with the front to provide information about tyre slip under power and braking, although this is only valid when the front is firmly planted on the ground. If the bike wheelies under power then the front wheel will slow and hence be an unreliable indicator of real ground speed. True ground speed sensors are available and are widely used with racing cars.

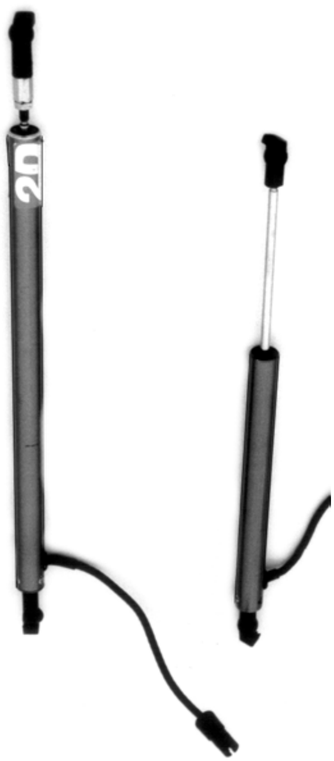
Throttle position is determined by means of a potentiometer (normally rotary) fixed to the carburettor operating mechanism. Principally used with engine parameters it is useful nonetheless to correlate with suspension displacements, helping to indicate areas of acceleration and braking.

Front and rear suspension movements are usually measured with linear potentiometers. At the front they are normally mounted to the lower fork yoke and the slider near the axle, but at the rear there is a wide variety of locations possible. Basically the sensor can be mounted between the main chassis and the swing-arm or mounted across the suspension unit. Depending on the angle of mounting the first method gives a good idea of wheel movement without any distortion due to suspension linkages etc., in general this is probably the most useful information. On the other hand when the sensor is mounted directly to the suspension unit it does not show us exactly what the wheel is doing, but it gives information that would probably be more useful to the manufacturer of the dampers.

The wheel movement potentiometers are made in a range of lengths and available travel to suit different installation requirements. The signals fed into the recording system are of the suspension displacement but can be processed by the analysis software to give suspension velocity. As shown in the suspension chapter, damping is dependent on damper velocity which thus gives an idea of the forces generated.

Linear potentiometers are a very simple and cost effective way of measuring suspension displacements, but there are much more expensive non-contact laser powered sensors available.

The data derived from suspension monitoring is invaluable for tuning a bike to specific circuits etc.. The overall range of suspension movement can be a very good guide to required spring rate, for example if the range of actual displacement is much less than the available movement it is an indicator that the spring rate is too high. If the range of movement tends to be all near the limit of either full compression or full extension then it is likely that the spring preload is in need of adjustment. There are many features in the data display that experienced people can use to help tune damping settings too. If the dive under braking occurs too quickly then increased bump damping at the front might be a solution but this may have to be compromised against damping requirements in other parts of the track.





**On the left are two different length displacement potentiometers typical of those used for suspension movement measuring. A typical installation on front forks is shown on the right. Note also the plastic ring around the slider tube, this is a low tech but very effective way of quickly detecting the maximum suspension compression that has taken place.**

Data can be collected on parameters indicative of steering and lean-in performance, although at the time of writing this is not very common, this might include:

- Steering angle.
- Steering torque.
- Roll rate.

Steering angle is relatively easy to measure, one possible method would be by the use of a rotary potentiometer. Such devices typically have a movement range of up to 270 degrees which is much higher than the total steering movement and so to improve accuracy and sensitivity it is sensible to “gear-up” the connection. This is easily done by means of small pulleys and a rubber belt drive.

Steering torque could be measured in a variety of ways depending on the manner of handle-bar fixation. For clip-on handle-bars the most suitable is probably by fitting strain-gauges near the mounting to the forks. Strain-gauges produce a signal which is dependent on the actual surface strain of the object to which they are attached. With prior calibration this information can be converted into steering torque.

Roll rate can also be measured by different methods, the most direct being by gyroscopes. Lean angle is difficult to measure directly but we can use the roll velocity and time information to calculate the angle by a mathematical technique called integration. Basically this sums the small angles moved in small time intervals during which we assume a constant roll velocity, however, there will always be small errors inherent in each calculation and so when we sum many time intervals these errors can easily add up to a value greater than the actual lean angle that we are trying to calculate. There are techniques that can be used to reduce these errors but accurate determination of lean angle is not without problems. This is not likely to be too much of a disadvantage in practice because knowledge of the instantaneous lean angle, whilst of general interest, is not really an important parameter for the purposes of setting-up a chassis. On the other hand, the roll rate, which can be measured, is a good indicator of lean-in performance especially when used in conjunction with measured steering torque.

Temperature measurement has obvious applications in relation to engine monitoring, and some form of contact measurement is both appropriate and easy to execute. However, there are three obvious applications for temperature measuring of chassis related parameters but only one of these is suitable for contact sensors.

- Suspension units.
- Brake rotors.
- Tyres.

It is only the suspension units that can be monitored by means of direct contact. Both the tyres and disks are rotating, which makes the job somewhat harder. For these applications it is usual to employ infrared detectors. A hot object radiates heat in the form of infrared radiation, the frequency of which is dependent on the surface temperature. Therefore, detectors can be placed to look the brake rotors and tyre surface without actually touching. This sounds easy but achieving the desired result is somewhat harder because the nature of the surface and type of material has a large influence over the infrared frequency for a given temperature. So the processing of the data must take these factors into account to obtain accurate results.

Many other parameters can be measured and whilst some would be of little use for track side setup, they may be invaluable information for refining component design. The measurement of material stress and flexure comes into this category. As mentioned earlier this is usually done with strain-gauges, physically these are small flat sensors with two wires attached, connected in a bridge circuit their internal resistance changes with flexure. Depending on the Young's modulus of the material the stress levels can be related to the output of the strain-gauge system. This information can be used to determine potential weaknesses as well as "over strong" components. Fundamental sources of poor handling can sometimes be pin-pointed by flexure information, for example excessive fork twisting can easily be picked up.

In a racing context the rider is arguably the most important component in the whole system and so it would be foolish to ignore the potential benefit that data acquisition could have for tuning the rider. Some of the measured parameters discussed above can be useful in this regard. The recorded data can highlight differences between riders, for example braking points and acceleration can be compared and referenced against split timing in various parts of the circuit. The differences in bike response between smooth riders and those somewhat more forceful in their approach can be easily distinguished by experienced eyes, apart from lap times such things are important for tyre life etc.. A very interesting possibility that I expect will see increasing use in the future is provided by GPS (satellite Global Positioning System), with this the exact position of a bike on a circuit can be mapped on a lap by lap basis. It will therefore be possible to compare corner lines between riders in a team and correlate these to split timings, a rider slower in a particular part of the track can then try out alternative lines to try and improve his own performance.

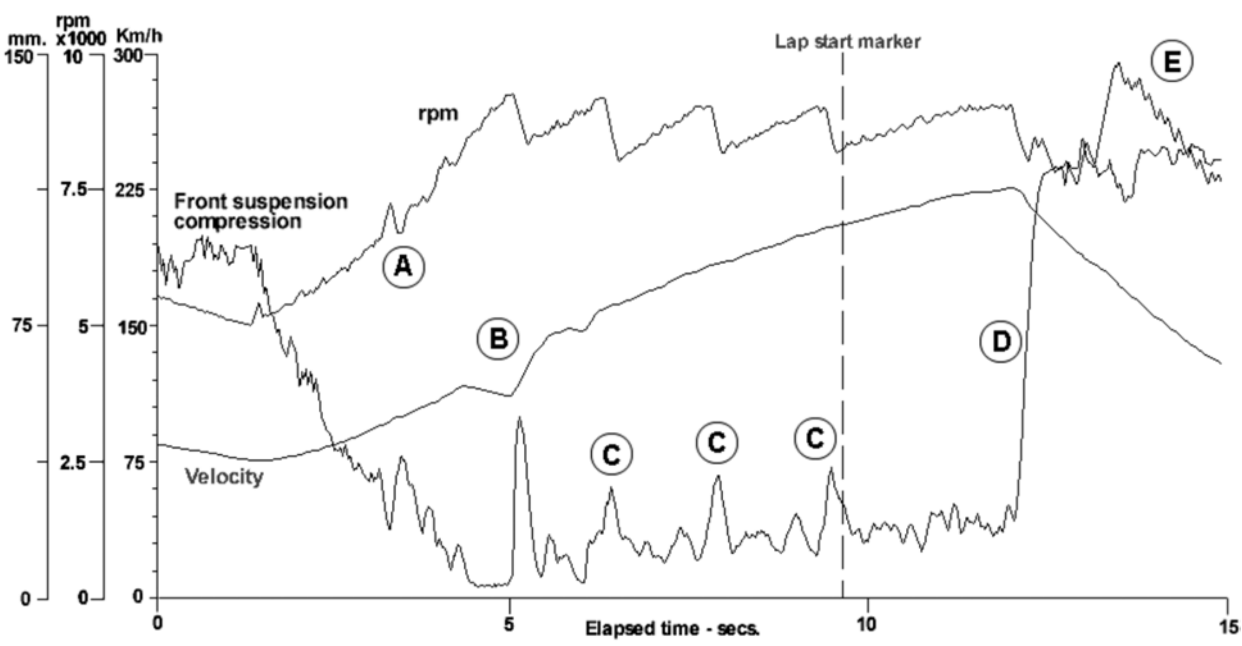



Fig. 15.1 **This 15 seconds of track data with only 3 parameters shown can yield an enormous amount of information when you know where to look.** (From data supplied by 2D)

Fig. 15.1 shows a typical sample of actual recorded data, for clarity only three parameters are used. In reality the analysis software will display the different parameters in colour and so more can be shown without interpretation difficulty. This example shows a motorcycle in the latter stage of a slow corner (75 km/h) then accelerating up to 225 km/h and then braking for the next corner. Let's look at some of the behaviour that these curves demonstrate.

Starting at the beginning of the data, the fork is compressed by about 95 mm., this is due to the extra load when cornering hard as explained in the suspension chapter on page 6.30, this is also when the bike is at its slowest speed. When the rider accelerates coming out of the bend, load transfers to the rear and so the forks start to extend but notice that just after 3 secs. **(A)**, there is a blip in both of the suspension and RPM curves. What happened here is that the rider applied just a little too much throttle and the rear tyre exceeded its maximum grip momentarily, causing the bike to slide out. The revs. rose quickly and the front fork extended briefly, as grip was regained the revs. dropped as the rear wheel adopted the road velocity and the suspension recovered.

At around 4.5 – 5.5 seconds **(B)**, we see that there is a dip in the front wheel velocity curve, this is a "wheelie" occurring. As the wheel leaves the ground it begins to slow, due mainly to air drag and residual brake friction, as it touches the ground again it is forced to regain speed much quicker. If we look at the RPM curve we can see that the wheelie starts as the engine nears maximum revs. and gets into the power band, the saw tooth effect of this curve shows the gear changing points, the wheelie begins to end as soon as the rider changes from second to third gear and power is removed. Returning to the velocity curve we notice a second smaller wheelie, but interestingly it ends before the next gear change.

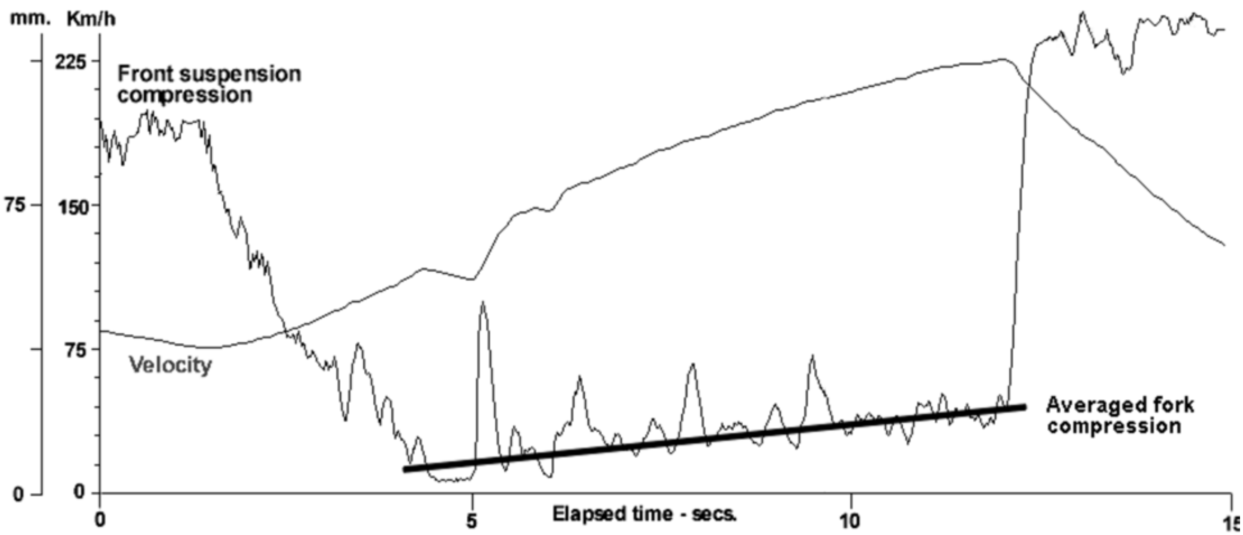
The fork compression also bears witness to the wheelie. Between about 4.5 – 5 secs. the fork compression curve is flat, this shows the topped-out position of the fork with the wheel airborne. Immediately afterwards there is a sharp spike which indicates the fork compression as the wheel hits the ground again. Note that from this time up to nearly 10 secs. **(C)**, there are three other compression spikes which coincide with each gear change. As the power is removed during the gear change period the rearward load transfer due to acceleration ceases and some load returns to the front, causing these short duration fork compressions.

Moving along to 12 – 12.5 secs. **(D)**, the speed begins to drop and the forks compress rapidly, this is the start of the braking period for the next bend. Near the end of the data **(E)**, there is a sudden rise in the RPM to a value higher than the previous gear change points, this rise occurred because the rider changed down a gear and released the clutch too early, over-revving the engine.

We can extract even more information from this data, by measuring the slope of the velocity and suspension curves at different points we can calculate the acceleration, braking deceleration and damper velocities. The latter being of importance to setting the suspension. Fortunately this type of calculation is rarely necessary because all good analysis programmes will perform these calculations for you.

It is all very interesting to have this data to see, as we have done, how the bike is behaving, but to justify the effort it has to be of help to get a race bike around a track quicker. This is usually done following feedback from the rider, when the cause of specific problems is sought. The data sample used above showed no particular problems, we can see that the forks only top-out when the wheel was in the air and under braking the maximum compression was about 110 – 120 mm. This should be compared with the available fork movement to check how close we are to bottoming the forks under the worst conditions.

The upper chart in fig. 15.2 uses the same data set as above to extract even more information. The lower chart is from a different bike at a different circuit, but the range of velocity is similar, from about 75 km/h to 225 km/h and both show the machines accelerating away from a slow corner. It is interesting to compare the superimposed line of the averaged fork compression during the acceleration period. The two bikes exhibit quite significant differences, the upper chart shows a gradually increasing front suspension compression as speed rises, whereas the lower data shows no similar tendencies.



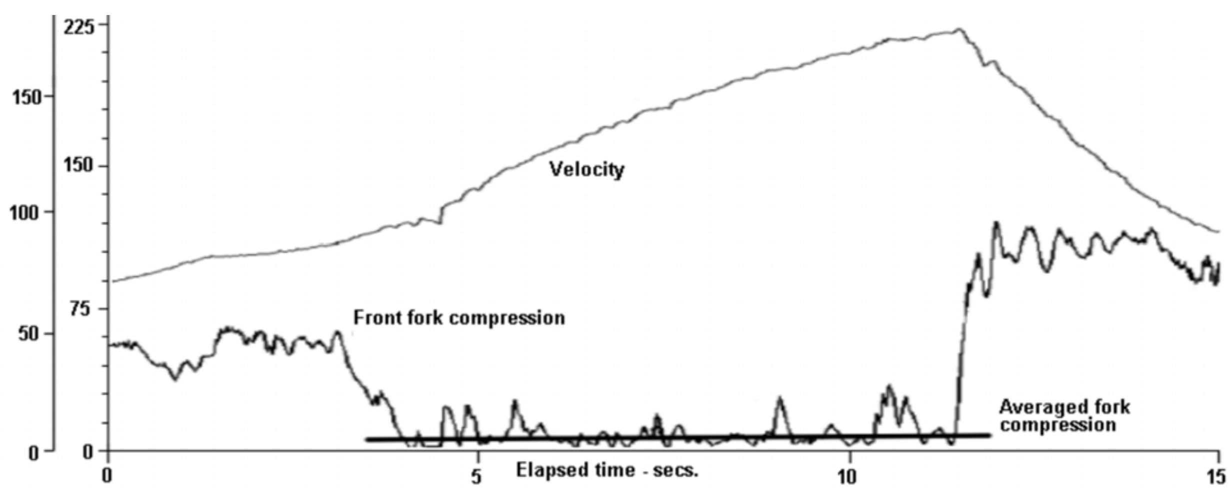




Fig. 15.2 **Measured data from two different race bikes. Note how the average fork compression increases with speed in the upper chart, from 112 to 225 km/h the suspension compresses by an extra 16 mm. The lower machine appears to have an unchanging front ride height, in the same speed range. Load transfer, suspension settings and aerodynamic effects all play a part in this aspect of chassis performance.** (From data supplied by 2D)

There is a 16 mm. extra compression, in the first case, as the speed doubles from 112 to 225 km/h. Under acceleration like this there are two main influences on front fork load which change with speed.

- As speed rises the acceleration reduces and with it the rearward load transfer, hence restoring some load to the front.
- Aerodynamic effects. The drag induced lift, explained in chapter 5, increases with speed and will unload the front, but this may be enhanced or reduced by the presence of other lift or downforce.

The combination of these two effects determines the variation in front load with speed. If we knew the appropriate physical data for this machine, we could easily calculate the suspension movement due to the changing load transfer. (*Comparing the amount of front dive under braking with the compression under acceleration the author has estimated that without aerodynamic effects the compression would have been around 60 mm. not the 16 mm. measured, thus leading to the conclusion that there is a considerable nett front aerodynamic lift. This is only an estimate but the owners of the machine should have the information to enable more precise calculation*.)

Load is not the only factor that determines fork movement, preload and spring rate are important as well and the lower chart in fact seems more indicative of excessive preload than it does of the load transfer and aerodynamic forces balancing each other so exactly.

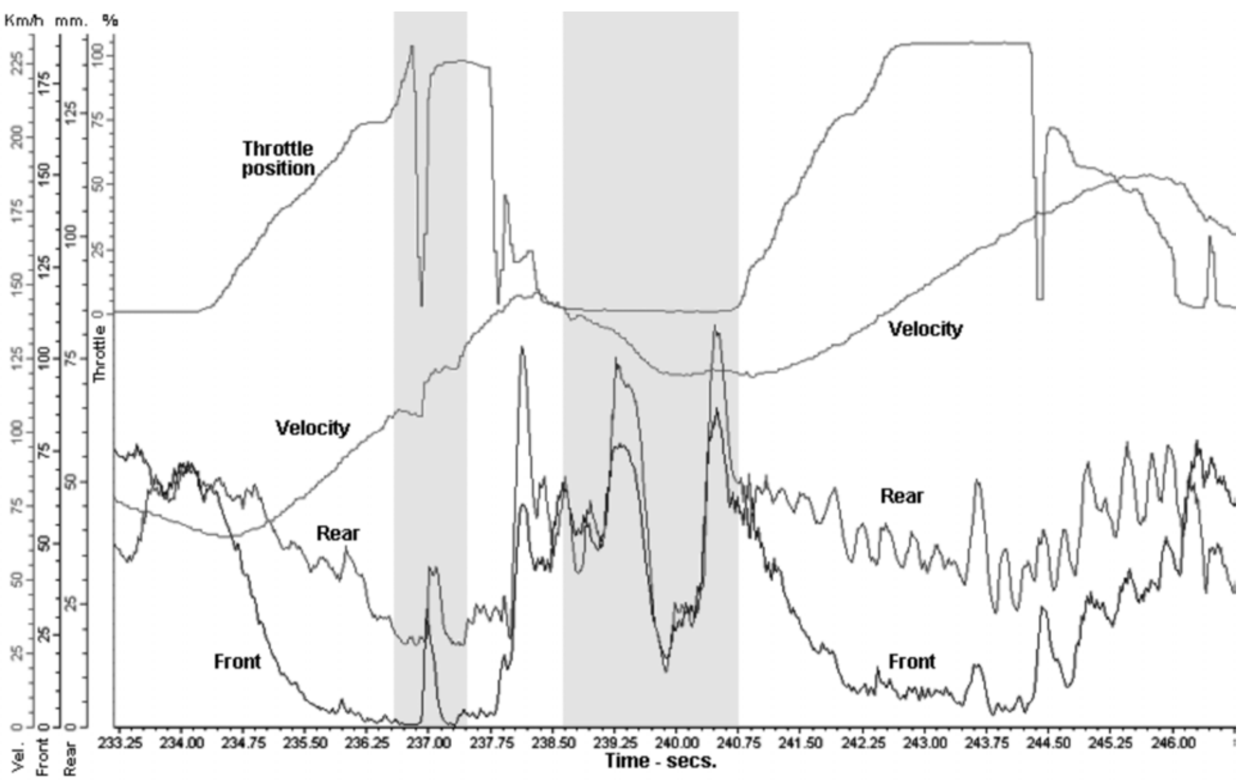



Fig. 15.3 **The shaded areas define aspects of this data sample which are discussed in the following text.** (Data from 2D)

Fig. 15.3 shows the throttle position and rear suspension data in addition to the front wheel velocity and front suspension movement shown in the previous charts. Consider the first shaded area, the speed trace shows us the signs of a wheelie similar to fig.15.1, but this time the wheelie is terminated only when the rider shuts the throttle quickly. Note also that, when the front wheel returns to the ground, that not only does the front fork compress but so does the rear. This may seem surprising but is quite normal. To see why, loosely hold a pencil horizontally in your fingers half way along its length. Lower it until one end hits the edge of a table or desk, and notice how the other end is forced downward.

The second shaded area shows the data during passage through a left / right chicane. Note the large dip in the suspension compression of both front and rear, this is due the extra compression, from cornering (see chapter 6, fig. 6.26), being relieved as the bike becomes more upright, additionally there is a centrifugal effect due to the roll velocity which can be quite high in the mid-point of the transition from one lean to an opposite lean. Compare this trace with fig. 6.27 from the suspension chapter (Fig. 6.27 shows compression downward and so the suspension traces are flipped vertically compared to fig. 15.3).

This brief look at some of the information that can be extracted from data recording shows the enormous potential that it has for quantifying aspects of chassis performance. For reasons of cost and availability of expertise, these techniques are currently the preserve of well funded race teams and manufacturers. Smaller race teams generally regard the cost as difficult to justify and prefer to use their funds more directly on bike related hardware. In some cases this may not be so cost effective as having the numerical data to guide the handling optimization decisions. Greater track time and more experimentation will be needed to extract the best from the bike, just like tuning an engine without a dynamometer. Fortunately, electronic and software products often become cheaper with time and so such systems may become more accessible in the future.

# Laboratory

As useful as data recording is from road and track testing, it has severe disadvantages also. In general it is very expensive in the long run, and not completely repeatable. The expense is often largely indirect, travel is often involved in order to find suitable terrain for specific types of test and which are then subject to the vagaries of the weather and moods of the test riders. Recently considerable progress has been made toward reproducing road testing in the laboratory. Capital cost of such equipment makes such testing the preserve of major factories or well funded race teams.

## Strength analysis

With the advent of small, inexpensive computers, the accurate analysis of strength and stiffness in our structures has become relatively straightforward, given a correct estimate of the applied loads. Using finite-element and similar analytical techniques, we can investigate resonance and vibration characteristics as well as both overall and detail stresses and deflections. The very speed of the computer enables us to try a large number of loading cases and to optimize suspension performance for various types of road surface. Our chief problem is to estimate or calculate the applied loads and there are several approaches. We can, for example, consider 'worst case' conditions and design for safety under specific maximum loads. This could mean deciding on the highest speed at which a head-on collision would not cause any significant damage, or the speed at which damage is limited to, say, the front wheel and fork. We can check other maximum-load conditions, too, such as hitting a pot-hole or brick in the road at top speed (perhaps also while cornering) or landing after a jump of a certain height with passenger and luggage. Given the worst-case conditions, these loads can be reasonably well calculated but selection of the conditions is somewhat arbitrary, although past history gives a good guide. If the cases are too severe, weight will almost certainly be excessive, whereas underestimation will result in a fragile structure, intolerant of hard use. Since metal fatigue is the most common cause of failure, in non-extreme circumstances, a study of the high-load conditions gives little indication of the life of a structure in normal service. (Fatigue, as explained in Chapter 10, is a result of pulsating stress.) Since conditions of use may vary widely from machine to machine, it is impossible to predict the exact future loading history of any particular motorcycle. The best we can do is to average out the expected life/load records. Such load cycle information can be obtained by use of a bike resembling the new design as closely as possible. This must be fitted with strain gauges and other transducers, then ridden under a wide variety of road conditions, with records kept of the time history of the loading. Given sufficient testing, we can thus build up a picture of the load cycles of a typical machine of that type over its expected lifetime. Additionally, data can be recorded of axle accelerations, these results can then be used to form the basis of the inputs to laboratory testing machines that can realistically load a motorcycle with completely repeatable test patterns, which mimic very closely a particular section of road or test track, to determine flexure and life expectancy characteristics.

Fig. 15.4 is an indication of the frequency of occurrence of different magnitude loads as found in practice.

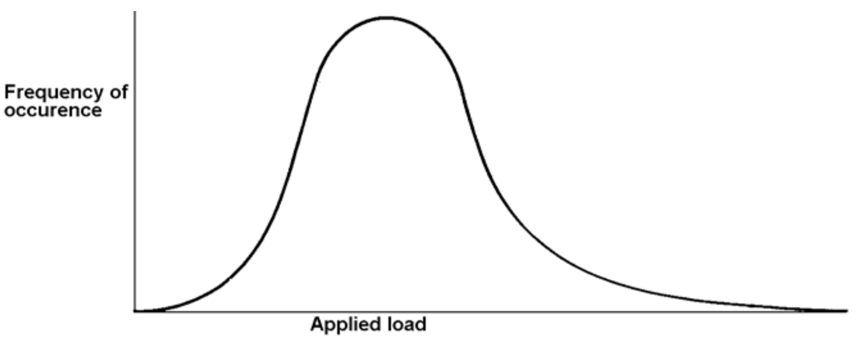



Fig. 15.4 **A general example distribution of applied loads and the occurrence of those loads. Mid range loads occur more often than extreme loads.**

Note that the middle range of load occurs most frequently, while both low and high values occur much less often. This is to be expected as low loading is produced at low speeds on smooth roads, while high loading is a result of severe conditions – both extremes being encountered for only a relatively small proportion of the machine's life. Mathematical techniques are available for the treatment of such test data and, together with stress analysis, they enable us to investigate the frame's fatigue behaviour. Fatigue analysis is based on summing the microscopic damage done during each load cycle, more damage is done at higher values of the load cycle. Such sophisticated testing methods are usually beyond the scope of the small chassis maker and the enthusiast building a one-off. Indeed, it is unlikely that even large manufacturers make full use of available techniques. However, good results can usually be achieved by rule-of-thumb and simpler calculations based on previous practice and experience. If the designer himself has the experience, so much the better. If not – and all designers must start somewhere – then a review of existing hardware is a good basis to build on.

After applying appropriate safety factors, allowable stress levels must be decided – which will enable us to determine the sizes of components (frame tubes or whatever). Safety factors are applied because of the possible severe consequences of failure, there is always a danger to life and limb if something breaks. Fatigue life is over dependent on load levels, a ten percent variation in loading may well cause a two to one variation in life expectancy. Safety factors also allow for our inability to predict future loads exactly, for some machines having a much harder life than average, and for any material flaws, stress concentrations or inaccuracies in calculation or manufacturing.

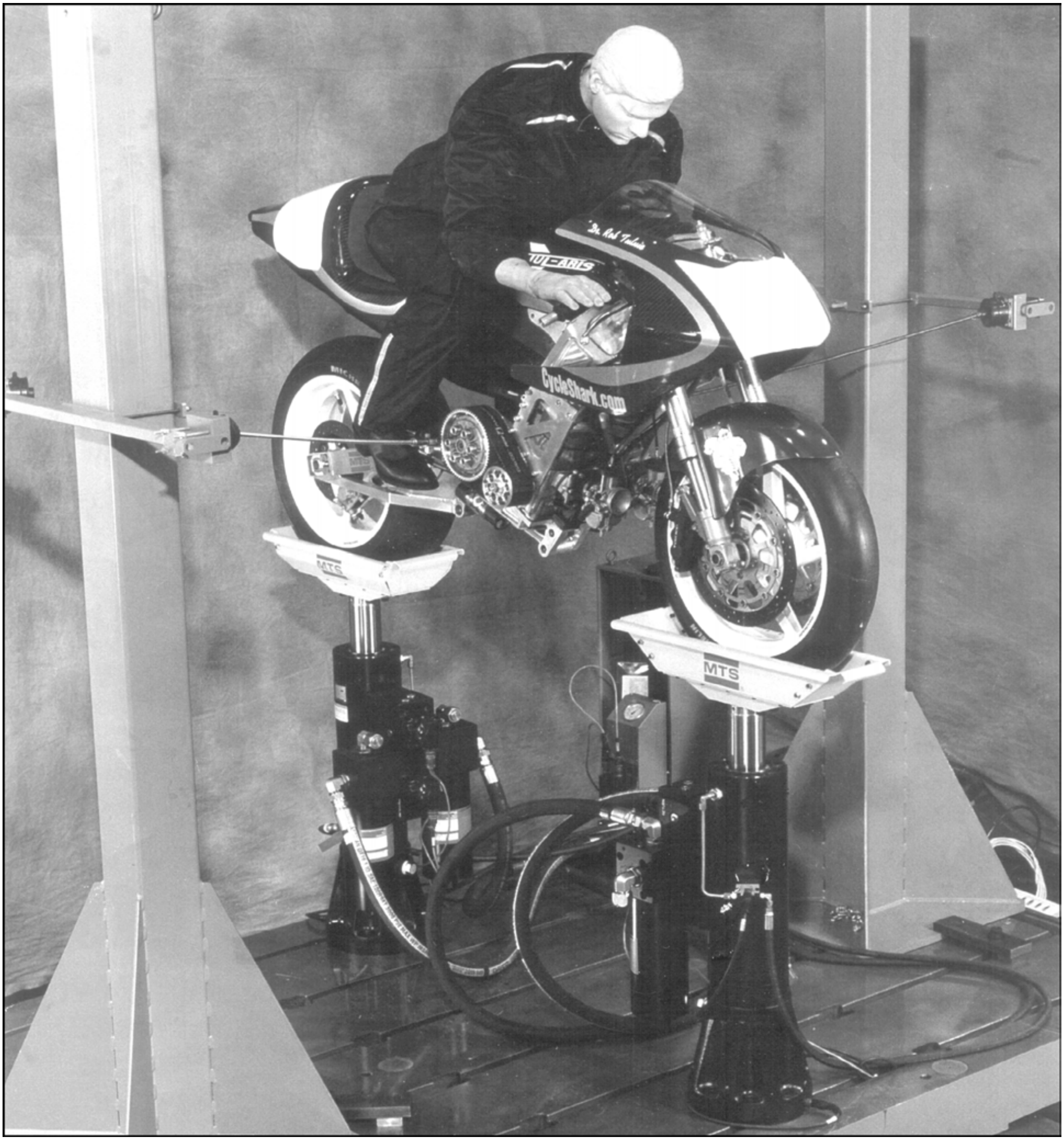



Fig. 15.5 **Called a "Two poster rig" this testing machine is designed to produce controlled vertical tyre loading on real motorcycles. The bike is supported by two powerful but fast acting servo-hydraulic rams. These are under the control of a computer programme and are vibrated to simulate real road surfaces, previously measured. Repeated tests can be done without further road testing. The side rods are to prevent the machine from falling over, without interfering with the vertical motions. The dummy rider is construction to mimic the dynamic behaviour of a real rider as closely as possible.** (Dr. Robin Tuluie, MTS Systems Corporation)

## Measurement and simulation

Enormous strides have been made in these areas over the recent past, with large testing machines being built that can test, in repeatable laboratory conditions, whole bikes in conditions very close to real road situations. MTS Systems Corporation in the USA are important constructors of such simulators and software (to whom thanks are due for permission to use the following illustrations), which can, in the laboratory, re-create the same damaging chassis loads and motions that the motorcycle experiences on the road or proving ground. With these simulators, durability and other forms of repeatable testing can be carried out significantly faster.

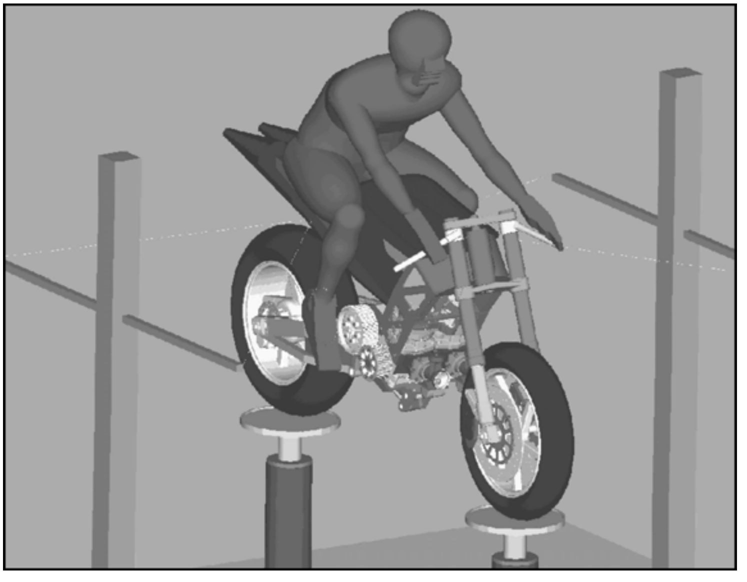


Fig. 15.6 **A computer simulation of the same bike on a test rig as shown in fig. 15.5. Comparing the results of simulations of such virtual models with those obtained on physical test rigs enable the virtual models to be tuned to a high degree of realism.**

(Dr. Robin Tuluie, MTS Systems Corporation)

The collection, processing and analysis of such data is a highly complex technical subject, and we can only just scratch the surface here. The new test systems would be largely impotent without the parallel development of advanced analysis techniques and software. The following is a very brief description of the principal steps involved to develop a system for testing and improving suspension settings and performance.

### Data collection

An example bike will be instrumented with various transducers, mainly accelerometers, at strategic locations. We would ideally like to directly measure what happens at the road/tyre interface, but this not currently feasible and so it is more usual to measure axle accelerations. These will be the principal parameters in later testing but the outputs from other transducers spread around the motorcycle will be used to help confirm that the test rig excites similar responses.

The motorcycle will then be ridden over the selected test course. Due to variations in the path followed by the test rider and variations in maintaining a repeatable speed, it is preferable that many similar runs are performed to get statistically significant data.

Modern data collection is generally digital in nature, which means that data measurement is not continuous, but is sampled at short time intervals. This sample rate puts a limit on the maximum data frequency that can be resolved. This frequency is known as the "Nyquist frequency" and is half of the sampling frequency. To prevent errors (called aliasing) in the subsequent transformation into frequency response data, it is very important to filter out any road input frequencies higher than the Nyquist rate. This must be done by means of "anti-alias" filtering at the data collection source, it can't be done by software later in the process, the information just will not in the data to do that.

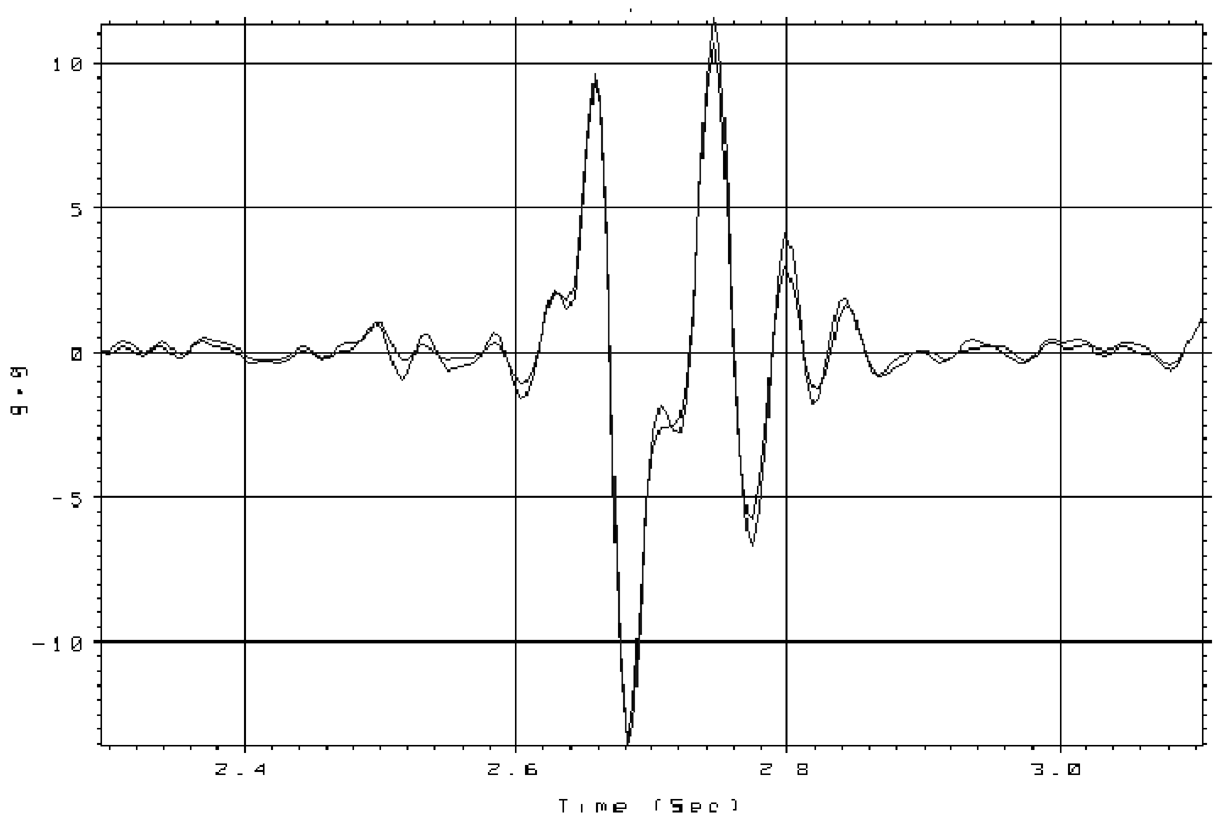



Fig. 15.7 **Showing the degree of accuracy with which the test rig in fig. 15.5 can reproduce real road inputs. There are two curves over-laid and it can be seen that there are only minor differences. One curve is of axle accelerometers during a road test and the other curve shows the same event simulated on the rig. The principle event in the centre of the curves was hitting a 50 mm. step at 48 km/h. The vertical axis is vertical acceleration, and time is shown on the horizontal.**

(Dr. Robin Tuluie, MTS Systems Corporation)

### Data processing

The collected data will generally go through some software transformation processes (Discrete Fourier, for example), which will show the axle input accelerations and the various motorcycle responses in terms of frequency. Applying the axle inputs to the previous measured characteristics of the test rig servo-hydraulic actuators, it is possible to calculate the inputs to the test rig that will reproduce the same axle accelerations as the road tests.

### Validation

The test bike will be mounted on the rig and excited. Data will be collect from transducers mounted as for the road tests. Standard statistical tests will be applied to this data to check for correlation with the road data, and if necessary adjustment will be made to the servo-hydraulic inputs to improve that correlation. Fig. 15.7 gives an idea as to the degree of accuracy possible, fine tuning could improve on that shown. One potential advantage of development using laboratory testing is the degree of repeatability when compared to road testing. For example the ***"standard deviation"*** of repeatability of the laboratory tests may typically be around 10% of that of the road tests. (***"standard deviation"*** is a measurement of the degree of variation in data samples.)

Tests will also be done to determine the sensitivity of the measurement system to small changes in the suspension settings and tyre pressures and compared to the sensitivity of the road tests. For example, what is the minimum variation in tyre pressure that will produce statistically significant changes in the measured accelerations?

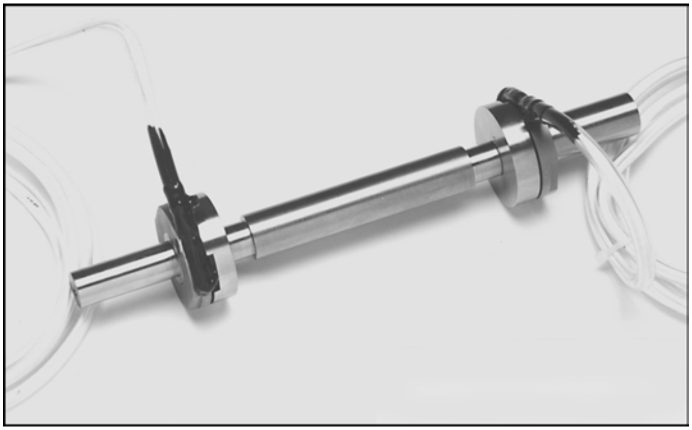
### Testing and development

Once the testing system has been validated, and there is confidence that road conditions can be accurately reproduced, then the actual development can proceed and various suspensions settings can be optimized. Final road testing can then be done to confirm the applicability of the settings to the real world. Testing can also be done in the laboratory beyond that possible on the road, manoeuvres that would be too dangerous for a test rider can be performed in complete safety.

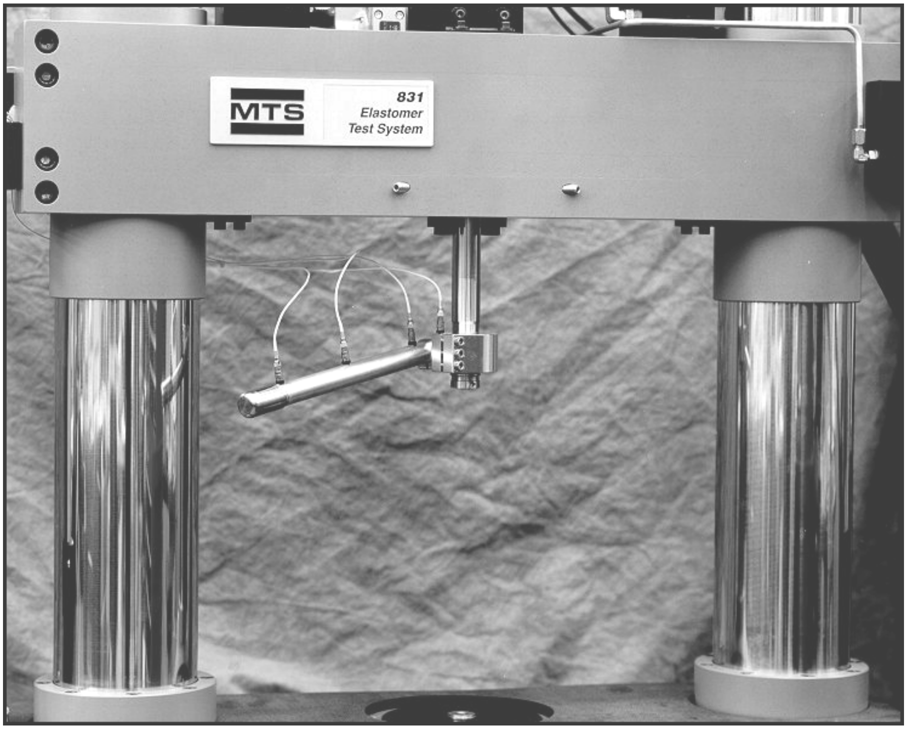
Although the above example was concerned with suspension development/settings, similar techniques can be used for other testing requirements. Long term durability for example. The addition of more servo-hydraulic actuators can increase the realism of some other types of test. Fore and aft axle actuators are often used as well as the vertical ones to simulate the structural loads of braking and acceleration, for example. The equivalent of say 150,000 kms. of road tests can be simulated in the laboratory in only a few weeks.

## Future development

Now that the motorcycle manufacturers have started to embrace this type of testing and simulation, we can expect to see new possibilities being developed. Proper use of these techniques will reduce the possibility of producing poor performing machines and new models can be brought to market faster. Coming on line in the near future will be machines capable of accurately simulating performance in a cornering attitude for example, with realistic treatment of the inertia forces. New transducers are continually being developed, like the wheel load transducer shown below.



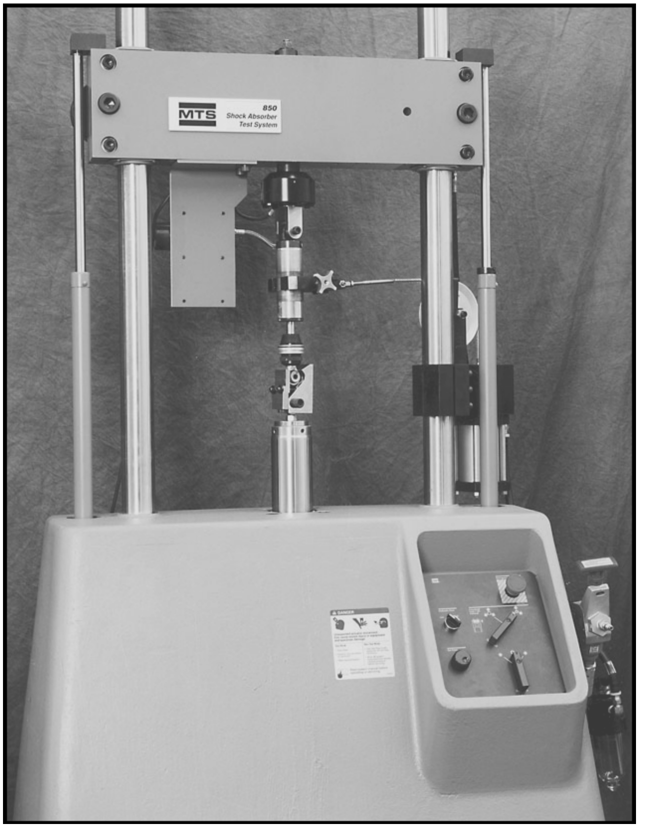
**Patented axle mounted wheel force transducers. These can measure three axle forces, up and down, fore and aft, and axial in addition to two axle bending moments. They are small enough to install with a wheel, and would be very useful when investigating some fundamental dynamic aspects of cornering etc.** (Dr. Robin Tuluie, MTS Systems Corporation)



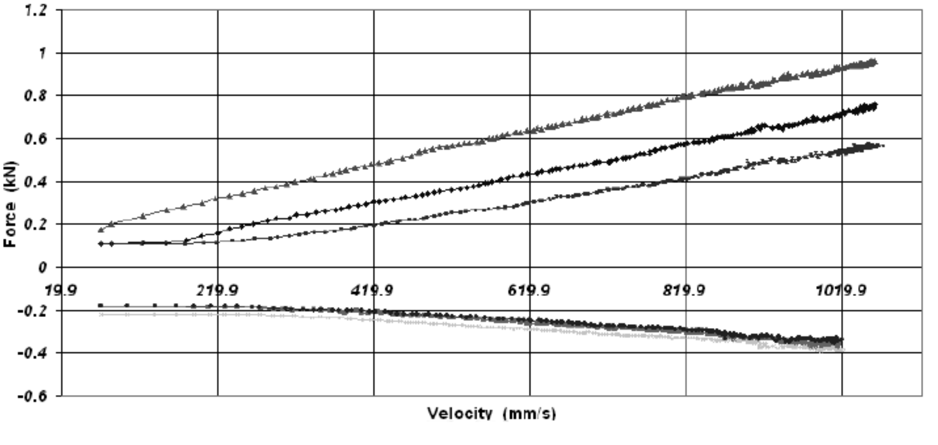

**With the right equipment even the vibration characteristics of individual components such as this clip-on handle bar can be tested and optimized. In this case the handle bar is subject to vertical excitation through a servo-hydraulic actuator. A wide frequency range can be scanned to mimic varying engine RPM., the main source of vibration.**

**When looking at vibration, it is important that the testing structure does not become part of the problem, hence the massive dimensions of this machine.**

(Dr. Robin Tuluie, MTS Systems Corporation)


**Suspension testing. On the left an USD fork leg under going test. To the right is a rear damper unit. In this case the visible side connection is a cable to apply a lateral load to the damper. The cable passes over a pulley and is tensioned by a pneumatic cylinder, these are partially hidden by the vertical columns. Hydraulic testing machines like this can apply a wide range of input displacements to the dampers. For example, step inputs, sine waves, variable frequency sine wave sweeps and white noise shaped to match typical road spectra. The traditional mechanical test machines used a crank mechanism (like a crank from a single cylinder engine) and this produced a pseudo-sine wave only, similar to the piston movement in an engine.** (Dr. Robin Tuluie, MTS Systems Corporation)




**Front fork damper plots, obtained from the machine shown above. Showing peak force against peak velocity for different settings with a sine sweep input.**

(Dr. Robin Tuluie, MTS Systems Corporation)

# 16 Practical frame building

This chapter is intended as a quick guide for those contemplating the construction of a one-off special or a relatively small production run. Obviously, it would take several complete books to cover the range of techniques and methods that could be employed for such a task, complete volumes are devoted to just welding technique. This is just an attempt to briefly review some aspects from my own experience that probably aren't covered in more general fabrication texts. To modify or construct chassis parts requires a certain degree of skill and knowledge, so unless you feel confident about your work, it would be better to consult a specialist. The result of getting it wrong can easily be a serious accident.

It is becoming more popular to have frames constructed of aluminium, particularly the twin-spar type. The techniques for producing such frames require much skill and as such are not recommended for the home constructor. Tips for the construction of these are not presented here.

The techniques used in large-scale production may differ considerably from those described here, because of the much larger investment available to a large manufacturer for automation. The differences apply to processes such as welding, tube manipulation (bending and end-profiling), bracket and sheet-metal pressing and jigging. The money spent on production facilities must be balanced against the extra construction time involved when using less-expensive equipment, and the final decision depends on the number of similar frames to be produced. For low-quantity frame production, we are usually restricted to the following choice of methods and materials:

- welded steel or aluminium tubing.
- welded steel, aluminium or stainless-steel fabricated backbones, monocoques and twin-spar.
- riveted or cast-aluminium backbones, e.g. Offenstadt and Seeley 500 Suzuki as shown elsewhere. This type is highly specialized and not really viable without a very well equipped workshop, and will not be discussed specifically.

## Welding

Two basic methods are available – electric and gas – each subdivided into a variety of techniques, which will now be described, together with their pros and cons.

### Electric

***Arc*** (stick electrodes). In common use on construction sites and cheap home-welding kits for do-it-yourself purposes, this is probably the least suitable technique for frame construction, although satisfactory results may be achieved provided the tubing is relatively thick-walled, say, 2 mm or more. It is true that the roadgoing versions of the famous Norton featherbed frame were welded in this way; but welding technology has advanced considerably in the intervening decades and it is likely that, if this frame was making its debut today, the following method would be used.

***MIG*** (Metal Inert Gas), often called $CO_2$ welding. Here the weld filler material takes the form of a wire, which is continuously fed into the joint by the welding machine and is shrouded by a flow of inert gas. This may be $CO_2$ or a mixture of argon and $CO_2$ (usually in the proportions of 80:20 or 95:5 respectively) and its purpose is to prevent oxidization of the hot welded material by blowing away the oxidizing air. Nearly all production frames in welded steel tubing are nowadays produced by this process, and it is easily automated. However, this is not to say that it is unsuitable for manual use on low production runs or one-offs. Although it is usually associated with steel tubing, it may also be used for aluminium and stainless steel.

The advantages include:

- Speed, which reduces not only labour costs but also distortion, since the total heat input is relatively low.
- Clean welds – no flux to be cleaned off or to adulterate the weld.
- Tolerance to operator skill – sound welds can be achieved with less experience than is required with some other methods.
- Gap filling is good, hence less time need be spent on joint preparation.

Disadvantages may be:

- Weld fillet is often convex, so may cause undue stress concentration in thin-wall tubing. This is less likely to be troublesome in triangulated or backbone frames than in the typical bent-tube type, but if used on 1.2 mm (18-gauge) or thicker tubing, it should cause few problems.
- The high capital cost of quality equipment may not be justified for the enthusiast wishing to experiment. Relatively cheap single phase machines are now available for home use, however, these vary in quality and so before any purchase it is advisable to arrange a trial on the type of work that you wish to do. The wire feed mechanism can a source of trouble on the cheaper units.

***TIG*** (Tungsten Inert Gas), also known as Argon-arc and, in America, as Heli-arc. In this system, an arc is struck between a tungsten electrode and the workpiece to provide the necessary heat, while the filler rod is fed in by hand. As in the MIG process, a gas shield keeps out the oxidizing air, however, in this case the gas is almost pure argon. This system can produce welds of higher metallurgical quality than the other methods described and may be thought of as a clean electrical version of an oxy-acetylene torch. In experienced hands it is very versatile, since it can be used for welding steel, stainless steel, aluminium, titanium and even the inflammable magnesium. Since this process takes longer than MIG welding, however, distortion may be greater, but extremely neat concave welds are possible, thus reducing stress concentrations. It is suitable for all thicknesses of tubing and excellent for sheet-metal applications such as fuel tanks, two-stroke exhaust boxes and stainless-steel or aluminium fabricated backbones. The results can be of very high quality. Unfortunately the capital cost of quality plant limits its use to those with sufficient throughput of work to warrant the expense.

## Gas (usually oxygen and acetylene)

***Fusion welding*** Here the flame is used to melt the parent metal while filler rod of similar composition is fed in by hand, as in the TIG system. Gas fusion welding is seldom used in frame building but is popular for the construction of fuel tanks (in both steel and aluminium) and exhaust systems, especially two-stroke expansion chambers. Weld quality, however, is inferior to that obtained by the TIG method but initial financial outlay is small. One possible advantage over TIG is that the heat is more widespread and so permits a gentler cooling in the weld area.

***Bronze welding*** (often incorrectly referred to as brazing) In the construction of special frames this has probably been the most widely used method, and is still used by many professional frame builders in preference to the electric methods. It is suitable for tubular steel structures, where one of its chief advantages stems from the lower temperatures involved. In all methods of welding, the heat reduces the strength of the parent metal in the vicinity of the weld, especially in the case of some of the higher-strength steels and aluminium alloys. In some applications, the full strength may be restored by subsequent heat treatment, but this is not usually practicable with a one-off motorcycle frame and would probably lead to distortion in any case. In bronze welding, however, the parent metal is heated to a temperature which, though high enough to melt the bronze filler, is well below its own melting point. The parent metal therefore retains much more of its original strength, which is one of its chief advantages. Various types of bronze are available, with different strengths and melting points, to suit different applications. With care, the weld fillet can be concave and broad based to give a smooth change of section and minimum stress concentration. For all these reasons, bronze welding is highly recommended by Reynolds – makers of the famous 531 manganese-molybdenum tubing, especially for thin wall sections. Although it is easy for an unskilled welder to make a joint by this method, considerable skill is required for a quality job. Temperature is critical – too cold and adhesion is poor, the joint weak, too hot and some of the elements in the bronze may vaporize off, while the filler material may penetrate deep into the grain structure of the parent metal, leaving it weak and brittle.

While many successful motorcycle frames have been welded by this method, it is this possibility of embrittlement that accounts for bronze welding not being approved for joints in the primary structure of aircraft. It is absolutely essential, when constructing a frame in this way, that no load is placed on the joint until it has cooled down completely – otherwise intergranular penetration will occur and subsequent failure will be almost inevitable.

## Distortion

The very process of welding causes distortion, so corrective measures must be taken to ensure accurate frame alignment. The causes of distortion are rooted in the cooling process of the weld, and proper welding sequence is as important to the finished product as is the basic design of the frame. Consider the simple weld shown in figure 16.1.

Fig 16.1 **When a weld cools it contracts. In this case the contraction will pull the vertical member to the right. The addition of the bracing member prevents distortion at the expense of residual stresses in the welds.**

While the weld material is still molten, the welded pieces remain at 90 degrees to one another. As the weld cools, after solidifying, the material contracts (at a rate determined by its coefficient of expansion) and tends to pull the joined components to the position shown. The linear contraction of the weld is, of course, only small but the movement at the end of the members is greatly multiplied by the leverage effect.

That is why weld materials with a low expansion rate cause fewer problems through distortion. Bronze, with its high coefficient of expansion, is at a disadvantage here. We can counter the effects of this type of distortion by putting other members in the structure, as shown in figure16.1. Unfortunately, this has the disadvantage of building in stress at the weld zones. As the weld cools it is prevented from contracting as before and thus becomes subject to tensile loads. These built-in stresses add to those induced by operational loads and so tend to weaken the chassis.

With a welded frame it seems, then, that we have to choose between distortion and built-in stress. To a large extent this is true but several techniques are available to alleviate the situation. If, in our original example, we tilted the vertical member to the left before welding, then the subsequent contraction would bring it back to 90 degrees – giving no distortion and minimum stress. If we consider the functional aspects of a chassis we see that fine tolerances are required in only a few instances, such as steering head to rear pivot housing, and the alignment of the final-drive sprockets. It is these relationships that are important, the path and location of the connecting structure are less critical. We can make use of this fact in designing our welding sequence. For example if we complete the welding of the connecting structure before we attach the steering head and/or rear pivot, then most of the possible distortion will have taken place before these components are added. Furthermore, if each joint is completed and allowed to cool before the next one is tackled, and the individual tubes are not jigged too rigidly, then there will be a minimum of built-in stress. The alternative method of tacking all joints first, then finish welding, may result in the more accurate positioning of individual tubes, but the welding stresses will be higher, particularly in the case of bronze welding. The author is not in favour of this method.

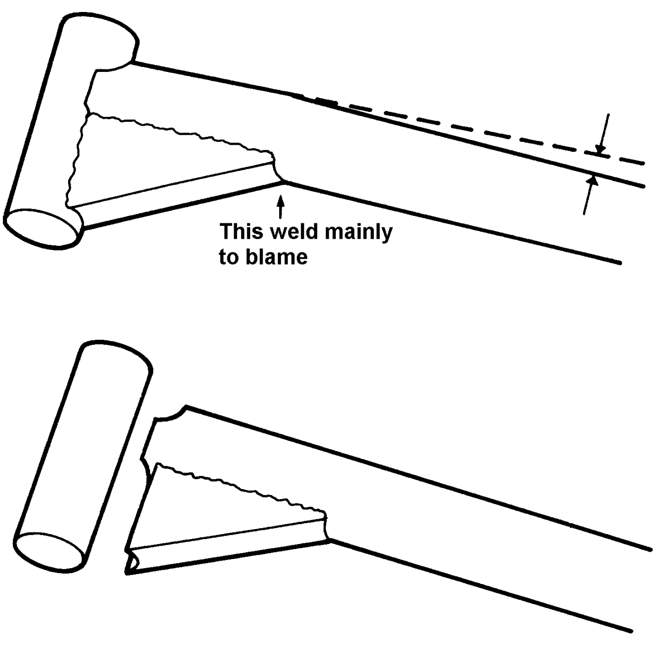



Fig 16.2 **The importance of welding sequence. Adding the gusset last (top) steepens the steering head on cooling, as shown. The simple solution to this problem was to attach the steering head last, after the top tube distortion had taken place (bottom).**

**In fact it may well have been better not to weld the gusset at the rear in any case.**

Fig. 16.2 shows a practical example of how welding sequence can affect final alignment. Originally – as shown in the first sketch – the backbone was welded to the steering head first, after which the gusset was added. On removing the frame from the jig, it was found that the steering-head angle would steepen. Investigation showed that the welding on the gusset had caused the backbone to bend slightly. The solution was simply to weld the gusset to the backbone first, so allowing the distortion to take place before the steering head was attached.

## Gussets

These are used at corner joints to stiffen the structure, spread loads and/or provide mounting points. Their design must be given careful consideration if excessive stress concentration and consequent weakening of the frame are to be avoided. We have already explained the benefit of attaching the gussets along the tube's neutral axis but there are additional methods of improving the assembly. For example, it is better to taper the ends of the gusset rather than cut them off abruptly. Here, ease of welding and manufacture conflict with the requirements of minimizing stress concentration. Sometimes, where similar gussets are used on both sides of a tube, they are formed by one folded pressing, so that the two sides are joined by a web. In this case it is often better not to complete the weld round the tube (i.e. at the ends of the web) as this would introduce more stress concentration.

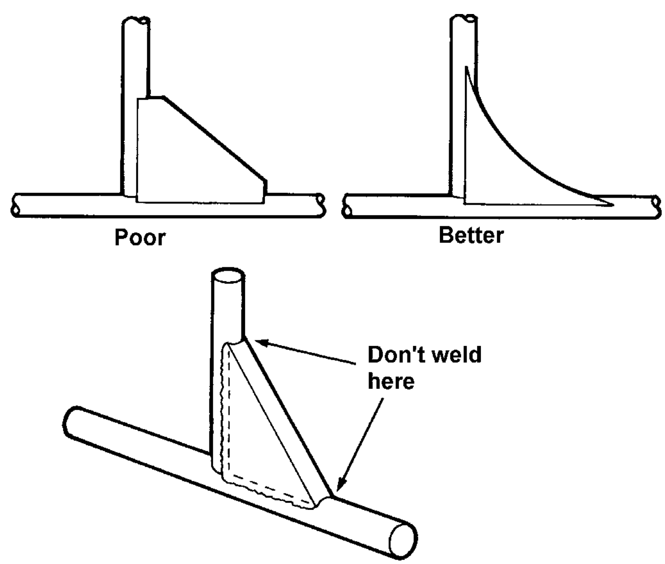



Fig 16.3 **Stress concentration is reduced if gussets are tapered rather than sharply cut off. A folded gusset should not generally be welded at the ends of the web.**

**The first example is the best from a production viewpoint and so the conflicts between fabrication and function need to be balanced correctly.**

The way in which a gusset stiffens a structure is not always understood. Imagine a tube attached to a rigid member and subjected to a bending force, as shown in figure 16.4. If there is no gusset to support it, the tube will bend over its entire length whereas, with a gusset, the bending is virtually restricted to the unsupported portion. The amount of flexure is proportional to the cube (third power) of the unsupported length, so reducing that by a half increases the stiffness by a factor of *eight.* Hence even small gussets can stiffen a multi-tube frame considerably. However, where a tube is stressed only in tension or compression a gusset can have no more than a minor effect. For that reason, they are seldom found on well-designed triangulated structures, except perhaps to provide mounting points.

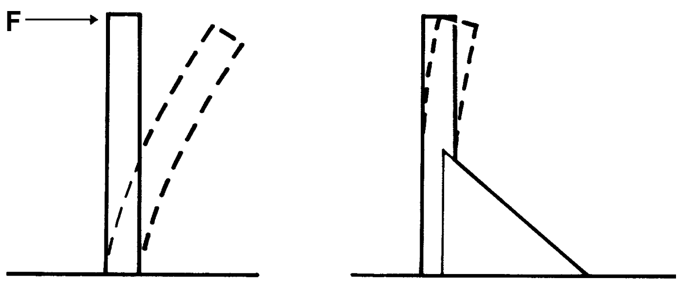



Fig 16.4 **A gusset restricts bending mainly to the unsupported length of the tube.**

## Jigging

A jig is simply a structure to hold the component parts of our frame in their correct relationship during welding. It may be simple or complex, depending on the quantity and quality of the frames to be produced. For one-off jobs or experimental development work, the most versatile system is to use a *rigid* flat surface as a base to support easily made sub-jigs. An old flat lathe bed or table from a milling machine is ideal for this purpose, as squaring off the sides is easy and the centre gap or T-slots are ideal for bolting on fixtures.

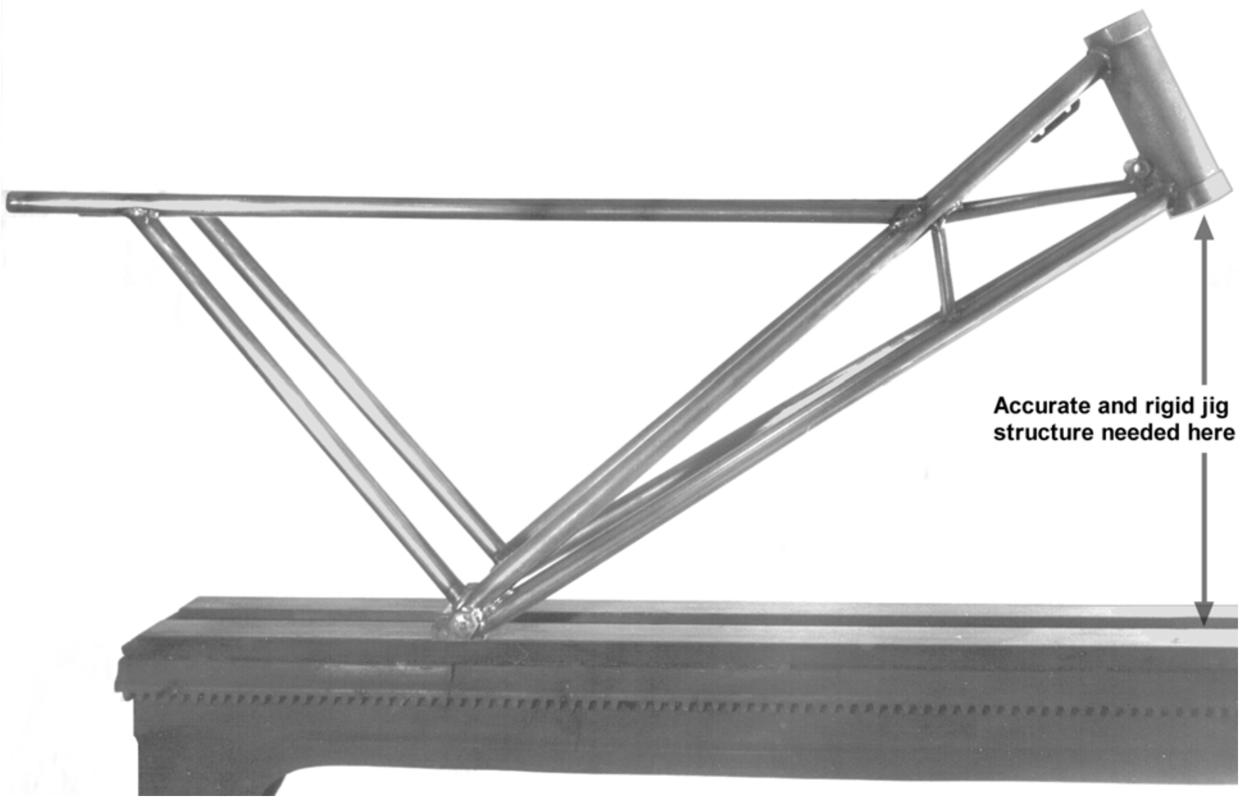



Fig 16.5 **Showing how an old flat lathe bed can provide a solid and accurate basis for a frame jig suitable for one-offs. It is also easy to see that if the frame were to be made in this position, then an additional tall jig structure would need to be made to support the head stock. This extra structure will reduce accuracy and rigidity.**

A final tip on jigging. It is natural to think of building a frame in the same orientation that it will adopt when finished. However, sometimes it is easier and more accurate to jig a frame in a different attitude to that which it will adopt on the road, even though you may find it a bit harder to visualize the placing of the various components. To jig a frame with its normal positioning may require more complex jigging, including a long, high steering-head fixture, particularly when a flat bed is used as a base. If the frame is properly drawn first (say to ¼ or ½ scale) then simpler jigging may be possible, with the steering head clamped directly to the bed, as shown below.

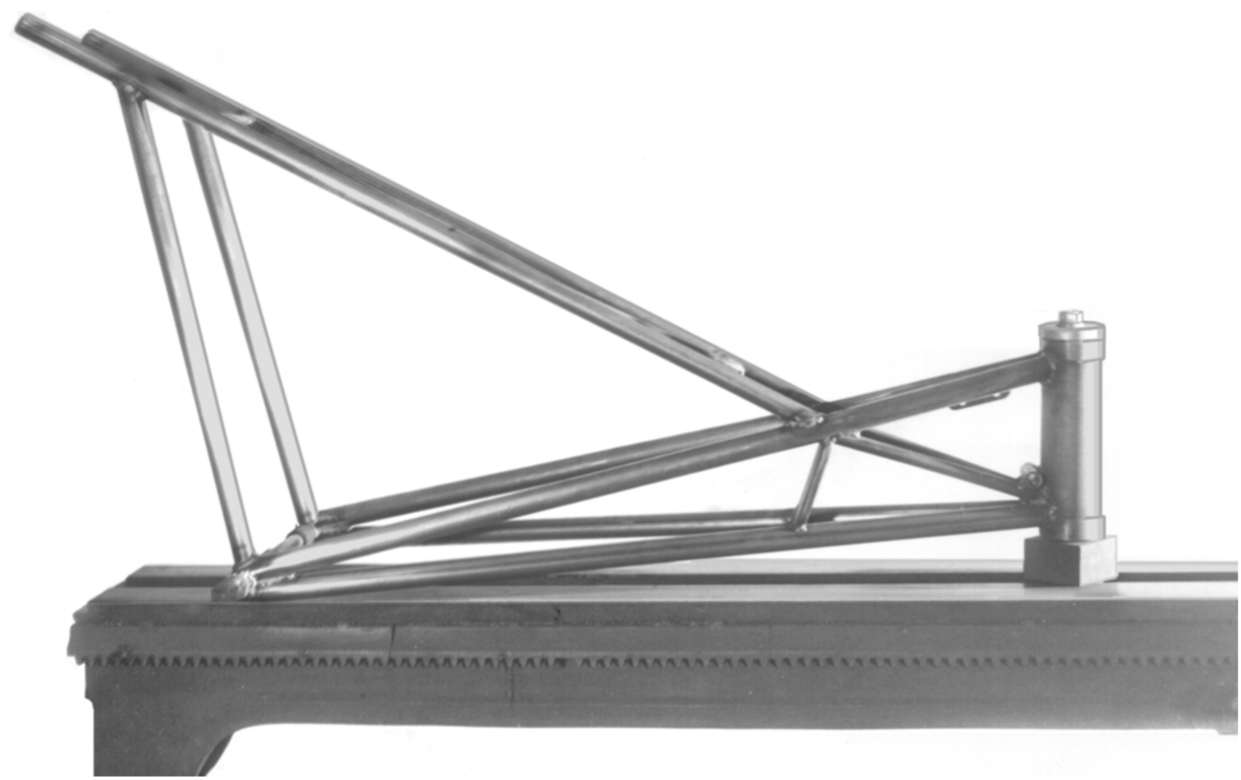


Fig 16.6 **Compare this with fig. 16.5. The head stock is simply bolted at right angles to the lathe bed with only a parallel spacer block between. This is both accurate and rigid. The down side is that it is more difficult to visualize the frame as you are fabricating it, and a taller support jig would be necessary to hold the seat tubes, but required accuracy here is of a lower order than between the head stock and the swing-arm pivot.**

Although all frame builders have their own preferences in choosing the type of jig to be used, there are some common requirements, the most important of which are accuracy and rigidity. Nowhere is accuracy more vital than in the relative positioning of the steering head and rear-suspension pivot. And if the jig is insufficiently rigid, then welding stresses and/or the weight of the frame may cause distortion and make a mockery of the original accuracy.

If at all possible it is worth while making the jig rotatable, as this gives full access to welded joints without contortions by the operator and so helps improve weld quality.

## Tube profiling

Except in the case of square or rectangular tubing, it is necessary to shape the tube ends to match the mating structure. This is often done with a milling cutter of appropriate diameter set at the required angle. While this method presents no problem for a well equipped workshop, it is seldom that a specials builder has access to the necessary facilities. There are available, tube notching saws. Basically just a normal hole saw with a supporting fixture for the tubing. Personally I've not been too impressed with these tools. In most cases, however, two straight saw cuts at the end of a round tube will produce a nicely fitting joint. It is not very high tech. but when hand fitting tubes to a one-off frame I've found this method to be about the quickest. If the cut profile does not match up exactly with the mating tube then a quick adjustment with a file or grinder is all that's necessary.

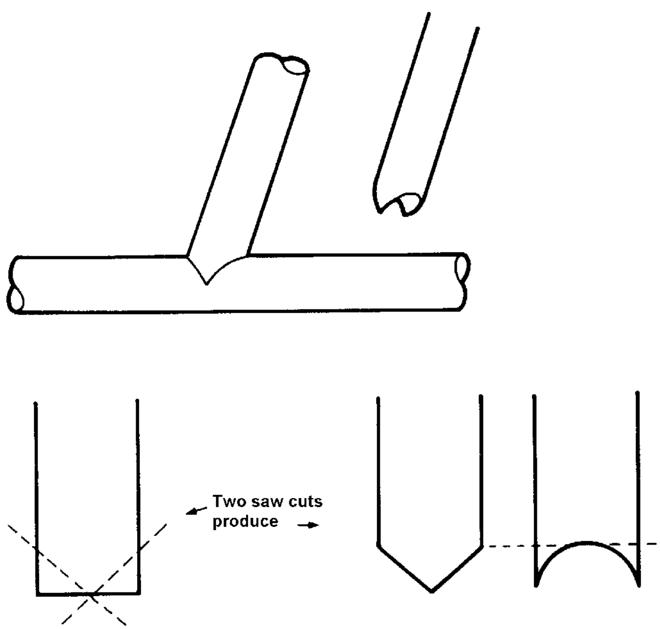
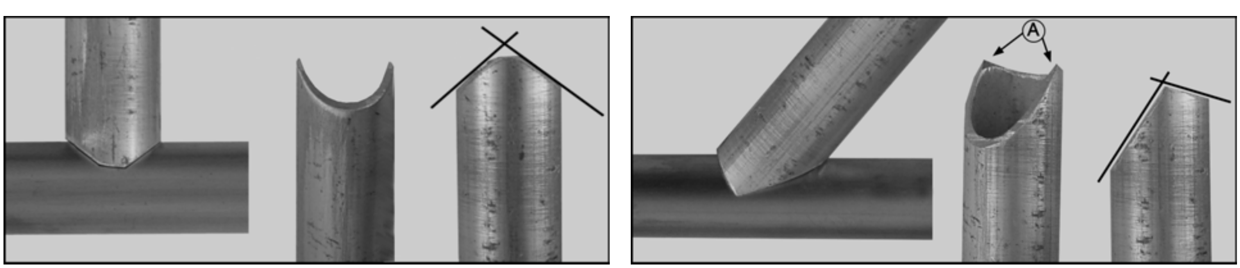



Fig 16.7 **Simple and quick tube profiling by two straight saw cuts.**



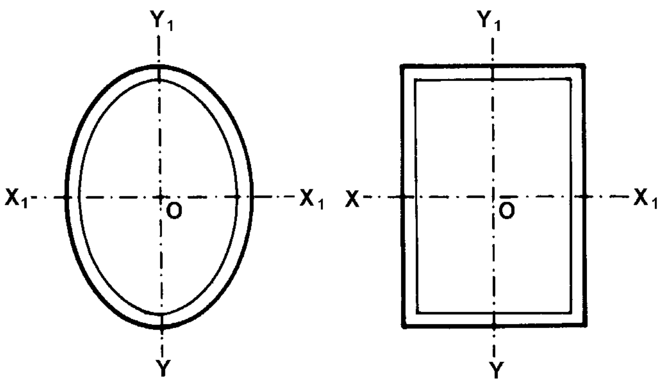

**Practical examples of tube profiling by two straight saw cuts. On the left we see symmetrical cuts to produce a right angle joint. The other example shows the two cuts at different angles to produce a joint of 50°. To obtain a perfectly fitted joint this example needed a small amount of adjustment with a half round file at points denoted by (A).**

## Tube types

Round-section tubing has long been the most common shape in frame production – and not just because it is cheaper than other types. It is also the best section for resisting torsional and compressive loads and is equally capable of resisting bending loads in whatever direction. However, other sections are sometimes used, mainly oval, square and rectangular, though occasionally we find tubing of taper section, as in the pivoted rear fork of some Velocettes. BMW have even gone to the expense of tapered oval tubing to match the tube properties to the applied loads, which is the chief justification for non-circular sections. When discussing structural features we saw that a tube's resistance to bending is a function of the second moment of area.

This depends on the tube shape and, except for a round section, differs about the neutral axis according to direction. In each case of fig. 16.8 the second moment of area is greater about the $XOX_1$ axis than about the $YOY_1$ axis, hence the resistance to bending is also greater about the $XOX_1$ axis.

Fig.16.8 **With oval and rectangular tubing, bending stiffness is greater about axis $XOX_1$ than $YOY_1$.**




So, if we know that the bending loads to which a frame member is subjected are mainly in one direction, we can tailor the tubing shape to achieve the most efficient structure. A typical example is a simple rear swing-arm, where any sideways load produces different bending moments about the neutral axes of the fork arms. See figure16.9. In this fairly typical layout – 325 mm. wheel radius and 250 mm. between fork arms – the vertical bending forces due to a sideways load are, as shown in the diagram, 2.6 times the horizontal forces.

Among the specialist manufacturers a popular tube for this duty was 51 x 25 mm x 1.5 mm (2 x 1 in. x16 $swg_4$) rectangular section. The relative second moments of area for this are 73600 $mm.^4$ and 23800 $mm.^4$ – i.e. a ratio of 3.1:1, which closely matches the load ratio of 2.6:1. But whereas the bending ***load*** acts at the wheel end of the fork, the maximum bending ***moment*** is at the pivot end.

Thus, if our fork is to be structurally balanced we should follow Velocette's lead and use taper tubing, though the extra expense is seldom thought to be warranted. The use of square-section tubing (which was popular in aluminium racing frames, for a while) is more difficult to understand and probably owes much to fashion. If we consider a 25 mm.– square tube of 1.5 mm. wall thickness (approximately 1" x 1" x 0.625") then this has a second moment of area of 13000 $mm.^4$ about any of its principal neutral axes. Now if we substitute a round tube of equal *weight* and wall thickness we need a diameter of 31.5 mm.

(approximately 1.25 in.), and the second moment of area of that is 15900 mm.[4] – i.e., 1.22 times that of the square tube. So, even on its most favourable axes, the square tube is less stiff in bending besides being less capable of handling torsional and compressive loads.

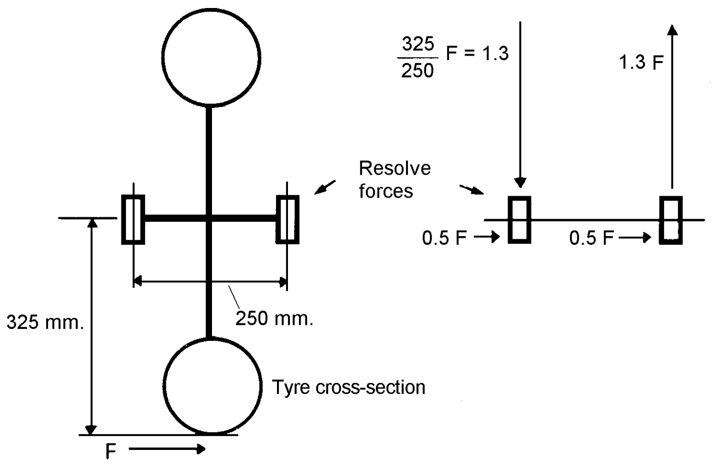



Fig 16.9 **In this end view of a typical but non-braced rear pivoted fork, the effect of a side force at the tyre contact patch is resolved into horizontal and vertical components on the fork arms.**

**This causes bending moments which are strongest at the pivot end.**

# Tube sizes

Tube diameter and wall thickness are determined by the size, weight and power of the machine, also by the design family of the structure – e.g. triangulated, backbone, etc. Consequently the following is intended only as a general guide.

## Triangulated (space) frames

As mentioned earlier, the stresses in the members of this type of frame are mainly tensile and compressive, hence large diameter is not necessary to provide adequate bending stiffness of the frame. More important is the cross-sectional area of the metal. Thus a 13 mm. diameter tube of 2 mm. wall thickness is interchangeable in most cases with a 25 mm. diameter tube of 1 mm. wall thickness. However, if a member is long in relation to its size then it may buckle under compression before the ultimate stress levels are reached. In that case the larger diameter tube is preferable, as it is for torsion stiffness. Whilst torsional stiffness of a triangulated frame is principally a function of the layout, the torsional properties of the tubes themselves also have an influence. Another reason for the use of round tube. Also, resonant vibrations create more of a problem with long, thin tubes. Typical sizes for steel tubing may range from 15 mm. diameter x 1.0 or 1.5 mm. wall. to 25 mm. diameter x 1.0 or 1.5 mm. wall. For the longest tubes needed on a motorcycle it is unlikely that diameters of more than 25 mm. would be needed. More commonly, we would expect to use tubes of, say: 20 to 25 mm. diameter.

## Bent multi-tubular frames

Since this chassis type gets its stiffness from a mixture of triangulation and the inherent bending and torsional stiffness of the tubes themselves, it is more difficult to recommend suitable sizes, these depend more on individual design. For the main structure, sizes may typically range from 22 mm. diameter x 1.5 to 2.0 mm., for smaller machines, to 38 mm. diameter x 1.2 to 2.0 mm. wall, for larger ones. Where the subframe supporting the seat is not intended to contribute to the main structural stiffness, it may be made from smaller tubing – e.g. 16 to 20 mm. diameter x 1.2 to 1.5 mm.

### Tubular backbone frames

Even on smaller machines, the main member is unlikely to be stiff enough if much under 50 mm. diameter x 1.2 to 1.5 mm. wall. On larger machines, though, there is little to be gained by going above about 75 mm. diameter x 1.2 to 2.0 mm. wall. With this diameter it is better to avoid excessively thin walls, otherwise cracking may result at joints as a result of local buckling. When designing this type of frame, great care must be taken to feed in the loads in such a way as to minimize the risk of such local buckling. This can be done by spreading the loads over a wide area, welding gussets and plates along the neutral axis wherever possible, and sometimes incorporating local reinforcement as shown in figure 16.10.

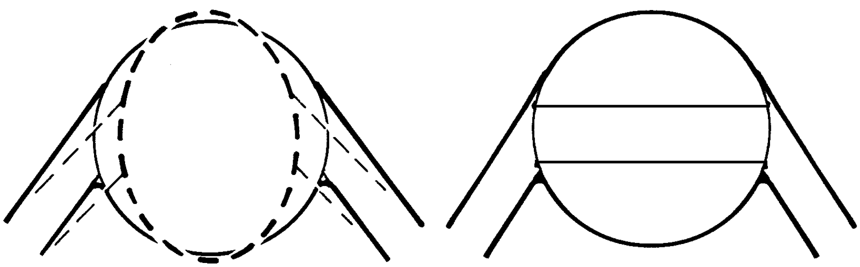


Fig 16.10 **Detail design is very important where loads are fed into thin-walled large diameter members. Shown here is how the insertion of a cross tube can prevent local buckling.**

### Fabricated backbones

Depending on detail design, 0.75 to 1.0 mm. steel sheeting may be used here, with possible thickening in regions of high stress or load application. Thicker material would be needed when made with aluminium, probably 1.2 to 1.5 mm. This type of frame derives its stiffness from its large cross-section, overall shape is not usually critical. Once more, however, the need to avoid local buckling cannot be over-emphasized and considerable thought must be given to all connection points if a satisfactory life is to be expected.

## Frame finishes

Once we have constructed our frame we usually need to give it some coating to prevent corrosion and enhance appearance. The finishes normally available include:

- Plating – chromium or nickel.
- Painting – a wide variety, including stove enamelling and epoxy powder.
- Plastic coatings.
- Anodizing – for aluminium parts

### Plating

Although attractive in appearance, this can be expensive if a first-class result is required. It also tends to highlight any visual flaws, such as lumpy welds and scratches. Acids from the plating process can become trapped in some tubes, either if the joints are not fully sealed by welding or if drain holes are not provided. By causing stress concentration, such holes can lead to premature fatigue failure. Trapped acids can eventually give rise to internal corrosion and can leach out at the welds, so causing rust areas and spoiling the finish.

A further risk with plating is hydrogen embrittlement. Here, hydrogen evolved during the plating process is trapped in the grain boundaries of the steel and can lead to failure. In this respect nickel plating is preferable to chromium. Indeed, in Formula 1 car racing chromium plating has long been banned on suspension parts for safety. Despite the potential pitfalls, it has to be said that Rickman Metisse frames, among others, have long been nickel plated, seemingly without trouble.

### Painting

New types of paint become available almost daily, hence advice should always be sought from the manufacturers. Paints such as cellulose were old favourites for tanks and other bodywork but modern health and safety regulations have generally caused a shift to water based paints. However, these paints do not always match up to the needs of the frame, where traditional stove enamelling produces the best all-round finish. The more modern electrostatically applied epoxy powders etc., have their devotees but, though quite resistant to damage, are less amenable to touching up when scratched.

### Plastic coating

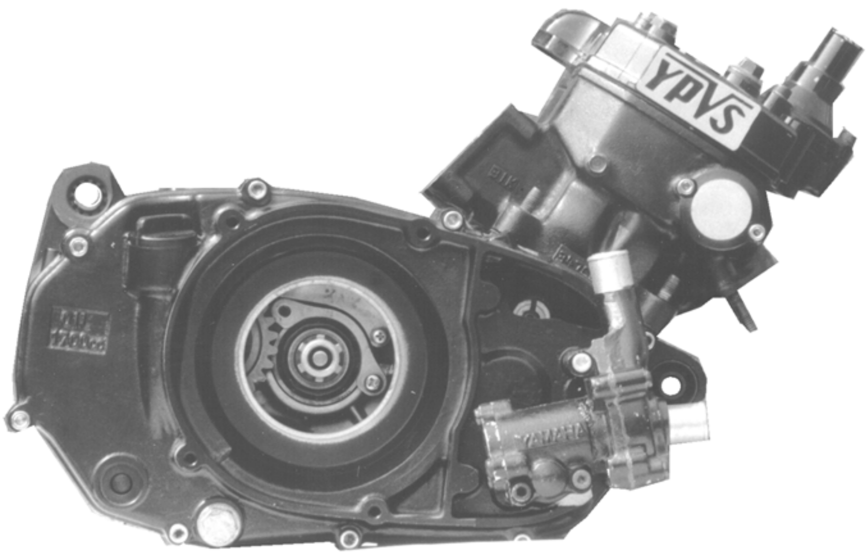
This gives an excellent finish when first applied, since it conceals flaws, but is less impressive in the long term because scratches cannot be so easily polished out or touched up as they can in paint. Furthermore, if damage should penetrate to the underlying metal, moisture may spread for a considerable distance under the adjacent coating, causing widespread corrosion and lifting the plastic. With paint, on the other hand, any corrosion is local to the damage area and easily repaired.

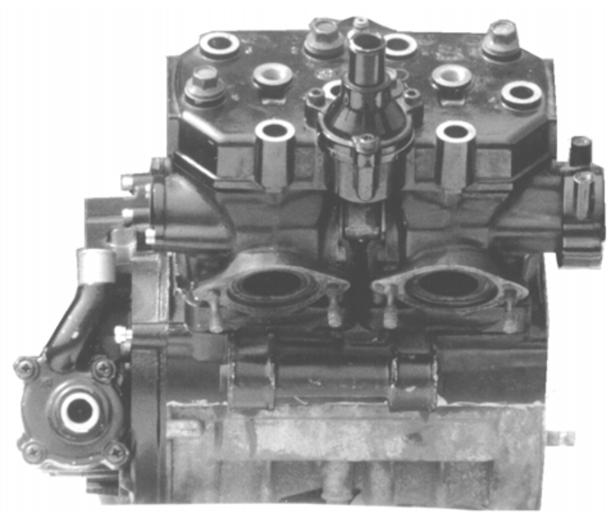
### Anodizing

Although some aluminium alloys have good corrosion resistance, the tell-tale white oxide powder on aluminium parts is an all-too-familiar sight, especially when a bike is ridden on salted winter roads. Anodizing, which involves immersion in an acid bath, is a protective process that prevents such corrosion by putting a tough oxide film on the surface. This oxide layer may be dyed – usually, grey, gold, red, blue or black – to provide an attractive appearance as well as protection. Some alloys benefit more than others from anodizing and wrought material usually responds better than cast.

## Design layout

When I was involved with making one off and small production runs of special frames, there was not the proliferation of desktop computers and cheap CAD software that there is today. In common with others all design work was done by hand on a drawing board, so if you don't have access to some of these modern design tools there is no need to despair, pencil and paper are all that's really necessary. There are many people that I know that claim that high end CAD software is as essential a tool as good fabrication equipment, welders and lathes etc. but generally these are people that use CAD in their daily employment and hence are skilled in the method. So if you are serious about building your own frame then use whichever design method is best for you, if you are familiar with CAD then it can be an extremely powerful tool and design aid but if you have never used it and your main concern is building and riding your own machine in a reasonable time period, then be aware that the time spent learning to use this type of software may well be longer than the time spent in the workshop actually building your dream. On the other hand if you plan on getting some parts machined elsewhere then having previously designed these parts on computer and produced compatible files, you might find that having them made at a CNC machining centre may well save you some money.



**Photos like these of a 250 Yamaha can be scaled from a reference dimension and overlaid on layout drawings. In the absence of engine drawings this technique can prove to be very useful. The photos are best taken from a long distance with a telephoto lens. Even with CAD, in the absence of engine drawings, photos like this can be scanned and overlaid on the drawings, or digitised with a drawing tablet.**

**Photo editing software can sometimes correct perspective distortion, to assist with this it is useful to include horizontal and vertical straight edges in the picture as a reference frame.**

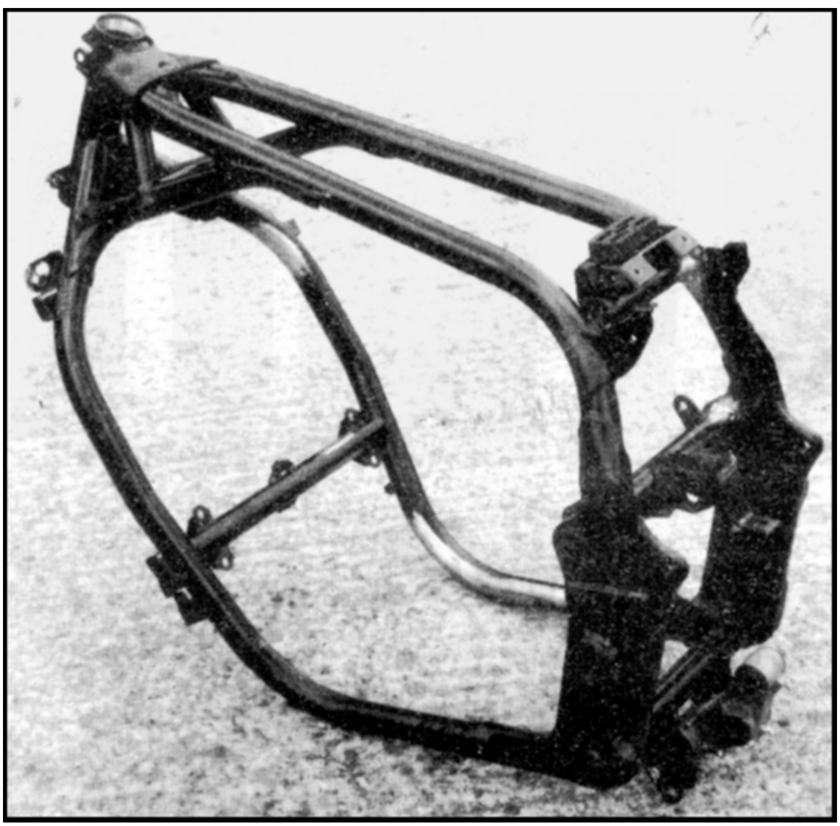
When laying out your design, whether by computer or drawing board the most difficult part to get accurate dimensions for is usually the engine-transmission unit. Sometimes you can be lucky and find dimensioned drawings in a service manual, even if not dimensioned these can be very useful as you can always measure a reference dimension and scale from there. If using a computer you can scan or digitize the image and do the necessary scaling, if doing it by hand then a simple method would be to size the image to the required scale by copying it on a magnifying photo-copying machine. In most cases it is likely that if you are building a one off special that the actual construction will be done around an actual engine and so using images scaled thus, will provide sufficient accuracy for your layouts.

A technique that I often used with success in the absence of engine drawings was to photograph the engine from the sides and front and rear. It is best to do this from as large a distance as possible to reduce perspective distortion, so a long telephoto lens is preferable if possible. I would then enlarge the photo to the required scale using some reference dimension. Again the scaling could be done with a photo-copier or computer. Modern photo editing software often has features that can be used to reduce the effects of perspective distortion, and so can be used to correct images that were not photographed from far away. It is useful to include horizontal and vertical straight edges in the picture to act as a reference for such correction.

# 17 Case study

This chapter describes some of the practicalities involved in trying to improve the stiffness of a standard road frame. The frame in question is a traditional bent tubular type in steel. This is likely to be the most common type of frame in need of such modifications.

I was asked to stiffen two frames for a 750 Kawasaki, the current model in the late 1980s. Despite favourable comments regarding the handling of this machine in many road tests, deficiencies were soon brought to light when subject to the much higher rigours of the race track. These particular frames were to be raced in a class where the rules forbade any modifications which entailed removing any parts, but it was permitted to add. As weight was obviously a priority, brute force was not the way to go, any added material had to earn its keep.



**The unmodified frame.**

**Lateral twisting of the head stock is only prevented by the relatively flexible two top tubes and the lengthy bottom tubes taking a curved path back to the swing arm pivot area.**

## Measurement

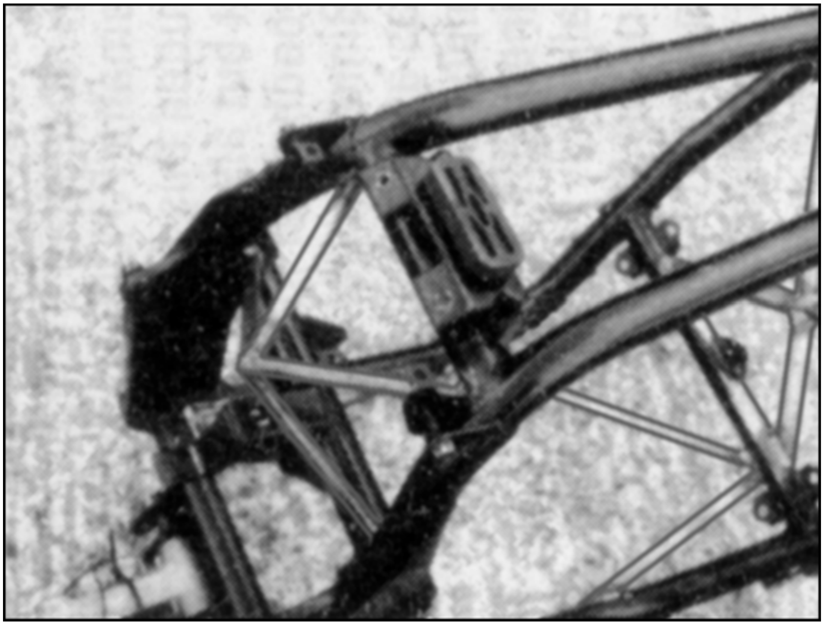
Before starting work on such a project it is very useful to measure the frame stiffness, as this way you can keep a check on the effectiveness of your work. This is not as difficult as it sounds, because you are only after a comparative figure and great accuracy is not necessary for our purposes.

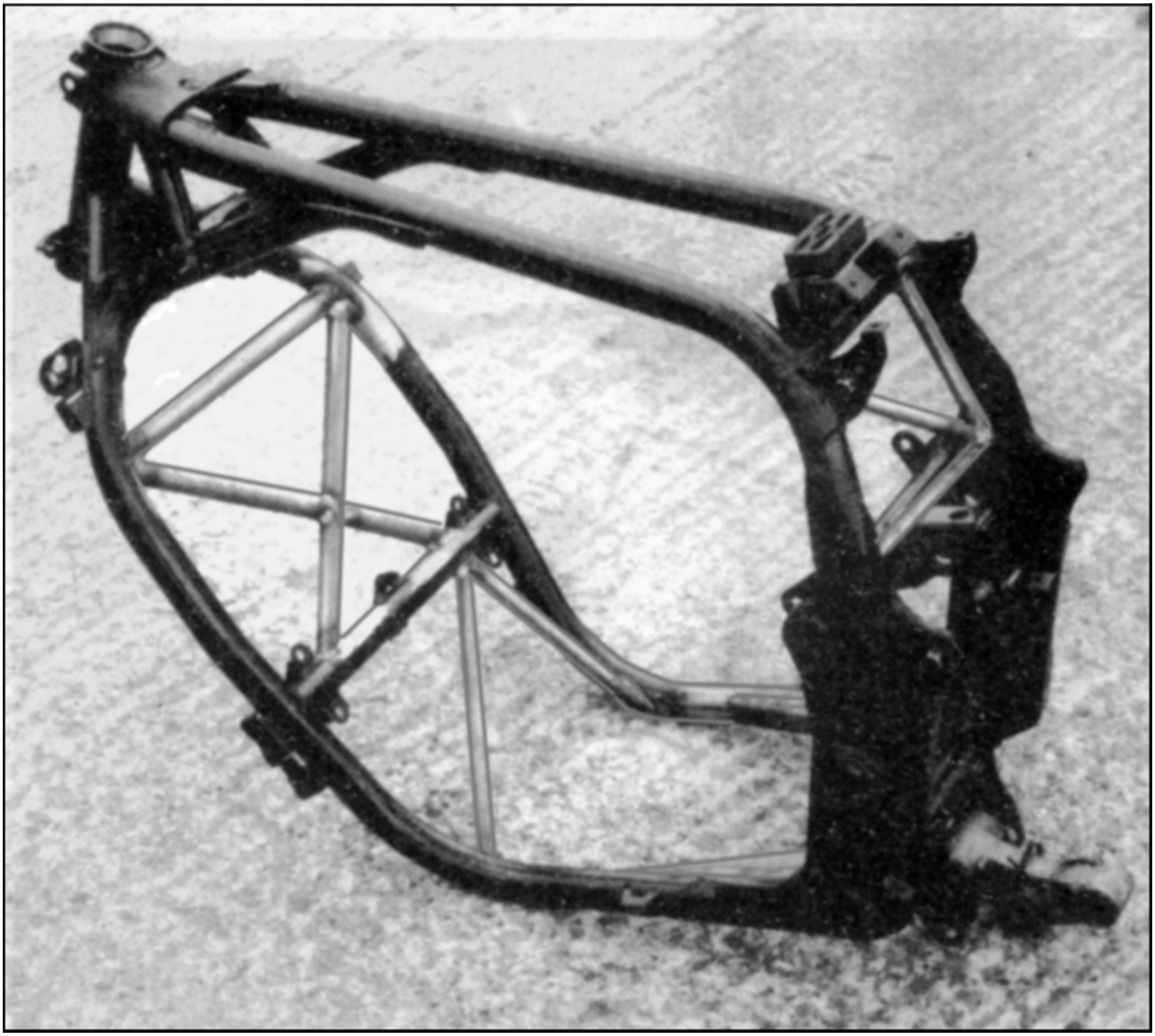
I used a heavy piece of tubing which was machined to be a good fit in the swing arm mounting of the frame, this piece of tubing is then fixed to a rigid and heavy piece of machinery in the workshop, although any solid object will do, such as a wall. For convenience of loading this mounting tube is located vertically, so that when the frame is mounted on it the frame lies horizontally. The frame can be loaded in torsion and lateral bending by applying a force to the end of another piece of tubing through the head-stock, this should be a good fit in the head-stock and if about three or four feet long it will be possible to significantly flex the frame with moderate hand pressure on the end of this tube. Frightening isn't it? Whenever I have done this in front of an audience, there is disbelief and amazement at the degree of deflection that can so easily be produced. If a constant load is applied through a spring balance always in the same place along the tube then we can compare the frame stiffness during the course of modification. Perhaps a more valuable consequence of this controlled loading is that we can actually see and measure the pattern of deformation within the frame. This makes it very easy to assess where it is most important to put bracing tubes and where it would be largely ineffective.

## Main frame

The photographs show the finished modifications that were done on this Kawasaki 750., and incorporated are examples of many of the techniques mentioned in other chapters.

**This pyramid above the swing arm pivot area is a very effective way of adding some torsional stiffness to an area subject to twisting deformation.**



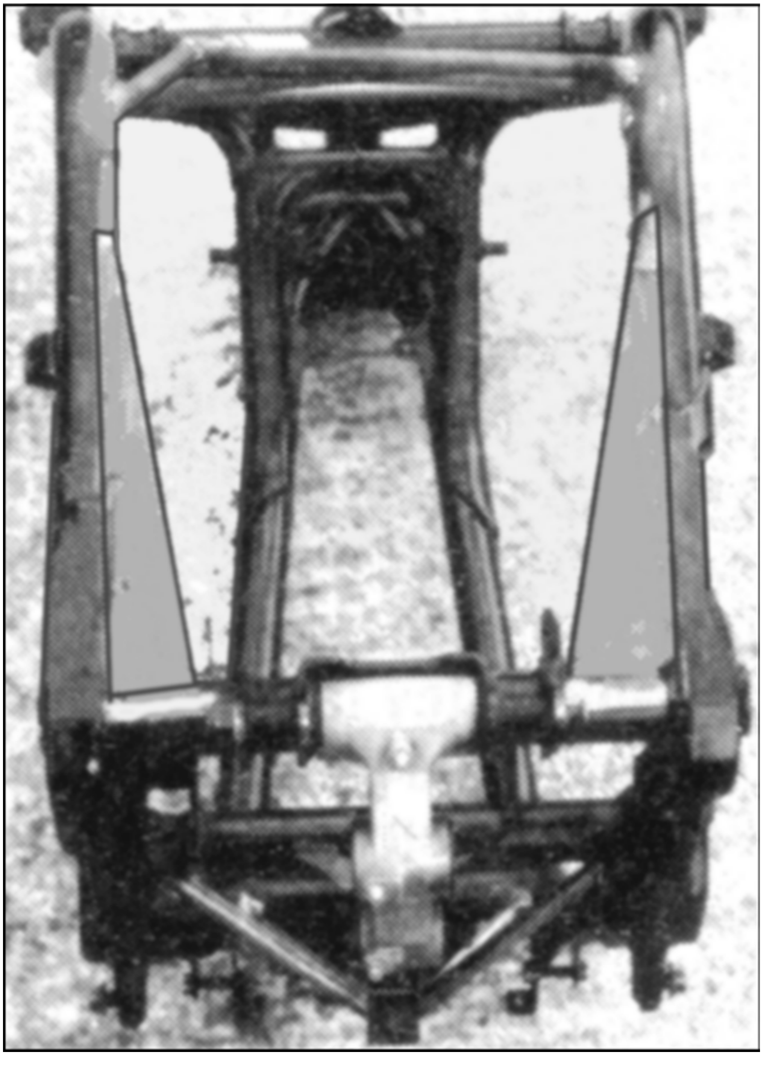


**The completed modified frame. Note the triangulation of the front down tubes, the asymmetric layout was determined by location of exhaust pipes and crankcase shape. The pyramid above the swing arm pivot provides in-plane torsional stiffness as well as lateral bracing.**

In front of the engine a large open area has been heavily triangulated, the actual layout of the addition tubes being dictated by the need to avoid the exhaust system and parts of the crankcase, there was not the space to use the pyramid method and all the bracing lay in a flat plane. This modification was particularly effective, the torsional stiffness of the bare frame being more than doubled.

The swing-arm pivot area is subject to high loads trying to deflect it in a fore and aft direction, these forces arise from the pull of the chain and any lateral loading on the rear wheel. This tends to try and twist the frame sections immediately above and below the swing-arm pivot, the section above is the most affected, simply because it is longer. To limit this flexing a pyramid was added. This is not always possible due to the location of large air-boxes and/or battery.

The large open four sided area under the engine could be seen to be lozenging when loaded on the stiffness testing rig, but the engine itself prevented any cross bracing and so the two rear corners were gusseted as much as room allowed. The similar area above the engine was approximating to a long thin triangle anyway, and so needed no similar treatment.



**The folded sheet metal gussets helped brace the large open area under the engine. Their size was limited by the crankcase and gearbox casings.**

## Engine mounting

So far we have considered only the bare frame, the engine unit is quite rigid and when bolted in place has great potential as an aid to stiffening the whole structure. Unfortunately, with modern bikes there are two factors that greatly reduce this effect. Firstly, the trend to rubber mounting, whilst good in other aspects, does nothing to help the handling. Secondly, to reduce production costs, mounting holes in crankcases are cast in rather than being drilled, this results in tapered holes that are a lose fit on the mounting bolts. These bolts are usually 10mm. in diameter, but the smallest part of the hole will be nearer 11 or 12mm. Considerable stiffening can occur if attention is paid to these two areas.

Rubber bushes can be replaced with specially machined aluminium ones, and the mounting holes can be carefully reamed out to either 11 or 12mm. (this job is best done with the motor in-situ, to ensure correct alignment) The engine bolts can then be replaced with ones of the correct diameter; I usually use stainless steel for these.

Both of these methods were applied to the Kawasaki, and in addition two detachable tubes were added between the outside head mounting studs and the front frame down-tubes, this largely braced the sides of the frame.

## Results

So how effective were these modifications? Well, less than 10% was added to the bare frame weight, which went from 12.7 kg. up to 13.8 kg. I cannot give you accurate figures for the stiffness increase because it became so stiff that most of the flexing was then taking place in my supporting jig, but in contrast to the unmodified frame, it was very hard to detect significant movement between any sections of the frame, when loaded in the jig.

If I was pushed to guess then I think that there was an improvement in torsion stiffness of between about seven to ten times.

## Material

Just a word about the size of tube needed for this type of bracing work. In the figures for the relative flexing of a tube in bending or tension/compression, we saw that the stress levels and deformation were minimal in the tension situation, and so quite small section tubes can be very effective. It is not necessary to use tube sizes similar to those already in use on the frame. Unless the bracing tube is long and subject to compression loads which may cause buckling then 13 to 15 mm. diameter with 1 or 1.5 mm. wall thickness should be more than adequate.

## Swing arm

It is all very well stiffening up the main frame loop in this way, but quite often it is the swing arm that is the major source of torsional movement. Unfortunately, I was not given the opportunity to either measure or modify the arm for these Kawasakis, so I don't know how the frame changes related to the whole assembly. Probably the most effective way to stiffen this item is with bracing similar to that used years ago on the Vincent and more recently the "cantilever" Yamahas.

In reality these are just versions of the pyramid that have been compromised by the need to avoid tubes going through the wheel, etc. There are a number of so-called "braced" swing arms on the market, but sad to say, many of them are of no more than cosmetic benefit. A lot of the flexibility in a swing arm is due to twist of the pivot tube, and hence those arms that only feature bits of added tube along the sides are not going to help much.

## Forks

The final link in the chain that is responsible for holding the wheels in line is of course the front forks. I'm not a great fan of telescopic forks but there are various ways in which they can be improved.

If money is no object, go out and buy top quality replacement USD units with large stanchion diameters and a large wheel spindle. If you are stuck with your originals then fit a brace above the wheel, but get a good quality one or don't bother. Like alternative swing-arms there are many ineffective ones on the market, make sure that it is rigid and equally important it must be accurately made or it may distort the fork alignment and prevent free movement of the sliders.

If you have the facilities then changing from the usual 15 or 17mm. diameter wheel spindle to a more rigid 20mm. one (like those used on some Italian machines) can be quite effective.

## Caution

Now just a word of caution. Frame stiffening as discussed will in most cases significantly reduce the stress levels in frame members as well as stiffening the whole structure, but there are occasions where the stiffening of one part of the frame may lead to increased risk of failure in another unstiffened area. A flexible frame acts as a spring and can absorb and reduce the effects of some types of loading, if only parts of the frame are stiffened then we may pass more load through to the unstiffened areas which may deform locally more than before even though the whole frame deforms less.

## Tuning

Well, now you have a rigid frame to work from, but that's all it is at the moment. Handling will probably have improved somewhat already but to get first class results you must start the fine tuning process. That is, selecting spring rates, matching tyres, changing geometry by moving the fork sliders in their yokes, etc. the list is endless.

# 18 Future developments

## The status quo

Throughout the history of motorcycle development there have been many brave attempts to advance the state of the art through radical changes in design. Yet it needs only a cursory glance at contemporary production machines to realize that these efforts have largely been in vain. In overall concept, today's bikes remain similar to those of yester-year, differing mainly in evolutionary improvement. Compare, for example, the featherbed Manx Norton of the early 1950s with the average present-day racer. Both have hydraulically damped telescopic front forks mounted on a high steering head; both have pivoted-fork rear suspension and both carry their considerable petrol load in the highest possible place – above the engine. The riding position is also unchanged. There have, of course, been vast improvements in performance during the several decades since the advent of the featherbed Norton – but they have been detailed rather than fundamental. For example, tyres have improved considerably in grip to the benefit of roadholding, cornering and braking. The brakes themselves have changed the most, with hydraulically operated discs superseding cable and rod operated drums on all high-performance machines. Suspension characteristics have been refined, the most significant trends being towards pressurized gas-filled shock absorbers, with shim stack control, externally adjustable damping and triangulated or braced rear forks with progressive rate linkages.

There is, of course, little point in embarking on a new design unless it is an improvement on the one it replaces – and improvement means different things in different markets (e.g. speed for racing, economy for commuting). But why have improvements come only through evolutionary changes and not through revolutionary new ideas? It seems inconceivable that the basic layout of the earliest motorcycles was so intrinsically excellent that fundamental improvement was impossible – much more likely that radical advances in design have been killed by commercial imperatives, even on competition machines. The major manufacturers are obliged to maximize the financial return on their activities – hence, in the heyday of the British industry, the entrenched but short-sighted reluctance to alter any design in such a way as to require substantial investment in new tooling. When rear suspension became popular (following racing successes) most manufacturers simply altered the rear end of the existing rigid frame so as to obviate re-jigging the front. Another example was the use of a common frame for engines of different capacities and types (singles and twins). It was hardly surprising that progress was slow and the industry foundered. Japanese manufacturers cannot be accused of reluctance to retool for new models. Indeed, they may have moved too far in the opposite direction, producing a rapid proliferation of models with few common parts, to the consternation of importers and spares stockists. Even from Japan, genuine technical advances (as opposed to marketing gimmicks) have been slow to appear, for the motorcycle market is essentially conservative and fashion-conscious, hence resistant to radical changes in design. The surest recipe for rapid acceptance of change is success in racing, but even in this intensely competitive branch of the sport radical advance comes slowly. One reason for this is the domination of racing by the major manufacturers, who prefer their track machines to bear at least a superficial resemblance to their catalogue roadsters. Paradoxically, another factor is the high ambition of the top riders, who are not willing to risk being uncompetitive during the development period of an unconventional machine, whatever its ultimate potential. Consequently, any such machine would have to be allocated to riders of lower status and – in competition with star riders on heavily sponsored factory bikes – would seem a failure. This persistence of the status quo is well illustrated by the history of endurance racing – which was initially the province of dedicated riders and dealers rather than factorybacked stars. Once the early problems of reliability were solved, the quest for enhanced all-round performance, free from conservative influences, led to an upsurge of technical innovation, and though some designs were a bit harebrained, others had appreciable merit. But then the big factories (notably Honda) recognized the publicity value of the championship and brought the full weight of their resources to bear on it – successfully of course. An unfortunate consequence was a stifling of innovation: the manufacturers succeeded with conventional, but highly developed, machines and so most other contestants reverted to that type. Perhaps another reason for slow fundamental change is that ultimate roadholding and hence cornering speed may be little affected by such a change. In the car world, cornering speeds have increased much over the past decades. Tyres played a large part in this but improvements in suspension geometry were necessary to allow these tyres to do their job properly. Aerodynamic down force then made an appearance with overwhelming superiority. On a bike, so long as the chassis is rigid enough to maintain wheel alignment and the springing and damping system is able to hold the wheels on the ground, then variations in suspension methods and/or geometry will have only minor effects on braking and cornering power. Hence modern racing lap times depend more on engine, machine weight and tyres than on revolutionary suspension layouts. The benefit, if indeed there is any, comes in the stability and feel or handling of the machine.

## Future possibilities

Predicting specific design trends is risky but in general there is unlikely to be any radical change in concept so far as mass-produced machines are concerned. As in the past, change will come through small steps on successive models. As always, we must look to small concerns and talented individuals for serious efforts to develop radical ideas (such as the Elf-sponsored French endurance racers from the 1980s). Unfortunately, however, few ideas of that sort ever find acceptance in the harsh commercial world.

Probably the only thing we can say with confidence about future design is that electronics are bound to play an increasing part. At the moment the main chassis related use of electronics is in ABS braking, but control over suspension response is an area that could be of great benefit.

Only a fool would predict more than that and so the following are not so much predictions but more a self indulgent short listing of some possibilities of interest to the author.

## Active suspension

The term ACTIVE has been coined because these systems rely on some external power source to respond ***actively*** to the suspension inputs. Whereas, the conventional designs, ***passively*** allow themselves to be pushed and pulled about under the action of the dynamic forces.

Active suspension offers the promise of a system that can block the passage of virtually all road disturbances without the usual compromises necessary to account for inertia effects under braking and acceleration.

The idea of active suspension is not new, Citroen have employed a form of slow acting active ride height adjustment for many years. The bump absorption and pitch and roll resistance being supplied by pneumatic springs and hydraulic dampers, with an active hydraulic system to maintain a pre-set average ride height for different vehicle loading.

Automotive Products demonstrated a similar but much faster acting system which could be set to control roll and pitch effects as well, the price paid for this being the higher power requirements of a much bigger hydraulic pump.

The most significant milestone came from Lotus Cars with the application of electronics to control hydraulic actuators that provided total suspension support without springs nor dampers. The drawing shows a simplified layout of this design.

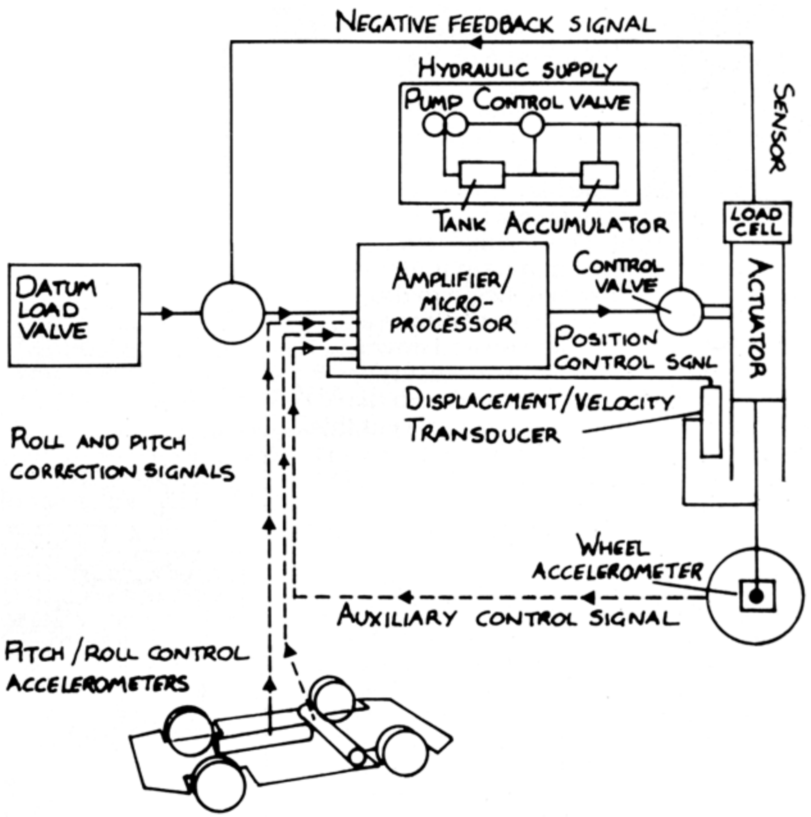



Fig. 18.1 **Simplified layout of the early Lotus active suspension system. This diagram only shows one wheel. Although much subsequent research has been done in the car field this type of system could usefully be applied to motorcycles.**

As the wheel moves relative to the body the transducers sensing load, wheel position, acceleration and velocity, send their signals to the computer together with the output of pitch and roll accelerometers mounted in the body. These signals are processed to determine whether the wheel movement is due to bumps, cornering roll or to acceleration/braking induced pitch. Signals are then passed to the control valves to leave, increase or decrease the amount of fluid in each actuator. The result is a system which can eliminate pitch, roll and ride height variations while at the same time giving a very comfortable ride. Often the requirements for a smooth ride are in conflict with those other needs, and so as usual a compromise must be struck. Active suspension can greatly reduce this need for compromise. Since the first Lotus designs were unveiled much progress has been made, particularly in the electronics and control algorithms and even though these developments have been mainly aimed at cars, it can be argued that motorcycles actually have more to gain from improved suspension response.

Motorcycles provide difficult challenges to the suspension designer for several reasons. Static load variation can be quite high, a mid range road model for example can have the overall weight easily doubled with the addition of rider and pillion with some luggage. The high CoG height to wheelbase ratio causes large load transfers under braking and acceleration as well as pitching tendencies in general. Add to this the pro-dive characteristics of the ubiquitous telescopic forks and a relatively low value for the ratio of sprung to unsprung mass ratio and we have a set of requirements that is very difficult to fulfil with a purely passive system.

However, if we program an active system to provide the necessary responses to the wheel movement and load sensing transducers, then we could have a pitch free, exceptionally comfortable ride with self levelling that would compensate for load changes and cornering forces.

This may sound very complicated and hence perhaps prone to trouble. Actually the hardware is no more complex than that long used on the Citroen cars, all the complex calculations and decisions are made by the ubiquitous silicon chip. Properly designed and programmed these devices can reliably process many times the amount of information needed for our machines. I think that the improvement in ride comfort and handling possible would be worth the complexity, but an additional price to pay for these benefits would be the high power required to drive the pump. It would probably be several BHP which may be unacceptable on all but the largest machines. It may be possible to use a simpler "semi-active" system with smaller power demands and still derive most of the benefits. The response time needed to deal with the problems of pitch, cornering and weight change is much slower than that for handling bumps. So perhaps we could separate these requirements and use some passive components as well to reduce the power requirement.

## Rheological Fluids

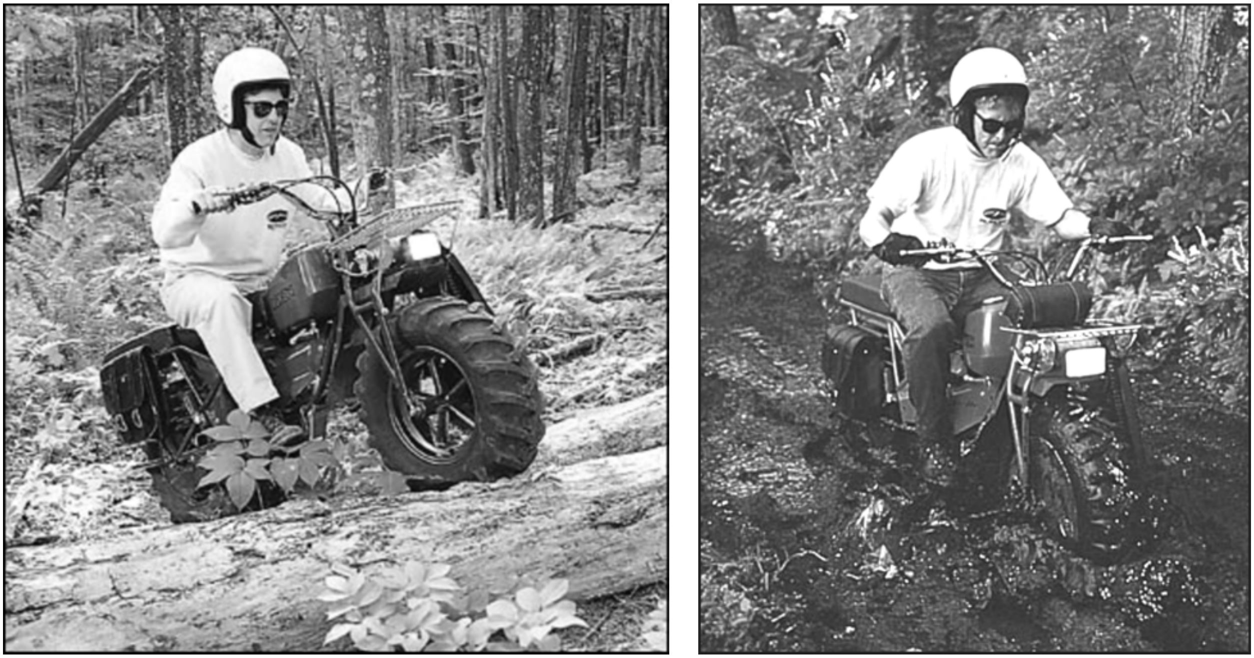
These are solids-filled fluids that change in viscosity relative to the level of electrical current or magnetic field induced in the fluid. The electrically controlled fluids are known as Electro-Rheological Fluids or ERF, and the magnetically controlled are called MRF or Magneto-Rheological Fluids. These fluids have obvious potential applications in adjustable dampers. The response time (in the order of 1 msec.) to change damping is fast enough to allow for in service continuously variable real time adjustment to suit changing road conditions, acceleration and braking, and cornering requirements.

Whilst not providing all the potential benefits of full active suspension they have the possibility of addressing the two main disadvantages of an active system – cost and the large power requirements. The electrical power needed for both the ERF and MRF systems is minimal. With appropriate control algorithms the damping could be controlled by micro-computer to optimize comfort and road-holding. There have been various promising developments in the car world for such systems but the author is unaware of any current motorcycle based research in this area.

A current problem with these fluids is limited life, they appear to "wear out" and lose their ability to thicken adequately. No doubt this problem will be solved with time.

## Two wheel drive (2WD)

Two wheel drive is far from a new idea and has obvious appeal to anyone that has ridden through mud or sand. Probably not the first to try it but the Canadian built Rokon has been produced since 1960 with chain driven 2WD. These specialized forest vehicles have very large balloon tyres and can easily be ridden over felled trees and other obstacles, even floated across rivers. Some other prototypes gear the front wheel to spin slightly slower than the rear, and have a free-wheel system, so that in normal conditions the drive is only through the rear wheel. When the rear loses enough grip and spins then the front wheel becomes driven also. The Rokon is the only 2WD machine that made it into production, and that uses a fixed drive ratio of 1:1 with a free-wheel system.



**The 2WD Rokon makes riding over logs and through deep mud look easy. The model on the right has sealed aluminium disc wheels which with the balloon tyres allows the machine to be floated across deep streams.** (Rokon)

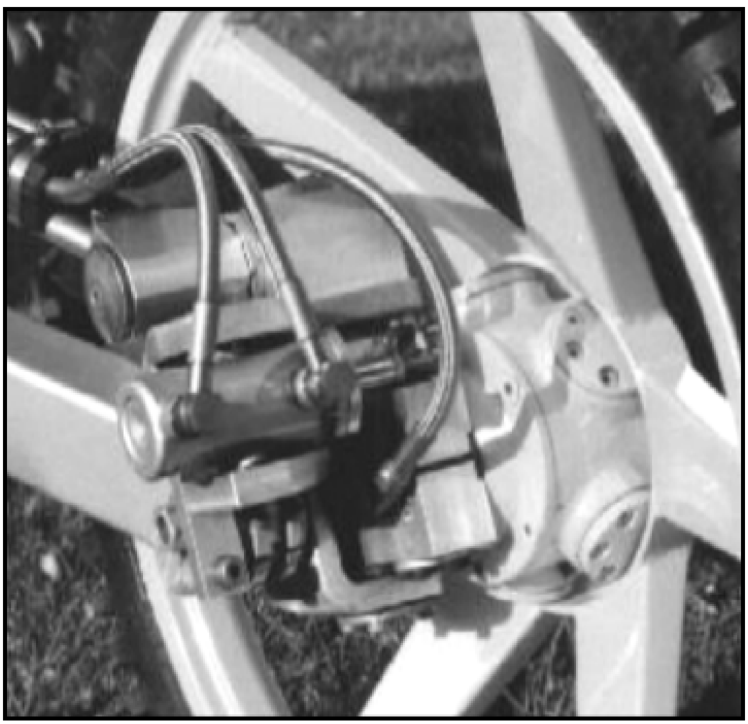
Trials riders, with rear wheel drive bikes, can probably surmount greater obstacles than is possible with the 2WD Rokon, but this is exceptional and average riders do not possess such skills, and it is for these that the 2WD system extends the possibilities.

Mechanically transferring power to the front wheel is not an easy task, particularly with telescopic forks, and if done by chain or belt results in a multitude of chains, sprockets and tensioners etc.. The Rokon uses front forks with no suspension to simplify the transmission of power, the large tyres provide sufficient cushioning. There have been designs that have used rotating shafts, often with some form of Hooke's or CV joint. Either of these systems are generally complex and unsightly, which might partially explain the reluctance of any manufacturer to test the market.

Other possibilities that would allow for easier installation are electric and hydraulic drive. An electric motor could be mounted in the hub of the front wheel and then it would only require a pair of wires to move with the suspension and steering motions. A generator or alternator would have to be driven by the engine to feed this motor. The main downside to this form of drive, which is used in other fields such as diesel-electric trains, is the size and weight of the motor and alternator. Even a modest power transmission to the front of say 3 to 5 BHP would require a very heavy motor, adversely affecting unsprung mass and steered inertia.

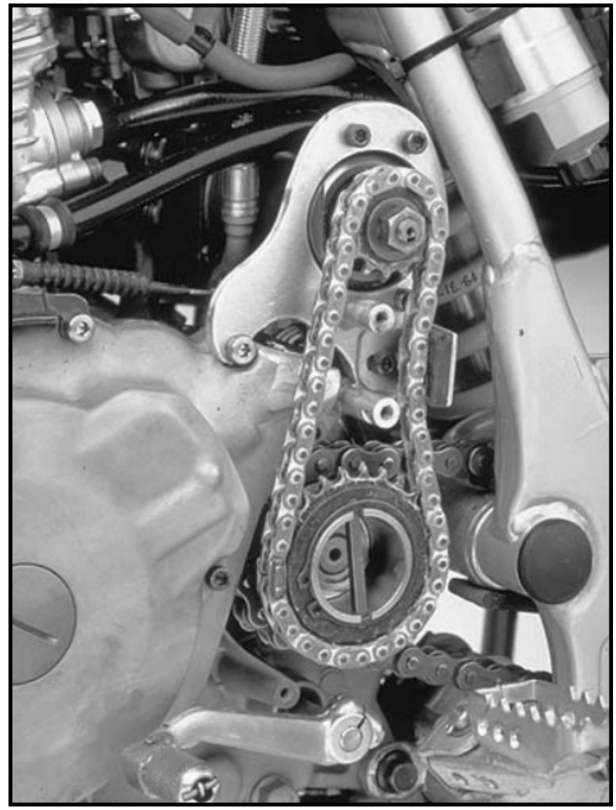
Hydro-static pumps and motors are used in various applications ranging from heavy earth moving equipment to aircraft, and are relatively small and light-weight if used at high pressure. In his teens the author started a 2WD project using such motors in both front and rear wheels, but his youthful enthusiasm was not matched by sufficient youthful funds and the project died. One advantage of such a drive system, whether used in a 2WD format or normal rear wheel drive only, is that continuously variable transmission ratios are possible (by dynamically changing the piston stroke of either or both of the motor or pump) which can be controlled automatically. Depending on the control algorithms, this could be used to provide maximum performance or gearing characteristics for maximum economy.

Australian Ian Drysdale completed such a 2WD off-road machine, which also featured hydraulic two wheel steer. To allow for steering and suspension movement it is necessary to use some form of flexible connection between the wheel mounted motors and the main chassis but flexible hoses become extremely stiff when pressurized. The Drysdale surmounted this problem by using swing-arm suspension at each end and using oil tight rotating joints mounted coaxially with the pivots for the arms. This machine reportedly worked extremely well in slippery conditions.




**The Drysdale 2WS and 2WD motorcycle. The five piston radial hydraulic motor can be clearly seen in the hub of the front wheel shown on the left. Note also the cross-wise cylinder, which is for steering. The photo to the right shows the rear wheel in a steered position. This machine now resides in the Donington Collection Museum.** (Drysdale)

More recently Ohlins/Yamaha have shown a very neat conversion to 2WD for motoX use. This uses conventional chain drive to the rear wheel with a small hydro-static motor in the front hub. An hydraulic pump is chain driven by an extension sprocket off the standard gearbox sprocket. The size of the motor and feed hoses indicate that this system is only intended to pass a relatively small amount of power to the front wheel. Even a small amount of drive from the front would probably be of great assistance in terrain where the front can get bogged down, but when we see that motoX riders keep the front wheel in the air for large periods of time it is only natural to query whether there is much benefit to be gained from this development at the top level of competition. It is probably the less skilled rider that would benefit the most in off-road terrain.

**Ohlins/Yamaha 2WD system. The photo on the left shows how the hydraulic motor is fitted to the front wheel hub. The size of the motor and the feed hoses indicates that only a small proportion of the engine power is directed to the front wheel. On the right is the detail of how the pump is driven at an increased speed from the rear drive sprocket.**

On the road and race-track modern bikes often have an excess of power over that which can be transmitted through just one rear tyre. At first sight it might seem that 2WD is an obvious answer, but more considered reflection indicates that it might not be so simple. To begin with, under conditions of high power delivery there is often little or no vertical load acting on the front wheel, in which case the front can't produce additional traction force. We would have to lower and/or lengthen current designs to reduce load transfer to take advantage of front drive. In a racing context this may slow handling to an unacceptable degree. Reducing the load transfer allows us to use some front drive but there will then be less load on the rear, reducing the maximum available drive from that end. Due to the non-linear grip/load characteristics of tyres, discussed elsewhere, there would probably be a nett gain in traction but this is unlikely to be large. In the same way that front only braking is directionally unstable (see chapter on stability), so too is rear wheel drive only and any transfer of drive from the rear to the front will work toward a more directionally stable machine.

The use of 2WD on tarmac may also introduce undesirable handling quirks, due to the interference that the driving forces would have on the steering. These forces act on other front wheel drive vehicles to a certain extent but it is our need to lean whilst cornering, and our dependence on the steering for balance that puts any two wheeled single track vehicle in a class of its own. To illustrate this problem, just think of the effects when the front brake is applied during cornering. The machine usually tends to stand up. If we were to drive the front wheel then the application of power would tend to create the opposite effect, and make the machine lean over farther. As we lean over in a curve the tyre contact patch moves around the tyre toward the inside of the bend, out of alignment with the steering axis. Any forces on the tyre will now create a torque acting about the steering axis, this torque acts on both wheels but the effects on the steering front are much greater. This torque produced by the driving force will act in a way to turn the steering to the outside, but as we steer by countersteering this will have the effect of leaning the machine into the bend. This is a reaction that in most cases would oppose our requirements, whilst in a curve we generally open the throttle on the exit when we wish to straighten up. Not only would these effects be evident when the throttle opening is changed, but a steady torque would be felt even on a constant power setting. We would be required to apply a compensating force to the handle-bars. This force will vary depending on the level of power being used. On the overrun a torque steering out of the bend would be needed, with the opposite action necessary for the power-on situation. The magnitude of these upsetting torques may be less under power than under braking, but we can usually avoid the need to brake in a corner. However, we cannot avoid the use of varying amounts of power in the bends and so the feel of the bike would be inconsistent. The modern trend toward ever wider tyres would aggravate this problem, because the contact patch can move even farther from the steering axis, so producing higher unsettling steering torques. We can conclude that the application of 2WD to road or racing machinery would introduce problems in need of development, additional to the obvious mechanical ones.

## Two wheel steering (2WS)

At the end of chapter 4 it was hinted at that 2WS might increase the maximum possible turn-in performance, because with front steering only, the rear tyre offers little help during the leaning transient.

2WD bikes are hard to find but examples of 2WS are even scarcer. The Drysdale featured above is the only one known to the author. Let's take a closer look at 2WS and see if it really has anything to offer future development. 2WS can take two basic forms;

- Same sense – both wheels turn in the same direction.
- Opposite sense – each wheel turns in an opposite direction.

The most obvious departure from the norm will be in the paths followed for a given steering angle, figs. 18.2 and 18.3 show the difference in the kinematic (low speed) turning circles to be expected. Fig. 18.2 assumes that the ratio between the front and rear steering angles is 1:1. Whereas the curves in fig. 18.3 show the normalized turn radii for different values of front steer angle over a wide range of front to rear steering angle ratio. The radii are normalized by dividing the actual radii by the wheel base.

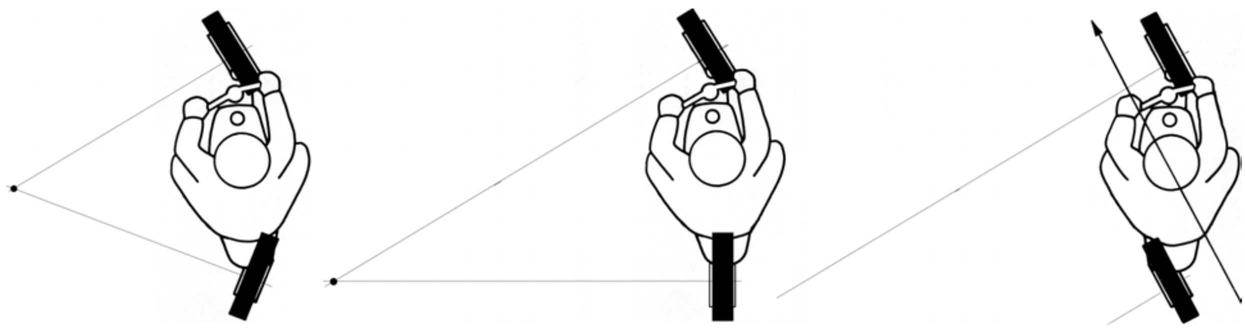


Fig. 18.2 **Two wheel steering.** *Left:* **The small turning radius of "opposite sense" 2WS is demonstrated, compare this with a normal front wheel only steer system,** *middle*. **"same sense" 2WS,** *right,* **with a 1:1 ratio between front and rear steering angles, clearly has the turn centre at infinity giving a straight line motion parallel to the wheel angles. The un-steered part of the bike would stay aligned with its original orientation. The machine would not negotiate a curve but could instead, adopt a directionally stable straight line attitude over a range of steer angles, resulting in the machine moving crab-wise as shown. The tendency to keep the wheels aligned with the chassis would be quite weak. The normal self aligning effects of trail would be satisfied as long as the two wheels remained parallel to the direction of travel.**

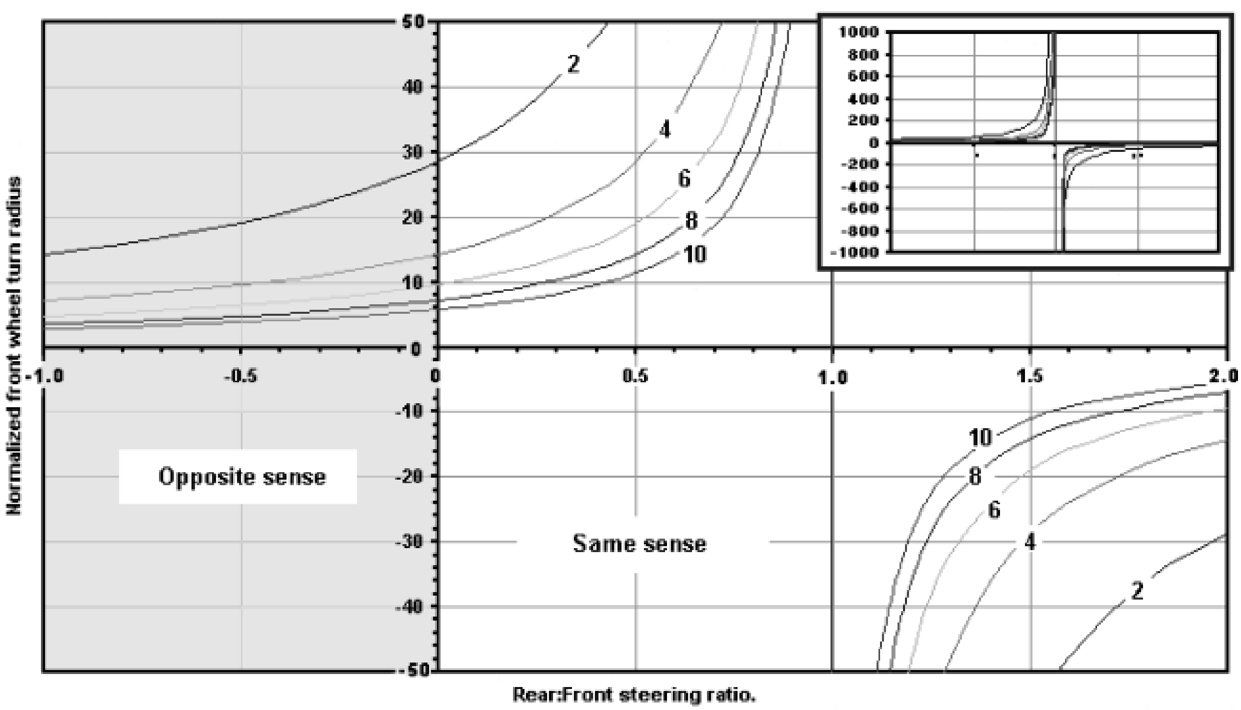



Fig. 18.3 **This plot shows the actual kinematic turn radii of the front wheel for different ratios between the steering angle of the rear and front wheels. –1.0 being opposite sense steering with equal steering angles, 0 is when the rear wheel is fixed and doesn't steer. The individual curves are for different front wheel steering angles as marked in degrees. The vertical scale has been changed in the inset to highlight the singularity that occurs with equal steer angles at each end.**

These curves show a singularity (the radius is + or - infinity) at a steering ratio of 1.0. Fig. 18.2 shows that the motorcycle just travels in a straight line at this ratio, i.e. it has an infinite radius, turning neither into nor out of the curve. When the ratio is less than 1.0 by even a small margin the machine will turn towards the wheel direction, but any ratio over 1.0 will result in a path away from the direction of the steer angle, represented by a negative radius in the plot.

Same sense steering has obvious benefits when parking in a tight space, the vehicle being able to move essentially sideways, but this is likely to be more useful with a car than a bike. Whereas, the tight turning circle of the opposite sense configuration could have useful low speed manoeuvring benefits, especially with a large heavy weight tourer.

We've seen in chapter 4 that low speed balance is a question of steering such that the line joining the two tyre contact patches moves back under the CoG. For an average machine with a longitudinal weight distribution of around 50/50, it is the path of the mid-point of the wheelbase that concerns us in this regard. This same criteria is also relevant to start the lean process during the countersteering phase of corner entry. Fig. 18.4 shows these paths for opposite sense 2WS, normal FWS and same sense 2WS.

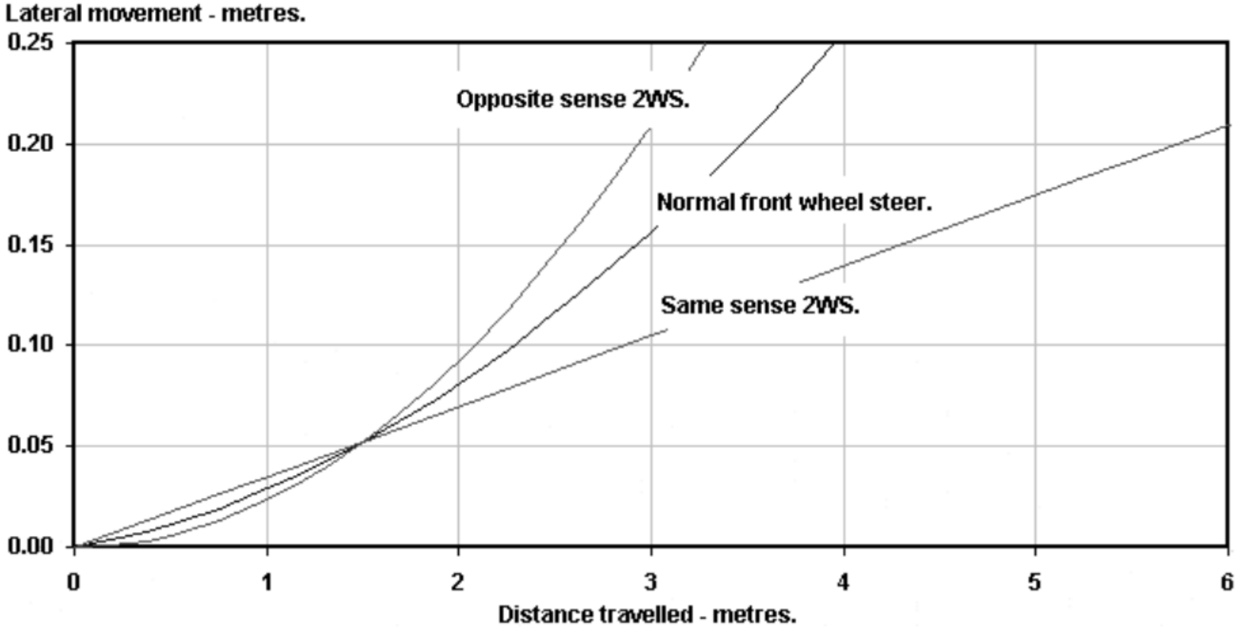



Fig. 18.4 **The kinematic (low speed) paths, of the mid-point of the wheelbase, travelled by the two opposite classes of 2WS and a normal front wheel only steer. All three curves are for a wheelbase of 1.5 metres and assume an instantaneously applied steering angle of 2°, and the two 2WS cases have front to rear steering ratios of 1:1. At the low values of steering angle used in normal riding (less than about 10°) the paths cross at one bike length.**

When the bike has travelled less than one wheelbase length from the application of equal steer angles we can see that same sense 2WS has the most agile lateral performance, but the situation reverses and the opposite sense arrangement comes out on top after this initial distance. At 5.5 km/h it takes 1 second to travel a wheelbase length of 1.5 metres. From a low speed balance view point, 1 second is quite a long time and important balance correcting manoeuvres need to be made within that time. It would seem therefore that very low speed balance might be enhanced with same sense 2WS. On the other hand the opposite sense system would initially suffer a delay (fig. 18.5) before the mid-point of the wheelbase would begin to move significantly, leading to a sluggish response.

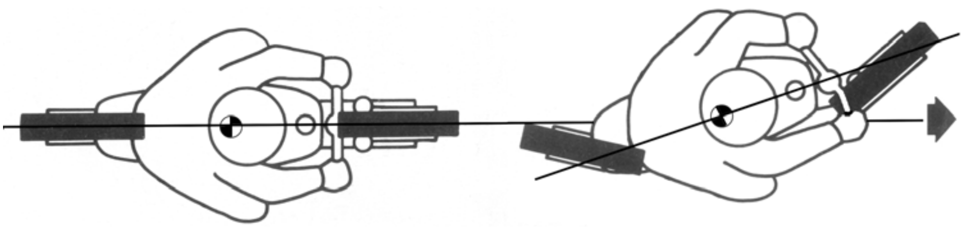
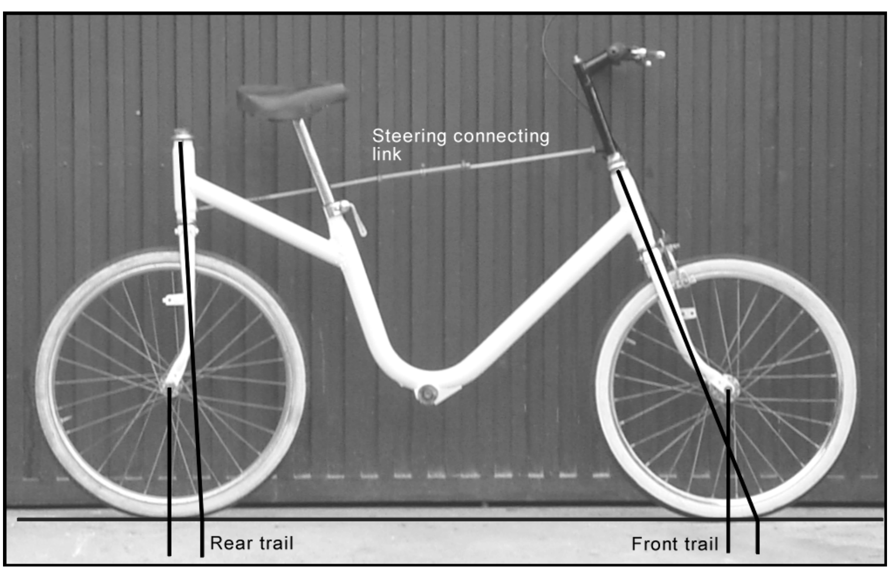
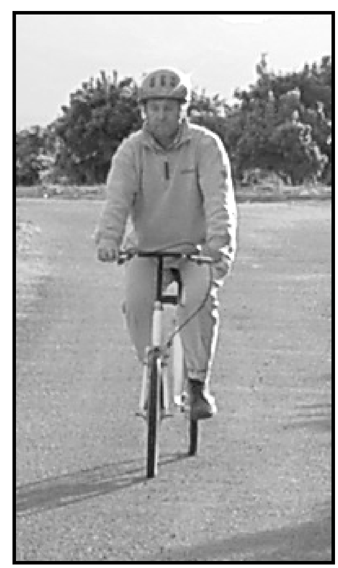


Fig. 18.5 **Opposite sense 2WS. At low speeds a balance correcting move would initially result in the rear moving outward by a similar amount to the inward moving front. The wheel-base mid-point would move very little under the CoG.**

As shown by the Drysdale example it doesn't have to be done that way. The Drysdale machine very cleverly circumvents these problems by allowing the front to have approximately 5° of steering angle before the rear begins to steer. Balance and counter-steering movements normally require less than 5° and so gentle riding is accomplished in a completely normal fashion. It is only during hard off-road riding that steering angles are large and that is when the Drysdale brings the rear steering into effect. It was reported that this machine was very agile and had a strong resistance against the rear sliding out, but it must be remembered that it was 2WD as well as 2WS.



**The author constructed this 2WS coasting bicycle to test the basic balance and rideability characteristics. The steering link could be mounted on either side at the rear so providing same sense and opposite sense steering as required. The steering ratio was also adjustable between 1:1 and 2:1 (rear wheel steered through half the angle of the front). The riding shot shows same sense steering at a 1:1 ratio. Note the crab like motion similar to that shown on the right in fig. 18.2.**

**The testing results were very much in accord with the expectations as outlined in the text.**

Low speed balance is controlled primarily by rider response, but at higher speeds the reactions of the rider are too slow, but fortunately, as we have seen, the effects of trail and gyroscopic reactions take over and give us an automatic mechanism of balance. At 220 km/h it only takes 25 msec. to travel the 1.5 m. wheelbase length and so in many cases the automatic responses will take place during the period when the opposite sense 2WS is more responsive. Whether this is beneficial or not is a very complex question and the answer depends on many factors; the nature of the balance disturbing force for example.

Whilst low speed balance is important most standard motorcycles perform adequately in that respect, but modern racing bikes do not always turn as quickly as needed. It was shown in chapter 4 that the ultimate limit on turn-in performance is front tyre grip during the transient manoeuvre. It would be interesting to investigate the possibilities offered by 2WS in this regard. It appears that neither same sense nor opposite sense steering offers the required response throughout the full duration of the lean-in phase. It may be that some form of steering control using fly-by-wire technology would be necessary to achieve the desired ends. Further research would be needed to investigate the behaviour in emergency situations like correcting a slide etc..

## Feet-Forward motorcycles. (FF)

Even though there had been many historic designs with a seating position more akin to that of a car than a conventional motorcycle, without doubt any resurgence of modern interest was given a boost by the Quasar design of the late Malcolm Newell.

**The Quasar in motion. Despite its weight and long wheel-base, 1800 mm., it had an agile cornering performance. Even with a relatively underpowered engine, the superior aerodynamics enabled a high top speed. Built to heavy construction standards, this machine has been shown to withstand frontal impacts better than conventional motorcycles.** (Clive Dixon)

As the name implies the key feature of this type of machine is that the rider’s feet are located substantially forward of the rest of his body. Many scooters and chopper styled machines fulfil this requirement but a substantial back-rest and low seating position are considered by FF aficionados as necessary to define the genre. The principal advantage of this class of machine is comfort, the reclining position and back-support relieving pressure from the wrists and arms. With a low seat height this layout can reduce frontal area and the more favourable shape can give a lower value for the Cd, even without any additional streamlining, thus resulting in faster and/or more fuel efficient vehicles. The principal down-side is that to achieve a low seating position it becomes necessary to increase length and hence weight of the machine, and this is even more likely when provision for a pillion passenger is included in the specification.

For many years this type of machine has been the preserve of a small band of enthusiasts but over the past decade there has been increasing interest from the major manufacturers, with several producing what are commonly referred to as “super scooters”, the Suzuki Burgman for example. These machines have quite low seating positions with a more reclining posture than traditional scooters, but are built considerably longer to achieve this. However, the lack of a full backrest and enclosing bodywork denies the rider the extra comfort otherwise possible from this seating layout.

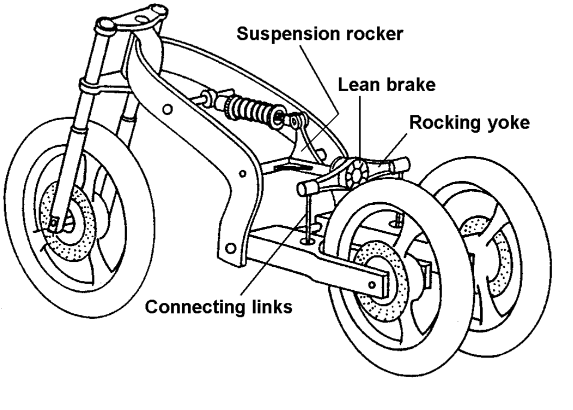
**Above, the Suzuki Burgman. The long wheelbase and small scooter wheels give a low reclining riding position, but the rider’s back rest is insufficient to give full back support. The pillion passenger is seated higher and benefits from a slightly bigger back rest.**

**The BMW C1 shown left. Fitted with full cross over seat belts this machine has been granted exemption from the helmet laws in some countries. Note the seat offering full back support and the side extensions to protect the rider’s shoulders.**

BMW have adopted a somewhat different approach, weather protection and safety being the obvious motives behind their C1, this has a fairly upright seating position similar to traditional scooters but the rider is provided with a fully supporting seat including head rest. The bodywork and windscreen add considerably more weather protection than normally available on a motorcycle or scooter. This machine is fitted with seat belts and has been granted exemption from the helmet laws in some countries.

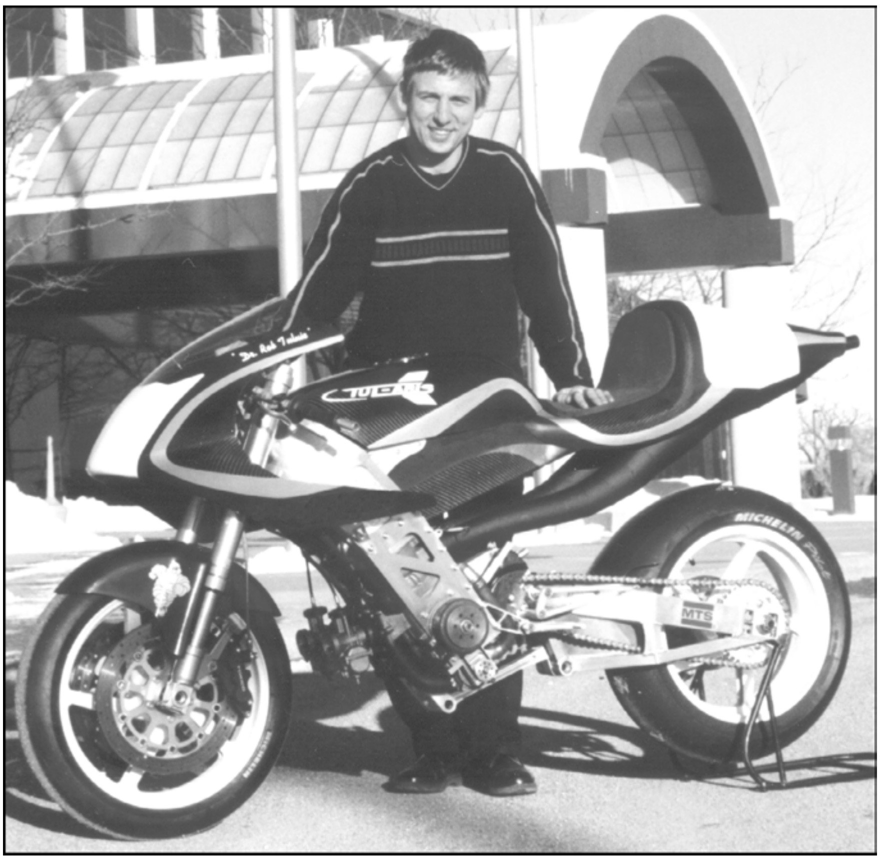
**Eco-mobile. The extremely long wheelbase can be clearly seen, as can the retractable out-rigger balance wheels. Powered by the flat 4 cylinder BMW engine, approximately 100 of these machines have been built in single *(left)* and dual *(right)* seat versions with fully supporting seats as in a car. On the right, designer Arnold Wagner demonstrates the capabilities of this interesting machine. Current models are capable of 300 km/h., largely due to the body shape.** (Peraves AG.)

The Swiss built Eco-mobile, in production since 1987, provides a well streamlined shape which gives total occupant enclosure like a car. It is of necessity long and heavy by motorcycle standards, 3.7 m. and 450 kg.. The full enclosure prevents the rider from extending his leg to maintain stationary balance and this machine is provided with retractable out-rigger wheels mounted amidships for this purpose. These detract somewhat from the neat appearance and streamlining of the body. Like the BMW C1, seat belts are fitted and helmet law exemption has been granted.




**The Calleja system with two rear wheels. Each wheel can freely move relative to the other to allow normal leaning in a corner, and a single suspension unit handles the bumps from either. This clever design does not increase overall width and would be ideal for enabling stationary balance for an enclosed machine.**

The Spanish designed Calleja has two independently sprung rear wheels, which allow full leaning for cornering. Unlikely to find much application on traditional motorcycles this design is an excellent solution to the problem of stationary balance for an enclosed motorcycle of either FF or traditional layout. The two rear swing-arms are connected by a pivoted balance bar to allow for the differential movement of the two arms, and this balance bar is fitted with a brake. The brake locks the two swing-arms together, thus holding the machine upright when stationary. Application of this brake can be rider controlled or under the command of a computerized system. Fitment of this design to a vehicle like the C1 for example, would allow a higher degree of side enclosure and weather protection, because riders would not need to put their feet down until it was time to dismount.




**Dr. Robin Tuluie shows off his TUL-ARIS racer. This machine gives a glimpse of the future of design and development methodology. It is not so prophetic as far as its physical layout is concerned which is fairly conventional, with the possible exceptions of the rear suspension linkage and the still secret handlebar vibration control. The futuristic aspect of this bike is the way that it was designed, analysed and tested.**

**The large manufacturers all now use various computer based techniques for design and testing but the TUL-ARIS is currently (2002) unique in the degree to which the most modern simulation and testing tools have been brought together and applied to one design, before and after construction. It is certain that such an approach will become more common with time. This will allow reduced testing costs and shorter development times.**

(Dr. Robin Tululie, MTS Systems Corp.)

# A1 Experiments with rake and trail

## Rake

The ideas suggested in chapter 3 in relation to the angle of the steering axis (rake) were subsequently put to the test by modifying a readily available standard production machine – a BMW R75/5. There were two advantages in the choice of this machine.

- The offset of the wheel spindle from the steering axis is divided almost equally between the offset in the yokes and that of the wheel spindle from the centre line of the fork sliders (figure A.1); the importance of this will become obvious later.
- The BMW was large and fast enough to make the results meaningful, which might have been less so with a slow, light machine such as a moped.

To keep other variables to a minimum, the original frame and suspension were retained and the wheelbase remained unaltered. Two non-standard rake angles were tried. In each case the ground trail was kept to approximately the same as the standard value (i.e. 89 mm.).

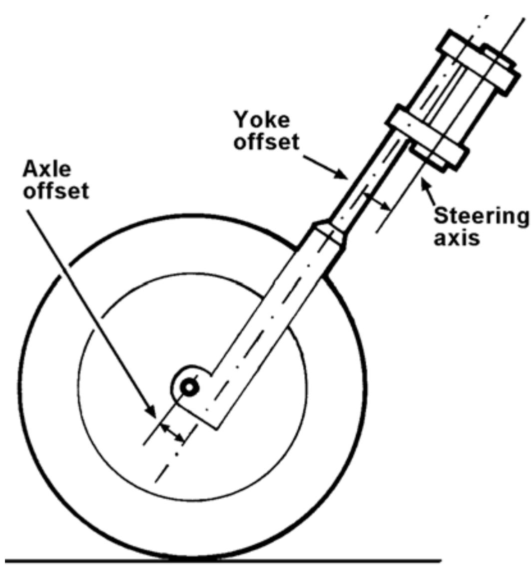



Fig. A1.1 **On the BMW R75/5, the total offset (wheel spindle from the steering axis) is divided approximately equally between that in the yokes (steering axis to fork legs) and that in the sliders (fork legs to spindle. This feature made these forks ideal for the tests as explained in the text.**

The first alternative set-up tried was with a rake of approximately 15 degrees and almost nil offset. This was achieved by bolting a superstructure to the frame to support the new headstock (see photo) and reversing the yokes, since their offset is very close to that of the wheel spindle, the overall offset was reduced virtually to zero. For the second setup, the rake was close to zero (i.e., vertical steering axis). This was achieved by reversing the complete front-fork assembly, thus giving the negative offset necessary to maintain the standard trail. The new headstock was supported by an extension of the original super-structure. In both cases the handlebar was pivoted in the usual place and connected to the fork by a ball-jointed link, a side effect of this was an adjustable steering ratio – i.e., for a given steering angle at the fork the angle needed at the handle-bar could be varied. With the 15-degree rake the bike had full road equipment, including lighting, so that it could be ridden under everyday conditions; indeed, five riders covered over 3000 kms. between them, including wet and dry going, bumpy country lanes, London traffic and motorway trips. Throughout this period, no steering damper was fitted.


**The standard BMW R75/5 used as a basis for the experiments in rake and trail, maker's figures were 27 degrees rake and 89 mm. ground trail.**

Although the results of these tests are essentially subjective and might be expected to depend on experience, personal preferences and preconceived ideas, there was in fact no divergence of opinion between the various riders. The initial testing was done on a bumpy, rutted country lane at speeds up to 80 km/h. Here the most noticeable effect was the insensitivity of the steering to ruts and bumps. Not only could the bike be ridden hands-off but at the same time it could be weaved from side to side across the ruts with little effort and with little detectable deflection of the steering. In corners, bumps had little effect, which was contrary to the behaviour of this particular machine before conversion, when it had a strong tendency, with no steering damper, to shake its head (sometimes violently) on bumpy corners. This lack of disturbance from longitudinal ruts was also confirmed on smoother roads at higher speeds, when the machine was ridden deliberately on the edge of painted white lines. Though unforeseen, this benefit is easily explained by reference to figure A.2. If we visualize a 90-degree rake (i.e. horizontal steering axis) we can see that the side of the rut gives rise to a moment about the steering axis that tends to steer the wheel back into the rut. With a vertical steering axis, however (zero rake), there is no effect on the steering, instead, the disturbance tends to cause the complete machine to lean into the rut. In this case, though, since the inertia of the whole bike is much higher than that of the front wheel alone, the effect on directional stability is considerably smaller and the rider is less aware of the rut. Thus the steeper the steering axis the smaller the disturbing effect.

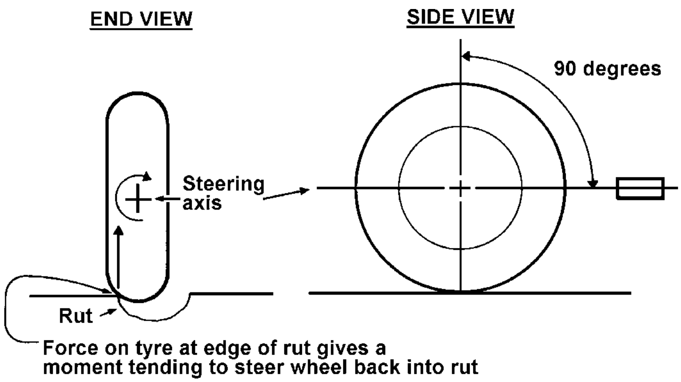


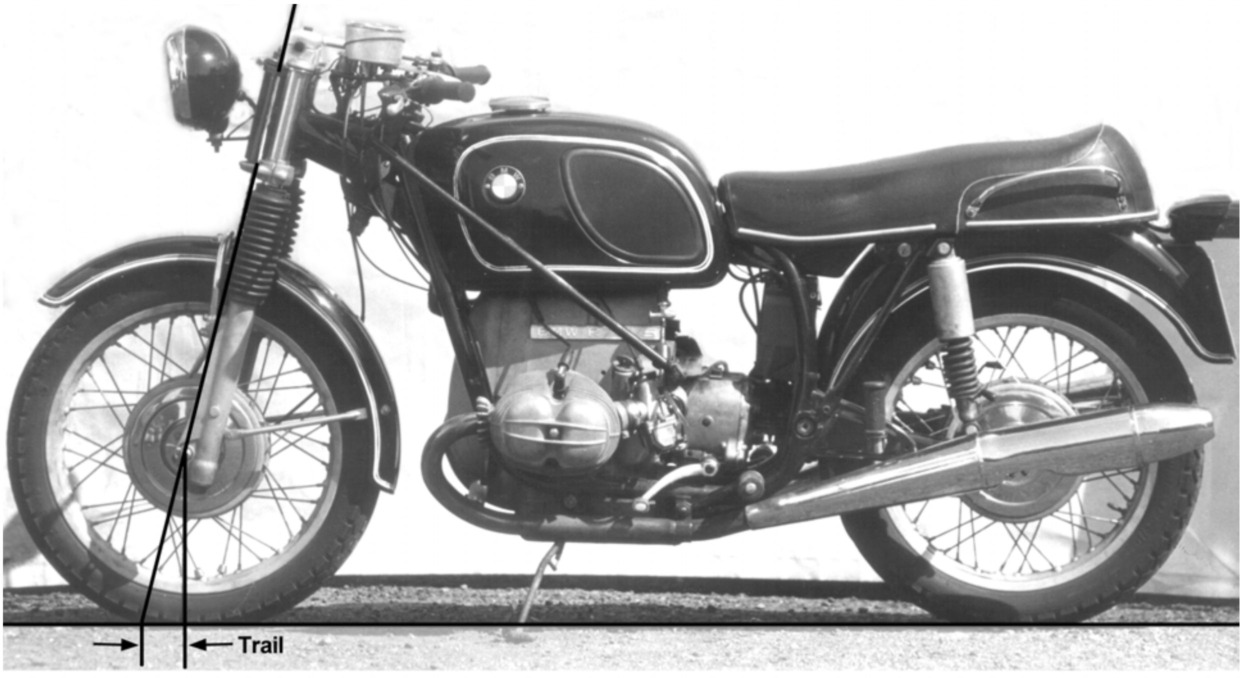

Fig. A1.2 **The effect of ruts on steering increases with rake angle, as shown in this exaggerated case. A vertical steering axis reduces the effect.**

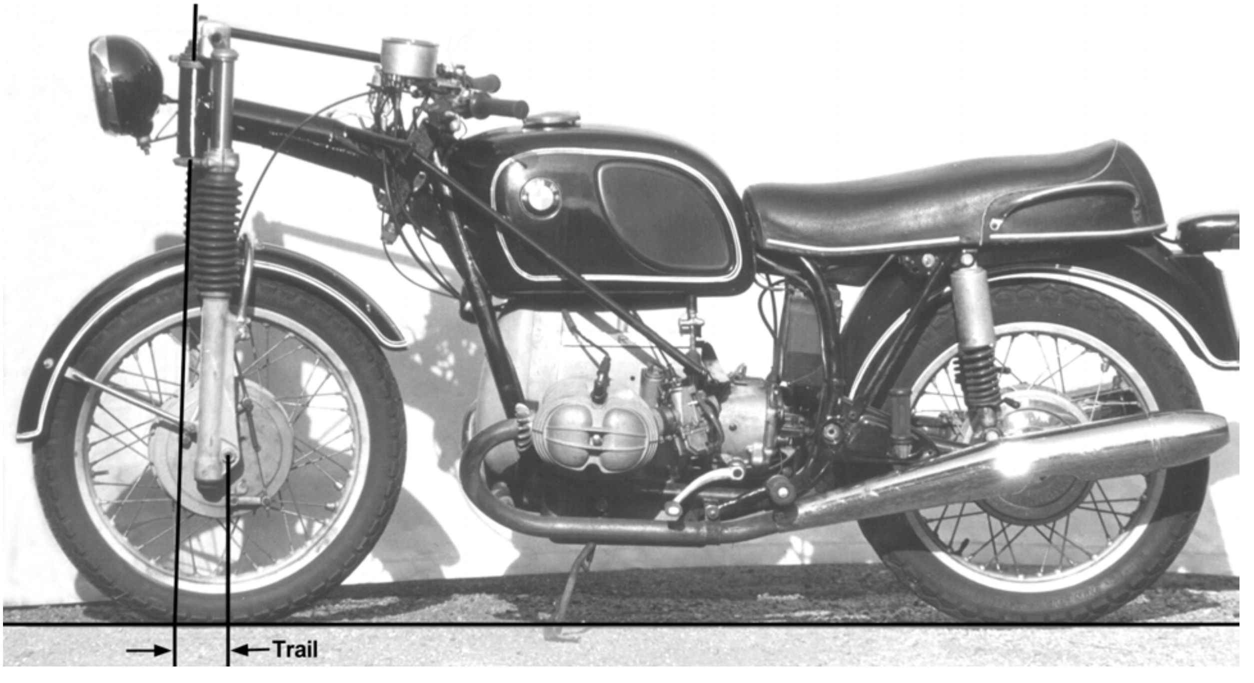
It was previously suggested that balance might be enhanced, particularly at low speeds, by steepening the head angle. To verify this, much riding was done at very low speeds and balance was indeed improved by the modifications. The machine could easily be ridden more slowly than when in standard trim before the rider had to put a foot down. (Of course, champion trials riders can balance a stationary machine indefinitely, but that is exceptional and most of us need to be moving slightly to maintain balance.) In heavy traffic, it was noticeably easier to trickle along slowly on the modified BMW, making it less tiring to ride from one side of London to the other.

When, without prior briefing, a novice was asked to try the machine, he commented on the surprising ease of moving off from rest, there was less wobbling than usually seen with a learner and his feet were quickly up on the foot-rests.

In some other sources it has been suggested that an unusually steep steering axis might induce wobbles at high speed. Nevertheless, with both the experimental rake angles on the BMW (15 degrees and zero) this was not noticed. With the handlebar released, the machine was ridden from approximately 160 km/h down to a walking pace and at no time was there any tendency to wobble or weave. With confidence built by several such runs, the handlebar was knocked to try to initiate a wobble. Whatever the speed, though, the disturbance was damped out very quickly. In standard trim (27-degree rake) this particular machine could develop a pronounced wobble when ridden no-hands at 50 to 65 km/h, though it was easily damped out by grasping the handlebar. Directional stability was always excellent and tremendous confidence was instilled in the rider at an early stage.

A further advantage of the steeper head angles was increased sensitivity of the front fork to small bumps. This results from reduced 'stiction' in the fork sliders as a consequence of the decrease in side loading. (The normal side-load component is approximately halved by reducing rake to 15 degrees and practically eliminated at zero rake.) In addition, this reduction in the side-load component is accompanied by an increase in the spring-load component as the fork is steepened – which gives the same effect as a lower spring rate. The effective rate varies little between zero and 15 degrees rake but is approximately ten per cent higher at 27 degrees. Similarly, the spring-load component of the braking force is reduced as the fork angle is steepened.




**The top photo shows the superstructure which gave an unaltered trail with zero offset and rake angle of 15 deg. The second shows the extended structure, with the reversed forks, giving equal trail with near zero rake angle.**

Moreover, since this spring force acts in concert with weight transfer to compress the fork, the reduction means less nose-diving. For this reason, the effective drop in spring rate was not detrimental and ride comfort was appreciably improved.

It was under braking, however, that a disadvantage was noticed, in the form of severe shuddering in the fork as the braking force tried to bend it backward. Naturally, this was more severe with the steering axis upright. Such juddering was entirely absent with the standard rake, though quite bad at 15 degrees.

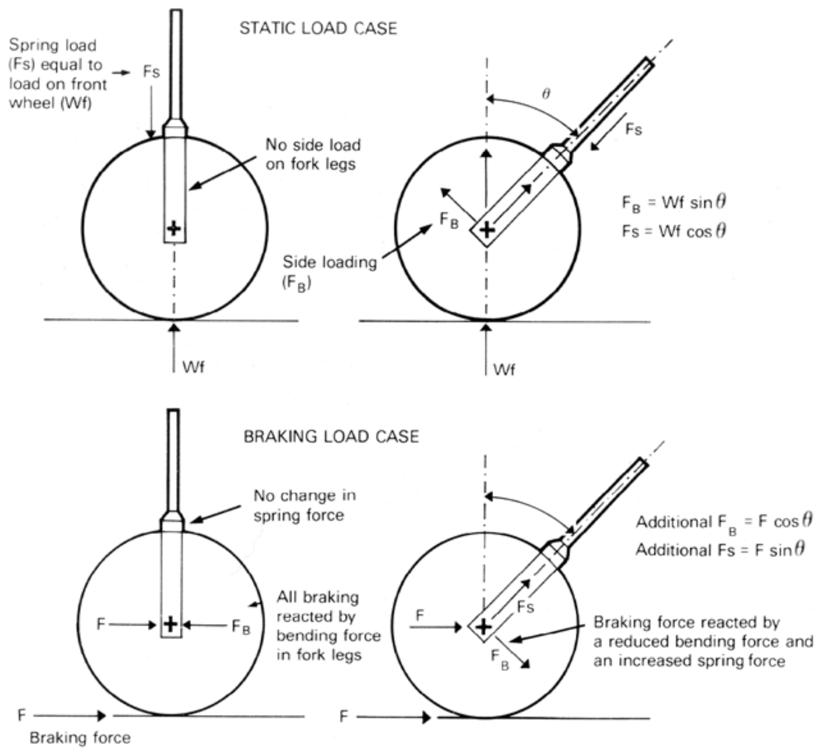
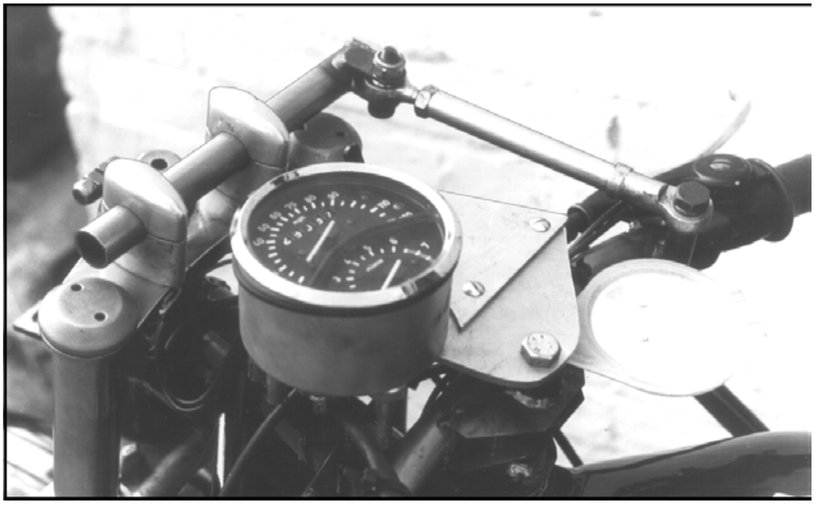



Fig. A1.3 **Steepening the steering head reduces the stiction in telescopic fork sliders, so improving sensitivity to small bumps. Nose diving under braking was also reduced.**

This effect apart, one of the most interesting results (mentioned by all the riders) was the surprisingly normal feel of the modified machine, with the steering pleasantly light at low speeds but always totally stable. No special riding technique was required and cornering was accomplished normally. The variable steering ratio mentioned earlier was tried from 1:1 (equivalent to conventional direct steering) to 1:2 (steering angle doubled from handlebar to fork). In normal riding (dry roads) it was difficult to detect the difference, only when manoeuvring at a standstill, using large steering angles, was the heavier feel of the 1:2 ratio noticeable.

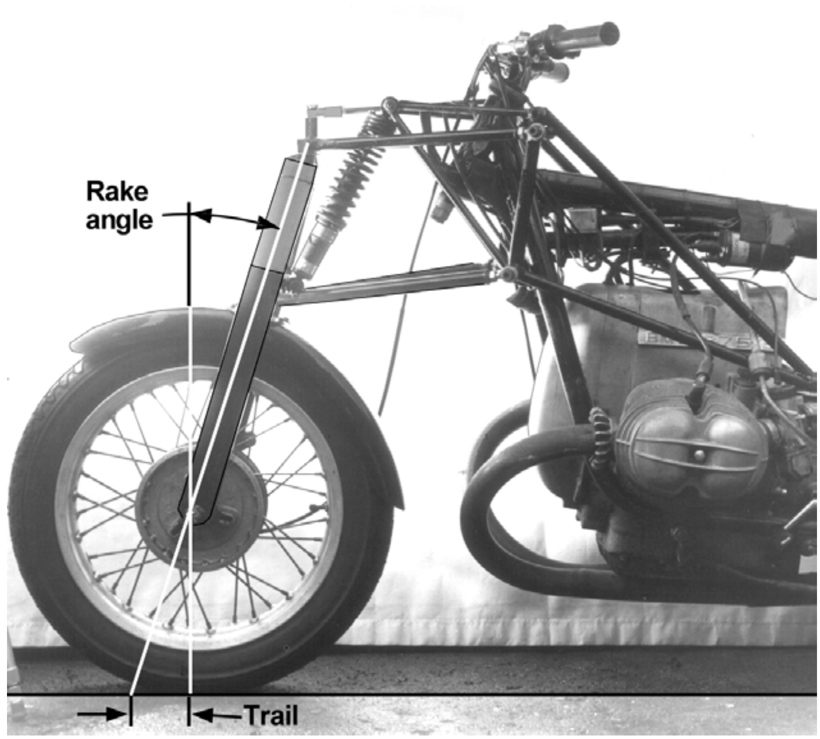
However, since the steeper rakes made the steering lighter anyway, even the effort required with the 1:2 ratio felt similar to that with the standard machine. Indeed, the reduced handlebar swing could be a bonus when designing a non-steerable fairing, as handlebar clearance usually results in a bulky shape if steering lock is not to be restricted.

**In both experiments the handlebar was connected to the fork by a rose-jointed link providing for steering ratios anywhere between 1:1 (direct) and 2:1 (geared up). This was accomplished by sliding the bar atop the forks toward greater or lesser radii.**



**With the modified BMW, the author demonstrates the stability achieved with the 15 degree rake angle.**

**To try trail values between 50 and 100 mm. with zero offset this adjustable double wishbone suspension system was made.**




## Trail

Throughout the rake experiments the standard ground trail of approximately 89 mm. was retained. Yet, it seemed reasonable to assume that the optimum trail (if there is such a thing) might vary with rake angle.

To test this, a double-wishbone front-suspension system was fitted to the BMW (see photo), making it possible to alter the trail by adjusting the wishbone lengths to give variations in rake. There was no offset, and trail values from 50 to 100 mm. were tried varying the rake angle either side of 15 degrees. Although the machine was perfectly ridable over the full range of settings, the front end proved livelier and the steering more sensitive to bumps as the trail was shortened (steeper rake), albeit never so much as with the standard set-up. In the upper range of trail settings the bike was very steady but could still be manoeuvred quickly. At about 76 mm. trail there was a tendency for the machine to lift itself out of a bend when cornered at moderate banking angles (say, 15 to 20 degrees), though no such effect was detectable at higher cornering speeds requiring, say, 35 to 40 degrees of lean. When a machine is banked into a bend, trail gives rise to two opposing effects: (1) directional stability, tending to make the machine run straight and (2) the self-steering effect (also dependent on rake and wheel diameter) mentioned in Chapter 3, tending to steer the machine into the bend. To achieve neutral handling, these effects have to be properly balanced (in concert with several other parameters), and the problem just mentioned, whilst not properly understood, is thought to have been caused by an unsuitable combination of these effects for that particular machine at a critical rake or trail. It is thought that at the greater bank angles, the self-steering effect would have outweighed the straight-ahead tendency.

# Conclusions

The scope of these experiments was limited by time and money. Nevertheless, the results indicate a need for more exhaustive and quantitative testing. These tests indicate that currently favoured geometry may be far from "optimum".

## Rake

From our experiments it seems there is nothing magical in the conventional rake angle of 26 to 28 degrees. Indeed, balance, stability and lightness of steering were all enhanced by steepening the angle. The greater improvement came from the first change (from standard to 15 degrees), the subsequent move to near zero rake producing only minor differences. Many effects of castor angle are approximately dependent on the cosine of that angle, the cosine of 15 degrees is 0.97 which is little different from the value of 1.00 for a vertical steering head. At 27 degrees the cosine reduces to 0.89, a more significant difference.

The only drawback noticed – juddering under braking – is a consequence of the poor structural integrity of headstock mounted forks as a type. It is not suggested that machines should be built with very steep steering axis, using a headstock mounted fork, because the consequent high, forward location of the headstock causes structural and styling problems. Much better to consider some form of hub-centre steering or other wishbone layout, such as that used on the Yamaha GTS, or other longitudinally stiff suspension system such as the BMW telelever.

## Trail

Apart from the need to avoid the critical situation mentioned, there seemed no obvious optimum value. Results were satisfactory throughout the full test range, so making personal preference the decisive factor.

# Post script

Since these tests were done (around 1984) there has been a gradual but continuous trend toward steeper rake angles in mainstream machines, particularly racing and sports bikes. This has been largely driven by the need for rapid turn-in performance. Weight distributions have moved forward partly by moving the riding position, this helps directly with turn-in but also makes it easier to use steeper rake angles and keep the handlebars in a similar relationship to the rider. General demands for better handling have led to telescopic forks much more rigid than their predecessors, and so the brake juddering problems experienced in the above tests are not a real problem. The steepest rake angles used on production machines are currently still over 20 degrees and so in the author's opinion still have a long way to go.

The results from the above trail experiments might seem to run counter to experience on the race track where trail is adjusted in increments of sometimes less than one mm. However, the situations are very different. These tests were principally concerned with general rideability, whereas on the race track the need is to provide the ultimate in performance and feel to match the requirements of riders very sensitive to minute changes.

# A2 Glossary of terms

| | |
|---|---|
| **ABS** | An anagram for the German words meaning Anti-locking brakes. Chapter 12. |
| **Acceleration** | Rate of change of velocity with time. Appendix A5. |
| **Active suspension** | A form of suspension without the usual springs and dampers. Computer controlled hydraulic rams support the vehicle with the minimum response to road shocks. |
| **Anti-dive** | A geometric suspension layout or mechanical arrangement that balances some or all tendency to dive under braking. Chapter 9. |
| **Anti-squat** | A geometric rear suspension layout that reduces the tendency for the rear to squat under acceleration. Chapter 9. |
| **Axis** | Normally a defined line about which some rotary motion takes place, a wheel spindle for example. |
| **Backbone** | A type of frame construction based on a large section member, circular or otherwise which forms the main structure of the chassis. Which generally passes over the engine. Chapter10. |
| **Bump** | The upward motion of the suspension due to hitting a raised road disturbance. |
| **C of G, CoG** | See Centre of Gravity. |
| **Camber angle** | The sideways inclination of a wheel, this is equal to the lean angle of the bike for the rear wheel. This can be modified to a small extent at the front wheel depending on rake and steering angles. Chapter 2. |
| **Castor angle** | See rake angle. |
| **Cd** | Aerodynamic drag coefficient. A normalized indicator of the aerodynamic drag that allows comparisons of drag between different shapes, regardless of size. Chapter 5. |
| **CdA** | The Cd multiplied by the frontal area, allows comparison of the actual drag between different machines, allowing for size, but not accounting for air density. Chapter 5. |
| **Centre of gravity** | A point on a body at which the whole weight can be considered to act, if supported at the CoG, gravity will produce no turning moments. Appendix. A7. |
| **Centre of pressure** | A point on an aerodynamic surface, at which the total aerodynamic force can be considered to act, producing the same forces and moments. Chapter 5. |
| **Centrifugal force** | The conceptual outwardly acting reaction force against centripetal force. Appendix. A5. |
| **Centripetal force** | A force that causes a moving object to move in a curved path. This force acts inward toward a turn centre and is at 90° to the instantaneous velocity of the object. Tyres provide this force with road vehicles. Appendix. A5., Chapter 2. |
| **Coefficient of drag** | See Cd. |

| | |
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| **Coefficient of friction** | The ratio between a friction force and the normal force. On a horizontal surface the normal force is the weight and the horizontal force required to move the object is the friction force. For most surfaces this ratio rarely exceeds 1.0 but modern tyres are sometimes capable of up to 1.3. Appendix. A5., Chapter 2. |
| **Compliance** | The opposite of stiffness, an indicator of the flexibility of some structural component. |
| **Coordinate systems** | A set of 2 or 3 mutually perpendicular axis which define a spatial reference system. In a 2 dimension system we usually use a horizontal X-axis and a vertical Y-axis. For 3 dimensional work with motorcycles the X-Y plane is horizontal. The X-axis is usually aligned fore-and-aft with positive towards the front, the Y-axis is across the bike with positive to the left, and the Z-axis is vertical with positive upwards. |
| **Couple** | See torque. |
| **Critical damping** | An amount of damping which absorbs the motion of an oscillating system just enough to prevent any oscillation. Chapter 6. |
| **Damping** | The dissipation of energy from a dynamic system. In a suspension system this is to prevent uncontrolled oscillations after a road disturbance. Intentional damping is usually by means of hydraulic dampers but friction and tyre hysteresis add to such damping. Chapter 6. |
| **Density** | The mass of some material per unit of volume. See also – Specific gravity. |
| **Dive** | The compression of the front suspension under braking. Chapter 9. |
| **Drag** | The retarding force on a vehicle. At high speed the aerodynamic force is the highest component of the total drag force. Rolling resistance and brake resistance also adds to the drag. Chapter 5. |
| **FF** | Anagram for a “Feet Forward” motorcycle. That is one in which the rider sits in a fashion similar to a car driver. This vehicle is usually lower than a conventional motorcycle. |
| **Force** | Force is defined in terms of its ability to change momentum. Appendix. A5. |
| **Heave** | The vertical component of suspension movement, pitch is the angular movement. Chapter 6. |
| **Hysterisis** | This is like a form of backlash in mechanism or material. For example, as you load a tyre it deflects but as you release the load it will not exactly follow the force-displacement pattern of the loading. This represents a loss of energy. |
| **Instantaneous force centre** | A point or axis that may be real or virtual. It is a point through which all the forces and moments in a mechanism can be assumed to act. It usually changes position as the mechanism is operated. Use in some methods for evaluating anti-dive and anti-squat characteristics. Chapter 9. |
| **Km/h, KPH** | Kilometres per hour. See also MPH. |
| **Laminar flow** | A fluid flow regime in which the fluid particles follow a smooth path rather than a turbulent one. See turbulent flow. Chapter 5. |

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| **Lbf.** | A unit of force in the imperial system. The weight of a lb. mass. Appendix. A5. |
| **Mass** | A measure of the amount of matter in an object. Appendix. A5. |
| **Mechanism** | Normally some form of mechanical structure that can pivot at some of the joints, so allowing a defined movement. The swing-arm for example. |
| **Moment** | See torque. |
| **Moment of inertia** | The angular equivalent of mass. Depends on the distribution of the mass. |
| **Monocoque** | A form of construction, commonly used on aircraft and racing cars. Less suitable for motorcycles because of the shape. Chapter 10. |
| **MPG** | Miles Per gallon, measure of fuel consumption. Litres/100 Km = 282.5/mpg |
| **MPH** | Miles Per Hour, measure of road speed. 1 mph = 1.61 Km/hour. |
| **Normal to** | At 90° to something. Weight acts “normal” to a horizontal surface. |
| **Pitch** | The angular orientation of a vehicle about a cross wise axis. Chapter 3. |
| **Pressure** | The force per unit of area. In a fluid the pressure acts equally in all directions. |
| **Radians** | A measurement of angle. Approximately equal to 57.3°. Appendix. 3. |
| **Rake angle** | The rear inclination of the steering axis from the vertical. Chapter 3. |
| **Rebound** | The suspension movement in extension. Opposite to bump. Chapter 6. |
| **Reynold’s Number** | A parameter which combines the effects of size and velocity to give an idea of the likely flow characteristics of fluids. Chapter 5. |
| **Roll** | The angular motion of a motorcycle about a longitudinal axis. Chapter 3 & 4. |
| **Scalar** | A parameter that only has a magnitude and no direction. See also vector. Appendix. 3. |
| **Single track vehicle** | A vehicle with all wheels in a single line or track. Normally two wheels as per motorcycles and bicycles. |
| **Slip angle** | The SAE definition is: “The angle between the longitudinal axis of the vehicle and the direction of travel of the centre of the tyre contact patch.” Chapter 2. |
| **Specific gravity** | A measure of the mass of a substance compared to the mass of the same volume of water under standard conditions. See also density. |
| **Spine frame** | See backbone. |
| **Spring rate** | The ratio of a spring compression or extension force to the displacement of the spring. This may not be constant throughout the working range of the spring, in which case the spring is called progressive or regressive. Chapter 6. |
| **Squat** | The rear suspension compression under the effects of acceleration. Chapter 9. |
| **Stability** | A measure of the ability of a system to return to a stable trim state, after some disturbance. Chapter 14. |
| **Steering head** | Also known as “head stock”. The part of a motorcycle designed to carry the steering axis bearings. |

| | |
|---|---|
| **Stiffness** | The ability of a structure or member to resist elastic deformation. Conceptually the same as spring rate, but the term is often used for constructions generally much stiffer than the springs. Chapter 10 & 13. |
| **Strain** | The proportional extension of material when subject to some force. Chapter 13. |
| **Streamline** | The path followed by a fluid particle. When the "streamline" follows an object placed in the flow we call it laminar flow, but when the motion is erratic the flow is turbulent. Chapter 5. |
| **Strength** | The level at which a structural member fails. Failure may be a total rupture, this is called Ultimate Tensile Strength (Stress) or UTS, or it may be when the material yields permanently – Yield Strength (Stress). Chapter 13. |
| **Stress** | The force applied to a structural member divided by the material cross sectional area. A measure of the intensity of the force. Chapter 13. |
| **Structure** | A combination of members (tubes, sheet metal, nuts and bolts etc.) that form a rigid assembly usually designed to support other objects. On a motorcycle the main structure are the main frame and swing-arm, etc. Chapter 10. |
| **Tangent to** | A line at 90° to the instantaneous radius of a curved path or velocity. |
| **Torque** | The angular equivalent of force. A twisting effort that is proportional to the length of the offset of the force from the point about which the torque acts. |
| **Track** | The lateral width between the wheels of a multi-track vehicle, not applicable to motorcycles except when fitted with a sidecar. |
| **Triangulation** | A technique of making efficient structures. Chapter 10. |
| **Trigonometric ratios** | Ratios between the lengths of the sides of right angled triangles, allowing for the calculation of angles. Appendix. 3. |
| **Turbulent flow** | A fluid flow regime in which the streamlines are erratic or turbulent. Chapter 5. |
| **Vector** | A quantity that has magnitude and a direction. See also scalar. Appendix. 3. |
| **Virtual arm** | An imaginary structural link pivoted at the virtual pivot or axis. Chapter 9. |
| **Virtual pivot** | See instantaneous force centre. Chapter 9. |
| **Viscosity** | A property of fluids that describes the ease with which adjacent layers can slide over one another. Chapter 6. |
| **Weight** | The gravitational force acting on a given mass. Appendix. 3. |
| **Wheelbase** | The linear horizontal distance between the front and rear axles. Chapter 3. |
| **Yaw** | An angular displacement of a vehicle about a vertical axis. Chapter 3. |

# A3 Units conversion

The original version of this book used the so-called "Imperial system" which had been traditionally used in English speaking countries. However, many of these countries have now completely changed over to the metric system (SI) and the others have been teaching it in schools for a long enough period for most readers to have some familiarity with it. As a result and also to try and ensure the usefulness of this book into the future it has been decided to feature the SI system in this edition. There are a few exceptions where, like with tyre dimensions, inch units are the natural ones to use because most tyres are still specified in those units. In a few other cases inch dimensions are shown along side the metric ones in the text, but this is not general. The following table is by no means exhaustive (it only includes units used in this book) and is intended only as a quick guide for those that need to view dimensions in the system most familiar to them.

| Type of unit | Imperial → Metric | Metric → Imperial |
|---|---|---|
| Length. | 1 inch → 25.4 mm.<br>1 foot → 0.3048 m.<br>1 mile → 1.609 km. | 1 mm. → 0.03937 inches<br>1 m. → 3.281 feet<br>1 km. → 0.6214 miles |
| Area | 1 in.$^2$ → 645.2 mm.$^2$<br>1 ft.$^2$ → 0.0929 m.$^2$ | 1 mm.$^2$ → 0.00155 in.$^2$<br>1 m.$^2$ → 10.76 ft.$^2$ |
| Velocity. | 1 ft/sec. → 0.3048 m/sec.<br>1 mph. → 1.609 km/h. | 1 m/sec. → 3.281 ft/sec.<br>1 km/h. → 0.6214 mph. |
| Acceleration. | 1 G → 9.807 m/sec.$^2$ | 1 G → 32.17 ft/sec.$^2$ |
| Mass. | 1 lb. → 0.4536 kg. | 1 kg. → 2.205 lbs. |
| Force. | 1 lbf. → 4.448 N.<br>1 lbf. → 0.4536 kgf. | 1 N → 0.2248 lbf.<br>1 kgf. → 2.205 lbf. |
| Moment of inertia. | 1 lb.ft$^2$ → 0.04214 kg.m$^2$ | 1 kg.m$^2$ → 23.73 lb.ft |
| Torque, moment. | 1 lbf.ft. → 1.356 Nm. | 1 Nm. → 0.7376 lbf.ft. |
| Spring rate. | 1 lbf./in. → 0.1751 N/mm. (kN/m)<br>1 lbf./in. → 0.01785 kgf./mm. | 1 N/mm. → 5.710 lbf./in.<br>1 kgf./mm. → 56.01 lbf./in. |
| 2$^{nd}$ moment of area | 1 in.$^4$ → 416200 mm.$^4$ | 1 mm.$^4$ → 2.402 x 10$^{-6}$ in.$^4$ |
| Stress | 1 psi → 0.0007031 kgf./mm.$^2$<br>1 ton/in.$^2$ → 1.575 kgf./mm.$^2$ | 1 kgf./mm.$^2$ → 1422 psi (lbf./in.$^2$)<br>1 kgf./mm$^2$ → 0.6349 tons/in.$^2$ |
| Young's modulus | 10x10$^6$ psi. → 0.7x10$^4$ kgf./mm.$^2$ | 1x10$^4$ kgf./mm.$^2$ → 14.22x10$^6$ psi. |
| Energy | 1 BTU → 1.055 kJoule | 1 kJoule → 0.9478 BTU |
| Power | 1 BHP → 0.7457 kW | 1 kW → 1.341 BHP |
| Pressure | 1 psi. → 0.06896 bar | 1 bar → 14.50 psi. |

# A4 Gyroscopic effects

Without doubt the most misunderstood aspect of basic dynamics is that of gyroscopic precession. To those that have witnessed demonstrations of this phenomenon the reactions may appear to be somewhat magical, often seeming to defy gravity itself. Those that have used a hand-held disc grinder will be familiar with the rather counter-intuitive feel that occurs when attempting to twist the machine. In reality gyroscopic effects are no more than an example of one of the basic laws of physics, known as *"Conservation of Momentum".*

Rotating objects such as motorcycle wheels and crankshafts are subject to the demands of the *conservation of momentum* but exactly how is somewhat less obvious than in the cases of linear or curvilinear motion as explained more fully in the appendix on basic mechanics. Each particle of the object has its own instantaneous linear momentum and could be considered individually and the overall effect determined by summing the separate changes in momentum. In this case the momentum of each particle is in a different changing direction and so it is rather difficult to visualize and discuss the situation on that basis. In order to provide an easy way of representing the momentum of rotating objects (called angular momentum) we adopt the convention that angular momentum acts along a straight line coaxial with the axis of rotation, as in fig. A4.1. This is a universal way of representing angular momentum but we must always remember that it is only a man-made convenience, the "real" momentum of individual particles is really still in the plane of the disc's rotation. As with the linear case above we have to apply some outside *force* to change the momentum as shown, in the case of angular momentum this *force* is in the form of torque, a *twisting force*.

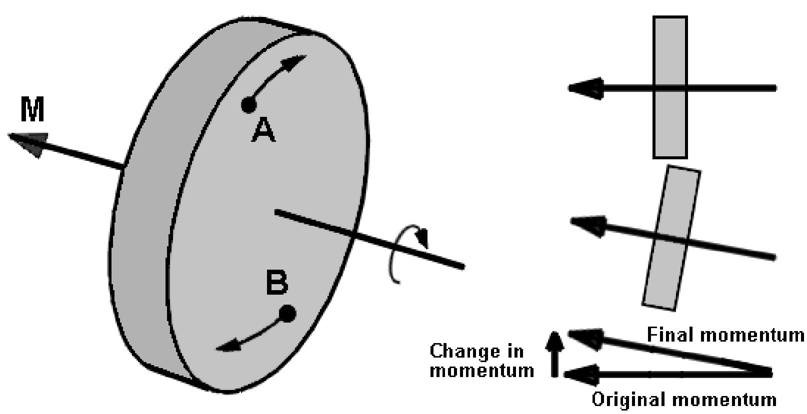



Fig. A4.1 **The momentum of particles in a rotating disc such as A & B is as shown, but it is much more useful and convenient to regard the overall angular momentum of the disc as M, coaxial with the axis of rotation. Consider the top view of the disc, as shown top right, we can see that if the axis of rotation is moved, as shown centre right, then the momentum will be changed also, lower right, although as with the linear case of fig. A5.3 the magnitude of the momentum has not changed, only the direction.**

Normally the equations describing gyroscopic motion are derived in physics books using the idea of momentum acting along the axis of rotation, but the mathematics of this are beyond this book and the explanations don't provide an intuitive feel for why precession works as it does. Presented here is an explanation, based on the instantaneous linear momentum of two particles contained within the rotating wheel, which clearly demonstrates the basic mechanism and direction of gyroscopic reactions.

Fig A4.2 shows a disc rotating, in a vertical plane, about its horizontal central axis. A torque **T** is applied as shown about the fore and aft horizontal axis. Now consider two opposite points, denoted by **A** & **B**, respectively at the top and bottom of the disc. The instantaneous velocity of **A** is directly forward and that of **B** is directly rearward, so these two particles have linear momentum in those directions. The rear view shows how the horizontal torque can be considered as two opposite forces, **F**, acting on these particles. Therefore, these forces will change the momentum of the particles as shown in the top views. **A** will be turned toward the left and **B** will turn to the right, or in other words the disc has been given an angular velocity, anti-clockwise about the central vertical axis (when viewed from above).

The torque about a horizontal axis has produced a velocity about a vertical axis, no wonder the reactions sometimes seem magical and counter-intuitive.

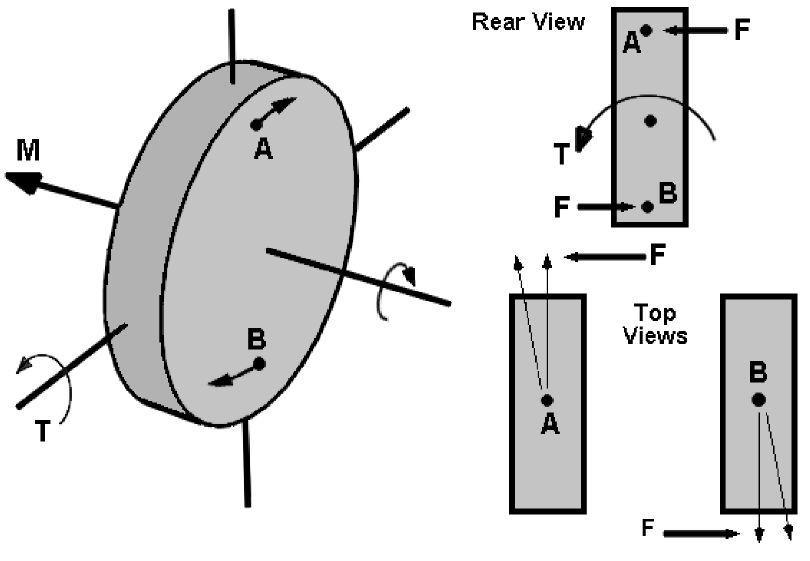



Fig. A4.2 **A torque** T **is applied as shown about the fore and aft horizontal axis, to a rotating disc. The instantaneous velocity of** A **is directly forward and that of** B **is directly rearward, so these two particles have linear momentum in those directions. The rear view shows how the horizontal torque** T **can be considered as two opposite forces,** F**, acting on these particles. Therefore, these forces will change the momentum of the particles as shown in the top views.** A **will be turned toward the left and** B **will turn to the right, or in other words the disc has been given an angular velocity, anti-clockwise about the central vertical axis.**

Gyroscopic reaction and precession are all about this relationship between a torque and an angular velocity each acting on a pair of mutually perpendicular axis, which are themselves both perpendicular to the axis of rotation of the object. If a torque is applied to one of this pair of axis then the object will generate an angular velocity about the other, this is known as precession. Conversely, if a disc is precessing about one of these axis then we know that a torque is being applied about the other.

This phenomenon is described by the simple formula $\mathbf{T} = I\omega_1\omega_2$

Where:

$\mathbf{T}$ = applied torque

$I$ = Polar Moment of Inertia of the disc about its axis of rotation

$\omega_1$ = Angular velocity about its axis of rotation

$\omega_2$ = Angular velocity about the precessional axis

So the gyroscopic effects are directly proportional to the angular momentum, $I\omega_1$ , of the rotating disc.

Note that this formula has been derived assuming that the precessional velocity $\omega_2$ is much less than the main rotational velocity $\omega_1$ .

This relationship has great influence over several aspect of motorcycle steering and behaviour in general. The directions of the torques and corresponding precession are very important for steering action and the maintenance of auto-stability. A steering torque toward the left hand side (anti-clockwise from the top) is balanced by a roll velocity to the right (clockwise from behind). A roll torque to the left (anti-clockwise from behind) will mean that there is a steering velocity to the left also (anti-clockwise from the top).

We know that we only turn the steering by a very small amount at normal riding speeds and so it follows that the steering angular velocity must be very low, and so the gyroscopic coupling between roll torque and steering velocity is relatively unimportant. However, during the lean-in process the roll velocity is significant and so the coupling between roll velocity and steering torque is of prime importance. This is further elaborated upon in chapter 4.

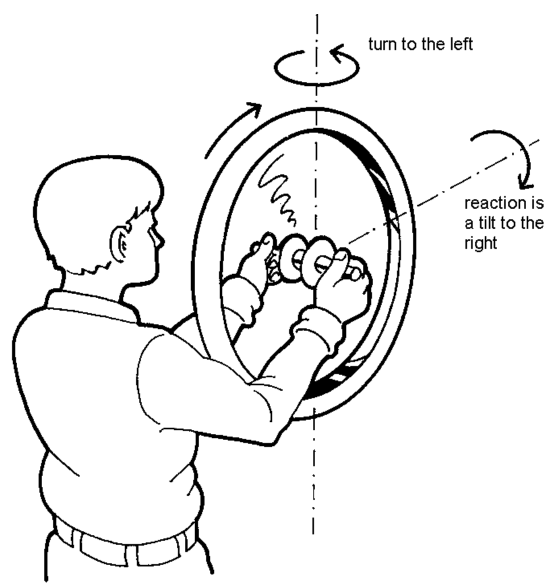



Fig. A4.3. **Gyroscopic precession. When a bicycle wheel, spinning as shown, is steered to the left (anti-clockwise from the top) it tilts strongly to the right (clockwise from behind). However, when it is tilted to the left (anti-clockwise from behind) it steers to the left (anti-clockwise from the top).**

There is another little talked about aspect of gyroscopic reaction that affects the final balanced lean angle of a bike. As we negotiate a turn the whole bike is subject to a yaw velocity equal to the angular velocity of the bike about the turn centre. Depending on the lean angle of the bike there will be a component of this yaw velocity about the local “vertical” or z axis of the bike, this must be supported by a roll torque acting away from the turn, which in turn must be balanced by an increased lean angle of the bike. In practice this only amounts to about 1 degree at maximum cornering speed.

As well as the angular momentum of the wheels the crankshaft and other rotating components of the bike make a contribution to the total. An across-the-frame engine with a forward running crankshaft will simply add to the effect from the wheels, a backwards running engine will obviously detract from those effects. In-line engines like BMWs and Guzzis react differently, in place of a roll torque their engines produce a pitch torque when cornering, the direction of which differs depending on the rotation direction of the engine and the direction of the turn, left or right.

# A5 Basic physics of motorcycles.

Whilst this book has been written to try and appeal to the technically interested rider as well as students and engineers, there is of necessity a certain level of mathematics and physics knowledge required to make the most of the text. This has been minimised as far as possible but there are a few basic concepts that will enhance understanding of the main chapters, and it is in the reader's own interest to try and get a good appreciation of some fundamental principles. This appendix is just to review a few basics and to refresh some long forgotten school work. If the thought of a trigonometry lesson fills you with horror then don't worry, there's no exam at the end. If the maths in the book is still not to your liking, then simply skip those parts, you'll still get a good idea of the subject from the descriptive parts of the text.

This appendix is also used to describe the nomenclature and other conventions as used in this particular book.

## Basic Trigonometry

A subject guaranteed to strike fear into the hearts of most people, but in reality this is simply a method by which we can easily refer to some properties of angles in general and right-angled triangles in particular, as far as this text is concerned. On a motorcycle, angles have relevance in various areas; we have the lean angle needed to balance, steering angle, rake or castor angle, swing-arm angle, slip-angle of the tyres and trigonometry is useful to help us with the resolution of forces (see below).

Fig A5.1 shows a simple right-angled triangle, the sides are labelled as a, b and c, the angle between sides a and b is the right-angle, and the angle between sides b and c is labelled as "θ" the Greek letter "theta". Don't be confused by Greek letters, they are traditionally used for angles and some other parameters and are no different from any other letter nor name used to refer to some quantity.

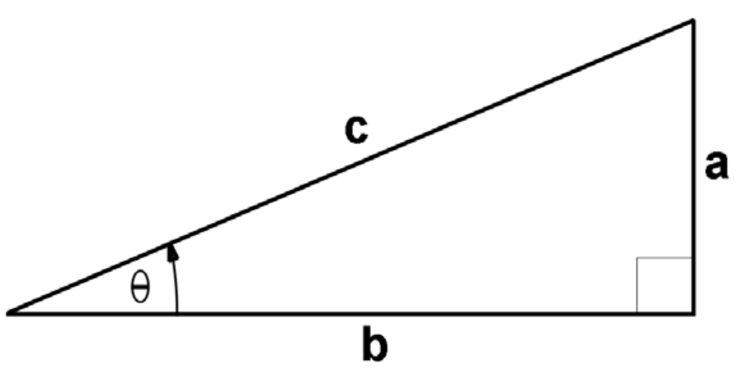



Fig. A5.1 **The basics of trigonometry are all contained in this simple triangle. The square in the angle between sides a & b indicates that the angle is a right-angle or 90 degrees. The three main trig. ratios can be expressed as:**

**sin $\theta$ = a/c**

**cos $\theta$ = b/c**

**tan $\theta$ = a/b**

There are three principal trigonometric ratios and three matching inverse ones, which just relate the ratios of the various sides to the angle “θ" in a way that is more convenient to handle.

- The sine of theta, sin θ, is just the ratio of the length of side “a” to the length of side “c” .
- The cosine of theta, cos θ, is just the ratio of the length of side “b” to the length of side “c”.
- The tangent of theta, tan θ, is just the ratio of the length of side “a” to the length of side “b”.

This is useful because the values of the sines, cosines and tangents of all angles have already been worked out for us, and so if we know the angle between two lines in such a triangle we can get the appropriate ratios of the lengths of all the sides. This is of particular value when we need to calculate the equilibrium of forces necessary to balance a bike when leant over in a turn, for example. The traditional way to get these trigonometric values was to look them up in a special book of tables or by using a slide rule. It is much easier now because every scientific calculator has the functions built in.

Both the sine and cosine can have any value in the range of –1.0 to +1.0. The tangent value can vary between –∝ (infinity) to +∝.

Some important values are as follows:

- When θ equals 0° then sinθ = 0, cosθ = 1.0, tanθ = 0
- when θ equals 45° then sinθ = 0.707, cosθ = 0.707, tanθ = 1.0
- when θ equals 90° then sinθ = 1.0, cosθ = 0, tanθ = ∝. (infinity)

The inverses of these trig. functions are known as arcsine (arcsin, asin or $\sin^{-1}$), arccosine (arcos, acos or $\cos^{-1}$) and arctangent (arctan, atan or $\tan^{-1}$), These are just the reverse, for example if we know the lengths of the sides we then use the inverse functions to establish the angle, for example arcsin 0.707 = 45°. These inverse functions are also found on electronic calculators.

## Units of angle

There are three units of measurement generally used to define angles, *degrees*, *radians* and *complete revolutions,* although we’re mainly concerned here with the first two. This is mentioned because some calculators require you to specify which of these units is in use for calculations of trig. values, and also as we shall see, the use of radians in various calculations often simplifies the expressions.

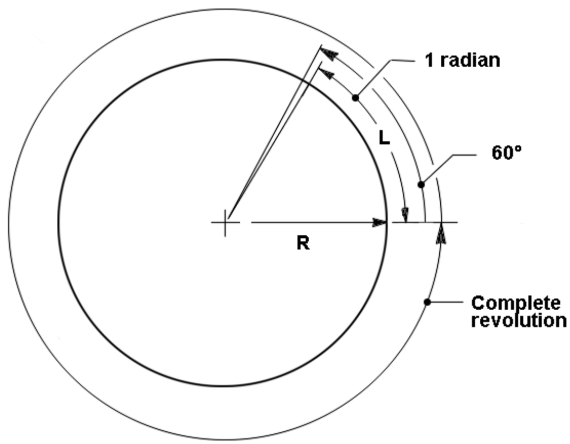



Fig. A5.2 **Showing the relationship between the three main methods of angular measurement. "L" is a length around the circumference of a circle. When L = R (the radius of the circle) then the angle is equal to 1 radian. There are 2π rads. in one complete revolution, and 57.3° in each radian.**

Most people are familiar with degrees and there are 360° in a complete revolution. Radians are sometimes referred to as a natural unit and do often occur naturally when deriving basic behaviour of many handling characteristics. It is therefore worth becoming familiar with their use. A radian is approximately equal to 57.3 degrees and is the angle formed by a segment of a circle where the length around the part circumference is equal to the radius. There are not an integral number of radians in a full circle, unlike degrees, instead there are 2π or 6.283.

Radians are often useful in calculations because for small angles (less than about 10°) the values of the sine and tangent are very close to the value of the angle itself when expressed in radians, this often allows for considerable simplification in some calculations with angles that are generally small, steering angles and slip angles being typical motorcycle examples.

- 10° = 0.1745 rads.
- sin 10° = 0.1736
- tan 10° = 0.1763
- cos 10° = 0.9848

So we see that at 10° the error in assuming that the angle in radians is equal to the sine and tangent is only about 1%, which reduces for smaller angles. It is also interesting to note that the cosine of small angles is very close to 1.0, the error being only 1.5% at 10°. In many cases the analysis of vehicle dynamics is greatly simplified by making these approximations where it is reasonable to assume that angular dimensions will be small.

## Velocity

Velocity is often mistaken to mean speed, that is the distance covered in a given time. The difference is that speed is regarded as a scalar quantity and velocity is a vector. With velocity it is not enough to specify the magnitude (speed) but we have to consider the direction also. Imagine travelling directly north at 100 km/h. and then turn a 90° corner to the east. At the beginning and end of the turn our speed remains constant at 100 km/h, but initially our velocity was 100 km/h north and 0 km/h east, but this changes to 0 km/h north and 100 km/h east at the end of the turn. This might seem an unimportant difference but is vital when evaluating any form of dynamic performance. The normal units of velocity when doing any such calculations are metres per second, m/sec. or m/s. (ft./sec. in the imperial system) and speed is usually considered in kilometres per hour, km/h. (MPH).

In addition to linear velocity we can have rotational or angular velocity. This tells us how fast something is spinning, but being a velocity there is a direction associated with it. However, as each particle in a spinning object is continually changing direction, it would be difficult to specify just one direction and so to overcome this we adopt the convenience of specifying that the direction is aligned with the axis of rotation. We can then consider changes in direction as changes in the orientation of the axis. In most calculations we use the units of radians per second, rads/sec., but also degrees/sec. and of course; RPM or revolutions per minute especially when talking about engines.

Linear velocity is denoted by the letter "v" in calculations, and angular velocity by the Greek letter omega "$\omega$".

## Acceleration

Acceleration is a measure of how quickly the velocity of an object changes, so it too is a vector, that is we have to consider its direction. Precisely, acceleration is the rate of change of velocity against time. For example, if an object is travelling at 10 m/sec. and takes 10 secs. to accelerate to 110 m/sec. in the same direction then the acceleration is (110 – 10) ÷ 10 = 100 ÷ 10 = 10 (m/sec.)/sec. usually written as m/sec.$^2$.

Returning to the 90° turn above, we note that the north bound velocity changed from 100 km/h down to zero, therefore there was an overall negative acceleration in that direction. On the other hand the east bound velocity changed from zero to 100 km/h and so there was an overall positive acceleration to the east. In fact if we looked at the motorcycle at each instant of its journey around the bend, we would see that the instantaneous acceleration was pointing directly into the centre of the curve. This is very important to the study of the cornering process and is sometimes called the cornering or lateral acceleration.

We must also consider angular acceleration, which is the rate of change of angular velocity with time and is expressed as rads/sec.$^2$.

Linear acceleration is denoted by "a" and angular acceleration by the Greek letter alpha "$\alpha$", which is similar to that used for infinity.

## Mass

Mass and weight are often confused, an important contributor to this is the misuse of the units used to measure the two quantities. We'll look at weight after considering the laws of Newton.

Mass is a measure of the amount of matter in an object, it is the same whether the motorcycle is here on earth, in outer space or on the moon, it is not affected by gravity. It is a scalar parameter, i.e. it has magnitude but no direction. The basic unit of mass is the kilogram, kg. and in the imperial system it is the pound, lb. although for some work the slug is used as it represents a larger mass. 1 slug = 32.2 lbs.

As we shall see mass opposes a change in its state of motion and as such is often referred to as *inertia,* the two words are basically equal in this context

We also have the rotary version of mass which is commonly called the *moment of inertia,* MoI, sometimes *polar moment of inertia* is used as well, although the alternative term of *2nd moment of inertia* would be more descriptive. This is not just a measure of the matter in a rotating object, but also of the distribution of that matter, specifically how far away is any particle from the axis of rotation. Any given particle contributes to the whole MoI by an amount equal to the mass of that particle multiplied by the square of its distance from the axis, ($mr^2$), to get the MoI of the whole object we need to sum the individual contributions of all the particles, this is usually done for different shapes by the use of a mathematical technique called the *calculus,* but that is beyond the scope of this short appendix. The units of MoI are kilogram.metres.squared or kg.$m^2$.

The letter “m” is used for mass, and “I” is used for MoI

## Momentum

Momentum brings together the effects of mass and velocity. Momentum is simply the product of mass and velocity, and hence is a vector quantity like velocity. If a car and a truck are travelling with the same velocity, we all know that it is harder to stop the truck or that it would do a lot more damage if it crashed into a building. This is because its momentum is greater due to its higher mass, even at the same velocity. The units of momentum are kg.m/sec. (lb.ft/sec. in the imperial system).

We’ve seen that we can have angular velocity and the angular equivalent of mass, so it follows that we can also have angular momentum. This is the angular velocity multiplied by the MoI and so has units of (kg.$m.^2$).rads/sec. but as radians are a dimensionless quantity we can write that simply as kg.$m.^2$/sec.

Momentum is easy enough to visualize with simple linear motion. Imagine a motorcycle travelling in a straight line on a level road, if all resistance to it’s motion were removed – no air drag, no rolling resistance, no friction etc, and no driving force either then the bike would continue on with the same velocity. Momentum is the product of an object’s mass and its velocity, so we can say in this case that its momentum remains constant. If we now acknowledge reality and allow drag etc. the bike will tend to slow down with time, that is, the application of external forces has resulted in a reduction of momentum. The reverse is true when we apply a driving force, the bike gains momentum due to that force. This is quite simple and intuitive, but velocity, as we have seen, is a vector quantity which means that it has a direction as well as a magnitude and hence momentum has a direction too. So momentum can be considered as having changed if the direction of motion has changed, even though the speed of the object remains constant. In line with the requirements of the ***conservation of momentum*** this means that some force is necessary to change direction too.

Consider again a bike with no drag nor driving force, travelling straight. Now imagine it travelling with the same speed after it has negotiated a 90 degree bend as in fig. A5.3. There will obviously be no longer any momentum in the original direction, $\mathbf{M_1}$, but there will be momentum of equal magnitude in the direction at right angles to it, $\mathbf{M_2}$. Therefore there has been a change in the momentum even though its magnitude is unchanged. This is due to the centripetal force from the tyres, **F**, continually pushing the bike towards the centre of the turn. Without the lateral force the bike will not have turned.

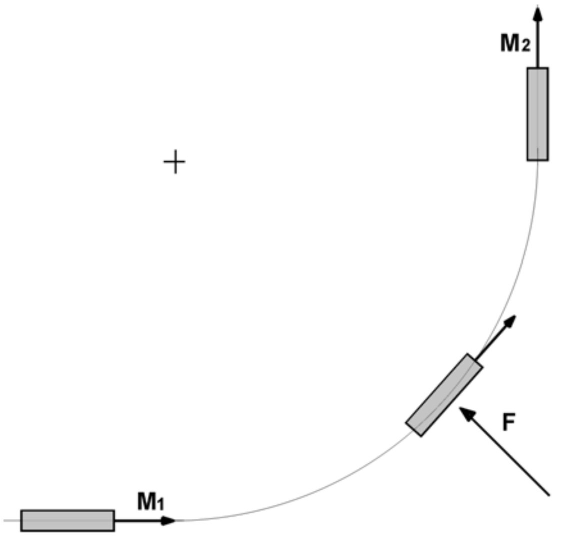



Fig. A5.3 **Prior to a 90 degree turn the bike is travelling to the right with a momentum of $M_1$ at the end of the turn the momentum is as $M_2$ with no residual momentum in the original direction to the right. To change the momentum in this way we have to apply a force F at right angles to the momentum direction at every point throughout the turn as shown. This force is the centripetal force from the tyres. If the force is always at right angles then the magnitude of $M_2$ will be as $M_1$ but in reality the turning tyres will create some drag force and so force F will not be exactly at right angles and the bike will have slowed, the magnitude of $M_2$ will have been reduced.**

# Newton's laws

The behaviour of most physical systems is controlled by a set of rules known as Newton's three laws, which were published in 1687. These are not some arbitrary man-made laws which decree how things must behave, rather they are statements which formalize how things have been observed, over a long period of time, to behave. In fact they cease to properly describe physical behaviour when objects travel very fast, that is when the velocity is a significant proportion of the speed of light. Even a GP 500 machine comes nowhere near this and so for all practical purposes we must consider these rules as unbreakable laws.

## The first law

In essence this can be stated as: ***An object will tend to maintain its present state of motion unless acted on by some external force.***

That is, if an object is stationary it will remain stationary unless acted on by a force. If it is moving then it will continue with the same velocity and direction unless acted on by a force.

This law is closely related to the concept of ***"Conservation of momentum."***

## The second law

Following on from the first law, if we apply a force to an object we change its state of motion, that is we accelerate it in the direction of the force. The second law relates this to the mass of an object.

***Force = Mass x Acceleration or F = ma***

which when rearranged is the same as;

***Acceleration = Force ÷ Mass or a = F/m***

The force and acceleration are in the same direction.

In other words to achieve any acceleration we must apply a force to an object. This not only applies to the obvious acceleration in a forward direction due to engine power, but to negative acceleration from braking and lateral acceleration when rounding a bend, a force must be applied and the acceleration achieved is dependent on the object's mass. If the mass is doubled the acceleration is halved. As we have seen; acceleration results in a change of momentum and so it follows that a force is necessary to change momentum. In fact we can also define ***Force = the time rate of change of momentum.***

This law also applies to rotational motion as well as linear, but in this case we must substitute the rotational equivalents of mass, force and acceleration. Thus;

***Torque = Moment of inertia x angular acceleration or T =*** $I\alpha$

This is a good example of the use of radians for angular measurement, this expression is valid when $\alpha$, the angular acceleration, is expressed in terms of radians/sec.$^2$, if we wish to use degrees/sec.$^2$ we have to multiply by the conversion factor and so the expression becomes $T = \pi I\alpha/180 = 0.01745\, I\alpha$

Not only is this important for the obvious cases of accelerating the wheels and crankshaft, but also for the less obvious situation of rolling and yawing a bike into a curve.

### The third law

Simply put this law states that; ***"Action and Reaction are equal in magnitude and opposite in direction."*** This means that if we apply a force to something then it pushes back with equal intensity. Through the tyres a bike pushes down onto the pavement, but the pavement must react by pushing back upward on the tyres. Imagine that the ground suddenly disappears, the bike will then start to fall because there is no longer any upward reaction from the road to balance the downward force of gravity. Now according to Newton's 2nd law, the gravitational force otherwise known as weight, will begin to accelerate the bike downwards.

All three laws are very important but this third one is vital when considering the actions between the road and the tyres, for example. We need to be very clear about the directions of the tyres pushing on the road, and the road pushing back on the tyres, not just the weight bearing case but under cornering, braking and accelerating.

## Force and weight

Force is defined in terms of its ability to change momentum, which also means to produce acceleration and so is also a vector quantity. However, there are countless everyday examples of forces that don't seem to be connected to any acceleration. For example, the simple act of pushing against the wall of a house doesn't involve acceleration, this apparent conflict is explained because Newton's third law shows us that the house pushes back in an opposite direction thus cancelling our pushing force. In other words there is no unbalanced or resultant force left to accelerate anything. This is a very important concept to grasp, we need to use it in virtually all aspects of evaluating dynamic vehicle behaviour. Motorcycle handling is about accelerations of several types, some linear and some angular, we have cornering acceleration, braking and roll acceleration just to mention three. The forces that produce these accelerations are the resultants of the combinations of numerous forces, which cancel and add to one another to form a balance. The unit of force is the Newton, N, which is the force which will give a mass of 1 kg. an acceleration of 1 m./sec.$^2$, the imperial system uses the poundal, pdl, which is the force that will give a mass of 1 lb. an acceleration of 1 ft./sec.$^2$

Weight is a force, but in everyday use it is most common to use the units of mass as if they were units of force, this is unfortunate because over the years it has been the source of much confusion and worst, has been the source of many engineering miscalculations. This occurs in both the metric and imperial based systems. Perhaps the source of this confusion is that the most convenient way to measure mass is to measure its weight, these two different parameters are related by the acceleration that the weight force (gravity) would give to the mass in free fall. This gravitational acceleration is close to 9.81 m./sec.$^2$, which means that a mass of 1 kg. must be subject to a force of 9.81 N, i.e. it weighs 9.81 N. here on planet earth.

The Newton and the poundal are both too small to be convenient for everyday uses such as shopping, and so we find that most weighing machines are calibrated in kg. and not newtons. It is therefore impossible to avoid such misuse and to reduce confusion, in a somewhat pragmatic fashion, it has become common in technical circles to add the letter "f" to distinguish force from mass. Thus a 1 kg. mass will weigh 1 kgf., (the same applies in the imperial system; 1 lb. mass weighs 1 lbf.). This kludge is not a real solution because it often leads to errors due to uncertainty about when and how to use the gravitational constant in calculations. This is easy to demonstrate with the expression for acceleration: ***Acceleration = Force ÷ Mass or a = F/m.*** If we use a force of 1 kgf. and a mass of 1 kg. then the acceleration will equal 1, but what units? We know that in m./sec.$^2$ the acceleration is 9.81 not 1. There are three common ways around this problem.

- Express the acceleration in relation to the acceleration of gravity, the units of acceleration then become "Gs". That is 1 kgf. gives an acceleration of 1 G to a mass of 1 kg.
- Express the mass in terms of different units, in the imperial system, an example is the previously mentioned "slug". So 1 lbf. will produce an acceleration of 1 ft./sec.$^2$ in a 1 slug mass.
- Introduce the gravitational constant as a fudge factor into the basic expression. So ***a = F/m*** becomes ***a = Fg/m***

The first has the advantage that the acceleration value will be the same regardless of the system of units in use, metric, imperial or other. It gives an intuitive feel for the degree of acceleration because it is some multiple of that due to gravity, with which we are all familiar.

The second is common in some specific spheres of engineering.

The third is also quite common but for a long time has been the source of countless errors in engineering calculations, some with serious consequences. The problem is that users of this method are often uncertain as to when to use "g" and whether it should be in the numerator or denominator.

It is very important to fully grasp the differences between weight and mass in order to get a good understanding of motorcycle behaviour.

In this book the unit of mass generally used is the kg. and that of force is the newton, N. There are some exceptions, mainly when displaying data from external sources which was supplied in other units.

## Moments, couples and torque

These three terms basically have exactly the same meaning although their use tends to be attached to specific topics. For example, "moment" tends to be used when discussing structural aspects and of course “torque” is used when talking about engine performance. In some other cases there seems to be no preference, “roll torque”, “roll moment” and “roll couple” are all used to describe the same thing.

Torque etc, is the rotational version of force, it is the measure of a twisting tendency created by an offset application of a force. Its value is determined by multiplying the force by the degree of offset as shown in the sketch below.

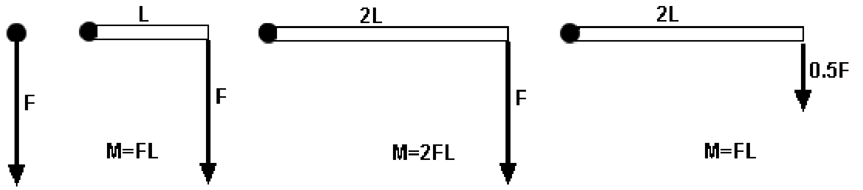



Fig. A5.4 **The first figure generates no moment about the black circle because there is no offset. The others show how the moment depends on both the force and the offset.**

The units of moment are newton.metres Nm. and is represented by the letter “M” although “T” is also used for torque.

## Centripetal & centrifugal force

Centripetal and centrifugal forces are the two opposing parts of the Newtonian concept that for every force there is an equal and opposite reaction, for the case of an object following a curved path. Centripetal is the inward force and for vehicles that is the lateral tyre force pointing toward the turn centre, centrifugal is the conceptual force trying to make the vehicle move outwards away from the centre.

As to whether centrifugal force exists is the subject of a long running argument between purists and pragmatists. The purists argue that there is no such thing as centrifugal force, and they have a good case, which is why I said conceptual force above. Without centripetal force the vehicle will go straight and this force is necessary to cause the forward momentum to change direction, in reality there is no force trying to make the vehicle move away from the turn centre, all it wants to do is carry on in its current direction.

Consider a mass on the end of a piece of string being swung around. Most people will talk about the tension in the string as being centrifugal force generated by the mass trying to fly away from the centre. The implied assumption being that the mass is pulling outward (centrifugal). Actually it is the string that is pulling the mass inward (centripetal) to change its direction of motion, which otherwise would be in a straight line. If we release the string, the mass doesn't fly outward, it simply continues in a straight line tangential to its instantaneous velocity at the moment of release. If the mass had been subject to a centrifugal force, then it would have flown radially outward from the centre. In fact its motion will be at right angles to the outward direction at that instant.

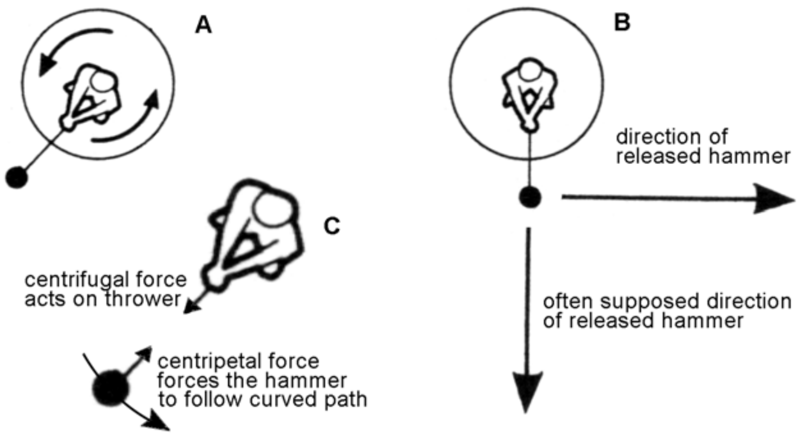



Fig. A5.5 **A hammer thrower swings the hammer around in a circle to build up the velocity, A.**

**At the point of release, B, the hammer continues in a straight line, tangent to the instantaneous velocity at that time.**

**C shows the meaning of centripetal and centrifugal forces.**

In vehicle terms there is no "force" trying to make the vehicle move outwards from a turn. What appears to be such a force is just its momentum wanting to continue as before, that is; continue in a straight line tangential to its instantaneous velocity. On the other hand we need a "real force" to push the machine around a bend, i.e. change the direction of its forward momentum. This force is supplied by the tyres and is known as centripetal force. Centripetal means "inward force" but as we know every force has to be balanced, in this case it is balanced by the changing momentum, however, it is often much more convenient to think in terms of an actual balancing "force", and so we have adopted the outwardly acting conceptual force called "centrifugal force". In other aspects of dynamics some people use (although its now largely regarded as an unnecessary technique) what's called the "d'Alembert" force which is just a way of expressing the reaction to a force needed to accelerate an object. "Centrifugal force" is just the "d'Alembert force" for the special case of a curved motion, it is the conceptual reaction force needed to balance the real Centripetal force.

We can say that the centripetal force acts inward at the tyre contact patch and hence causes a rolling moment about the CoG. Alternatively, it is often, though less correctly, said that centrifugal force acts outwards at the CoG which causes a rolling moment about the tyre contact patch. The end result is the same, but most people are more at home with the latter approach.

My own personal preference is to only use the term centripetal force, but to impose that on the reader would needlessly create difficulties for some, therefore in this book there is a mixture of the two terms. “Centrifugal” is used in cases were most will find that concept easier to accept, the balance of forces that control lean angle being a good example.

## Addition and resolution of velocities and forces

Vector quantities can be added and broken down into components as long as due regard is paid to the direction. This is simple to visualize when forces for example are inline. Consider the case of braking whilst still applying power. We normally regard the driving force as positive and braking as negative. If we add these two forces the difference between them will be the nett driving or braking effort. When forces or velocities are not in line we can still add them but the method is not quite as straightforward, fig A5.6 shows how we do it.

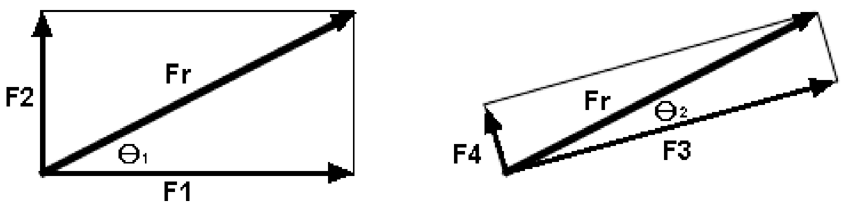



Fig. A5.6 **Addition of forces. In both cases the resultant force Fr is identical, but the two forces that make up the same Fr are different, both in magnitude and direction.**

If we represent the forces to be added by lines indicating the direction of the forces and the lengths of the lines being proportional to the magnitudes, then we can find the resultant as shown in the figure. On the left the two forces F1 and F2 are added to produce a resultant force Fr as show. In the second case the resultant is exactly the same but the component forces F3 and F4 are different, there is an infinity of force combinations that can produce the same final force. In these examples F2 is at right angles to F1 as is F4 to F3, but this is not necessary we can sum forces regardless of the orientation between them.

This is where trigonometry is of practical use to us. In the first example we see that $\tan\theta_1$ = F2/F1 and so by using the inverse we can calculate the angle between Fr and F1 by $\theta_1$ = arctan(F2/F1), and between Fr and F3 by $\theta_2$ = arctan(F4/F3). Note that as the angle gets smaller F3 gets closer to Fr and F4 gets closer to zero.

If there are an infinite combinations of forces that can make one resultant, as above, then it follows that we can break any resultant force down into an infinity of components, this is usually of most value to us when we resolve a force into two components at right angles. For example let's consider the case of evaluating the loading on the front fork when just subject to the static weight of the bike, fig. A5.7

We need to know just how much of this weight has to be supported by the springs and how much acts in a bending manner. The sketch shows how the total fork load can be separated into these two components. This is also another application of simple trigonometry. In this case we know $\theta$, because it is the rake angle, and we can measure the weight and so we also know **Ft.**

**Fb** and **Fs** can be calculated from **Fs = Ft.cos** $\theta$ and **Fb = Ft.sin** $\theta$**.**

At a rake angle of 25° **cos** $\theta$ **= 0.906** and **sin** $\theta$ **= 0.423.**

so **Fs = 0.906.Ft** and **Fb = 0.423.Ft**

In other words for a typical rake angle the load supported by the front springs is about 91% of the supported weight but about 42% of the weight value acts to bend the forks frontward.

The above examples show the addition and resolution of forces but exactly the same methods apply to all vector quantities and there are other examples using velocity in the subject chapters, particularly when discussing slip angles and velocities in the tyre chapter.

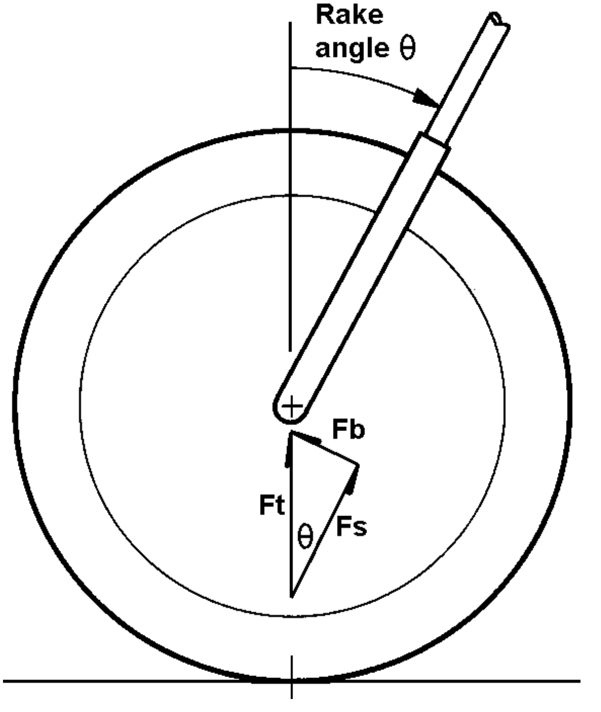



Fig. A5.7 **A typical motorcycle example where it is useful to resolve a force into a pair of components at right angles to one another. Ft is the total vertical force pressing upward at the axle to support the front weight of the bike (minus the front wheel). However, this force is not acting inline with the fork movement, only some will be going into compressing the springs and some will be going into trying to bend the fork legs. So we need to break the force down into two components, one acting on the springs, Fs, and the other as a bending force, Fb.**

## Work, energy and power

Work in a technical sense is identical to energy.

Work involves movement and is the product of the force required to move an object and the distance that it moves. Fig. A5.8 shows a moving motorcycle, this has a total drag force of **Fd**, including rolling resistance etc., to keep it moving the engine must supply a final force at the tyre / road interface that is equal and opposite to **Fd**, call it **Ft**. To move over a distance of L the engine must do **Ft.L** amount of work. The amount of energy expended or work done does not depend on how long it takes. We have to talk about power when time is a factor.

Power is defined as the time rate of doing work, that is; how much work is done in a unit of time. In our example case if it takes time **t** to cover the distance **L** then the power needed is **P = Ft.L/t.**

There are many types of energy but we are principally concerned with potential energy and kinetic. As the name suggests, potential energy describes the potential that the state of an object has for doing work. The most common example is when something has been raised, then the force of gravity can do some work if the object is allowed to drop. Kinetic energy is energy of motion, and depends on the mass and speed. When a motorcycle is stationary at the top of a hill it has some potential energy referenced to the bottom of the hill, now if the motorcycle is allowed to free roll down the hill the bike will gather speed, as it does so the potential reduces, it has just done work by accelerating the machine which gains it as kinetic energy.

Work and energy have units of newton.metres, Nm which are also called joules, J. Power has the units of joules/sec. which is expressed as Watts, W.

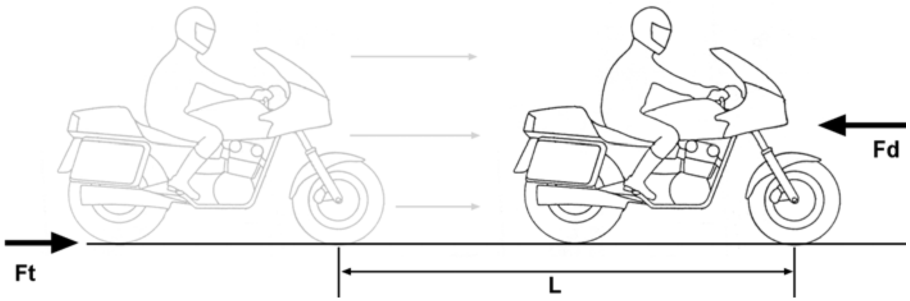



Fig. A5.8 **Energy at work. The tyre or traction force, Ft, balances the overall drag force, Fd, to move the bike. To move it a distance of L the engine must do Ft.L amount of work.**

Work, energy and power are scalar quantities, they have magnitude only but no direction. This might seem strange because both force and distance are vectors, but there are two methods of vector multiplication and one yields scalars and the other vectors. Imagine pushing a motorcycle on a level road, you will have done some work to cover say 100 metres, but if you turn around and push back to the start the work will not be returned, in fact you'll have to do as much work again to get back, therefore work is not affected by direction and so neither is power.

## Nomenclature and sign conventions

Because this book is aimed at the average motorcyclist with a technical interest as well as the engineer, I have tried to use a nomenclature that is as easily understood as possible. It is common in more academic works to use a whole host of Greek letters to denote various parameters especially all the different angles of interest on a bike. This is good when there are a lot of equations using a mixture of angles, for example: rake, roll, steering, slip –angles, when these are used together we need a system of symbols to distinguish each one, but this book is not like that and I know that most people are confused when they see a lot of unfamiliar symbols. Therefore, to keep it simple, the symbol theta, $\theta$, is used to denote any angle, be it rake or roll etc.. The only other strange symbols used are omega, $\omega$, for angular velocity, alpha, $\alpha$, for angular acceleration, sigma, $\sigma$, for stress and infinity is $\propto$.

I have also adopted a similar pragmatic approach to the sign conventions used, especially when talking about roll, steering and slip angles. It is normal to adopt a coordinate system that has positive angles to one side and negative angles to the other, an exception to this is camber angle which distinguishes positive and negative by whether the wheel leans into of out from the vehicle, this obviously has no application to a single track vehicle. In order to make as much sense to the majority of readers the reference for deciding a positive or negative angle, in this text, is whether or not the angle is in the direction of the curve. For example a positive lean angle is considered to be one that leans into the centre of the desired turn. A positive steer angle is when the handlebars are turned to the left for a left hand bend, a negative or countersteering angle is when the handlebars are pointing left when the bend is to the right.

## Normalization

Various data are presented in what's known as a normalized form. A common example is the coefficient of friction. To measure this we would place a vertical load on a object and measure the horizontal force needed to move it. We might also do this at several different vertical loads. So our results would be in the form of an horizontal force against a vertical force, to make it easier to compare different materials and different surfaces we divide the horizontal force by the vertical and so we get a non-dimensional parameter that equals 1.0 when the two forces are equal. This is the coefficient of friction and the process to get it is called *normalizing*.

In the same way it is possible and often useful to normalize various parameters and the tyre camber and steering stiffnesses are an example widely used in this book. In this case when it is necessary to get an actual value for the tyre camber force we just multiple the normalized value by the vertical load.

The aerodynamic drag coefficient, the Cd, is another well used case. The Cd allows us to compare different shapes without having to consider size.

# A6 Analysis of mechanisms

Whole books have been written on this subject and so we can only scratch the surface, but all that is necessary here, is to review the basics behind some of the concepts used in chapters 3 and especially 9. We just need to understand the ideas behind terms like; virtual pivots, virtual swing-arm and instantaneous force centre.

Consider what's called a four bar linkage as in fig. A6.1, top left.

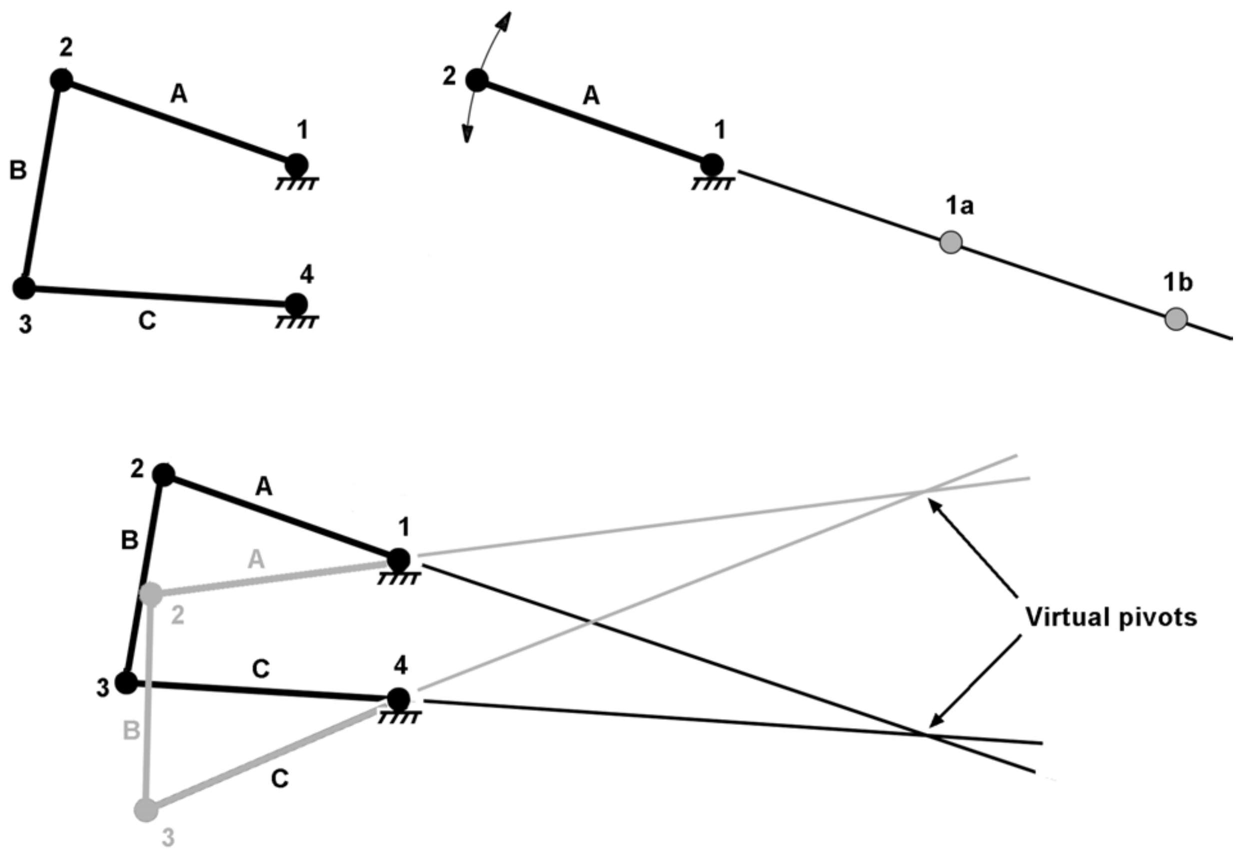



Fig. A6.1 **Motion of a 4 bar mechanism shown top left. Top right: isolates link A, if the pivot is moved anywhere along a co-linear line as shown for example at 1a or 1b, the instantaneous motion at 2 will be unchanged. The lower diagram shows how a virtual pivot is defined and also demonstrates clearly how it changes position as the mechanism is moved about its real pivots.**

The pivots in each of the four corners allow the links to move in a defined manner. Pivots 1 & 4 are fixed to some reference object (the main frame of a car or motorcycle) and so links A & C are free to rotate such that points 2 & 3 describe arcs of circles. Let's isolate link A and study its motion independently. At any instant in time the velocity of point 2 will be at right angles to the link A itself, so as long as the pivot point 1 is aligned with the link A (along the drawn line), the instantaneous motion of point 2 will be unchanged. This will also apply to link C. Therefore, if we draw lines co-linear with links A & C as in the lower sketch then if they meet, this will be a unique point such that, if both of the links were extended to it (that is; pivots 1 & 4 become coincident at that point) the instantaneous motion at the other ends of the links, points 2 & 3, will be unchanged. This point is called the virtual centre or virtual pivot, and is the only point that satisfies the condition that the motion of points 2 & 3 remain unchanged at any instant. When the links A & C are parallel we can't drawn in the virtual centre but in fact it acts as if it were at infinity in line with the links.

The significance of this virtual centre is that we could replace the original 4 bar linkage with a 3 link version pivoted at the virtual centre. This then becomes the only connection between the reference frame and the mechanism, and its motion characteristics can easily be evaluated. Perhaps of more use to us is that as this becomes the only connection, the virtual pivot is the only point that forces can be passed between the links and the reference frame. Additionally as this is a pivot point it cannot pass any moments either. The virtual centre is also called the "Instantaneous Force Centre" or IFC and sometimes the "Zero Moment Centre", for obvious reasons.

These properties of the IFC provide us with some simple graphical methods for the analysis of suspension behaviour, in particular squat and dive characteristics. These methods are detailed in more depth in chapter 9.

# A7 CoG and mass distribution of rider

This appendix is just a brief guide to calculating the CoG position of a motorcycle with and without rider. The CoG is an imaginary point through which we consider all the weight of the bike to act. That is to say, for many calculation purposes we can consider the machine to be composed of a point weight equal to the machine weight but all acting at the CoG. For all practical purposes this is the same as the mass centre and the terms are often used interchangeably, even though strictly speaking they have different meanings.

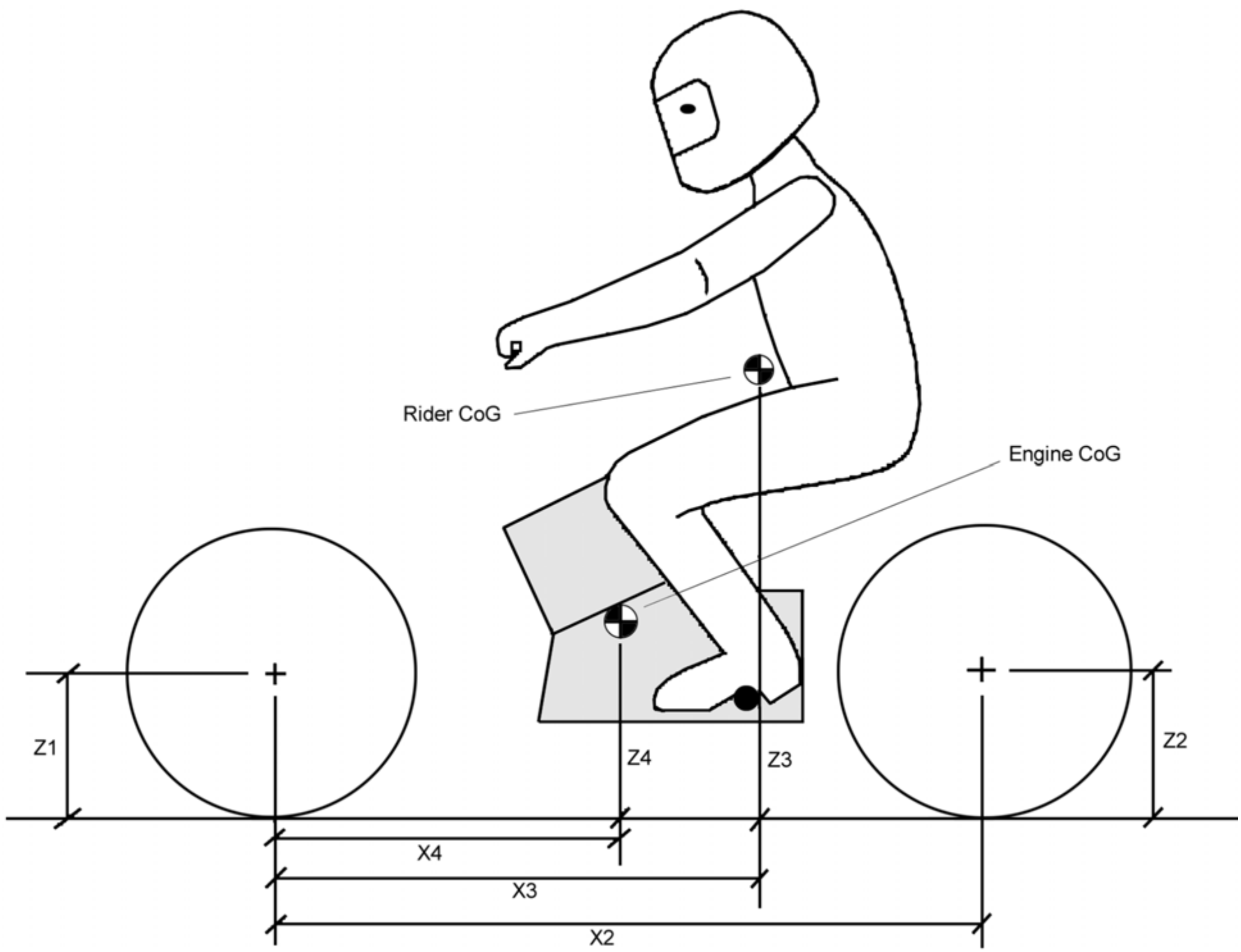



Fig A7.1 **This sketch shows the important parameters for calculating the CoG of a machine with four components.**

**Viz: Two wheels, engine / transmission and a rider. Other component parts are added in the same way.**

As the CoG is a point in space it needs three coordinates to uniquely specify it. However, on a motorcycle we can usually make the simplifying assumption that it is on the vertical centre plane of the machine. In that case we only need to consider two dimensions, that is, the longitudinal and vertical coordinates. To calculate the CoG position we only need to realize that the motorcycle would be balanced in any position if it were suspended at the CoG. That is, there is no nett gravitational moment about that point. If we start with a convenient reference point and balance the total moment against that of the CoG then we can easily determine the CoG location. We need to do this twice, once to get the longitudinal dimension and then again to get the vertical. Fig. A7.1 shows a simplified bike, with only the two wheels, engine and rider but this is adequate to show the method. I normally choose the ground contact point of the front wheel as the reference and the fig. shows the coordinates from that point.

Basically we must equate the moment about this point of the total weight acting at the CoG with the sum of all the moments of all the individual components. Hence:

$$W_t.X_{CoG} = W_{rw}.X_2 + W_r.X_3 + W_e.X_4 \ldots\ldots\ldots$$

$$or$$

$$X_{CoG} = \frac{W_{rw}.X_2 + W_r.X_3 + W_e.X_4 \ldots\ldots\ldots}{W_t}$$

where:

| | | |
|---|---|---|
| $W_t$ | = | total weight |
| $W_{rw}$ | = | rear wheel weight |
| $W_r$ | = | rider weight |
| $W_e$ | = | engine weight |
| $X_{CoG}$ | = | horizontal position of CoG from front ground contact point |
| $X_2$ *etc,* | = | horizontal position of the CoG of the separate components as shown |

We add all the other components of the machine in like manner. Just add the weight of a part multiplied by is distance from the reference point. Note that we didn't include the front wheel, that is because we choose the front wheel as our reference point. We calculate the vertical CoG location in the same way but just substitute the vertical distance of the individual parts. In this case we must add the front wheel because our reference was at ground level not at axle height. In practice the easiest way to do this is to setup a spreadsheet on a computer.

At the design stage we have no option but to estimate or measure the weights and CoG location of each part and calculate as above. However, if we want to know the CoG position of an existing bike then the easiest and most accurate way is by measurement. The horizontal position is the easiest to obtain, we only need to get the weight supported at each end and then:

$$X_{CoG} = \frac{L_r.WB}{W_t}$$

where:

$L_r$ = load supported by rear wheel

$WB$ = wheelbase

Measuring the vertical CoG position on an existing bike can be done in a variety ways but is more difficult than obtaining the horizontal value, especially with the rider on board. Probably the easiest way is to lift the rear wheel onto a raised block and then weigh each end again. Depending on the height of the block there will be a reduction in rear wheel load.

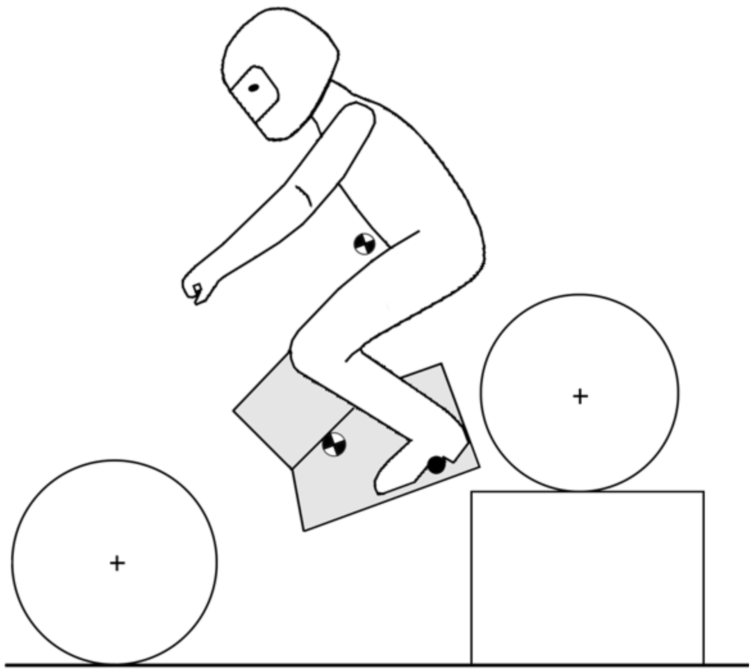


Fig A7.2 **To measure the vertical CoG position, one method is to weigh each end of the bike with one end raised onto a block.**

In this case we calculate the CoG height according to:

$$H_{CoG} = \frac{WB.(L_r - L_{r2})}{W_t.\tan(\theta)} + R_{av}$$

where:

$H_{CoG}$ = height of CoG

$L_r$ = load supported by rear wheel when level

$L_{r2}$ = load supported by rear wheel when lifted onto block

$R_{av}$ = average radius of the wheels

$\theta$ = angle of lifted bike [ = arcsin(*height of block/WB*) ]

This formula can be simplified if we use a block height which is one third of the wheelbase. For example, if the wheel base is 1500 mm. make the rear wheel lift equal to 500 mm. Then:

$$H_{CoG} = \frac{2 \cdot 828.WB.(L_r - L_{r2})}{W_t} + R_{av}$$

To assist with the estimation of the CoG of the rider I find it useful to use an articulated model of a rider in combination with some averaged values of the distribution of masses between the limbs. The following figures show such a model made from sheet plastic (actually an old helmet visor), and typical mass distribution. Remember that humans vary in shape and so these figures are only a guide.

Fig A7.3 **Articulated rider model made from helmet visor.**

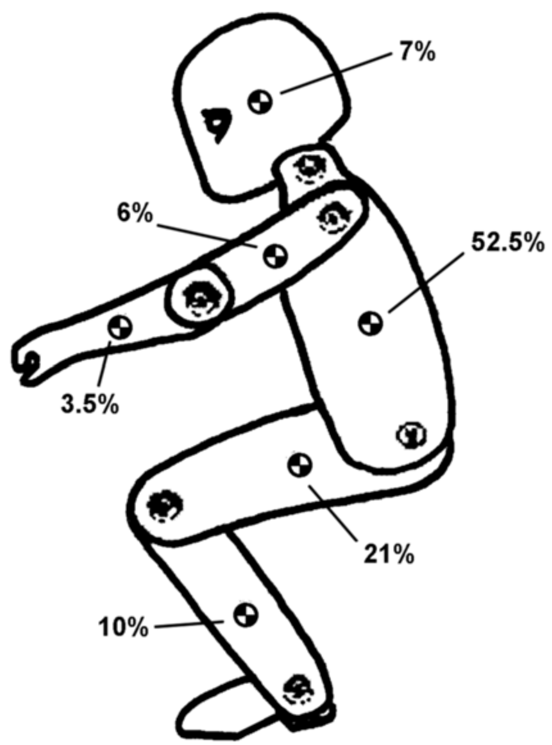



Fig A7.4 **Typical mass distribution of rider.**

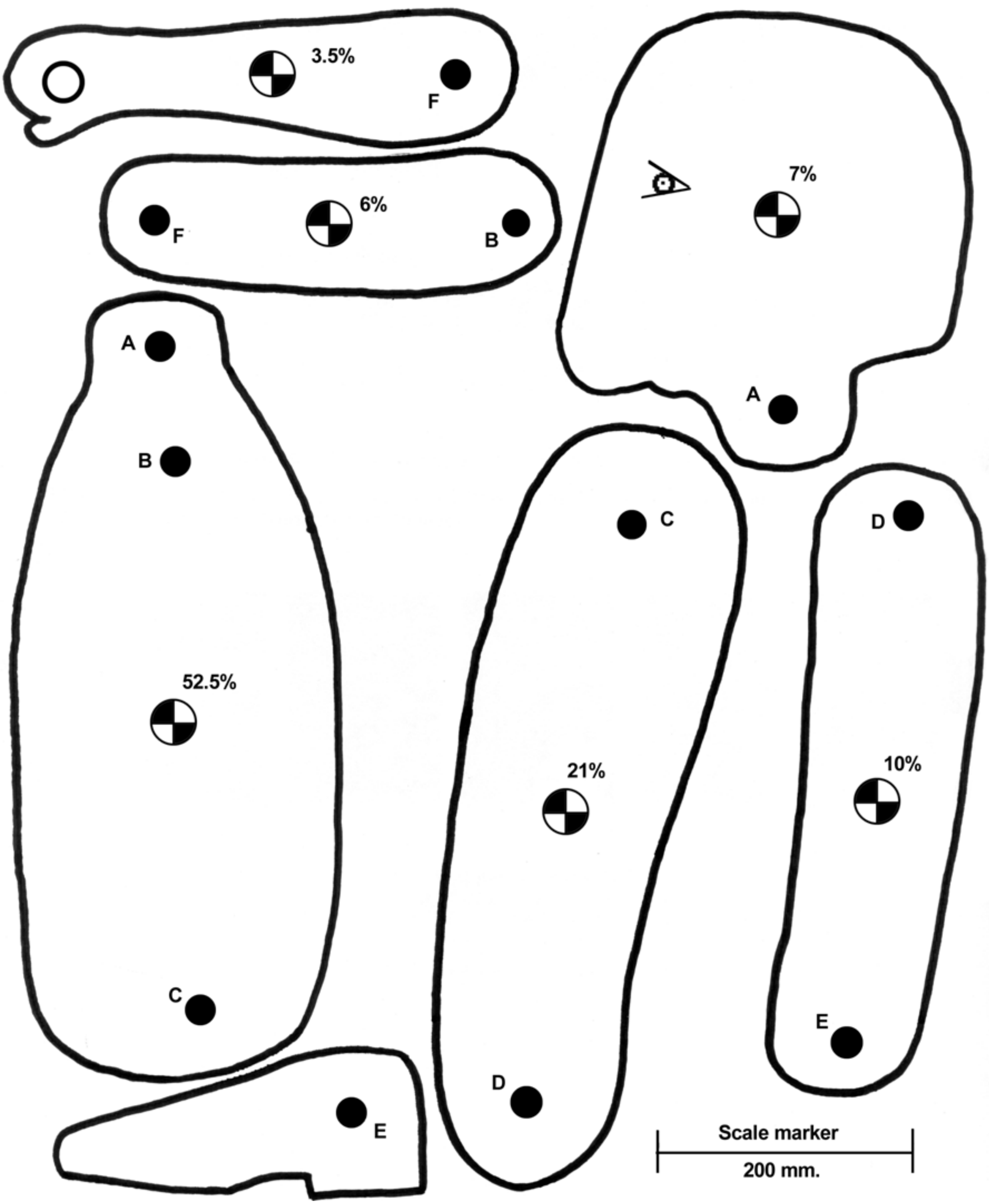
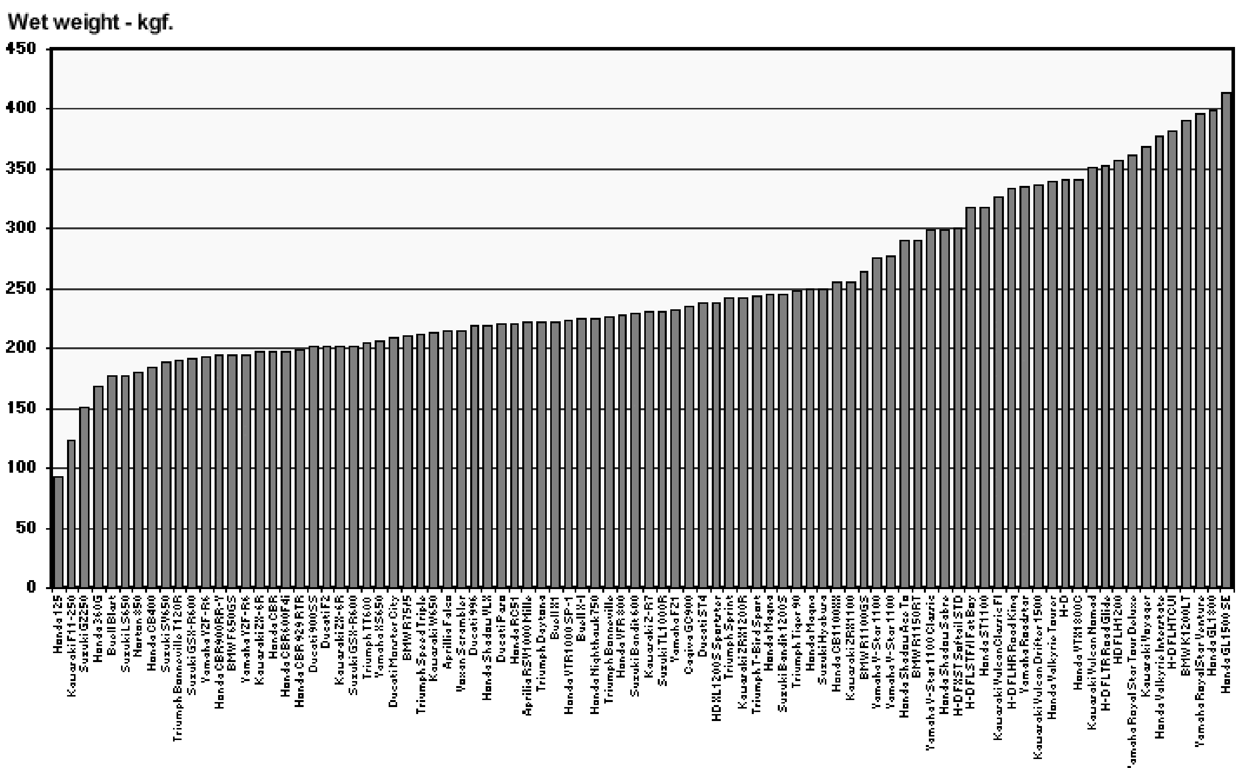


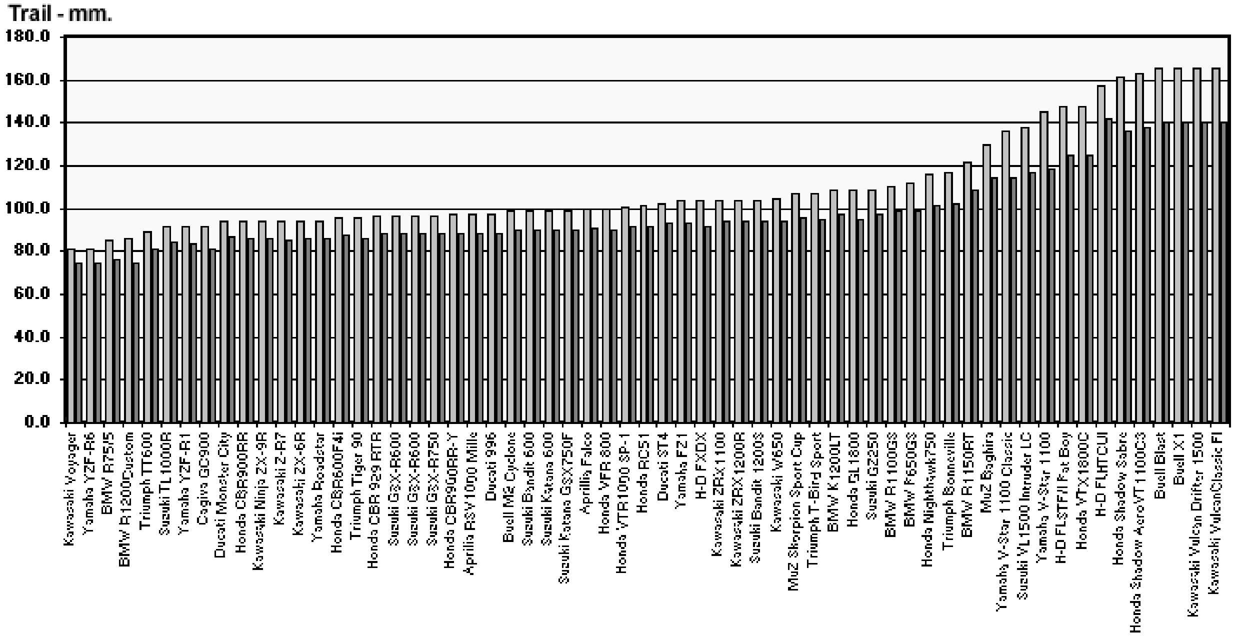

Fig A7.5 **Photocopy or trace these templates to make a model as in fig. A7.3. Pin or pop-rivet the joints so that the model can be adjusted to suit the riding position of the bike under consideration. The CoG of the rider can then be estimated by adding the components from each body part when appropriately aligned.**

# A8 Typical data

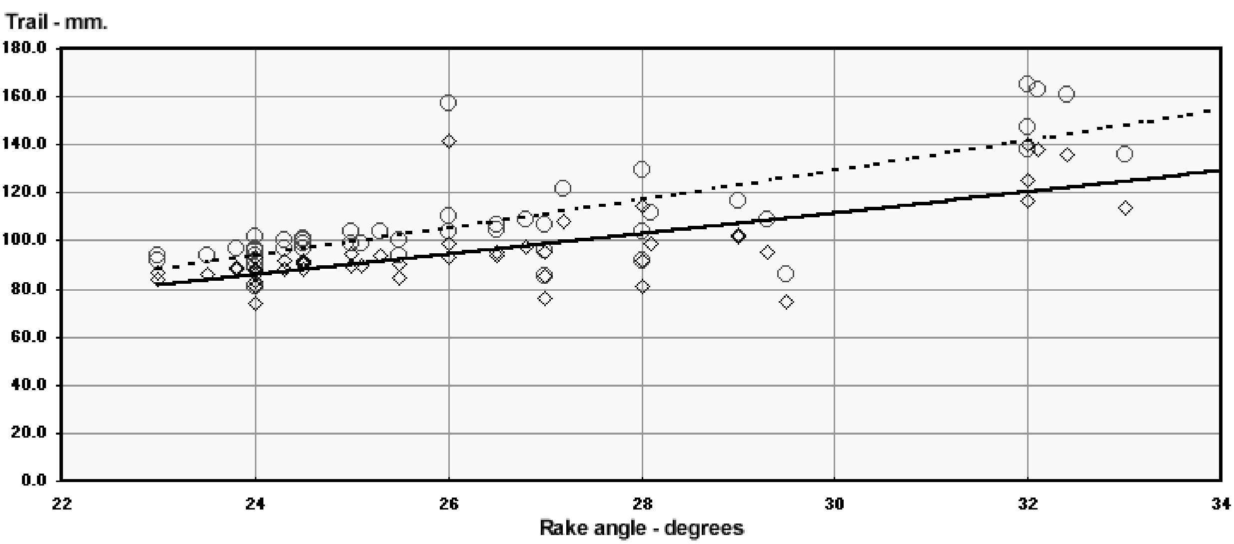
This appendix lists various data from a range of typical motorcycles. Most are models from 1998 onwards, although a small number date back to the 1970s, and are generally of 650 cc. or greater capacity but with a small number of lower capacity machines. The data has been compiled from various sources (the majority from the American “Motorcycle Consumer News”). Some data is as supplied by the manufacturers who like to cheat where possible, for example the wheelbase of chain driven machines can vary by the amount of their chain adjustment, and some quote the most forward position in order to enhance a sporty image. The CoG height and dependent data is probably subject to greater measurement error than the other parameters. Nevertheless, the following is an interesting collection of parameters as a guide to typical values. The data is presented in a series of ordered bar and cluster charts to provide an easy visual appreciation of the data ranges.



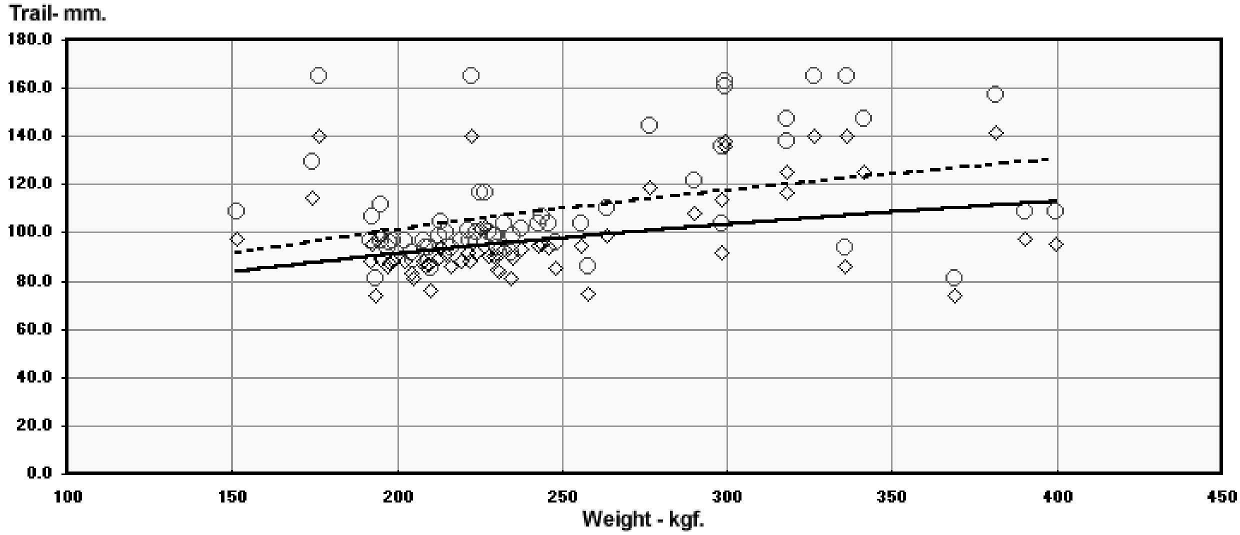

**Wet weights of a range of motorcycles. Note that a large number of examples range between 180 and 250 kgf. The range over 250 kgf. is inhabited by large touring machines and cruisers, and below 180 kgf. is generally the preserve of the smaller capacity bikes. The perception of weight is affected by the ease of stationary balance and this depends on CoG height as well. The “perceived weight index” chart, at the end, probably gives a better idea of this.**



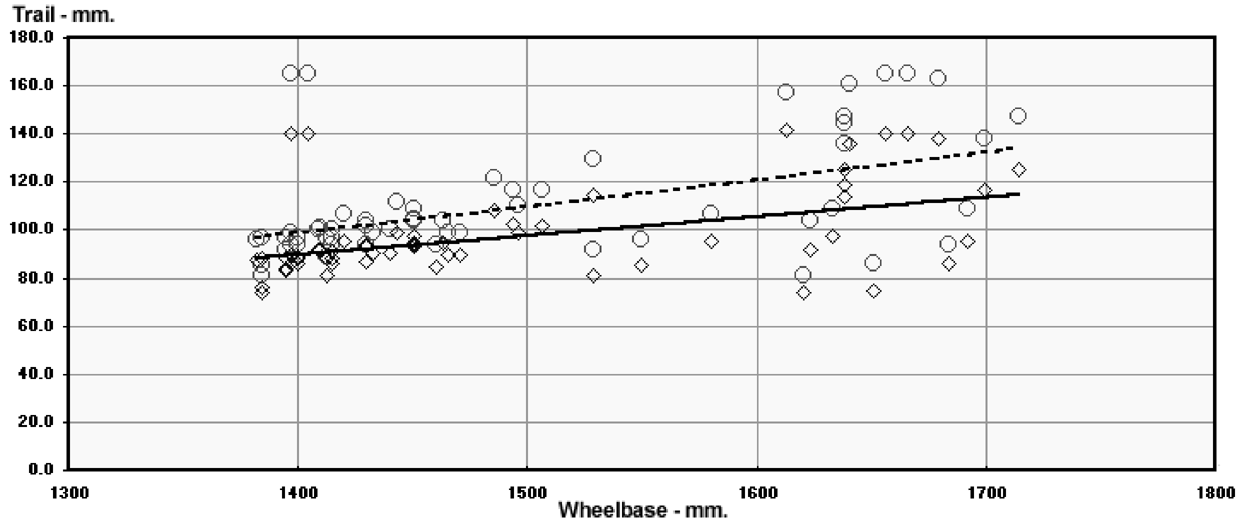

**The light coloured bars indicate the ground trail and the dark bars show the real trail. The majority of values are in the range or 90 – 110 mm. for ground trail and 80 -- 100 mm. real trail. Generally it is the cruiser machines that have much larger trail values, these machines also tend to have greater rake angles for styling reasons.**



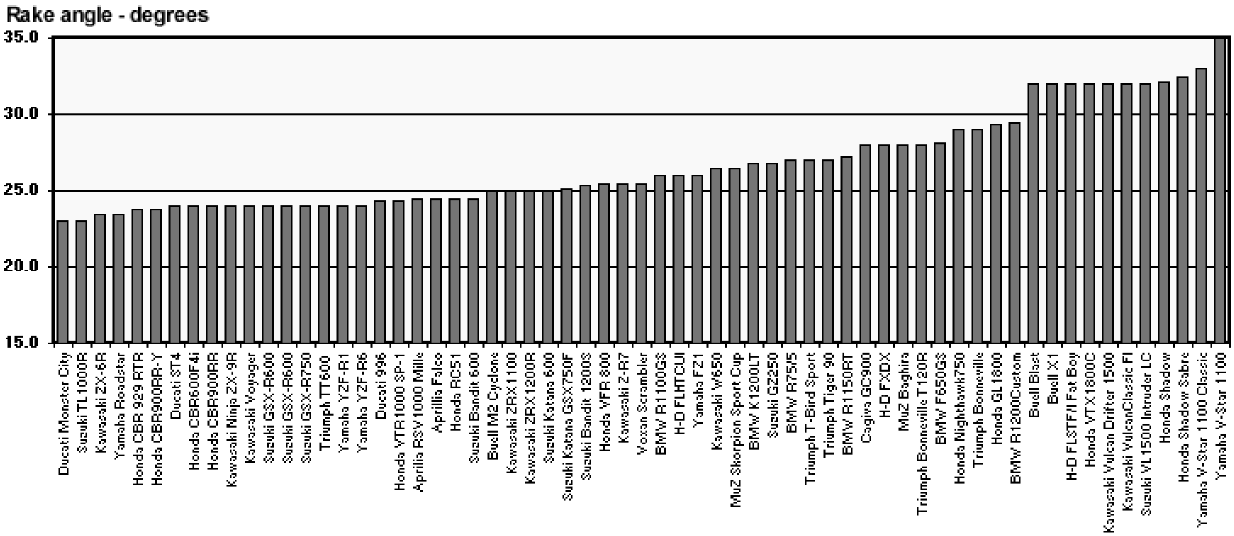

**Cluster chart of trail against rake angle, the circular symbols are ground trail and the others are real trail. Trend lines have been added, the solid one is for real trail. Note that both real trail and ground trail tend to increase with rake angle.**



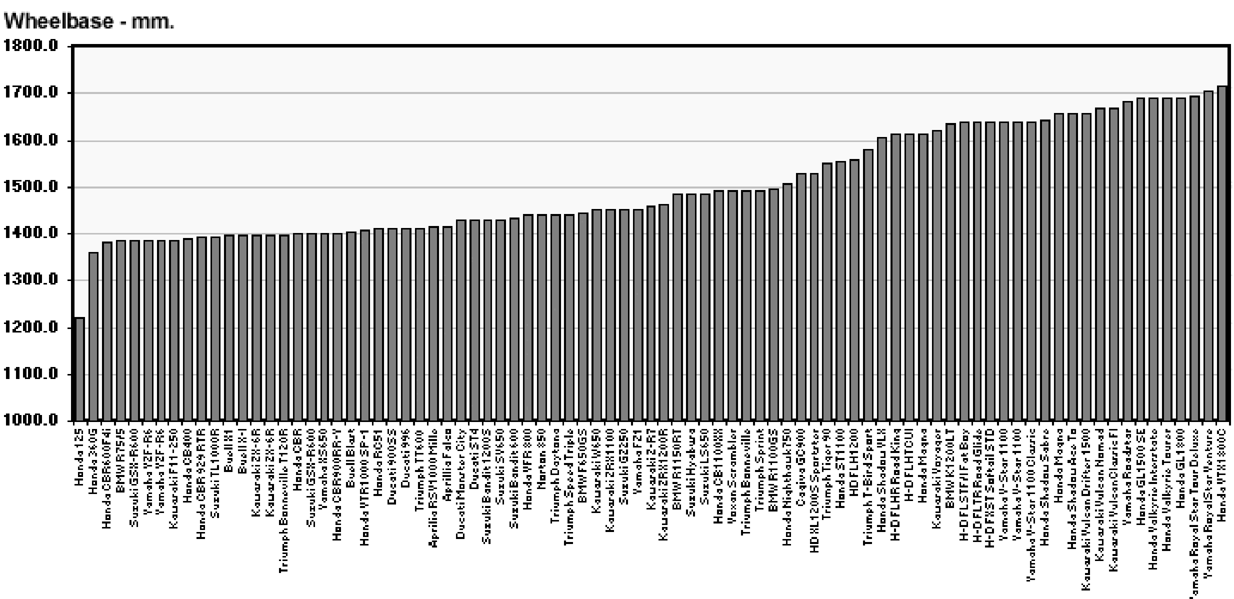

**Using the same symbols as the previous chart, this shows the trail compared to weight. Although there is more data scatter the trend lines indicate that in practice more trail is generally used on heavier bikes.**



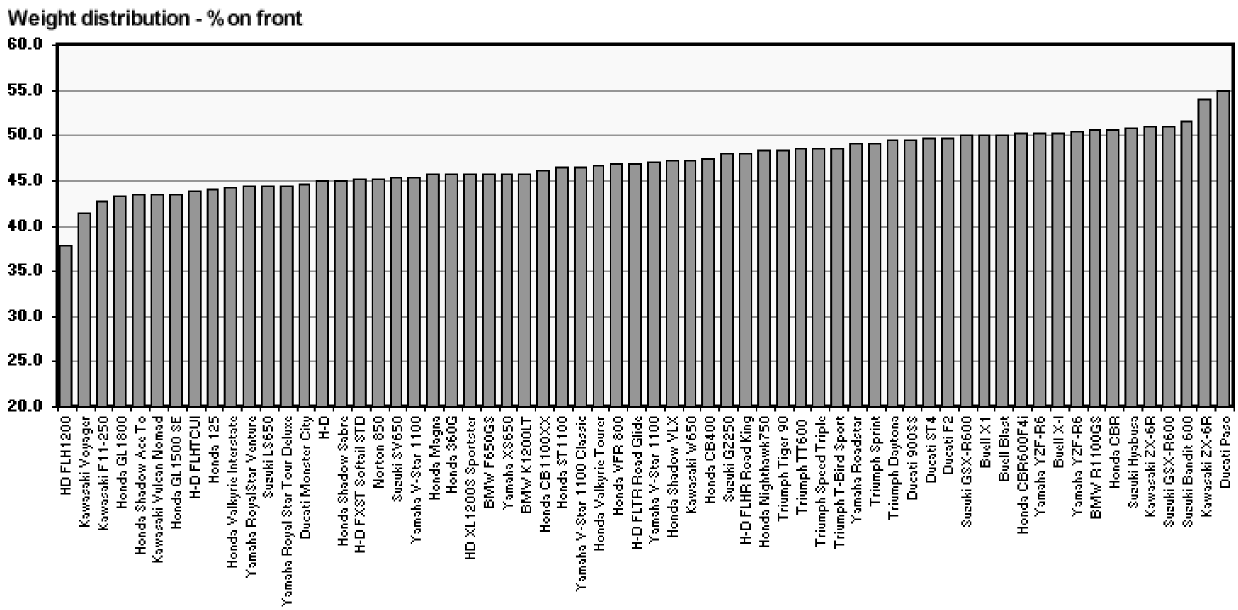

**Trail against wheelbase, as with the trail relationship to weight there is a degree of scatter but the trend lines show a general increase in trail with greater wheelbase.**



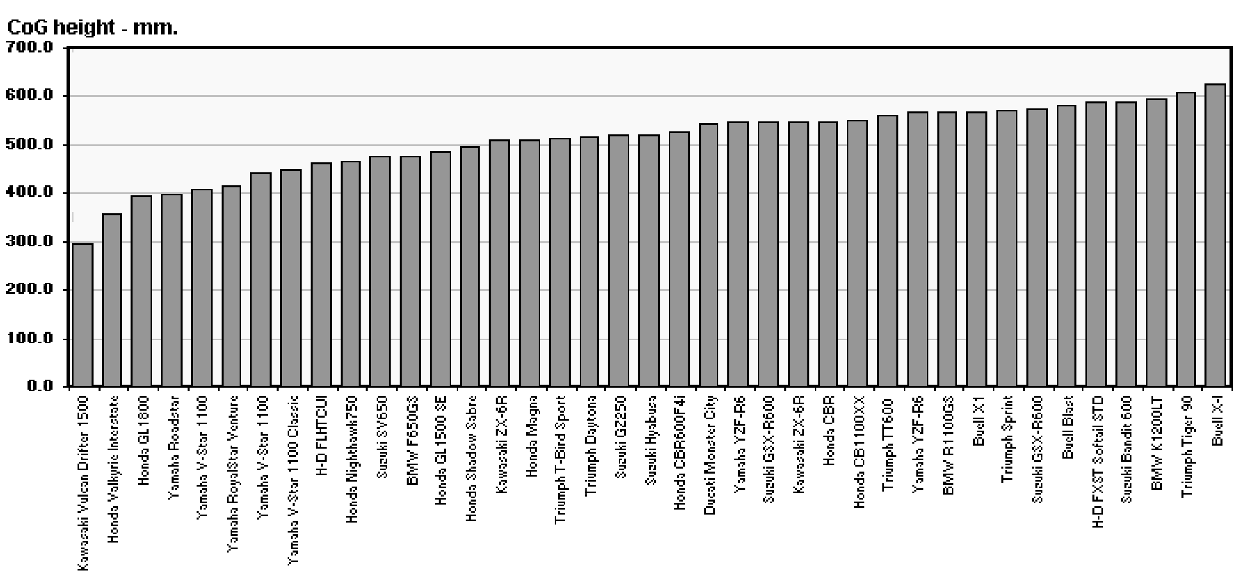

**The rake angles of various motorcycles. The vast majority fit into the range of 23-27 degrees, with modern sports bikes at the lower values. Generally heavy-weight tourers and cruisers go for angles above about 29 degrees.**



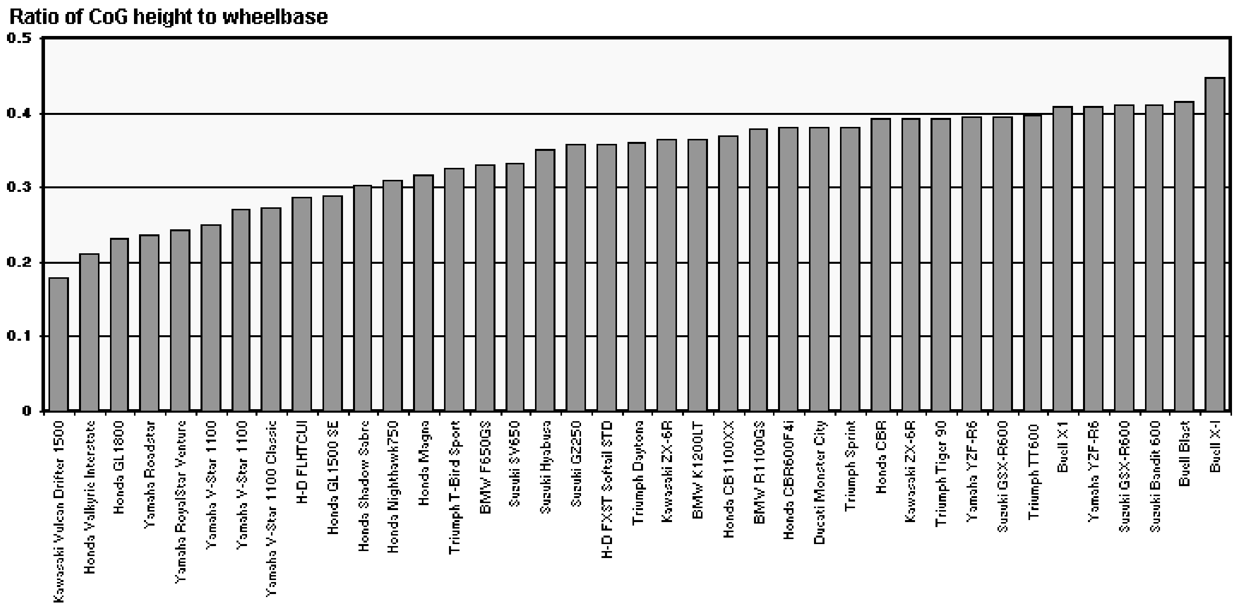

**Modern sports bikes have short wheelbases starting from about 1380 mm. to help turn quickly. Average general machines tend to be around 1450 – 1500 mm., and some large tourers go a bit over 1700 mm.**



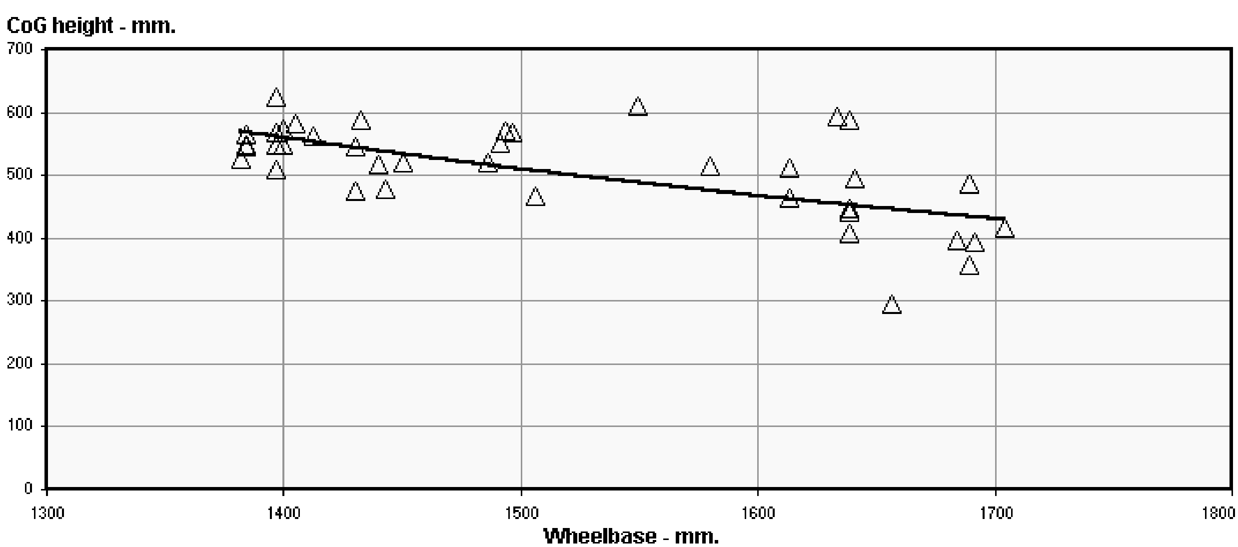

**This shows the percentage of the weight carried by the front wheel. As with all the characteristics show here, this is of the bike alone without the rider aboard. Mostly 40-50% are the most common values. Recently there has been a trend to increase this value, especially on high performance sports machines for stability and steering response. Cruisers have a more rearward weight bias and hence give a lower percentage on the front.**



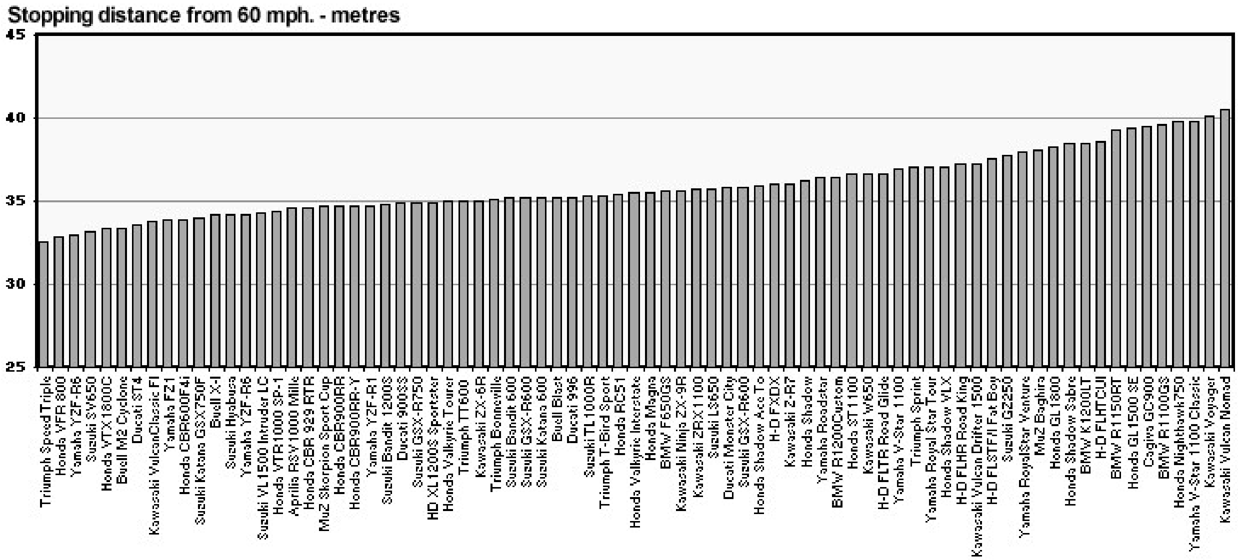

**CoG height for various machines. There is a reasonably even distribution between 400 – 600 mm.**




**This uses the same sample of bikes as in the previous chart, but relates the ratio of the CoG height to the wheelbase. This parameter varies by a two to one ratio over the set of sample bikes. The CoG height will be increased significantly with the rider on-board. This parameter is important because it affects load transfer under braking and acceleration.**

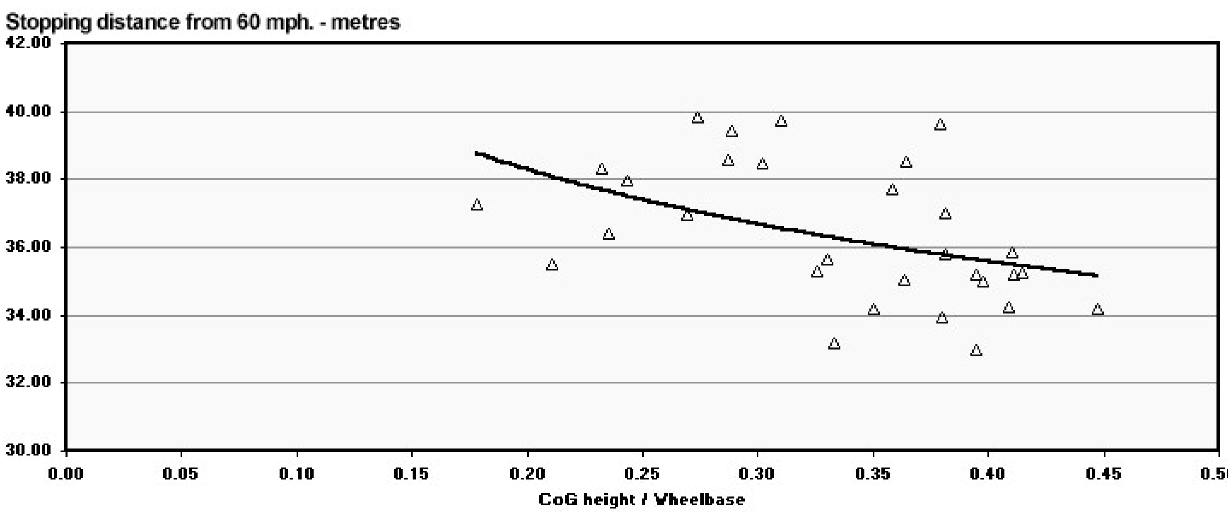



**Cluster chart showing the relationship between CoG height and wheelbase. As we might expect the longer bikes tend to have lower CoG heights, a trend that will continue with a rider seated. Longer bikes tend to have lower seat heights.**

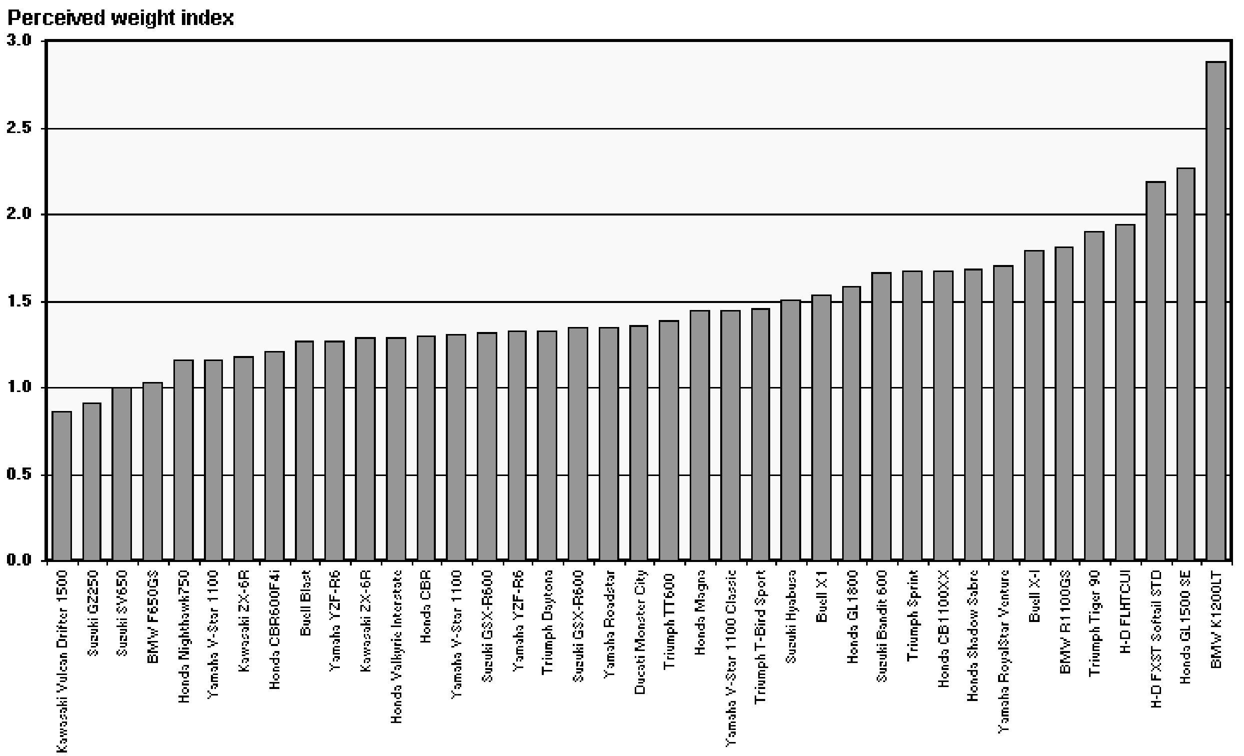



**60 mph to zero (96.6 km/h) stopping distances. These have been measured from a computer controlled radar gun and the braking was started at 65 – 70 mph to avoid reaction times etc. from affecting the results. Aerodynamic drag will have a minimal effect on stopping distance at these speeds.**

(Data from Motorcycle Consumer News)




**Stopping distances against the ratio of CoG height to wheelbase. This chart would be more meaningful if the CoG height included the rider but that information was not available. There is a lot of scatter but the trend seems to be that the bikes with a higher CoG/wheelbase ratio tend to stop in a shorter distance. These bikes are more likely to be sports machines.** (Data from Motorcycle Consumer News)




**The perceived weight of a bike depends very much on the CoG height as well as the true weight itself, and inertia effects vary as the square of the CoG height. The author's "Perceived weight index" combines these factors into a guide number to give a rough idea of how the bike will feel at rest, balanced by the rider. This is only a rough guide because seat height and the angle at which the rider's leg meets with the ground also have an effect.**

**Note particularly, the difference between the BMW K1200LT and the Honda GL1800. The Honda is a little heavier at 400 kgf. in place of the BMW's 390 kgf. but the Honda has a lower CoG and its "index" value is 1.6 against the BMW at 2.9. magazine road tests confirm that the Honda feels lighter than the BMW.**