1

Introduction and History

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1.1 Introduction

To understand vehicle performance and cornering, it is essential to have an in-depth understanding of the basic geometric properties of roads and suspensions, including characteristics such as bump steer, roll steer, the various kinds of roll centre, and the relationships between them.

Of course, the vehicle is mainly a device for moving passengers or other payload from A to B, although in some cases, such as a passenger car tour, a motor race or rally, it is used for the interest of the movement itself. The route depends on the terrain, and is the basic challenge to be overcome. Therefore road characteristics are examined in detail in Chapter 2. This includes the road undulations giving ride quality problems, and road lateral curvature giving handling requirements. These give rise to the need for suspension, and lead to definite requirements for suspension geometry optimisation.

Chapter 3 analyses the geometry of road profiles, essential to the analysis of ride quality and handling on rough roads. Chapter 4 covers suspension geometry as required for ride analysis. Chapter 6 deals with steering geometry. Chapters 6–9 study the geometry of suspensions as required for handling analysis, including bump steer, roll steer, camber, roll centres, compliance steer, etc., in general terms.

Subsequent chapters deal with the properties of the main particular types of suspension, using the methods introduced in the earlier chapters. Then the computational methods required for solution of suspension geometry problems are studied, including two- and three-dimensional coordinate geometry, and numerical iteration.

This chapter gives an overview of suspensions in qualitative terms, with illustrations to show the main types. It is possible to show only a sample of the innumerable designs that have been used.

1.2 Early Steering History

The first common wheeled vehicles were probably single-axle hand carts with the wheels rotating independently on the axle, this being the simplest possible method, allowing variations of direction without any steering mechanism. This is also the basis of the lightweight horse-drawn chariot, already important many thousands of years ago for its military applications. Sporting use also goes back to antiquity, as illustrated in films such as *Ben Hur* with the famous chariot race. Suspension, such as it was, must have been important for use on rough ground, for some degree of comfort, and also to minimise the stress of the structure, and was based on general compliance rather than the inclusion of special spring members. The axle can be made long and allowed to bend vertically and longitudinally to ride the bumps. Another important factor in riding over rough roads is to use large wheels.

Suspension Geometry and Computation J. C. Dixon
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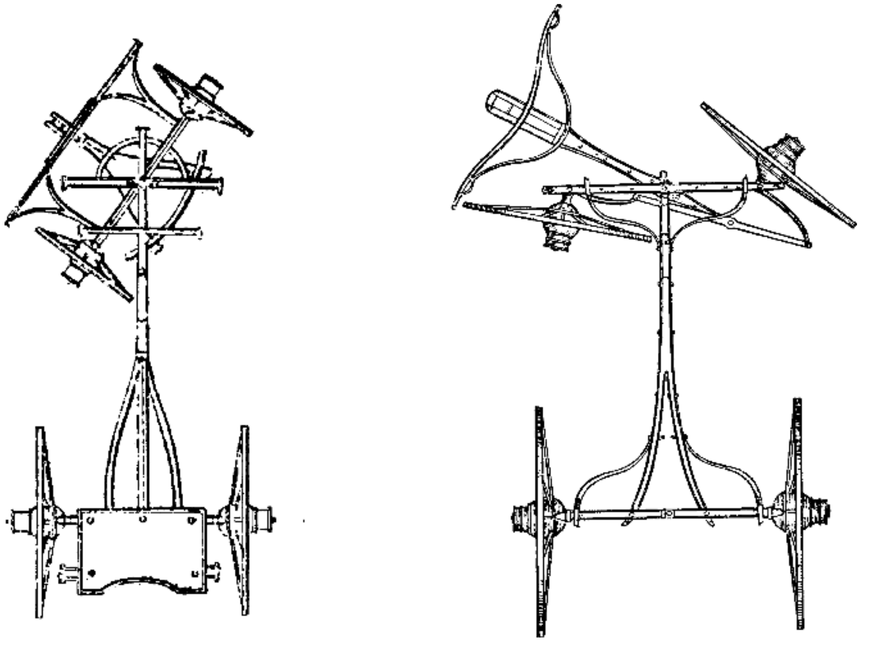

Figure 1.2.1 Steering: (a) basic cart steering by rotating the whole axle; (b) Langensperger's independent steering of 1816.

For more mundane transport of goods, a heavier low-speed two-axle cart was desirable, and this requires some form of steering mechanism. Initially this was achieved by the simple means of allowing the entire front axle to rotate, as shown in Figure 1.2.1(a).

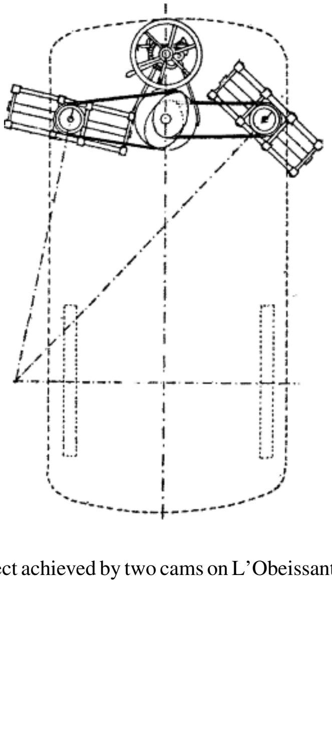

Figure 1.2.2 Ackermann steering effect achieved by two cams on L'Obeissante, designed by Amedée Bollée in 1873.

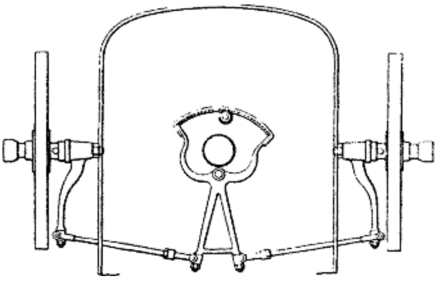

Figure 1.2.3 Ackermann steering effect achieved with parallel steering arms, by using angled drive points at the inner end of the track rods: 'La Mancelle', 1878.

Steering by the movement of the whole axle gives good geometric positioning, with easy low-speed manoeuvring, but the movement of the axle takes up useful space. To overcome this, the next stage was to steer the wheels independently, each turning about an axis close to the wheel. The first steps in this direction were taken by Erasmus Darwin (1731–1802), who had built a carriage for his doctor's practice, allowing larger-diameter wheels of great help on the rough roads. However, if the two wheels are steered through the same angle then they must slip sideways somewhat during cornering, which greatly increases the resistance to motion in tight turns. This is very obvious when a parallel-steered cart is being moved by hand. To solve this, the two wheels must be steered through different angles, as in Figure 1.2.1(b). The origin of this notion may be due to Erasmus Darwin himself in 1758, or to Richard Edgeworth, who produced the earliest known drawing of such a system. Later, in 1816 Langensperger obtained a German patent for such a concept, and in 1817 Rudolf Ackermann, acting as Langensperger's agent, obtained a British patent. The name Ackermann has since then been firmly attached to this steering design. The first application of this steering to a motor vehicle, rather than hand or horse-drawn carts, was by Edward Butler. The simplest way to achieve the desired geometry is to angle the steering arms inwards in the straight-ahead position, and to link them by a tie rod (also known as a track rod), as was done by Langensperger. However, there are certainly other methods, as demonstrated by French engineer Amedée Bollée in 1873, Figure 1.2.2, possibly allowing a greater range of action, that is, a smaller minimum turning circle.

The 'La Mancelle' vehicle of 1878 (the name refers to a person or thing from Le Mans) achieved the required results with parallel steering arms and a central triangular member, Figure 1.2.3. In 1893 Benz obtained a German patent for the same system, Figure 1.2.4. This shows tiller control of the steering, the common method of the time. In 1897 Benz introduced the steering wheel, a much superior system to the tiller, for cars. This was rapidly adopted by all manufacturers. For comparison, it is interesting to note that dinghies use tillers, where it is suitable, being convenient and economic, but ships use a large wheel, and aircraft use a joystick for pitch and roll, although sometimes they have a partial wheel on top of a joystick with only fore–aft stick movement.

1.3 Leaf-Spring Axles

Early stage coaches required suspension of some kind. With the limited technology of the period, simple wrought-iron beam springs were the practical method, and these were made in several layers to obtain the required combination of compliance with strength. These multiple-leaf springs became known simply as leaf springs. To increase the compliance, a pair of leaf springs were mounted back-to-back. They were curved, and so then known, imprecisely, as elliptical springs, or elliptics for short. Single ones were called

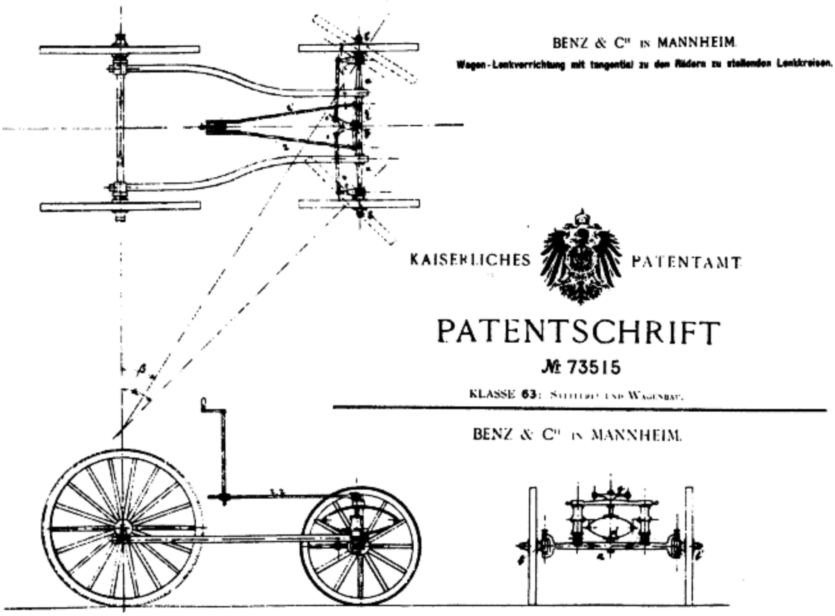

Figure 1.2.4 German patent of 1893 by Benz for a mechanism to achieve the Ackermann steering effect, the same mechanism as La Mancelle.

semi-elliptics. In the very earliest days of motoring, these were carried over from the stage coaches as the one practical form of suspension, as may be seen in Figure 1.3.1.

The leaf spring was developed in numerous variations over the next 50 years, for example as in Figure 1.3.2. With improving quality of steels in the early twentieth century, despite the increasing average

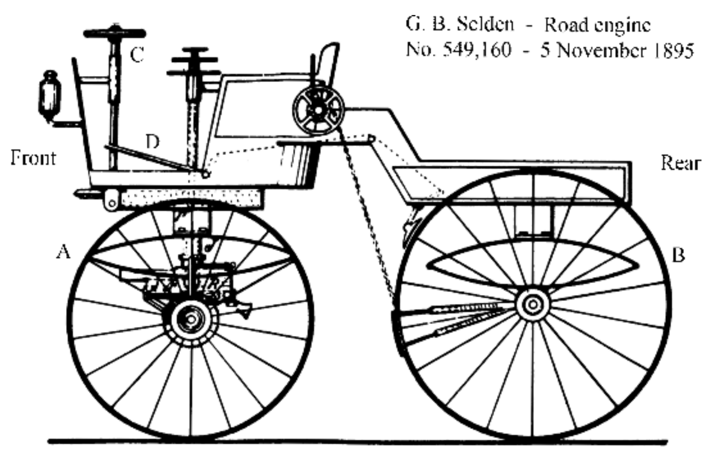

Figure 1.3.1 Selden's 1895 patent showing the use of fully-elliptic leaf springs at the front A and rear B. The steering wheel is C and the foot brake D.

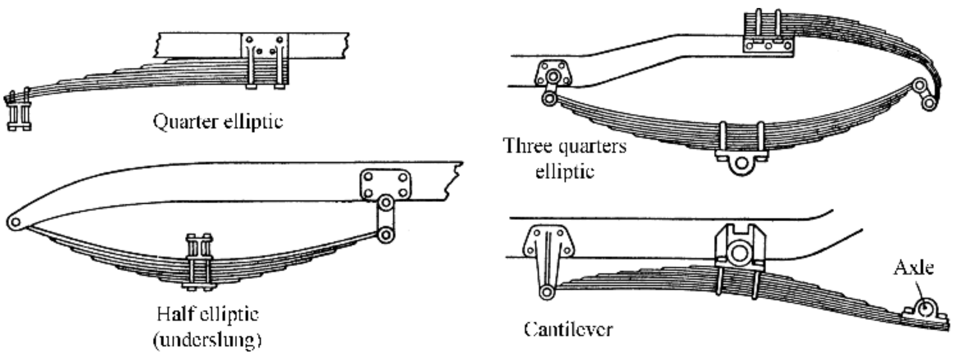

Figure 1.3.2 Some examples of the variation of leaf springs in the early days. As is apparent here, the adjective 'elliptical' is used only loosely.

weight of motor cars, the simpler semi-elliptic leaf springs became sufficient, and became widely standardised in principle, although with many detailed variations, not least in the mounting systems, position of the shackle, which is necessary to permit length variation, and so on. The complete vehicle of Figure 1.3.3 shows representative applications at the front and rear, the front having a single compression shackle, the rear two tension shackles. A very real advantage of the leaf spring in the early days was that the spring provides lateral and longitudinal location of the axle in addition to the springing compliance action. However, as engine power and speeds increased, the poor location geometry of the leaf spring became an increasing problem, particularly at the front, where the steering system caused many problems in bump and roll. To minimise these difficulties, the suspension was made stiff, which caused poor ride quality.

Figures 1.3.4 and 1.3.5 show representative examples of the application of the leaf spring at the rear of normal configuration motor cars of the 1950s and 1960s, using a single compression shackle.

Greatly improved production machinery by the 1930s made possible the mass production of good quality coil springs, which progressively replaced the leaf spring for passenger cars. However, leaf-spring use on passenger cars continued through into the 1970s, and even then it functioned competitively, at the rear at least, Figure 1.3.6. The leaf spring is still widely used for heavily loaded axles on trucks and military vehicles, and has some advantages for use in remote areas where only basic maintenance is possible, so leaf-spring geometry problems are still of real practical interest.

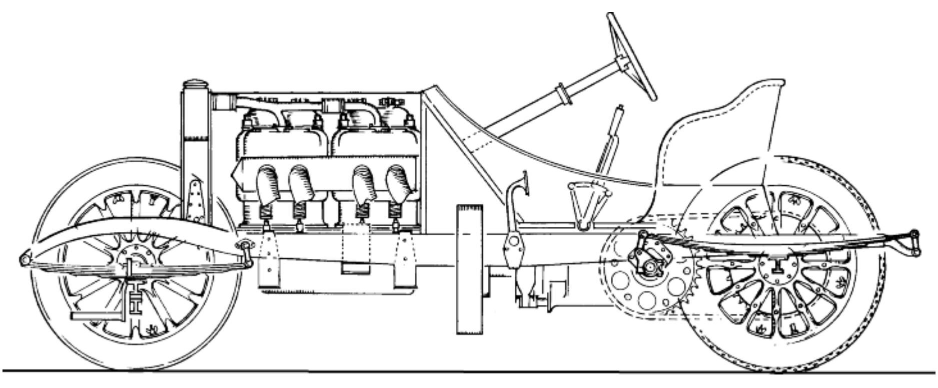

Figure 1.3.3 Grand Prix car of 1908, with application of semi-elliptic leaf springs at the front and rear (Mercedes-Benz).

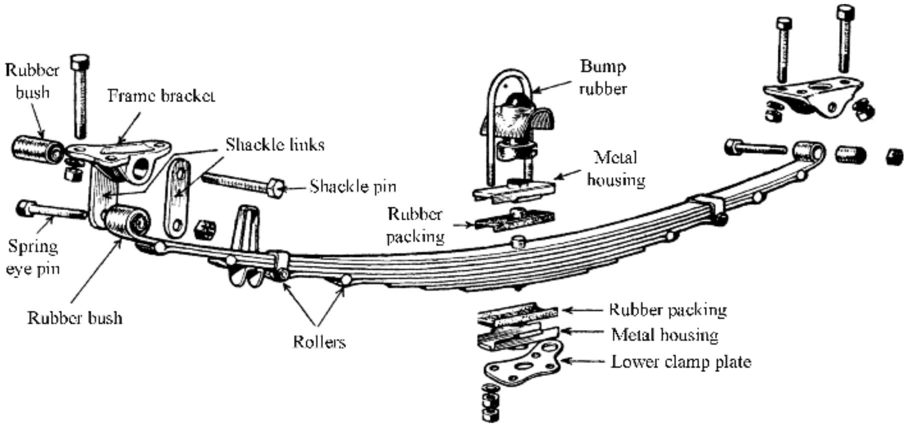

Figure 1.3.4 A representative rear leaf-spring assembly (Vauxhall).

At the front, the leaf spring was much less satisfactory, because of the steering geometry difficulties (bump steer, roll steer, brake wind-up steering effects, and shimmy vibration problems). Figure 1.3.7 shows a representative layout of the typical passenger car rigid-front-axle system up to about 1933. In bump, the axle arc of movement is centred at the front of the spring, but the steering arm arc is centred at

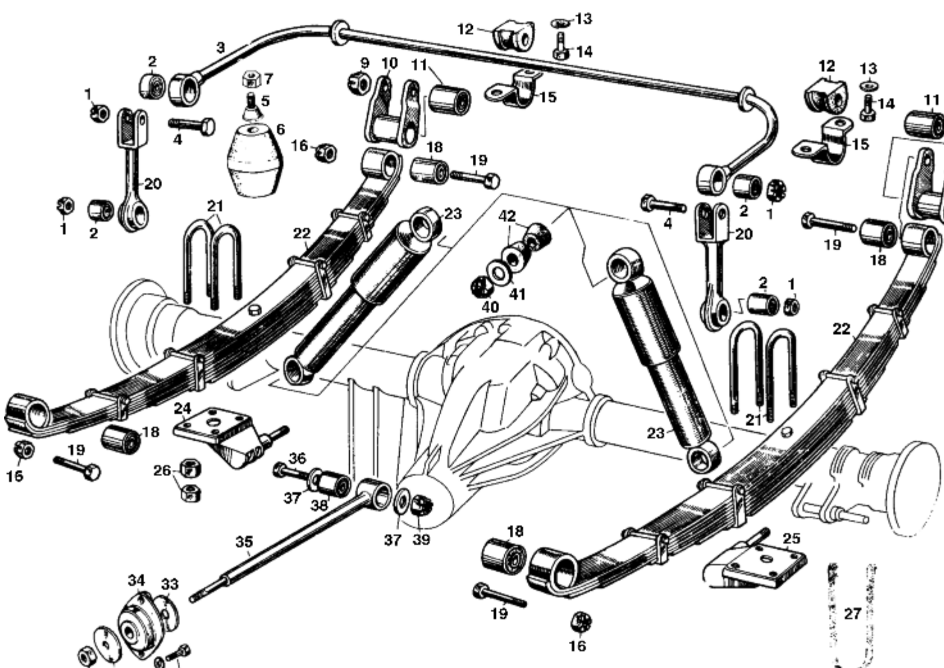

Figure 1.3.5 A 1964 live rigid rear axle with leaf springs, anti-roll bar and telescopic dampers. The axle clamps on top of the springs (Maserati).

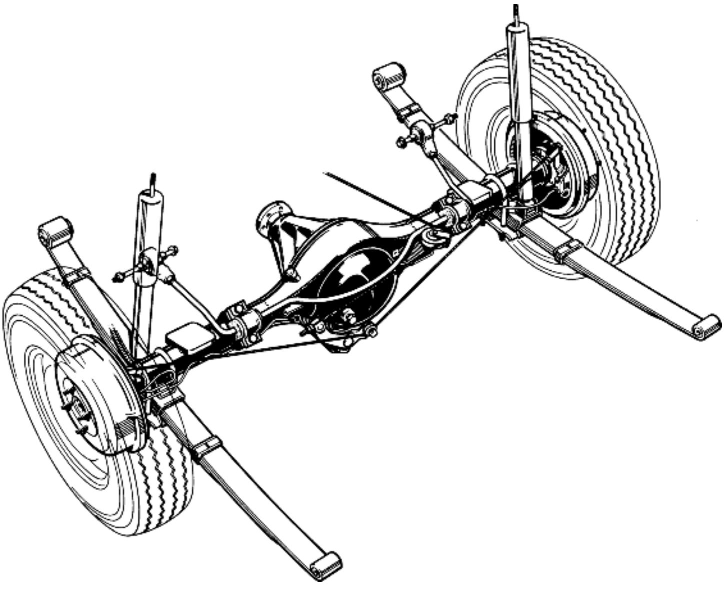

Figure 1.3.6 Amongst the last of the passenger car leaf-spring rear axles used by a major manufacturer was that of the Ford Capri. Road testers at the time found this system in no way inferior to more modern designs.

the steering box. These conflicting arcs give a large and problematic bump steer effect. The large bump steer angle change also contributed to the shimmy problems by causing gyroscopic precession moments on the wheels. Figure 1.3.8 shows an improved system with a transverse connection.

Truck and van steering with a leaf spring generally has the steering box ahead of the axle, to give the maximum payload space, as seen in Figure 1.3.9. In bump, the arc of motion of the steering arm and the axle on the spring are in much better agreement than with the rear box arrangement of Figure 1.3.7, so bump steer is reduced. Also, the springs are likely to be much stiffer, with reduced range of suspension movement, generally reducing the geometric problems.

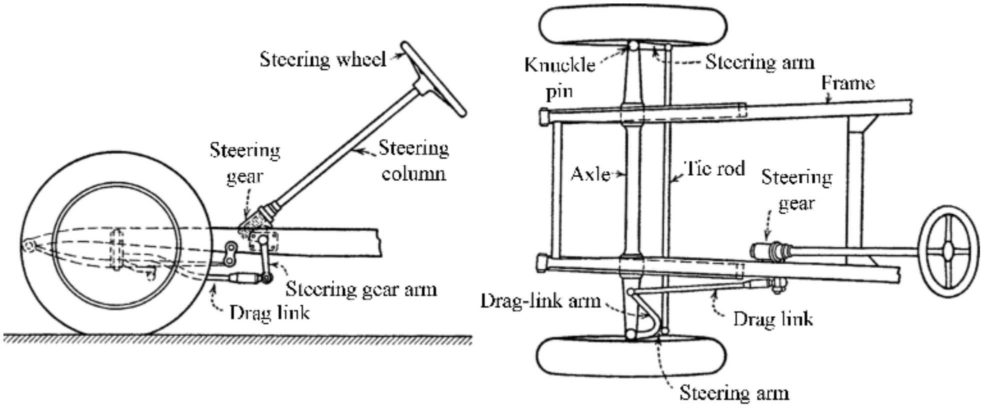

Figure 1.3.7 Classical application of the rigid axle at the front of a passenger car, the normal design up to 1933. Steering geometry was a major problem because of the variability of rigid axle movements.

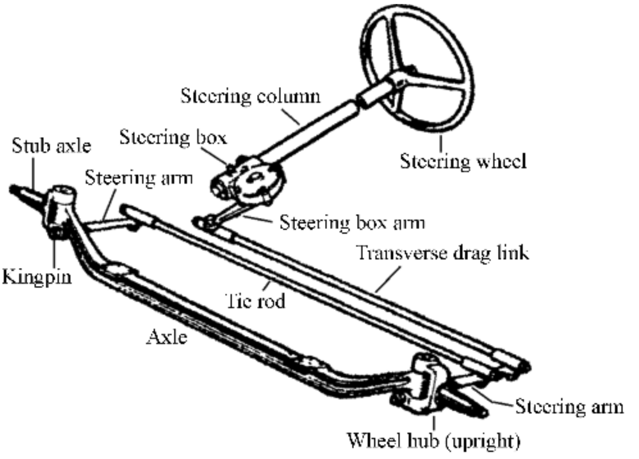

Figure 1.3.8 Alternative application of the rigid axle at the front of a passenger car, with a transverse steering link between the steering box on the sprung mass and the axle, reducing bump steer problems.

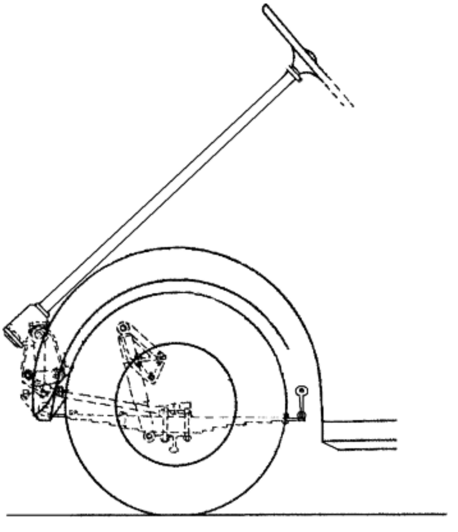

Figure 1.3.9 Van or truck steering typically has a much steeper steering column with a steering box forward of the axle, as here. The steering geometry problems are different in detail, but may be less overall because a stiffer suspension is more acceptable.

1.4 Transverse Leaf Springs

Leaf springs were not used only in longitudinal alignment. There have been many applications with transverse leaf springs. In some cases, these were axles or wheel uprights located by separate links, to overcome the geometry problems, with the leaf spring providing only limited location service, or only the springing action. Some transverse leaf examples are given in Figures 1.4.1–1.4.4

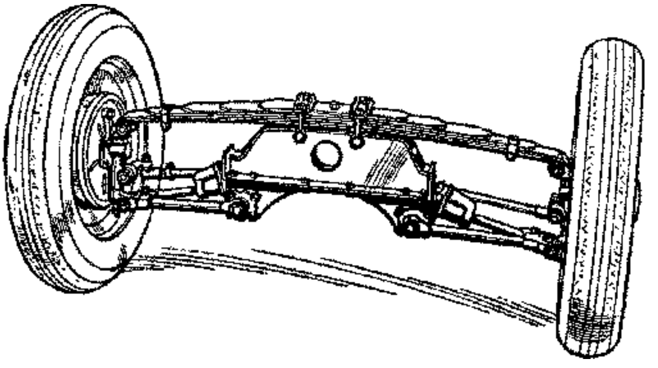

Figure 1.4.1 A transverse leaf spring at the top also provides upper lateral and longitudinal location on this front axle, with a lower wishbone (early BMW).

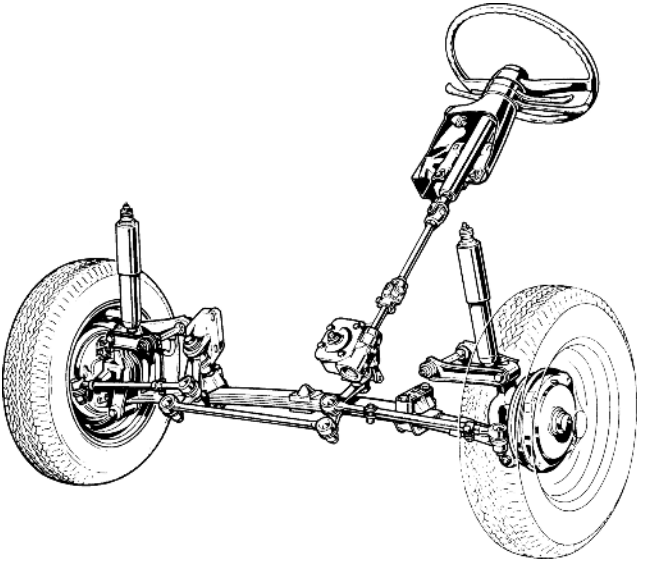

Figure 1.4.2 This more modern small car front suspension has a transverse leaf spring at the bottom with an upper wishbone (Fiat).

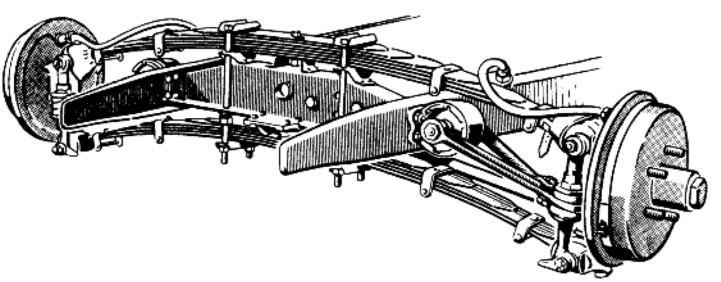

Figure 1.4.3 Two transverse leaf springs providing complete hub location acting as equal-length wishbones without any additional links (1931, Mercedes Benz).

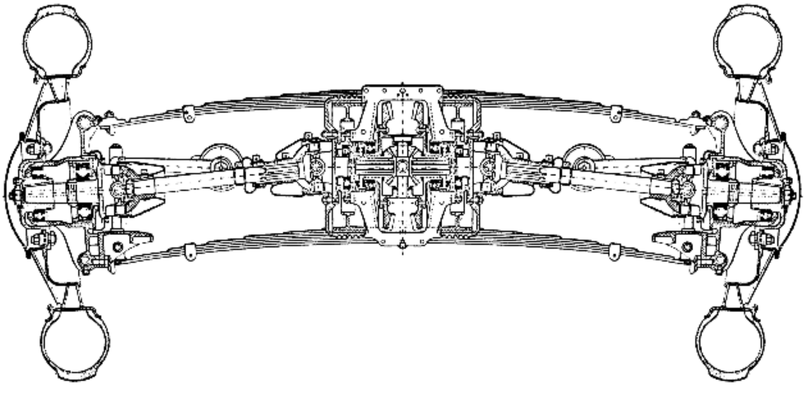

Figure 1.4.4 The Cottin-Desgouttes of 1927 used four leaf springs on the driven rear axle, in a square configuration, also featuring inboard brakes. The parallel pair of springs at the top or bottom acted as equal-length wishbones, with length equal to three-quarters of the cantilever length. The driveshaft length can be chosen to match this length, to minimise the plunge requirement at the splines. The wheels have 2.5° static camber but do not rotate in bump (zero bump camber).

1.5 Early Independent Fronts

Through the 1920s, the rigid axle at the front was increasingly a problem. Despite considerable thought and experimentation by suspension design engineers, no way had been found to make a steering system that worked accurately. In other words, there were major problems with bump steer, roll steer and spring wind-up, particularly during braking. Any one of these problems might be solved, but not all at once. With increasing engine power and vehicle speeds, this was becoming increasingly dangerous, and hard front springs were required to ameliorate the problem, limiting the axle movement, but this caused very poor ride comfort. The answer was to use independent front suspension, for which a consistently accurate steering system could be made, allowing much softer springs and greater comfort. Early independent suspension designs were produced by André Dubonnet in France in the late 1920s, and a little later for Rolls-Royce by Donald Bastow and Maurice Olley in England. These successful applications of independent suspension became known in the USA, and General Motors president Alfred P. Sloan took action, as he describes in his autobiography (Sloan, 1963).

Around 1930, Sloan considered the problem of ride quality as one of the most pressing and most complex in automotive engineering, and the problem was getting worse as car speeds increased. The early solid rubber tyres had been replaced by vented thick rubber, and then by inflated tyres. In the 1920s, tyres became even softer, which introduced increased problems of handling stability and axle vibrations. On a trip to Europe, Sloan met French engineer André Dubonnet who had patented a successful independent suspension, and had him visit the US to make contact with GM engineers. Also, by 1933 Rolls-Royce already had an independent front suspension, which was on cars imported to the USA. Maurice Olley, who had previously worked for Rolls-Royce, was employed by GM, and worked on the introduction of independent suspensions there. In Sloan's autobiography, a letter from Olley describes an early ride meter, which was simply an open-topped container of water, which was weighed after a measured mile at various speeds. Rolls-Royce had been looking carefully at ride dynamics, including measuring body inertia, trying to get a sound scientific understanding of the problem, and Olley introduced this approach at GM. In 1932 they built the K-squared rig (i.e. radius of gyration squared), a test car with various heavy added masses right at the front and rear to alter the pitch inertia in a controlled way. This brought home the realisation that a much superior ride could be achieved by the use of softer front springs, but soft springs caused shimmy problems and bad handling. Two experimental Cadillac cars were built, one using Dubonnet's type of suspension, the other with a double-wishbone (double A-arm) suspension of GM's design. The engineers were pleased with the ride and handling, but shimmy steering vibration was a persistent problem requiring intensive development work. In March 1933 these two experimental cars were demonstrated to GM's top management, along with an automatic transmission. Within a couple of miles, the 'flat ride' suspension was evidently well received.

March 1933 was during the Great Depression, and financial constraints on car manufacturing and retail prices were pressing, but the independent front suspension designs were enthusiastically accepted, and shown to the public in 1934. In 1935 Chevrolet and Pontiac had cars available with Dubonnet suspension, whilst Cadillac, Buick and Oldsmobile offered double-wishbone front suspension, and the rigid front axle was effectively history, for passenger cars at least. A serious concern for production was the ability of the machine tool industry to produce enough suitable centreless grinders to make all the coil springs that would be required. With some practical experience, it became apparent that with development the wishbone suspension was easier and cheaper to manufacture, and also more reliable, and was universally adopted.

Figure 1.5.1 shows the 1934 Cadillac independent suspension system, with double wishbones on each side, in which it may be seen that the basic steering concept is recognisably related to the ones described earlier. As covered in detail in Chapter 6, the track-rod length and angle can be adjusted to give good steering characteristics, controlling bump steer and roll steer. The dampers were the lever-operated double-piston type, incorporated into the upper wishbone arms. Such a system would still be usable today.

Figure 1.5.2 shows the Dubonnet type suspension, used by several other manufacturers, which was unusually compact. The wheels are on leading or trailing arms, with the spring contained in a tube on the

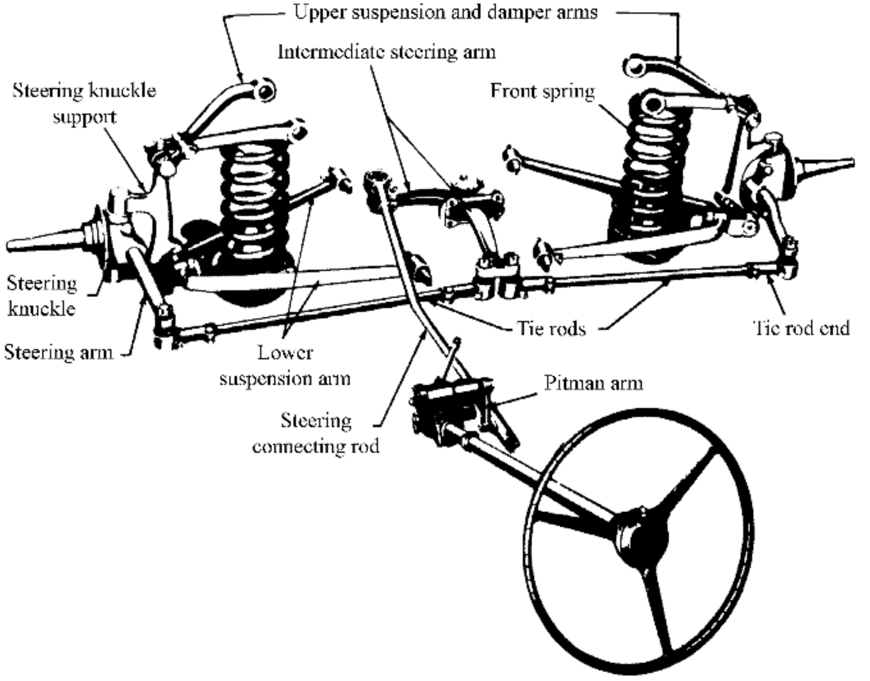

Figure 1.5.1 The new Cadillac steering and independent suspension of 1934.

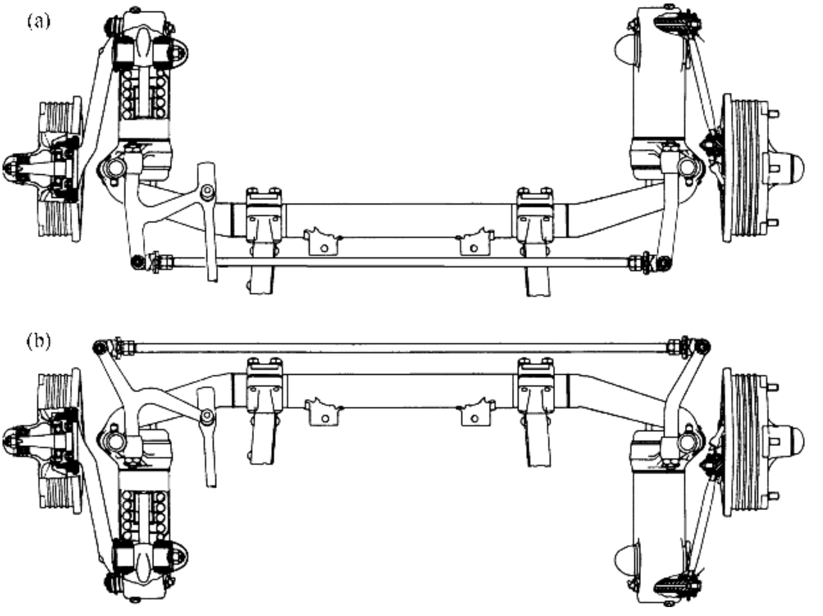

Figure 1.5.2 The Dubonnet type suspension in plan view, front at the top: (a) with trailing links; (b) with leading links (1938 Opel).

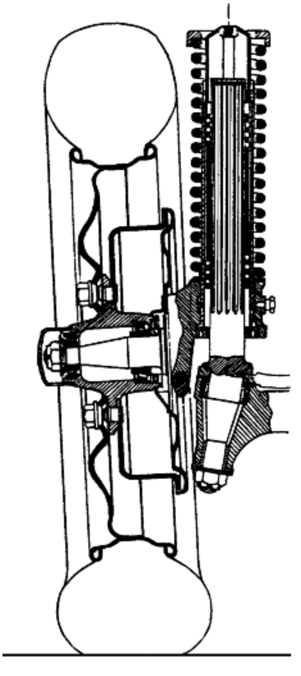

Figure 1.5.3 Broulhiet ball-spline sliding pillar independent suspension.

steerable part of the system. The type shown has a single tie rod with a steering box, as was usual then, but the system is equally adaptable to a steering rack. The Ackermann effect is achieved here by angling the steering arms backwards and inwards in Figure 1.5.2(a) with the trailing arms, or forwards and outwards in Figure 1.5.2(b) with the leading arms. The steering action is entirely on the sprung mass, so there is no question of bump or roll steer due to the steering, and there are no related issues over the length of the steering members. Bump steer effects depend only on the angle of the pivot axis of the arms, in this case simply transverse, with zero bump steer and zero bump camber. Other versions had this axis at various angles. The leading link type at the front of a vehicle gives considerable anti-dive in braking, but is harsh over sharp bumps. The trailing-arm version is better over sharp bumps but has strong pro-dive in braking.

Another early form of independent suspension was that due to Brouhliet in France, who used sliding splines, with ball bearings for low friction, for the suspension action, Figure 1.5.3, again allowing the steering to be entirely sprung, eliminating the steering problems of the rigid front axle.

1.6 Independent Front Suspension

Some independent suspensions have already been shown. Section 1.4 illustrates some with transverse leaf springs. Section 1.5 shows two from the mid 1930s – the Dubonnet, now effectively defunct, and the double wishbone which was the *de facto* standard front suspension for many years, although now that could be perhaps be said instead of the strut and wishbone. Subsequent to the leaf spring, torsion bar suspensions were quite common. However, the modern independent suspension is almost invariably based on the coil spring, with location by two wishbones (A-arms) or by a strut with one wishbone at the bottom.

Figure 1.6.1 shows a sliding pillar suspension, not representative of common modern practice, but this was an early success of some historical interest. With the spring and damper unit enclosed, it was very

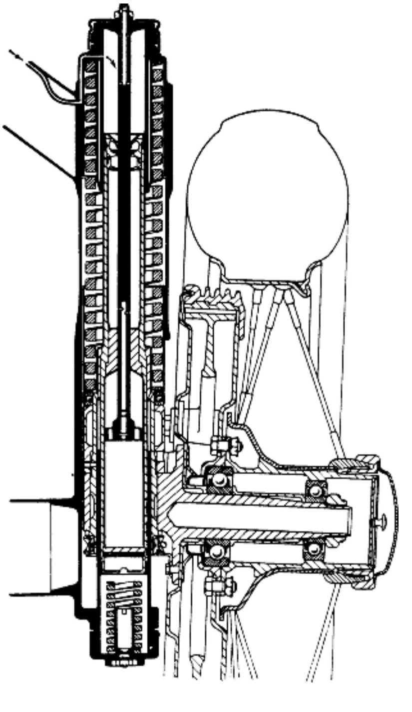

Figure 1.6.1 Sliding pillar front suspension (Lancia Lambda).

reliable, particularly compared with other designs of the early days. When introduced, this was regarded by the manufacturer as the best suspension design regardless of cost.

Figure 1.6.2 shows various versions and views of the twin parallel-trailing-arm suspension, which most often used torsion-bar springing in the cross tubes. On the Gordon-Armstrong this could be supplemented with, or replaced by, coils springs used in compression with draw bars, with double action on the spring. Again, this was a very compact system. The steering can be laid out to give zero bump steer, as in the Aston Martin version of Figure 1.6.2(d), or even with asymmetrical steering as in Figure 1.6.2(a).

A transverse single swing-arm type of front suspension can be used, as in Figure 1.6.3, but with lower body pivot points than the usual driven rear swing axle, giving a lower roll centre. There is a large bump camber effect with this design, such as to effectively eliminate roll camber completely. Steering is by

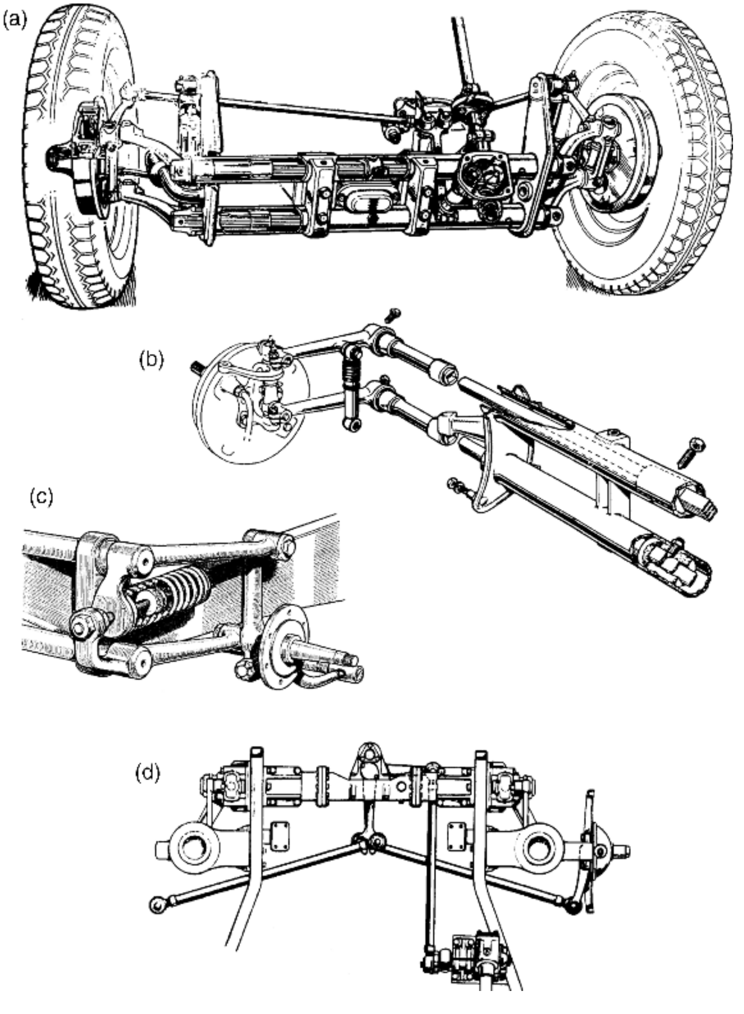

Figure 1.6.2 Parallel-trailing-arm front suspension: (a) general front view (VW); (b) rear three-quarter view of torsion bar type construction; (c) parallel-trailing-arm suspension with laid-down compression coil spring and tension bar (Gordon-Armstrong); (d) steering layout for parallel-trailing-arm system, plan view (early Aston Martin).

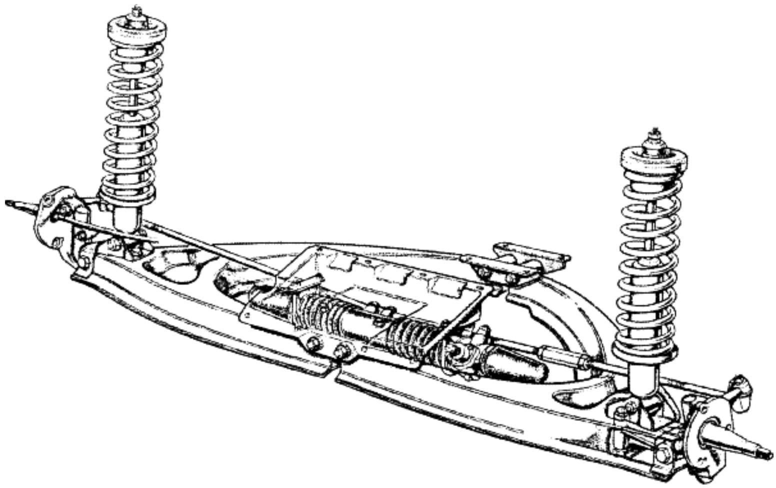

Figure 1.6.3 Single transverse swing-arm independent front suspension with rack-and-pinion steering (1963 Hillman).

rack-and-pinion, with appropriately long track rods, giving no bump steer, this requiring the pick-up points on the rack to be aligned with the arm pivot axes in the straight-ahead position. Unusually, the track-rod connections are on the rear of the rack, which affects only the plan view angle of the track rods, and hence the Ackermann factor.

The Glas Isar had double wishbones, as seen in Figure 1.6.4, but the upper wishbone had its pivot axis transverse, so in front view the geometry was similar to a strut-and-wishbone suspension. The steering system is high up, and asymmetrical. Analysis of bump steer requires a full three-dimensional solution, but with the asymmetrical steering on this design there could be problems unless the track-rod connections to the steering box arm are aligned with the upper wishbone axes.

Some early double-wishbone systems were very short, particularly on racing cars, as in Figure 1.6.5. With the relatively long track rod shown there would have been significant bump steer, which could have been only marginally acceptable by virtue of the stiff suspension and small suspension deflections. This makes an interesting contrast with the very long wishbones on modern racing cars, although in that case the deflections are still small and it is done for different reasons.

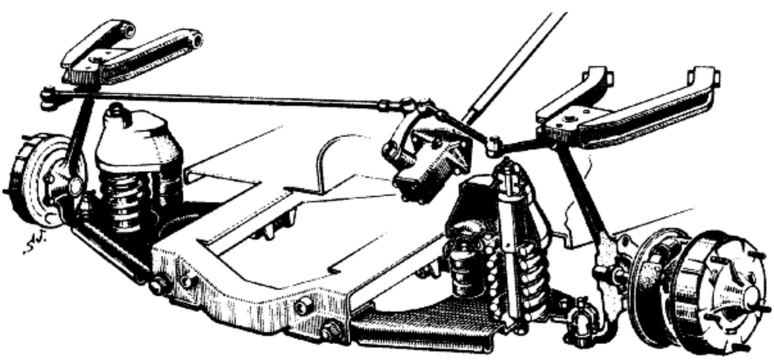

Figure 1.6.4 A double-wishbone suspension in which the wishbone axes are crossed (Glas Isar).

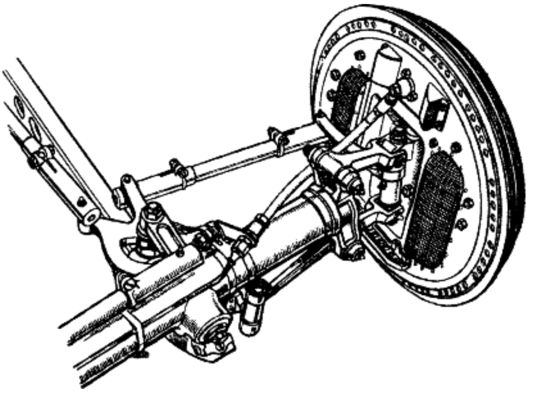

Figure 1.6.5 An early double-wishbone system with very short arms (Mercedes Benz). The suspension spring is inside the transverse horizontal tube.

Figure 1.6.6 shows an engineering section of a fairly representative double-wishbone system, with unequal-length arms, nominally parallel in the static position. As is usual with double wishbones, the spring acts on the lower arm at a motion ratio of about 0.5. The steering axis is defined by ball joints rather than by the old kingpin system, with wide spacing giving lower joint loads.

In Figure 1.6.7, a more recent double-wishbone system, the upper wishbone is partially defined by the rear-mounted anti-roll bar. The steering arms are inclined to the rear as if to give an Ackermann effect, but the track rods are also angled (see Chapter 5). The offset connections on the steering box and idler arm give some Ackermann effect.

The modern double-wishbone system of Figure 1.6.8 is different in that the spring acts on the upper wishbone, with vertical forces transmitted into the body at the top of the wheel arch in the same way as for a strut. However, despite the spring position this is certainly not a strut suspension, which is defined by a

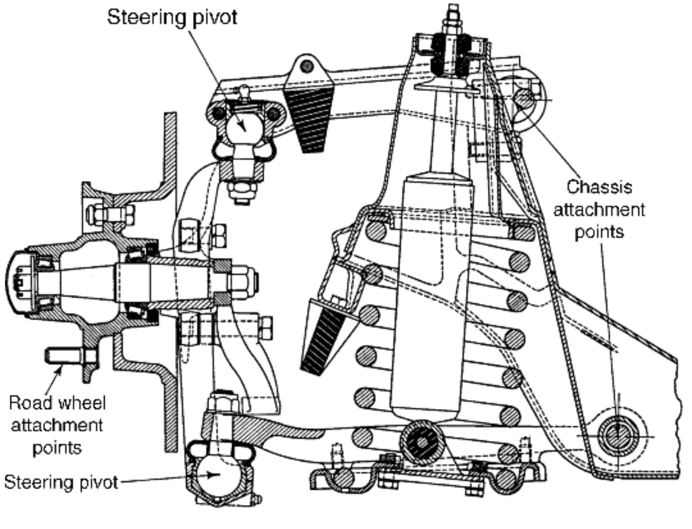

Figure 1.6.6 Traditional configuration of passenger car double-wishbone suspension, with the spring and damper acting on the lower wishbone (Jaguar).

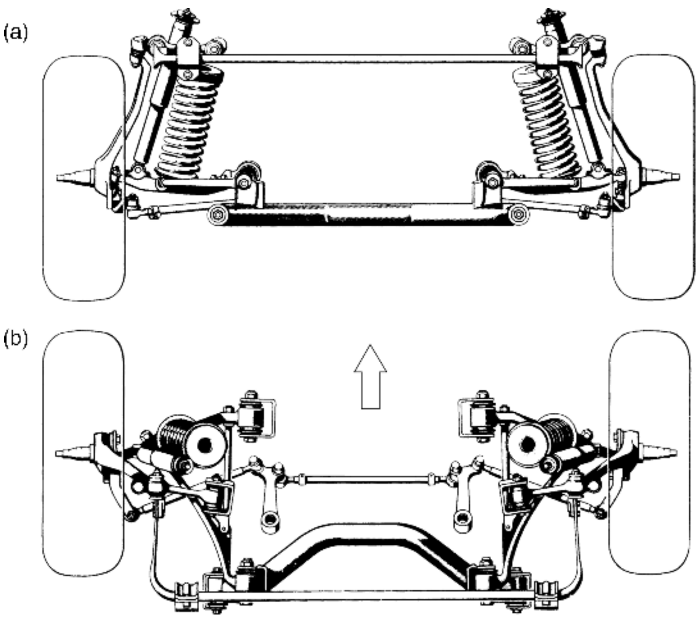

Figure 1.6.7 A passenger car double-wishbone system with a wide-base lower wishbone, and the upper wishbone partially defined by the anti-roll bar (Mercedes Benz).

rigid camber connection between the wheel upright and the strut. The steering connections are to the ends of the rack, to give the correct track-rod length to control bump steer.

The commercial vehicle front suspension of Figure 1.6.9 is a conventional double-wishbone system with a rear-mounted anti-roll bar, and also illustrates the use of a forward steering box and steep steering

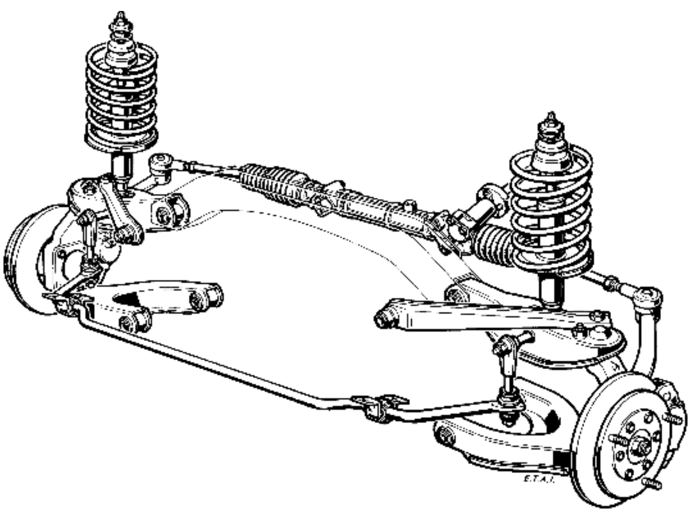

Figure 1.6.8 A representative modern double-wishbone suspension, with spring and damper acting on the upper arm (Renault). This is not a strut suspension.

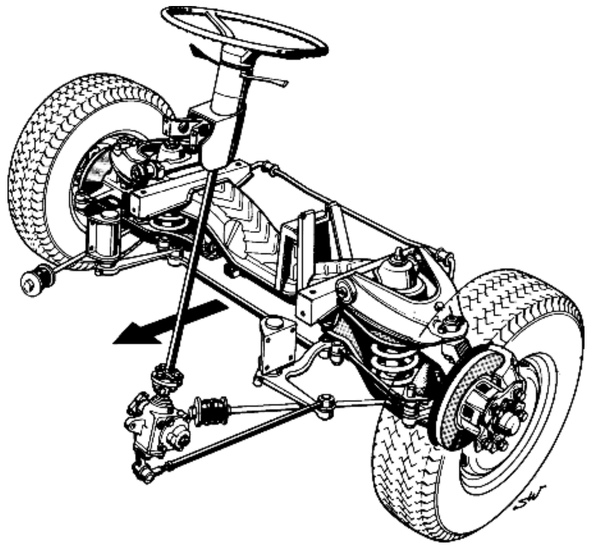

Figure 1.6.9 A double wishbone system from a light commercial vehicle, also illustrating a forward steering system (VW).

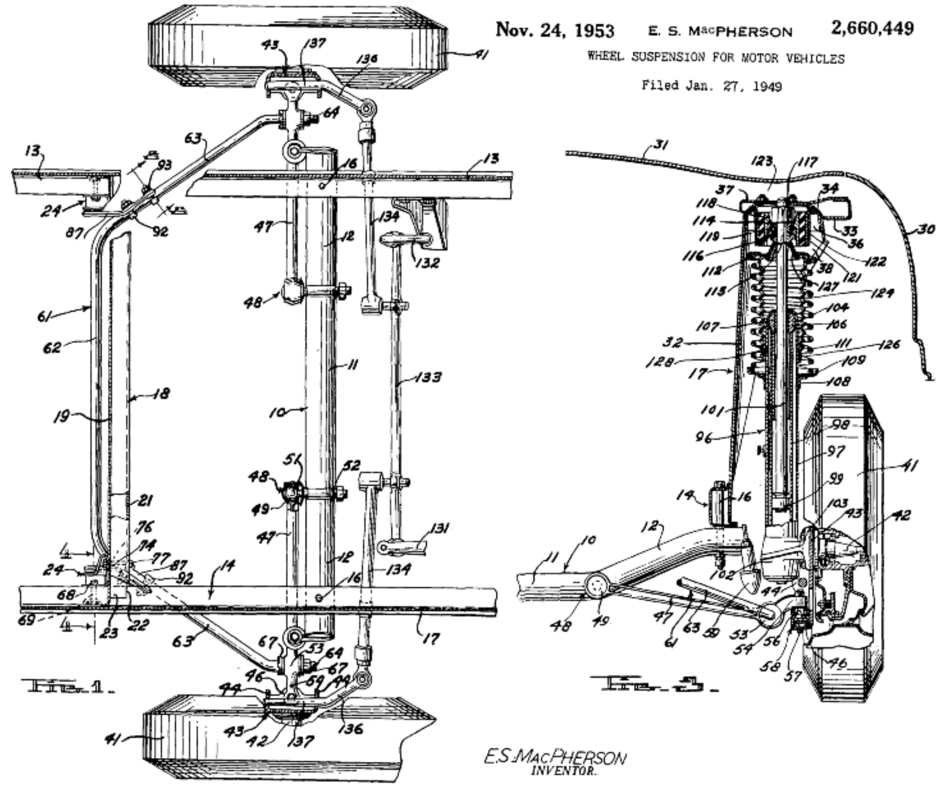

Figure 1.6.10 MacPherson's 1953 US patent for strut suspension (front shown).

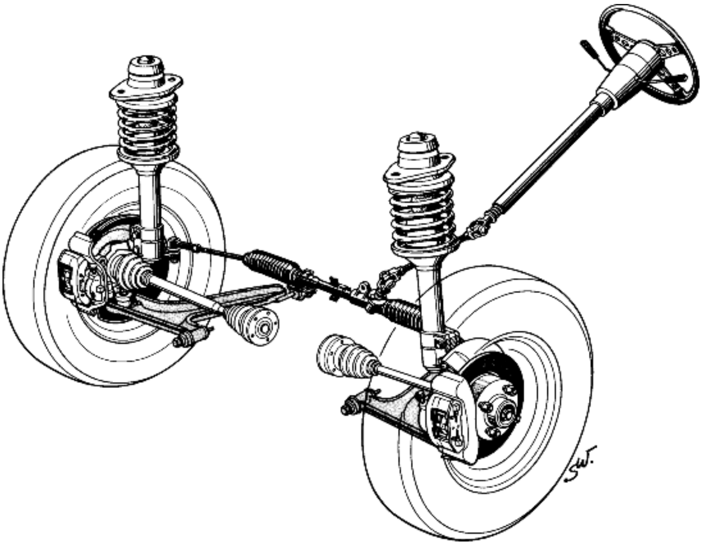

Figure 1.6.11 Passenger car strut suspension with wide-based lower wishbone and low steering (VW).

column on this kind of vehicle. Again, the tie rod and idler arm allow the two track rods to be equal in length and to have correct geometry for the wishbones.

Finally, Figure 1.6.10 shows the MacPherson patent of 1953 for strut suspension, propsed for use at the front and the rear. A strut suspension is one in which the wheel upright (hub) is controlled in camber by a rigid connection to the strut itself. This was popularised for front suspension during the 1950s and 1960s by Ford, with the additional feature that the function of longitudinal location was combined with an anti-roll bar. Strut suspension lacks the adaptability of double-wishbone suspension to desired geometric properties, but can be made acceptable whilst giving other benefits. The load transmission into the body is

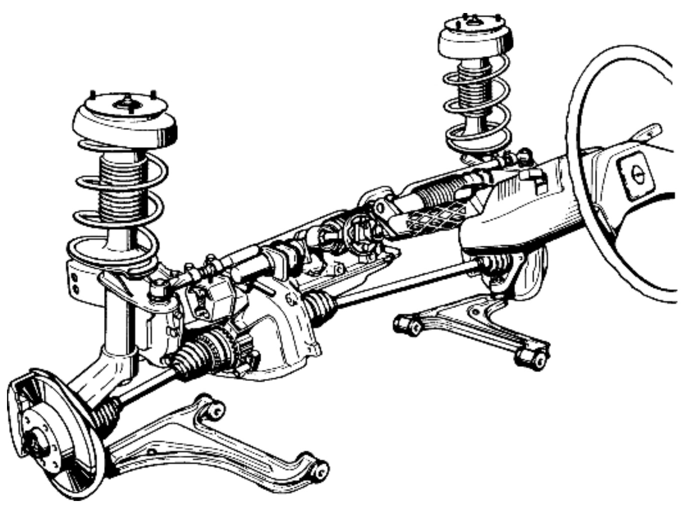

Figure 1.6.12 Passenger car strut suspension with wide-based lower wishbone and high steering (Opel).

widely spread, and pre-assembled units of the suspension can be fitted to the body in an efficient manner on the final assembly line.

Figures 1.6.11 and 1.6.12 give two more modern examples, in fact of strut-and-wishbone suspensions, normally just called strut suspensions, as now so commonly used on small and medium passenger vehicles. The two illustrations show conventional struts. In contrast, a Chapman strut is a strut suspension with a driveshaft, the shaft providing the lateral location of the bottom member, this requiring and allowing no length change of the driveshaft, eliminating a number of problems such as sticking splines under load (see Figure 1.10.7). In Figure 1.6.11 the steering rack is low, close to the level of the wishbone, and the track-rod connections are, correspondingly, at the ends of the rack, to give the correct track-rod length. In Figure 1.6.12 the steering is higher, at the level of the spring seat, so for small bump steer the track rods must be longer, and are connected to the centre of the rack.

1.7 Driven Rigid Axles

The classic driven rigid rear axle, or so-called ‘live axle’, is supported and located by two leaf springs, in which case it is called a ‘Hotchkiss axle’, as shown previously in Figures 1.3.5 and 1.3.6. Perhaps due to the many years of manufacturers’ experience of detailing this design, sometimes it has been implemented with great success. In other cases, there have been problems, such as axle tramp, particularly when high tractive force is used. To locate the axle more precisely, or more firmly, sometimes additional links are used, such as the longitudinal traction bars above the axle in Figure 1.7.1, opposing pitch rotation. These used to be a well-known aftermarket modification for some cars, but were often of no help. In other cases, the leaf springs have been retained as the sole locating members but with the springing action assisted by coils, as in Figure 1.7.2, giving good load spreading into the body.

However, with the readier availability of coil springs, in due course the rear leaf-spring axle finally disappeared from passenger cars, typically being replaced by the common arrangement of Figure 1.7.3, with four locating links, this system being used by several manufacturers. The two lower widely-spaced parallel links usually also carry the springs, as this costs less boot (trunk) space than placing them directly on the axle. Lateral positioning of the axle is mainly by the convergent upper links, although this gives rather a high roll centre. The action of axle movement may not be strictly kinematic, and may depend to some extent on compliance of the large rubber bushes that are used in each end of the links.

The basic geometry of the four-link system is retained in the T-bar system of Figure 1.7.4, with the cross-arm of the T located between longitudinal ribs on the body, allowing pivoting with the tail of the T, connected to the axle, able to move up and down in an arc in side view. This gives somewhat more precise location than the four-link system, and requires less bush compliance for its action, but again the roll centre is high, satisfactory for passenger cars, but usually replaced by a sliding block system for racing versions of these cars.

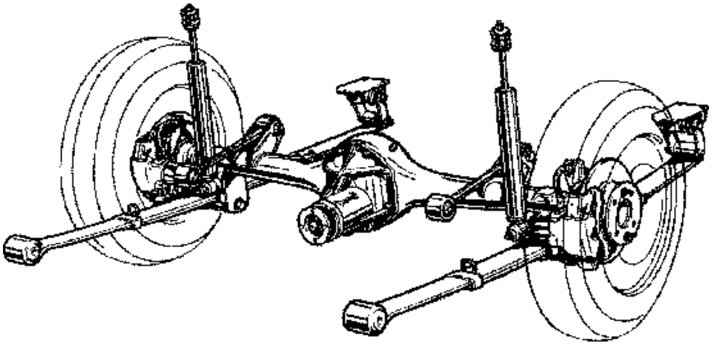

Figure 1.7.1 Leaf-spring axle with the addition of traction bars above the axle (Fiat).

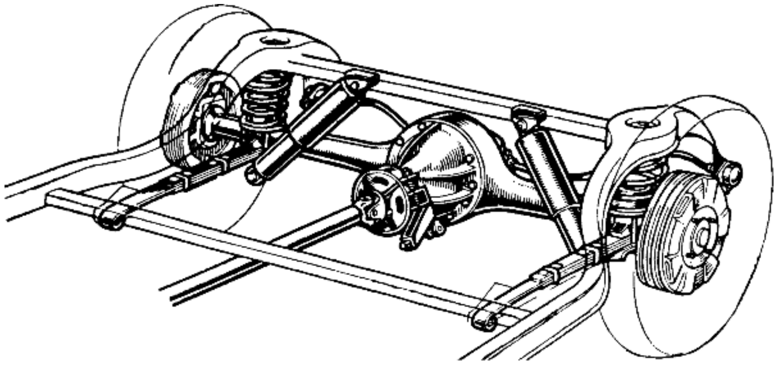

Figure 1.7.2 Leaf-spring axle with additional coil springs (Fiat).

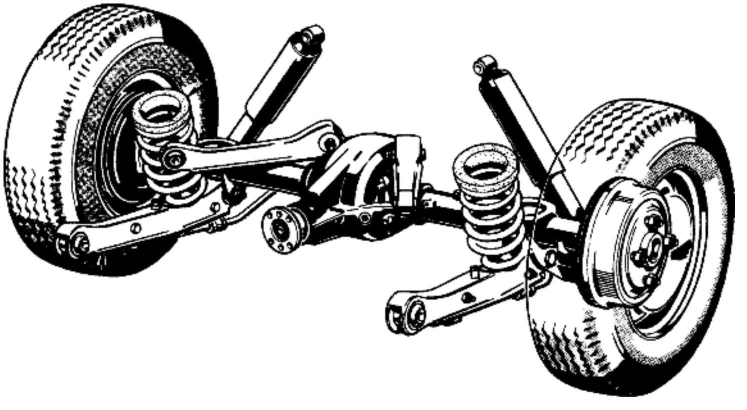

Figure 1.7.3 Widely-used design of four-link location axle (Ford).

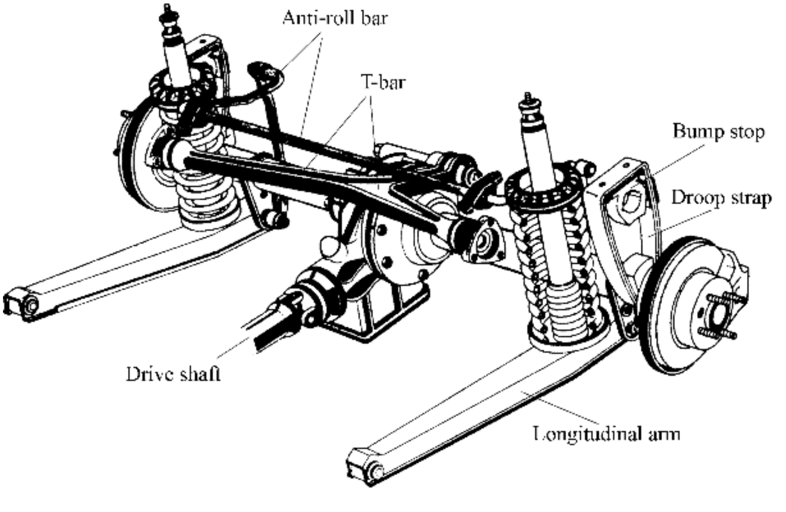

Figure 1.7.4 Alfa-Romeo T-bar upper lateral location.

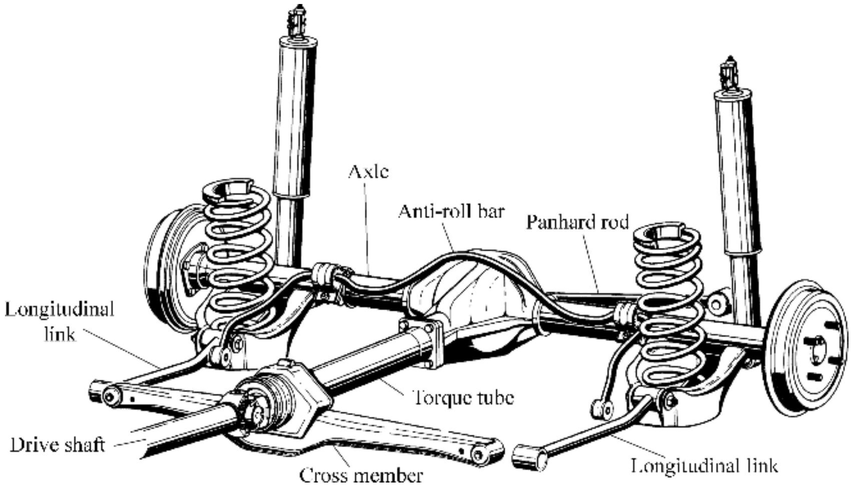

Figure 1.7.5 Rear axle with torque tube and Panhard rod (Opel).

There have, of course, been many other lateral location systems for axles, including the Panhard rod and the Watt's linkage, shown in Figures 1.7.5 and 1.7.6, respectively.

The rigid axle is sometimes fitted with a rigid tube going forward to a cross member in which it can rotate as in a ball joint. This, perhaps confusingly, is called a 'torque tube', presumably because it reacts to the pitch effect of torque in the driveshafts acting on the wheels. It does give very good location of the axle in pitch. Additional lateral location is required at the rear, such as by a Panhard rod as in Figure 1.7.5. In the similar torque tube system of Figure 1.7.6 the rear lateral location is by a Watt's linkage. Generally, the torque tube arrangement is a superior but more costly design used on more expensive vehicles.

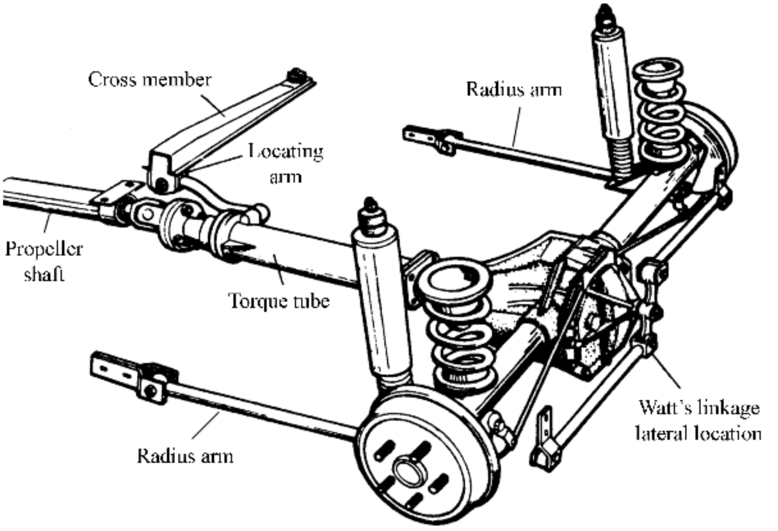

Figure 1.7.6 Axle with torque tube and Watt's linkage (Rover).

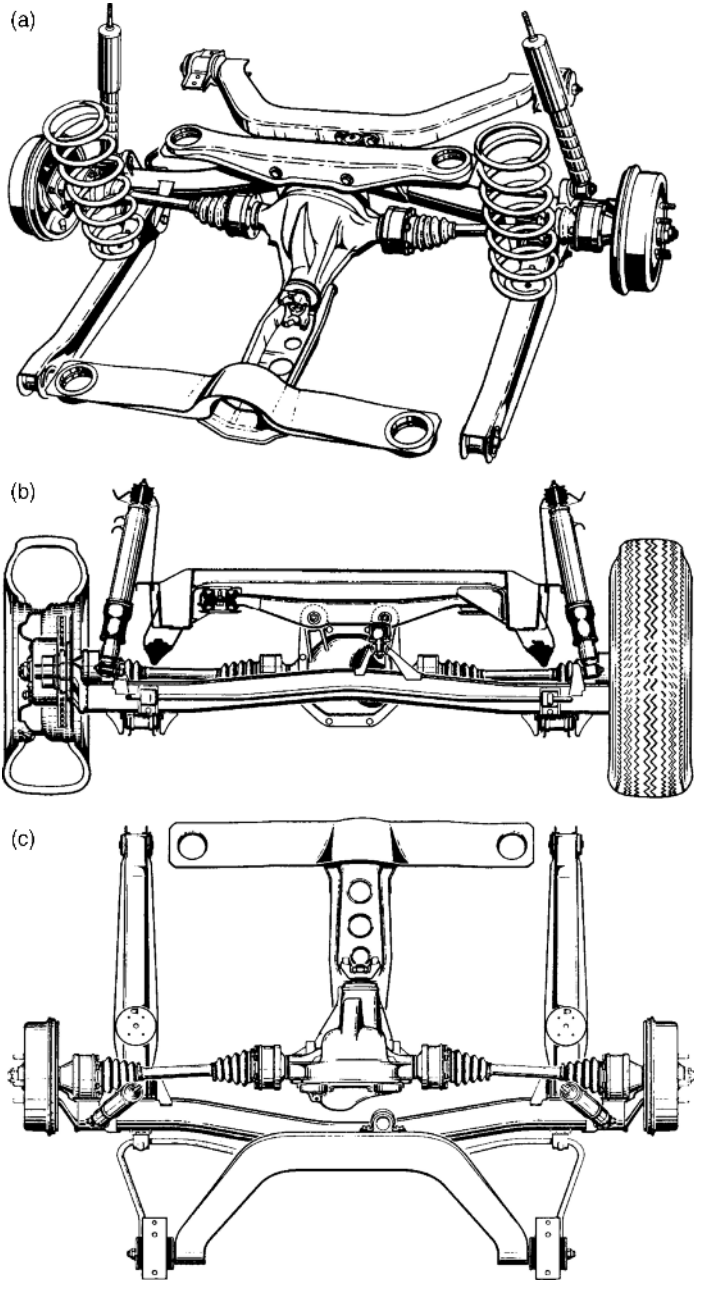

Figure 1.8.1 De Dion axle: (a) front three-quarter view; (b) rear elevation; (c) plan view (1969 Opel).

1.8 De Dion Rigid Axles

The de Dion design is an old one going back to the earliest days of motoring. In this axle, the two wheel hubs are linked rigidly together, but the final drive unit is attached to the body, so the unsprung mass is greatly reduced compared to a conventional live axle, Figure 1.8.1. Driveshaft length must be allowed some variation, for example by splines. The basic geometry of axle location is the same as that of a conventional axle.

Figure 1.8.2 shows a slightly different version in which the wheels are connected by a large sliding tube permitting some track variation, so that the driveshafts can be of constant length.

In general, the de Dion axle is technically superior to the normal live axle, but more costly, and so has been found on more expensive vehicles.

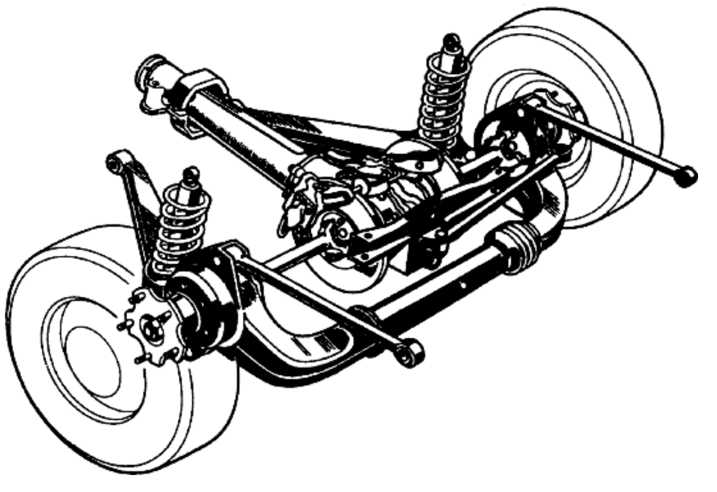

Figure 1.8.2 De Dion axle variation with sliding-tube variable track (1963 Rover).

1.9 Undriven Rigid Axles

Undriven rigid axles, used at the rear of front-drive vehicles, have the same geometric location requirements as live rigid axles, but are not subject to the additional forces and moments of the drive action, and can be made lower in mass. Figure 1.9.1 is a good example, with two lower widely-spaced

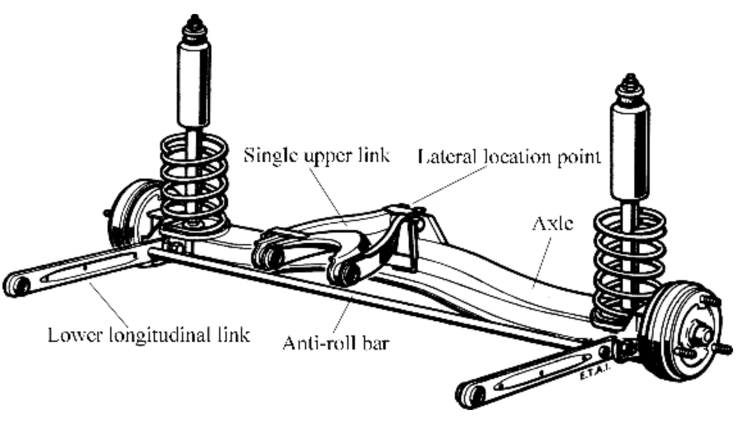

Figure 1.9.1 Simple undriven rigid axle (Renault).

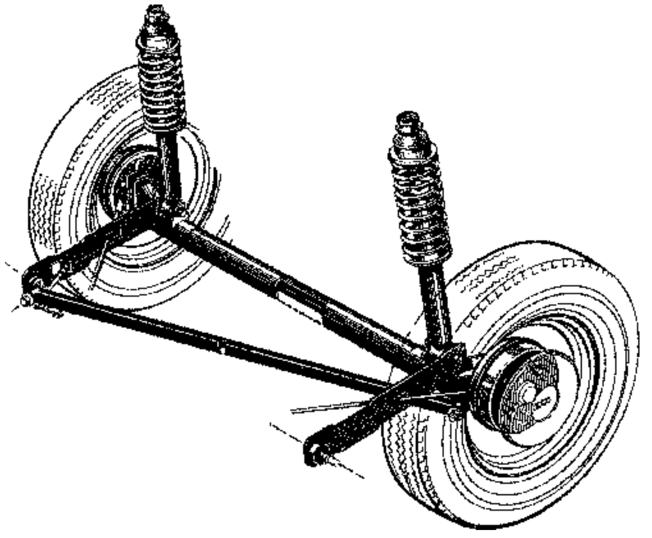

Figure 1.9.2 Undriven rigid axle with diagonal lateral location (Audi).

longitudinal arms and a single upper link for lateral and pitch location. The arms are linked by an anti-roll bar. The roll centre is high.

In Figure 1.9.2, lateral location is by the long diagonal member. This form eliminates the lateral displacements in bump of the Panhard rod. If the longitudinal links are fixed rigidly to the axle then the axle acts in torsion as an anti-roll bar, the system then being a limiting case of a trailing-twist axle.

The undriven axle of Figure 1.9.3 has location at each side by a longitudinal Watt's linkage, giving a truer linear vertical movement to the wheel centre than most systems, and also affecting the pitching angle of the hub, introducing an axle torsion anti-roll bar effect, but in a non-linear way.

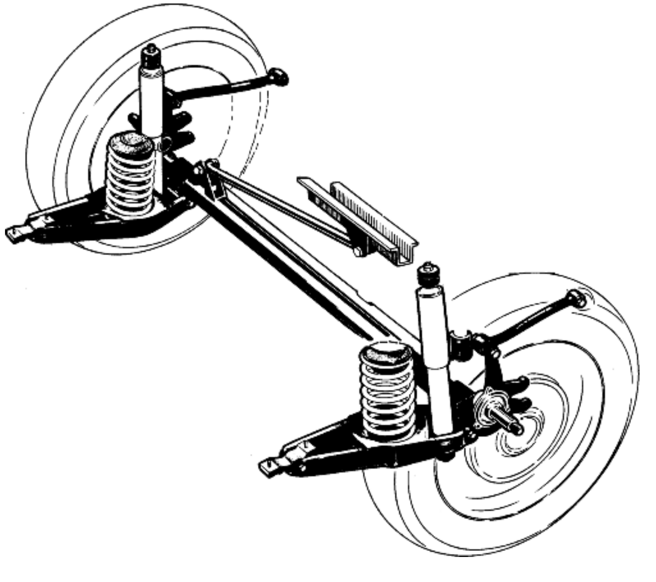

Figure 1.9.3 Undriven rigid axle with longitudinal Watt's linkages and Panhard rod (Saab).

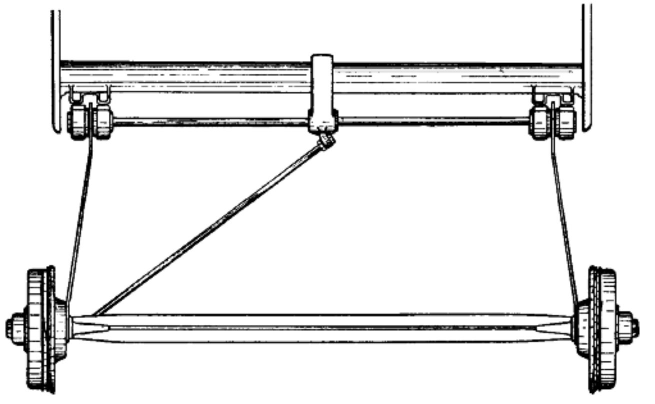

Figure 1.9.4 Rigid axle with diagonal lateral location and torsion bar springing, plan view (nascent trailing-twist axle) (Citroën).

Figure 1.9.4 shows a system with trailing arms operating half-width torsion bars, and with a short diagonal link for lateral location. If the trailing arms are fixed to the axle then in roll the axle will deflect torsionally, giving an anti-roll bar effect, whilst the thin trailing arms deflect relatively freely. This is another example of a limiting case of a trailing-twist axle, with the cross member at the wheel position.

Figure 1.9.5 shows a modern rigid axle with location system designed to give controlled side-force oversteer, the axle being able to yaw slightly about the front location point, according to the stiffness of the bushes in the outer longitudinal members.

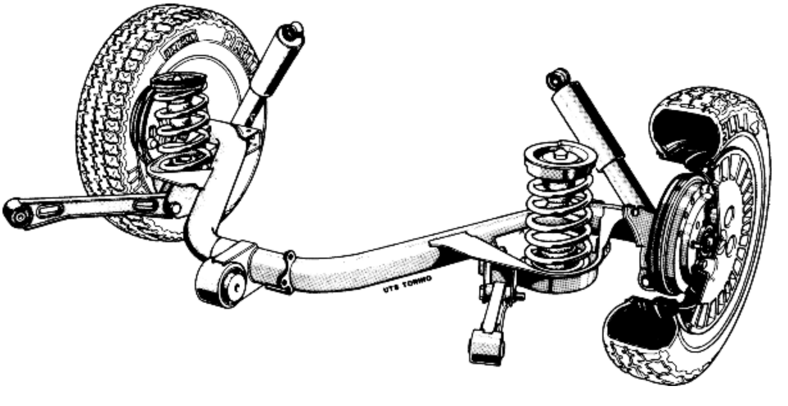

Figure 1.9.5 Tubular-structure undriven rigid axle with forward lateral location point and two longitudinal links (Lancia).

1.10 Independent Rear Driven

In the early days, most road vehicles had a rear drive, using a rigid axle. There were, however, some adventurous designers who tried independent driven suspension, such as on the Cottin-Desgoutes, which was shown in Figure 1.4.4. The most common early independent driven suspension was the simple swing axle, which has the advantage of constant driveshaft lengths, and low unsprung mass. The driveshafts can

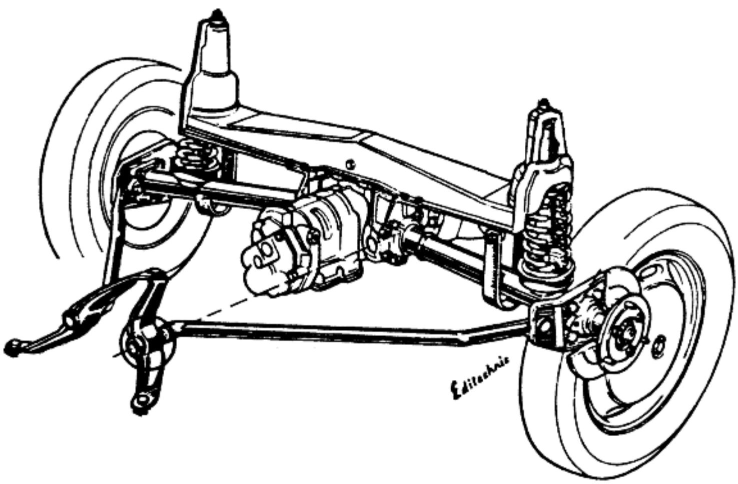

Figure 1.10.1 Swing axle with long leading links for longitudinal location (Renault).

swing forward so they require some extra location. Initially, a simple longitudinal pivot was used. Sometimes the supporting member had pivot points both in front of and behind the driveshaft. Figure 1.10.1 shows one with a single, forward, link. The swing axle has a large bump camber and little roll camber. The roll centre is not as high as with many rigid axles, but it is more of a problem because with a high roll centre on independent suspension there is jacking, which *in extremis* can get out of control with the outer wheel tucking under.

To overcome the problem of the roll centre of the basic swing axle, a low-pivot swing axle may be used, as in Figure 1.10.2, now requiring variable-length driveshafts by splines or doughnuts. The bottom pivots are offset slightly, longitudinally. This is still considered to be a swing axle because the axis of pivot of the axle part is longitudinal.

The obvious alternative to the swing axle is to use simple trailing arms, with the pivot axis perpendicular to the vehicle centre plane and parallel to the driveshafts. Again, this requires allowance for length

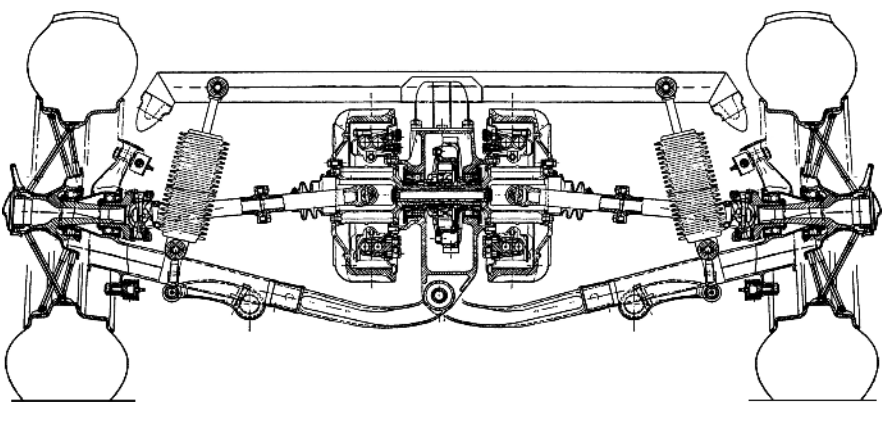

Figure 1.10.2 Low-pivot swing axle with inboard brakes (Mercedes Benz).

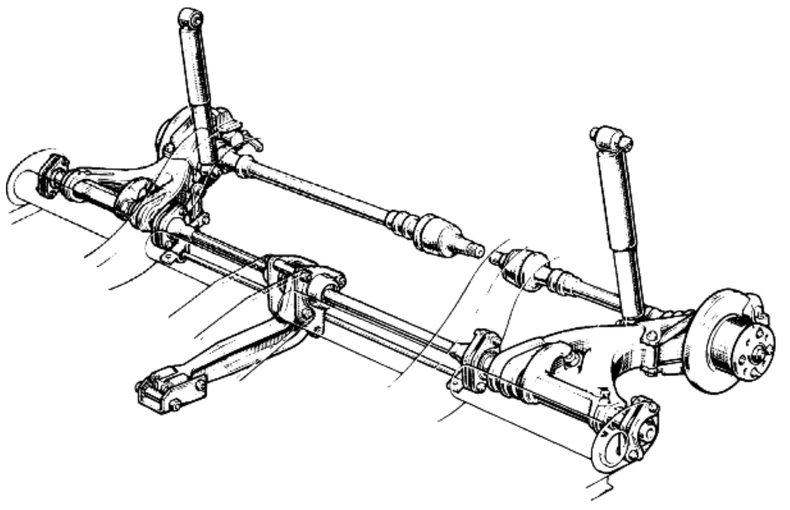

Figure 1.10.3 Plain trailing arms with 90° transverse axis of pivot (Matra Simca).

variation, a significant complication, Figure 1.10.3. In the example shown, the springing is by half-width torsion bars anchored at the vehicle centreline. There is also an anti-roll bar.

The next development, introduced in 1951, was the semi-trailing arm in which the arm pivot axis is a compromise between the swing axle and the plain trailing arm, typically in the range 15° to 25°, as in Figure 1.10.4. A more recent and simpler semi-trailing arm system is shown in Figure 1.10.5. Bump camber is greatly reduced compared with the swing axle.

To control the geometric properties more closely to desired values, a double wishbone system may be used, although this is less compact and on the rear of a passenger car it is detrimental to luggage space, but it is very widely used on sports and racing cars. Figure 1.10.6 shows an example sports car application, where the camber angle and the roll centre height were made adjustable.

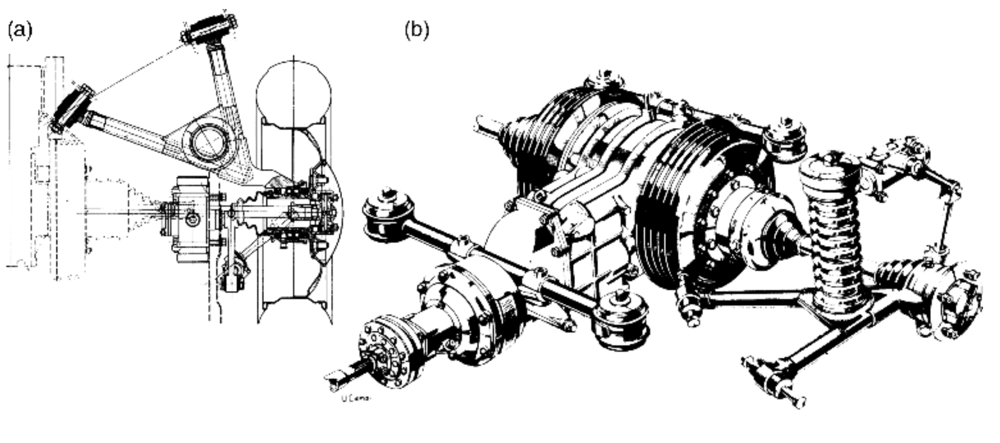

Figure 1.10.4 The first semi-trailing-arm design, also with transaxle and inboard brakes: (a) plan view; (b) front three-quarter view (1951 Lancia).

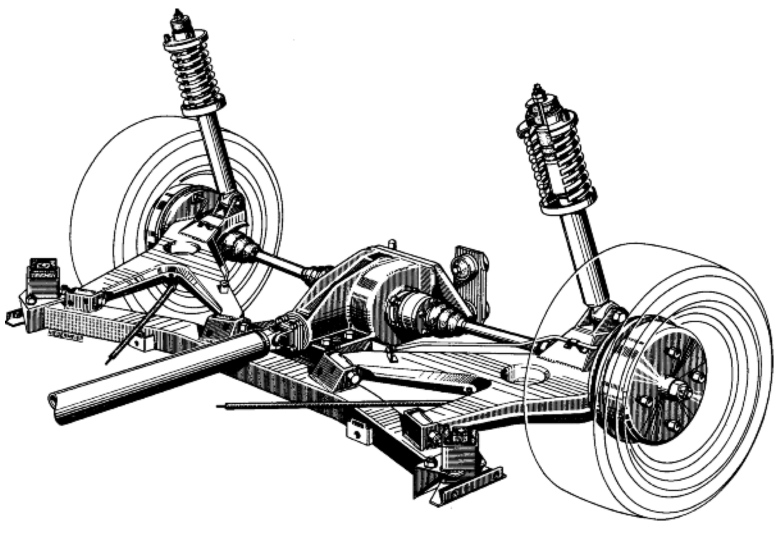

Figure 1.10.5 Semi-trailing arms (BMW).

The Chapman strut is a strut suspension in which the lower lateral location is provided by a fixed-length driveshaft. Figure 1.10.7 gives an example. Lower longitudinal location must also be provided, as seen in the forward diagonal arms which also, here, carry the springs.

Figure 1.10.8 shows the 'Weissach axle', which uses controlled compliance to give some toe-in on braking, or on power lift-off, for better handling.

A relatively recent extension of the wishbone concept is to separate each wishbone into two separate simple links. There are then five links in total, two for each wishbone and one steer angle link. This system

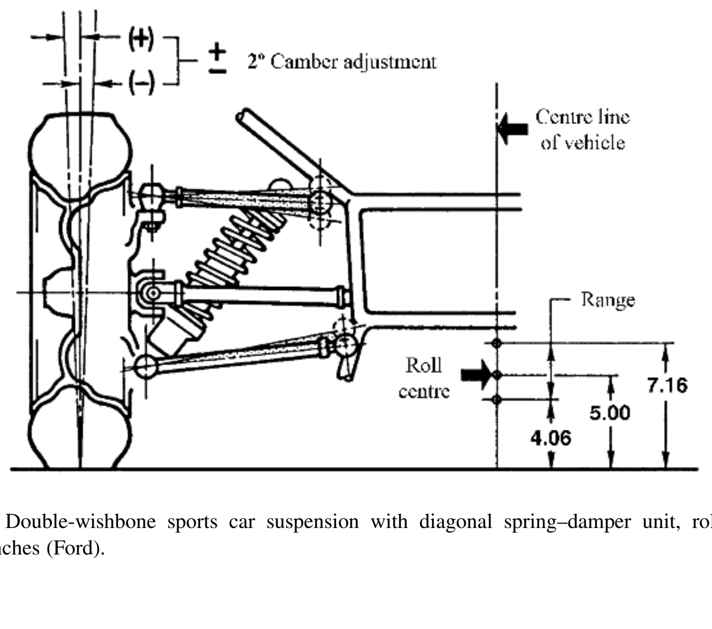

Figure 1.10.6 Double-wishbone sports car suspension with diagonal spring–damper unit, roll centre height dimensions in inches (Ford).

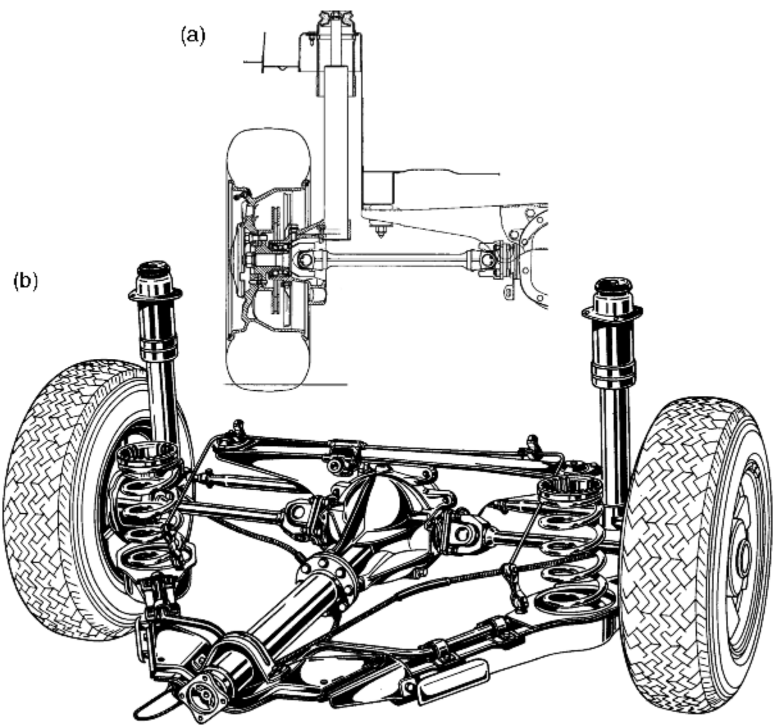

Figure 1.10.7 Chapman strut with front link for longitudinal location: (a) rear elevation; (b) front three-quarter view (Fiat).

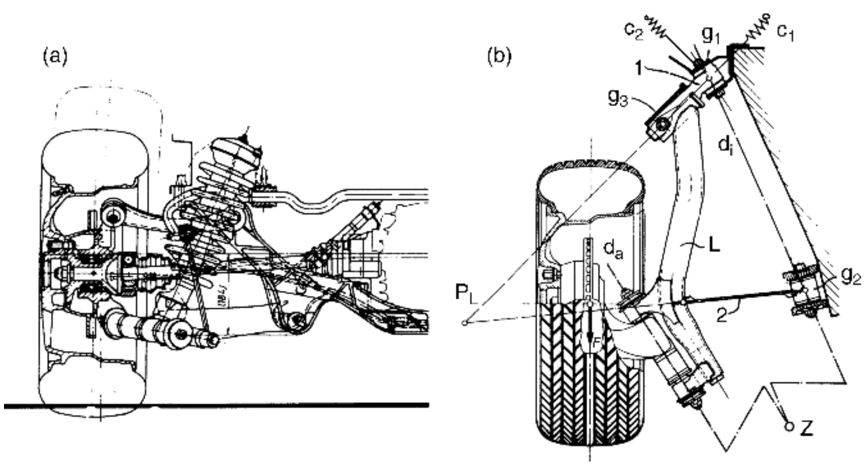

Figure 1.10.8 'Weissach axle' (Porsche).

has been used at the front and the rear, and, with careful design, makes possible better control of the geometric and compliance properties. Figure 1.10.9 shows an example. The advantages seem real for driven rear axles, but undriven ones have not adopted this scheme. The concept has also been used at the front for steered wheels.

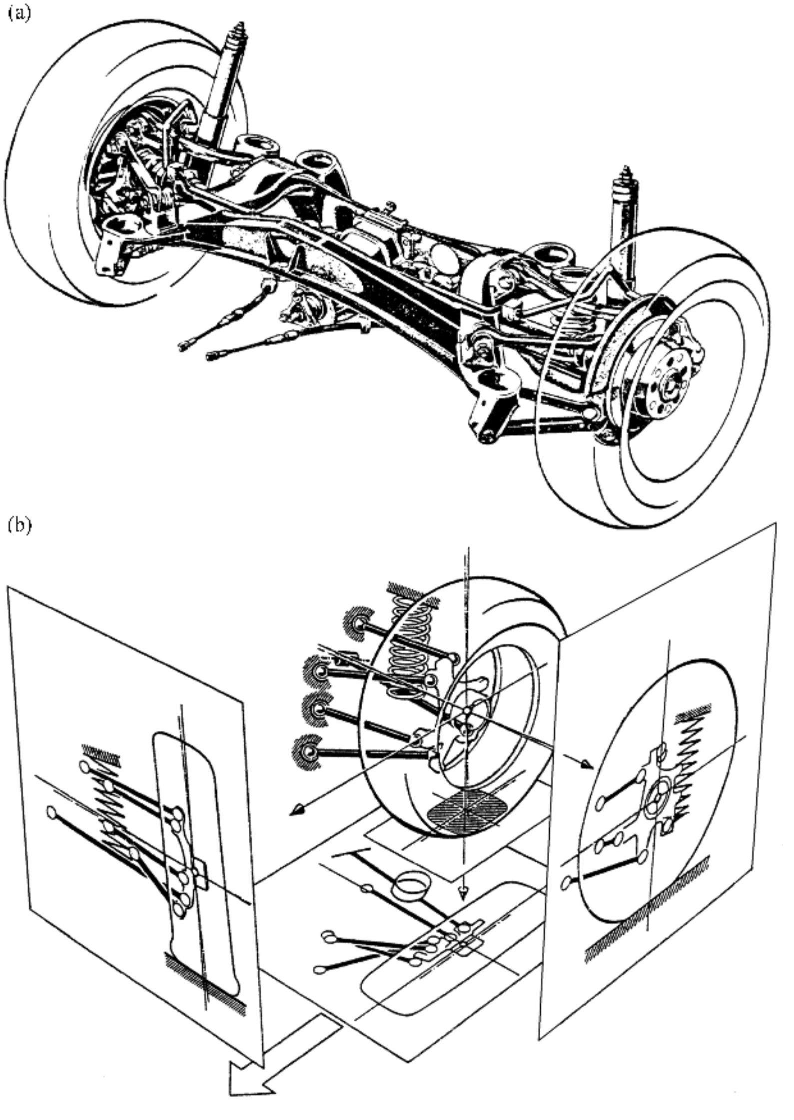

Figure 1.10.9 Five-link ('multilink') suspension: (a) complete driven rear-axle unit; (b) perspective details of one side with plan and front and side elevations (Mercedes Benz).

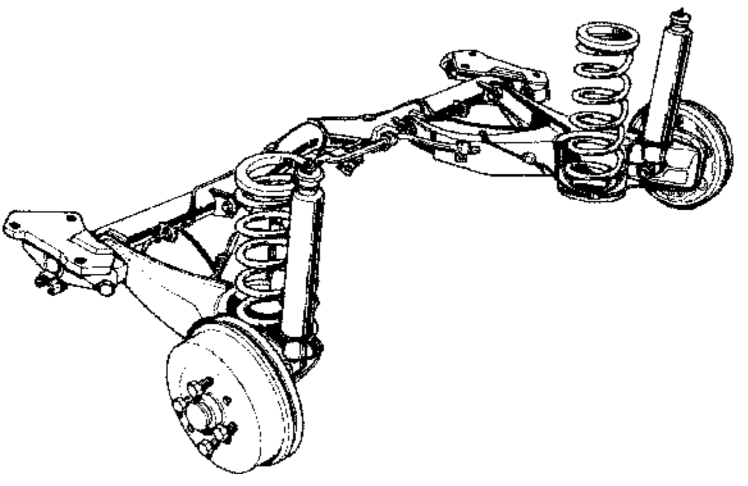

Figure 1.11.1 Plain trailing arms, 90° pivot axis, coil springs (Simca).

1.11 Independent Rear Undriven

At the rear of a front-drive vehicle it seems quite natural and easy to use independent rear suspension. Figures 1.11.1–1.11.4 give some examples.

The plain trailing arm with transverse pivot at 90° to the vehicle centre plane has often been used. The original BMC Mini, on which it was used in conjunction with rubber suspension, was a particularly compact example. A subframe is often used, as seen in Figure 1.11.1. Vertical coil springs detract from the luggage compartment space, so torsion bars are attractive. For symmetry, these have a length of only half of the track (tread), which is less than ideal. Figure 1.11.2 shows an example where slightly offset full-length bars are used. The left and right wheelbases are slightly different, but this does not seem to be of practical detriment.

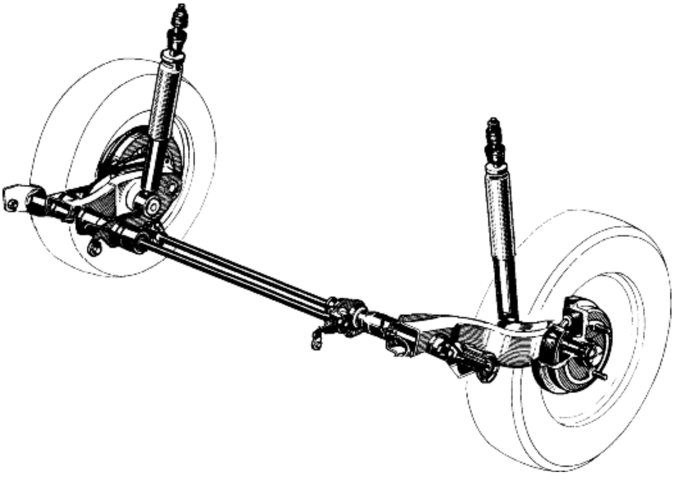

Figure 1.11.2 Plain trailing arms, 90° pivot axis, offset torsion-bar springs, unequal wheelbases (Renault).

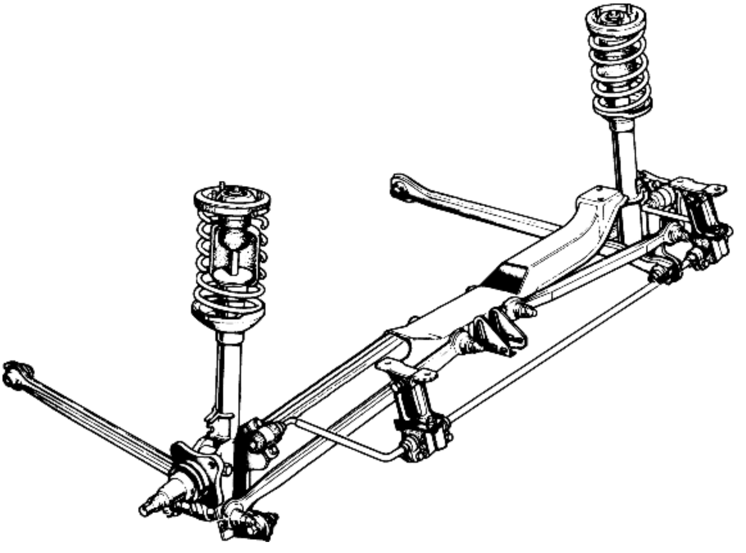

Figure 1.11.3 Strut suspension with long twin lateral/yaw location arms and leading link for longitudinal location (Lancia).

Figure 1.11.3 shows an independent strut rear suspension, the wheel-hub camber and pitch being controlled by the spring–damper unit. Twin lateral arms control scrub (track change) and the steer angle. The anti-roll bar has no effect on the geometry. Figure 1.11.4 shows another strut suspension, with a single wide lateral link controlling the steer angle.

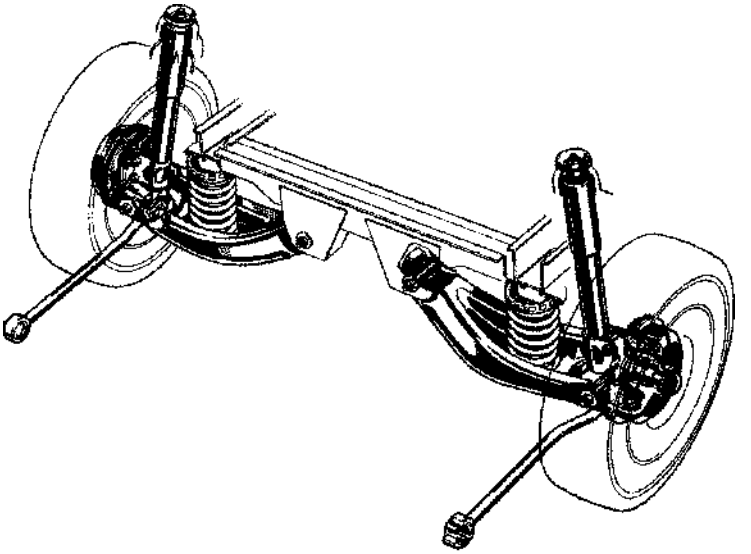

Figure 1.11.4 Strut suspension with wide lateral links and longitudinal links (Ford).

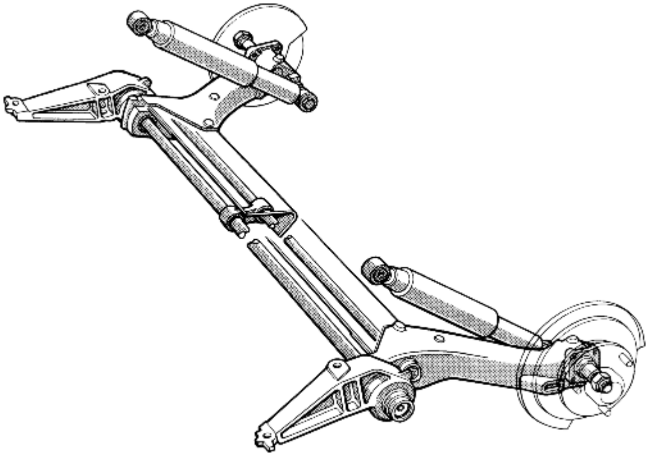

Figure 1.12.1 A compound crank axle using torsion-bar springs (Renault).

1.12 Trailing-Twist Axles

The 'trailing-twist' axle, now often known as the 'compound-crank' axle, is illustrated in Figures 1.12.1–1.12.3. The axle concept is good, but the new name is not an obvious improvement over the old one. This design is a logical development of the fully-independent trailing-arm system. Beginning with a simple pair of trailing arms, it is often desired to add an anti-roll bar. Originally, this was done by a standard U-shaped bar with two mountings on the body locating the bar, but allowing it to twist. Drop links connected the bar to the trailing arms. A disadvantage of this basic system was that the anti-roll bar transmitted extra noise into the passenger compartment, despite being fitted with rubber bushes. This problem was reduced by deleting the connections to the body, instead using two rubber-bushed connections on each trailing arm, so that the bar was still constrained in torsion. This was then simplified

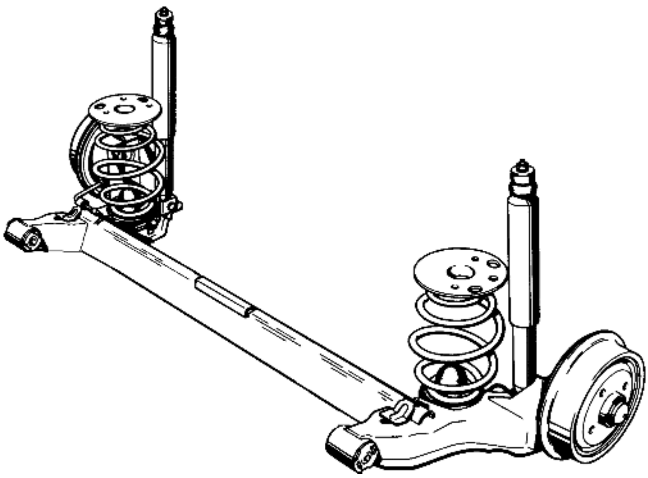

Figure 1.12.2 A compound crank axle with coil springs (Opel).

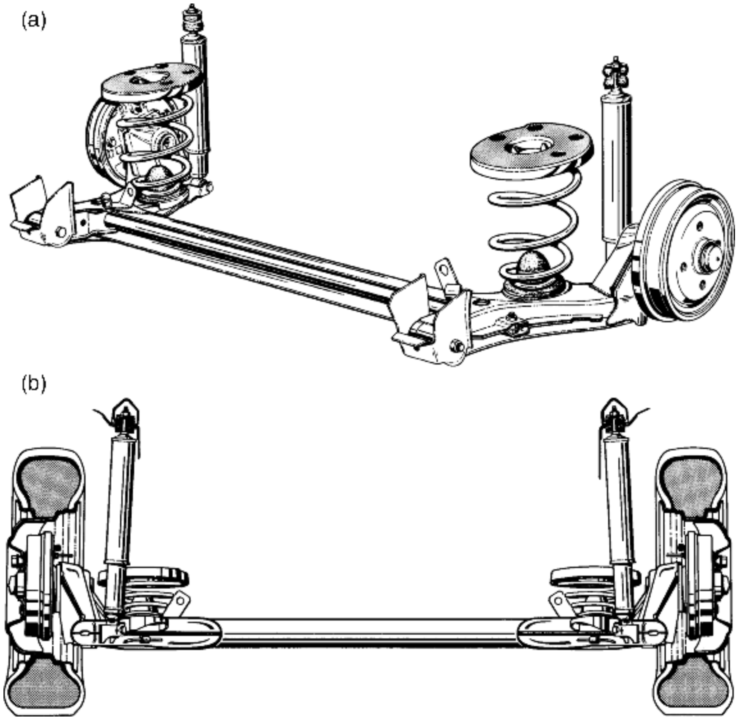

Figure 1.12.3 A compound crank axle similar to that in Figure 1.12.2, also showing in (a) the body fixture brackets, and in (b) the front elevation (Opel).

mechanically by making the two arms and the bar in one piece, requiring the now only semi-independent axle to flex in bending and torsion. This complicated the geometry, but allowed a compact system that was easy to install as a prepared unit at the final assembly stage. As seen in the figures, to facilitate the necessary bending and torsion the cross member of the axle is an open section pressing. The compound crank axle is now almost a standard for small passenger cars.

1.13 Some Unusual Suspensions

Despite the wide range of conventional suspensions already shown, many other strange and unusual suspensions have been proposed, and patented, and some actually used, on experimental vehicles at least, the designers claiming better properties of one kind or another, sometimes with justification. For most vehicles, the extra complication is not justified. Some designs are presented here for interest, and as a stimulus for thought, in approximate chronological order, but with no endorsement that they all work as claimed by the inventors. Proper design requires careful consideration of equilibrium and stability of the mechanism position, the kind of analysis which is usually conspicuously lacking from patents, which are generally presented in vague qualitative and conceptual terms only. As the information about these suspensions is mainly to be found in patent applications, engineering analysis is rarely offered in their support, the claims are only qualitative. The functioning details of actual roll angles and camber angles and the effect on handling behaviour are not discussed in the patents in any significant way.

There are many known designs of suspension with supplementary roll-camber coupling, that is, beyond the basic roll camber normally occurring. So far, however, this has not been a very successful theme. The proposed systems add cost, weight and complexity, and often have packaging difficulties. Also, there does not appear to have been a published analysis of the vehicle-dynamic handling consequences of such systems to provide an adequate basis for design. Typically, where the expected use is on passenger vehicles, the intention is to reduce or eliminate body roll, or even to bank the body into the corner, with claims of improved comfort. Where racing applications are envisaged, the main concern is elimination of adverse camber in roll.

Most of the suspensions discussed are of pure solid mechanical design. Such suspension design tends to add undesirable weight, is usually bulky, and is therefore space-consuming with packaging problems. Camber adjustment accomplished by lateral movement of the lower suspension arms causes adverse scrub effects at the tyre contact patch which is probably not acceptable.

In the Hurley suspension, shown in Figure 1.13.1, a pendulum drives four hydraulic master cylinders in two independent circuits to body roll and camber slave cylinders, the intention being to control both body roll and wheel camber. Either system could be used alone. A serious problem here, as with many other designs, is that an apparently reasonable kinematic design may fail to operate as expected, dynamically. In this case, force and energy are to be provided by the pendulum. Instead, body roll and wheel camber may simply force the pendulum over to the other side, unless it is sufficiently massive. One may also doubt the practicality of the general configuration for normal service. However, the Hurley design is of great historical interest.

In the Bennett 'Fairthorpe TX-1' suspension, Figure 1.13.2, the basically independent suspension has trailing arms, but the wheel hubs have additional camber freedom by rotation on the arms, and there is a crossed intercoupling not unlike cruciate ligaments. Double bump gives zero camber effect. Cornering body roll is said to give wheel camber into the curve. Only in single wheel bump is the camber characteristic considered to be less than ideal. An experimental version in racing was claimed to work well. The characteristics are basically like a rigid axle.

In the Drechsel suspension, shown in Figure 1.13.3, the body is suspended from an intermediate member, a mobile subframe, at each axle. The suspension is connected to the subframe. The body is intended to roll into the corner, pivoting like a pendulum on the subframe, simultaneously adjusting the wheel camber. Energy is provided by the outward motion of the body, which in effect provides a massive

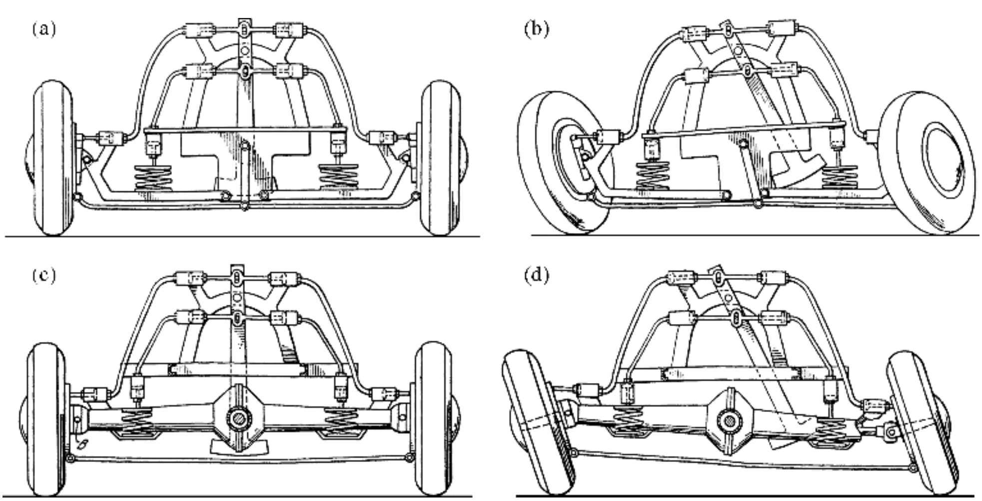

Figure 1.13.1 Hurley suspensions, all in front view: (a) applied to independent suspension; (b) independent in right-hand turn; (c) applied to an axle; (d) axle in right-hand turn (Hurley, 1939).

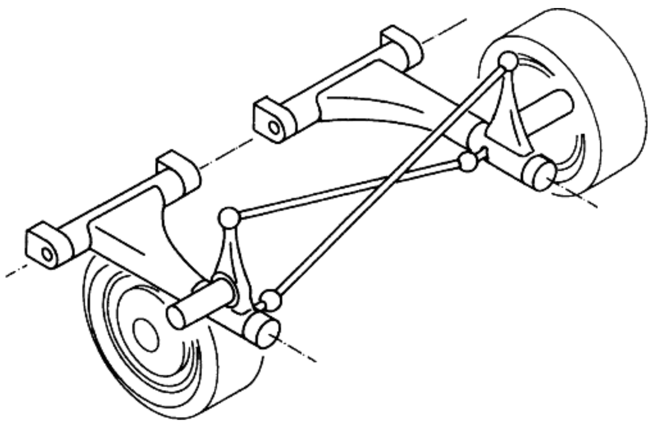

Figure 1.13.2 'Fairthorpe TX-1' suspension (T. Bennett, 1965).

pendulum as an improvement to the Hurley separate-pendulum concept. There are two geometric roll centres, one for the body on the subframe, the other for the suspension on the subframe.

The Trebron double roll centre (DRC) suspension, Figure 1.13.4, due to N. Hamy, is similar in operation to the Drechsel concept, but with changes to the details. An adapted passenger car test vehicle operated successfully, demonstrating negated chassis roll and camber change in cornering.

In the Bolaski suspension, shown in Figure 1.13.5, the operating principle is again similar to that of Drechsel, using the body as a massive pendulum, but in this case the main body rests on compression links, and in cornering is intended to deflect the central triangular member which in turn adjusts the wheel camber by the lower wishbones. Bolaski limits his invention for application to front suspensions by the title of this patent.

In the Parsons system, Figure 1.13.6, each axle has two mobile subframes. The body and the two subframes have a common pivot as shown, but this is not an essential feature. The front suspension design is expected to use struts. Each upper link, on rising in bump, pulls on the opposite lower wishbone, changing the camber angle of that side. On the rear suspension design, with double wishbones, in bump the rising lower link pushes the opposite upper wishbone out, having the same type of camber effect. In the double-wishbone type shown, the spring as shown would give a negative heave stiffness, but this would be used in conjunction with a pair of stiff conventional springs to give an equivalent anti-roll bar effect. Of course, the springs shown are not an essential part of the geometry.

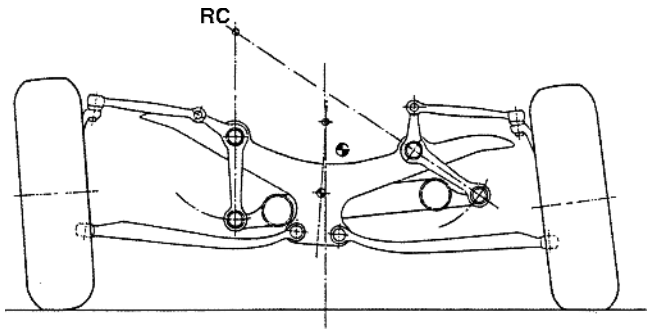

Figure 1.13.3 Drechsel suspension, rear view, left-hand cornering (Drechsel, 1956).

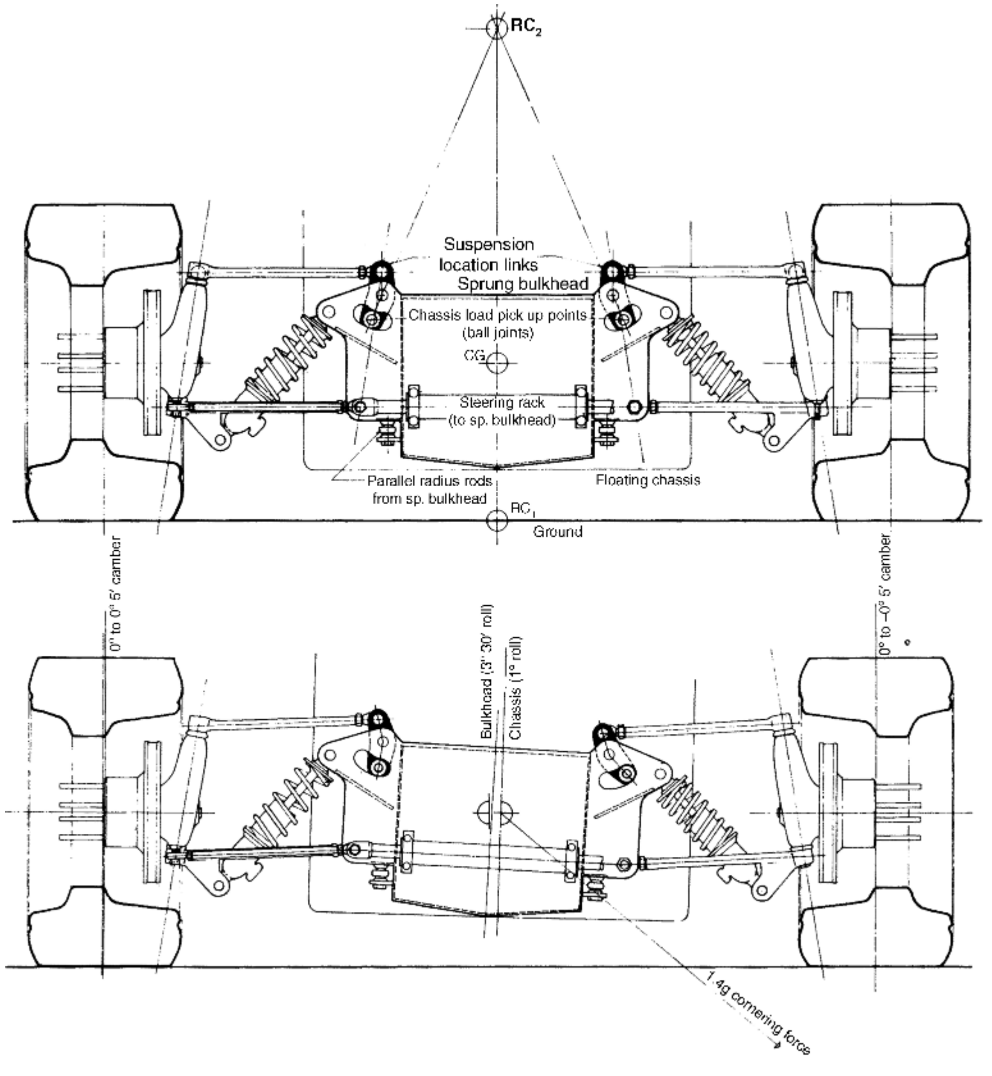

Figure 1.13.4 Trebron DRC suspension rear views: (a) configuration, (b) in turn, (Hamy, 1968).

The Jeandupeux system, shown in Figure 1.13.7, has one mobile subframe at each axle, as a slider, carrying the lower two wishbone inner pivots, giving simultaneous lateral shifting of the lower wishbones. In addition there are extra links connecting to the upper wishbones. In cornering, the wheel lateral forces produce forces in the lower wishbones, which move the slider subframe across, turning the vertical rocker, and pulling the centre of the V across, tending to lift or lower the opposing wheels, hence opposing the body roll and opposing camber change. In other words, there is a direct, basically linear, relationship between lateral tyre force and consequent load transfer through the mechanism, to be compared with the natural value for a simple suspension. With correct design, there should be no roll in cornering. Double bump causes no camber because of the equal parallel wishbone design.

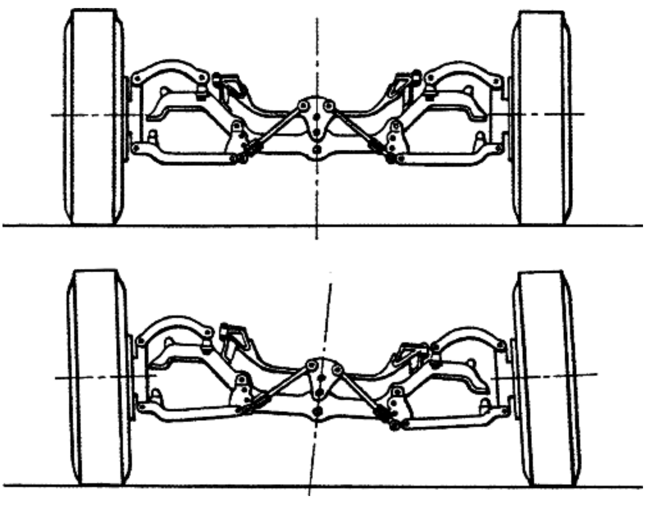

Figure 1.13.5 Bolaski suspension (Bolaski, 1967).

In contrast to the earlier suspensions, all claimed to be passive in action, the Phillippe suspension, Figure 1.13.8, is declared to be an active system with power input from a hydraulic pump, with camber control by hydraulic action on a slider linking the inner pivots of the upper wishbones. This is direct active control of the geometry, quite different from the normal active suspension concept which replaces the usual springs and dampers in the vertical action of the suspension. The light pendulum shown acts only as a sensor.

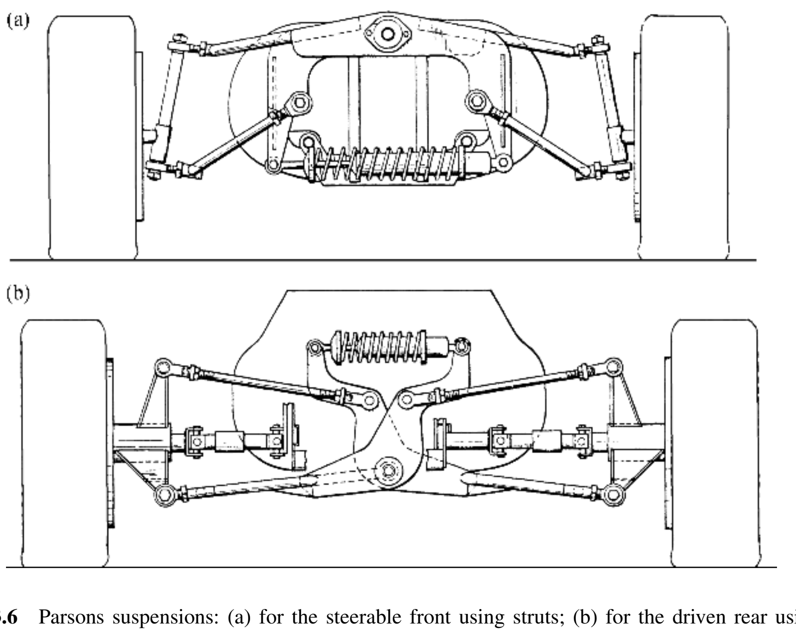

Figure 1.13.6 Parsons suspensions: (a) for the steerable front using struts; (b) for the driven rear using double wishbones. Additional springs and dampers would be used on each wheel (Parsons, 1971).

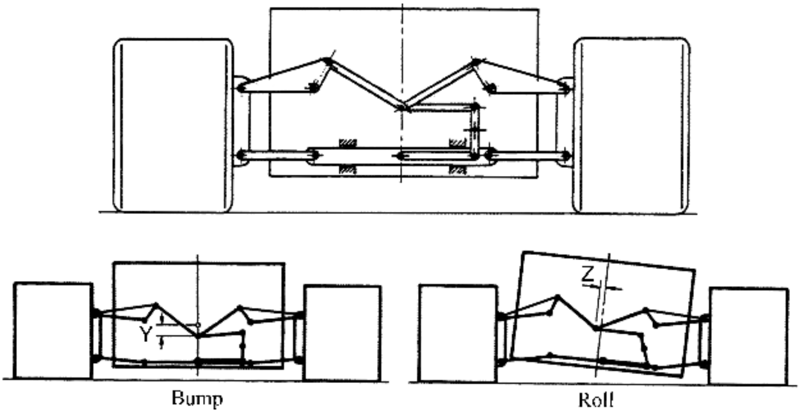

Figure 1.13.7 Jeandupeux suspension: (a) configuration; (b) kinematic action in bump and roll without camber change (Jeandupeux, 1971).

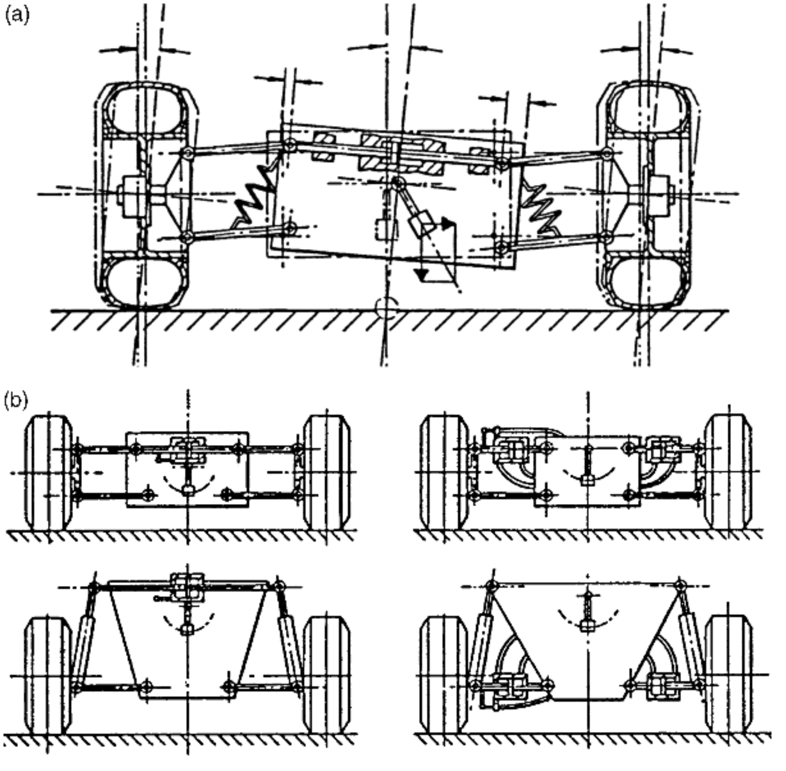

Figure 1.13.8 (a) Phillippe suspension (Phillippe, 1975). (b) Variations of Phillippe suspension.

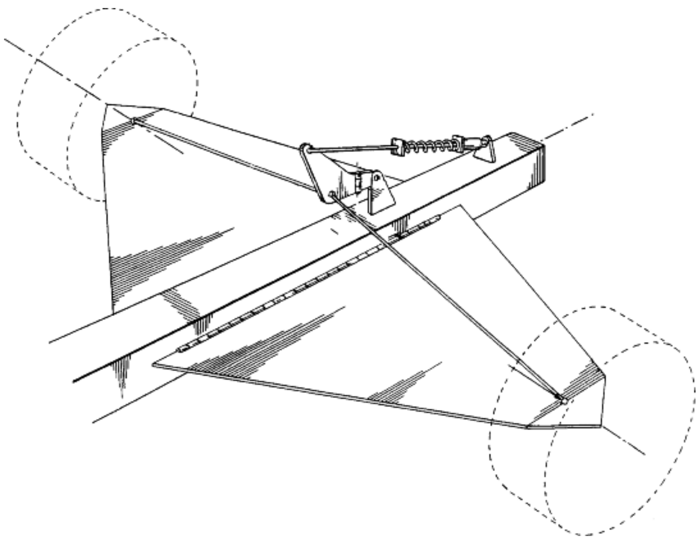

Figure 1.13.9 Pellerin suspension (Pellerin, 1997).

The Pellerin suspension is aimed specifically at formula racing cars which use front suspensions which are very stiff in roll. These have sometimes used a mono-spring–damper suspension, the so-called 'monoshock' system, which is a suspension with a single spring–damper unit active in double bump, with nominally no roll action in the suspension mechanism itself. The complete front roll stiffness then depends mainly on the tyre vertical stiffness with some contribution from suspension compliance. In the Pellerin system, Figure 1.13.9, the actuator plate of the spring–damper unit is vertically hinged, allowing some

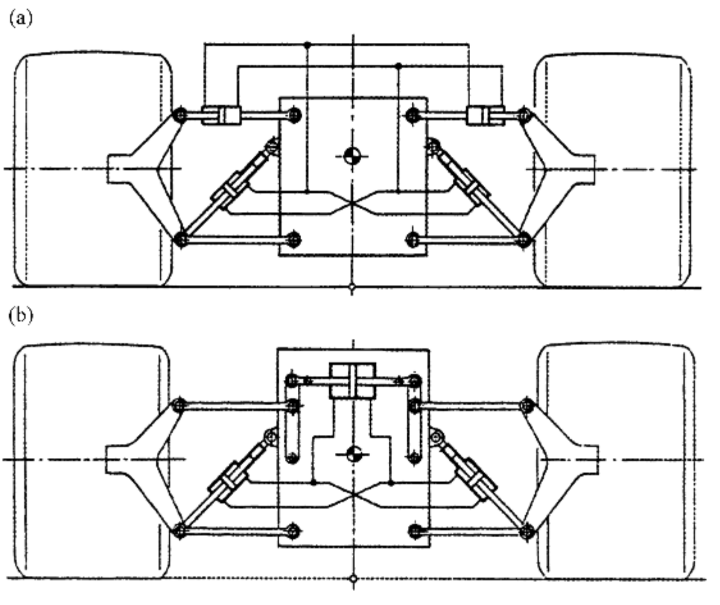

Figure 1.13.10 Weiss PRCC suspension (springs not shown): (a) version 1, camber cylinders in wishbones; (b) version 2, camber cylinder between the two wishbone pivots (Weiss, 1997).

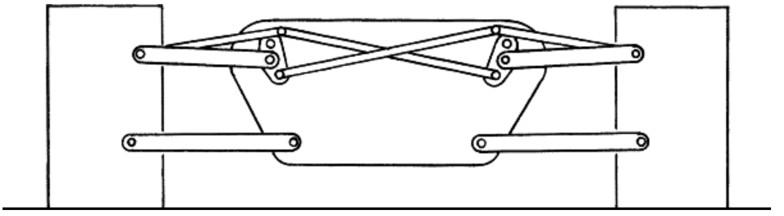

Figure 1.13.11 Walker 'Camber Nectar' suspension (Walker, 2003).

lateral motion of the pushrod connection point. The roll stiffness of the suspension mechanism is then provided by the tension in the hinged link, and is dependent on the basic pushrod compression force, which increases with vehicle speed due to aerodynamic downforce.

The Weiss passive roll-camber coupled (PRCC) system, shown in Figure 1.13.10, uses hydraulic coupling of the wheel camber angle with body roll, in several configurations. Normal springs are retained. Essentially, in cornering the tyre side forces are used to oppose body roll, and body forces oppose adverse camber development, mediated by the hydraulics. Simple double bump moves the diagonal bump cylinders, but these are cross coupled, so in this action they merely interchange fluid.

Finally, Figure 1.13.11 shows the Walker 'Camber Nectar' intercoupling system simply operating between the upper wishbones, each of which is hinged at its inner end on a vertical rocker, the lower end of which is coupled to a drive rod on top of the opposite wishbone. The basic wishbone geometry is such as to give about 50% compensation of roll camber by the basic geometry and 50% by the action of the extra links, so that double bump and roll both give zero camber change.

The above examples give at least a flavour of the many unconventional systems that have been proposed. Many others may be found in the patent literature.

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