

1

Introduction

1.1 History

The current world-wide production of vehicle dampers, or so-called shock absorbers, is difficult to estimate with accuracy, but is probably around 50–100 million units per annum with a retail value well in excess of one billion dollars per annum. A typical European country has a demand for over 5 million units per year on new cars and over 1 million replacement units, The US market is several times that. If all is well, these suspension dampers do their work quietly and without fuss. Like punctuation or acting, dampers are at their best when they are not noticed - drivers and passengers simply want the dampers to be trouble free. In contrast, for the designer they are a constant interest and challenge. For the suspension engineer there is some satisfaction in creating a good new damper for a racing car or rally car and perhaps making some contribution to competition success. Less exciting, but economically more important, there is also satisfaction in seeing everyday vehicles travelling safely with comfortable occupants at speeds that would, even on good roads, be quite impractical without an effective suspension system.

The need for dampers arises because of the roll and pitch associated with vehicle manoeuvring, and from the roughness of roads. In the mid nineteenth century, road quality was generally very poor. The better horse-drawn carriages of the period therefore had soft suspension, achieved by using long bent leaf springs called semi-elliptics, or even by using a pair of such curved leaf springs set back-to-back on each side, forming full-elliptic suspension. No special devices were fitted to provide damping; rather this depended upon inherent friction, mainly between the leaves of the beam springs. Such a set-up was appropriate to the period, being easy to manufacture, and probably worked tolerably well at moderate speed, although running at high speed must have been at least exciting, and probably dangerous, because of the lack of damping control.

The arrival of the so-called horseless carriage, i.e. the carriage driven by an internal combustion engine, at the end of the nineteenth century, provided a new stimulus for suspension development which continues to this day. The rapidly increasing power available from the internal combustion engine made higher speeds routine; this, plus the technical aptitude of the vehicle and component designers, coupled with a general commercial mood favouring development and change, provided an environment that led to invention and innovation.

The fitting of damping devices to vehicle suspensions followed rapidly on the heels of the arrival of the motor car itself. Since those early days the damper has passed through a century of evolution, the basic stages of which may perhaps be considered as:

- (1) dry friction (snubbers);
- (2) blow-off hydraulics;
- (3) progressive hydraulics;
- (4) adjustables (manual alteration);
- (5) slow adaptives (automatic alteration);
- (6) fast adaptives ('semi-active');
- (7) electrofluidic, e.g. magnetorheological.

Historically, the zeitgeist regarding dampers has changed considerably over the years, in roughly the following periods:

- (1) Up to 1910 dampers were hardly used at all. In 1913, Rolls Royce actually discontinued rear dampers on the Silver Ghost, illustrating just how different the situation was in the early years.
- (2) From 1910 to 1925 mostly dry snubbers were used.
- (3) From 1925 to 1980 there was a long period of dominance by simple hydraulics, initially simply constant-force blow-off, then through progressive development to a more proportional characteristic, then adjustables, leading to a mature modern product.
- (4) From 1980 to 1985 there was excitement about the possibilities for active suspension, which could effectively eliminate the ordinary damper, but little has come of this commercially in practice so far because of the cost.
- (5) From 1985 it became increasingly apparent that a good deal of the benefit of active suspension could be obtained much more cheaply by fast auto-adjusting dampers, and the damper suddenly became an interesting, developing, component again.
- (6) From about 2000, the introduction, on high-price vehicles at least, of controllable magnetorheological dampers.

Development of the adaptive damper has occurred rapidly. Although there will continue to be differences between commercial units, such systems are now effective and can be considered to be mature products. Fully active suspension offers some performance advantages, but is not very cost effective for passenger cars. Further developments can then be expected to be restricted to rather slow detail refinement of design, control strategies and production costs. Fast acting control, requiring extra sensors and controls, will continue to be more expensive, so simple fixed dampers, adjustables and slow adaptive types will probably continue to dominate the market numerically for the foreseeable future.

The basic suspension using the simple spring and damper is not ideal, but it is good enough for most purposes. For low-cost vehicles, it is the most cost-effective system. Therefore much emphasis remains on improvement of operating life, reliability and low-cost production rather than on refinement of performance by technical development. The variable damper, in several forms, has now found quite wide application on mid-range and expensive vehicles. On the most expensive passenger and sports cars, magnetorheologically controlled dampers are now a popular fitment, at significant expense.

The damper is commonly known as the shock absorber, although the implication that shocks are absorbed is misleading. Arguably, the shocks are 'absorbed' by the deflection of the tires and springs. The purpose of dampers is to dissipate any energy in the vertical motion of body or wheels, such motion having arisen from control inputs, or from disturbance by rough roads or wind. Here 'vertical' motion includes body heave, pitch and roll, and wheel hop. As an agglomeration of masses and springs, the car with its wheels constitutes a vibrating system that needs dampers to optimise control behaviour, by preventing response overshoots, and to minimise the influence of some unavoidable resonances. The mathematical theory of vibrating systems largely uses the concept of a linear damper, with force proportional to extension speed, mainly because it gives equations for which the solutions are well understood and documented, and usually tolerably realistic. There is no obligation on a damper to exhibit such a characteristic; nevertheless the typical modern hydraulic damper does so approximately. This is because the vehicle and damper manufacturers consider this to be desirable for good physical

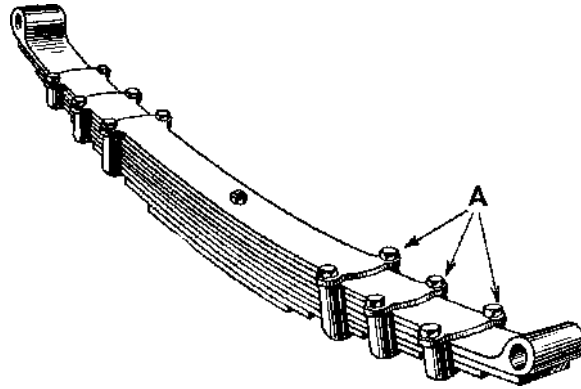


Figure 1.1.1 Dry friction damping by controlled clamping (adjustable normal force) of the leaf spring (Woodhead).

behaviour, not for the convenience of the theorist. The desired characteristics are achieved only by some effort from the manufacturer in the detail design of the valves.

Damper types, which are explained fully later, can be initially classified as

- (a) dry friction with solid elements;
 - (i) scissor;
 - (ii) snubber;
- (b) hydraulic with fluid elements;
 - (i) lever-arm;
 - (ii) telescopic.

Only the hydraulic type is in use in modern times. The friction type came originally as sliding discs operated by two arms, with a scissor action, and later as a belt wrapped around blocks, the 'snubber'. The basic hydraulic varieties are lever-arm and telescopic. The lever-arm type uses a lever to operate a vane, now extinct, or a pair of pistons. Telescopic, now most common, are either double-tube or gas-pressurised single-tube.

The early days of car suspension gave real opportunities for technical improvement, and financial reward. The earliest suspensions used leaf springs with inherent interleaf friction. Efforts had been made to control this to desirable levels by the free curvature of the leaves. Further developments of the leaf spring intrinsic damping included controlled adjustment of the interleaf normal forces, Figure 1.1.1, and the use of inserts of various materials to control the friction coefficients, Figure 1.1.2.

Truffault invented the scissor-action friction disc system before 1900, using bronze discs alternating with oiled leather, pressed together by conical disc springs and operated by two arms, with a floating body. The amount of friction could be adjusted by a compression hand-screw, pressing the discs together more or less firmly, varying the normal force at approximately constant friction coefficient. Between 1900 and 1903, Truffault went on to develop a version for cars, at the instigation

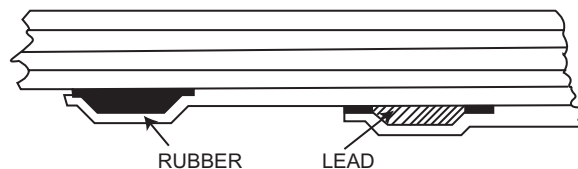


Figure 1.1.2 Leaf spring inserts to control the friction coefficient and consequent damping effect.

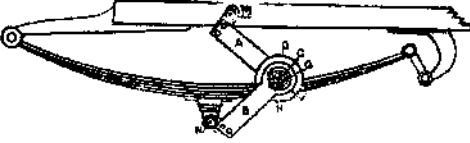
GREATEST SPEED EVER ATTAINED
IN HISTORY OF THE WORLD

At Brescia, Italy, on Sept. 4th, Lancia, with his 80 H. P. Fiat won the Italian road race, beating the most renowned racing cars of Europe.

230 Miles in 3 Hours 10 Minutes 56 Seconds

Making the unheard-of long distance average of 72 MILES AN HOUR, beating all railroad records. This was only possible by use of the famous

Truffault-Hartford Suspension



RECENT VICTORIES

French Eliminating trials (Thery—"Richard-Brasier," 80 H. P.) Beat nearest competitor 25 mins.

Gordon-Bennett Cup (Thery—"Richard-Brasier," 80 H. P.) Wrested cup from Germany and the famous Mercedes.

Mile Standing Start (Baras—"Darracq" 80 H. P.) World record lowered 5 secs.

Flying Kilometre (Rigoly—"Gobron-Brille"). 104 miles an hour.

Harry Harkness, Long Branch, Aug. 17, (60 H. P. "Mercedes"). Cut the world 50 mile track record 16 mins. 27 secs.

Providence, Sept. 10, (H. L. Bowden—90 H. P. "Mercedes"). 2-10 mile track records lowered.

"Diminishes in a remarkable degree the wear on the tires."—*Michelin*.
"Was able to drive at highest speed over all obstructions."—*Thery*.
"Remarkable and perfect suspension."—*Mercedes Co.*

Although indispensable for racing cars, its greatest utility is for ordinary touring cars or runabouts.

Are you not tired of being jolted and jarred?

Can be applied to any car in a few hours.

Address E. V. HARTFORD, 67 Vestry St., New York City
Telephone, 3754 Spring

Figure 1.1.3 An advertisement from 1904 for the early Truffault designed dry friction scissor damper manufactured by Hartford.

of Hartford in the US, who began quantity production in 1904, as in Figures 1.1.3–1.1.5. Truffault, well aware of the commercial potential, also licensed several other manufacturers in Europe, including Mors and Peugeot in France, who also had them in production and use by 1904. A similar type of damper was also pressed into service on the steering, Figure 1.1.6, to reduce steering fight on rough roads and to reduce steering vibrations then emerging at higher speeds and not yet adequately understood.

Figure 1.1.7 shows an exploded diagram of a more recent (1950s) implementation from a motorcycle. This is also adjustable by the hand-screw. Subsequent to the Truffault–Hartford type, The Hartford Telecontrol (the prefix *tele* means remote) developed the theme, Figure 1.1.8, with a convenient Bowden cable adjustment usable by the driver *in situ*. A later alternative version, the Andre Telecontrol, had dry friction scissor dampers, but used hydraulic control of the compression force and hence of the damper friction moment.

In 1915, Claud Foster invented the dry friction block-and-belt snubber, Figure 1.1.9, manufactured in very large quantities by his Gabriel company, and hence usually known as the Gabriel Snubber. In view of the modern preference for hydraulics, the great success of the belt snubber was presumably based on low cost, ease of retrofitment and reliability rather than exceptional performance.

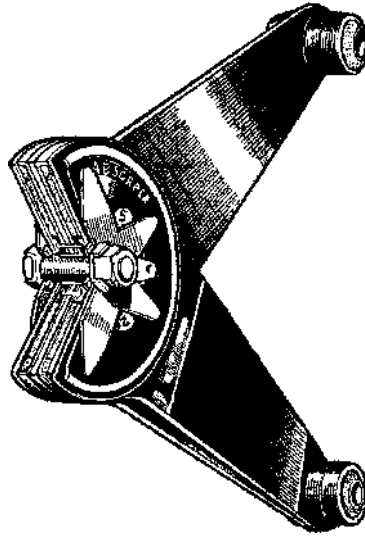


Figure 1.1.4 The Andre-Hartford scissor-action dry friction damper.

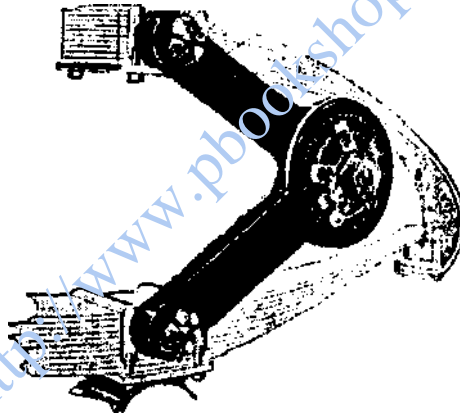
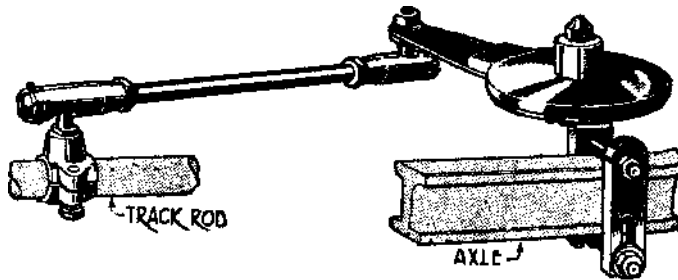


Figure 1.1.5 Installation of a dry-friction scissor damper on three-quarter-elliptic leaf springs (from Simanaitis, 1976).



A Damping Device for Preventing Wheel Wobble on Rigid-Type Front Axles.

Figure 1.1.6 Use of the Truffault-Hartford rotary dry friction damper on steering.

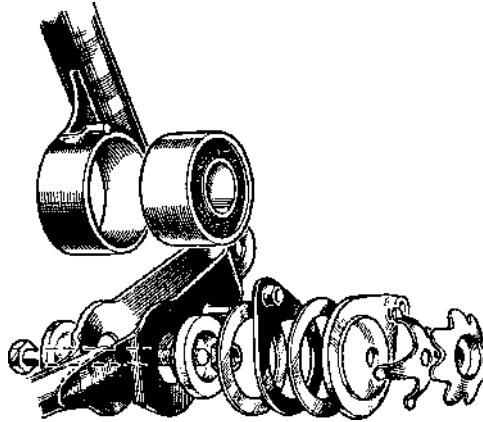


Figure 1.1.7 The Greeves motorcycle front suspension from around 1950 had a rubber-in-torsion spring, using an integral rotary dry friction damper easily adjustable by hand.

The spring-loaded blocks are mounted on the body, in particular on the chassis rails in those days, with the leather belt being fixed to the wheel upright or axle. In upward motion of the suspension, the snubber has no effect, but the spring-loaded blocks take up any slack. Any attempt by the suspension to extend will be opposed by the belt which has considerable friction where it wraps over itself and around the blocks. Hence the action is fully asymmetrical. The actual performance parameters do not seem to have been published. Some theoretical analysis may be possible, derived from the standard theory of wrapped circular members, with friction force growing exponentially with wrapping angle, for prediction of the force in relation to block shape, spring force and stiffness and belt-on-belt and belt-on-block coefficients of friction. The overall characteristic, however, seems to be an essentially velocity-independent force in extension, i.e. fully asymmetrical Coulomb damping. The characteristics could have been affected in service conditions by the friction-breaking effect of engine vibrations.

An early form of hydraulic contribution to damping was the Andrex oil-bath damper, Figure 1.1.10. This had metal and leather discs as in the dry damper, but was immersed in a sealed oil bath. There may also have been a version with separated metal discs relying on oil in shear. Another version, Figure 1.1.11, was adjustable from the dashboard, with oil pressure transmitted to the dampers to control the normal force on the discs, or perhaps in some cases to adjust the level of oil in the case. The pressure gauge in Figure 1.1.11 suggests that this type was controlling the normal force.

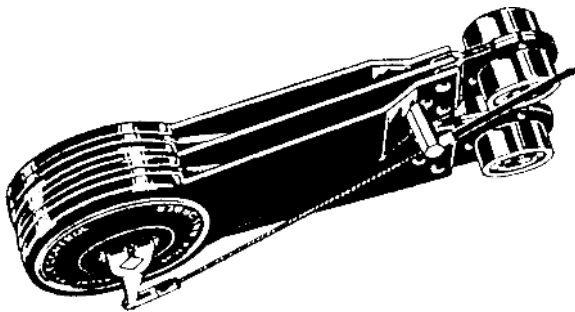


Figure 1.1.8 The Hartford Telecontrol damper was adjustable via a Bowden cable, and hence could be controlled easily from the driving seat, even with the vehicle in motion.

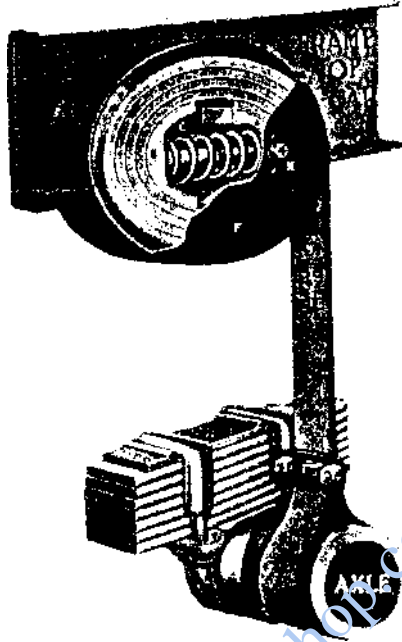


Figure 1.1.9 The Gabriel Snubber (1915) used a leather strap sprung metal or wooden blocks to give restraint in rebound only (from Simanaitis, 1976).

The early development timetable of dampers thus ran roughly as follows:

- 1901: Horock patents a telescopic hydraulic unit, laying the foundations of the modern type.
- 1902: Mors actually builds a vehicle with simple hydraulic pot dampers.
- 1905: Renault patents an opposed piston hydraulic type, and also patents improvements to Horock's telescopic, establishing substantially the design used today.
- 1906: Renault uses the piston type on his Grand Prix racing cars, but not on his production cars. Houdaille starts to develop his vane-type.
- 1907: Caille proposes the single-lever parallel-piston variety.

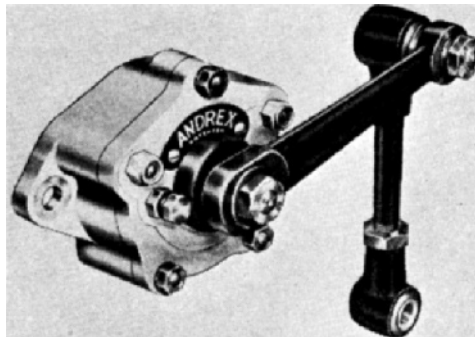


Figure 1.1.10 The Andrex multiple discs-in-oil-bath damper.

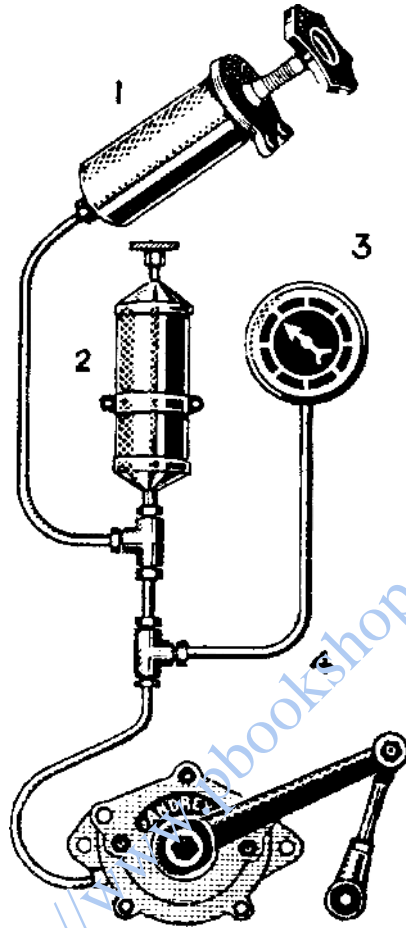


Figure 1.1.11 The adjustable version of the Andrex oil-bath damper included pump, reservoir and pressure gauge.

1909: A single-acting Houdaille vane type is fitted as original equipment, but this is an isolated success for the hydraulic type, the friction disc type remaining dominant.

1910: Oil damped undercarriages come into use on aircraft.

1915: Foster invents the belt 'snubber' which had great commercial success in the USA.

1919: Lovejoy lever-arm hydraulic produced in the USA.

1924: Lancia introduces the double-acting hydraulic unit, incorporated in the front independent pillar suspension of the Lambda. The Grand Prix Bugatti uses preloaded nonadjustable drum-brake type.

1928: Hydraulic dampers are first supplied as standard equipment in the USA.

1930: Armstrong patents the telescopic type.

1933: Cadillac 'Ride Regulator' driver-adjustable five-position on dashboard.

1934: Monroe begins manufacture of telescopics.

1947: Koning introduces the adjustable telescopic.

1950: Gas-pressurised single-tube telescopic is invented and manufactured by de Carbon.

2001: Magnetorheological high-speed adjustables introduced (Bentley, Cadillac).

The modern success of hydraulics over dry friction is due to a combination of factors, including:

- (1) Superior performance of hydraulics, due to the detrimental effect of dry Coulomb friction which is especially noticeable on modern smooth roads.
- (2) Damper life has been improved by better seals and higher quality finish on wearing surfaces.
- (3) Performance is now generally more consistent because of better quality control.
- (4) Cost is less critical than of old, and is in any case controlled by mass production on modern machine tools.

During the 1950s, telescopic dampers gradually became more and more widely used on passenger cars, the transition being essentially complete by the late 1950s. In racing, at Indianapolis the hydraulic vane type arrived in the late 1920s, and was considered a great step forward; the adjustable piston hydraulic appeared in the early 1930s, but the telescopic was not used there until 1950. Racing cars in Europe were quite slow to change, although the very successful Mercedes Benz racers of 1954–55 used telescopics. Although other types are occasionally used, the telescopic hydraulic type of damper is now the widely accepted norm for cars and motorcycles.

It was far from obvious in early days that the hydraulic type of damper would ultimately triumph, especially in competition with the very cost-effective Gabriel snubber of 1915. The first large commercial successes for the hydraulic types came with the vane-type, developed from 1906 onwards by Maurice Houdaille. The early type used two arms with a floating body, a little like the dry friction scissor damper. The later type still used vanes, but had a body mounted on the vehicle sprung mass, operated by an arm with a drop link to the leaf spring suspension, Figures 1.1.12–1.1.14.

The 1919 Motor Manual (UK, 21st edition) devoted less than one of its three hundred pages to dampers, suggesting that the damper was not really considered to be of great importance in those days, stating that:

These devices, of which there are a great number on the market, are made for the purpose of improving the comfortable running of the car, more especially on roughly-surfaced roads. The present system of springing is

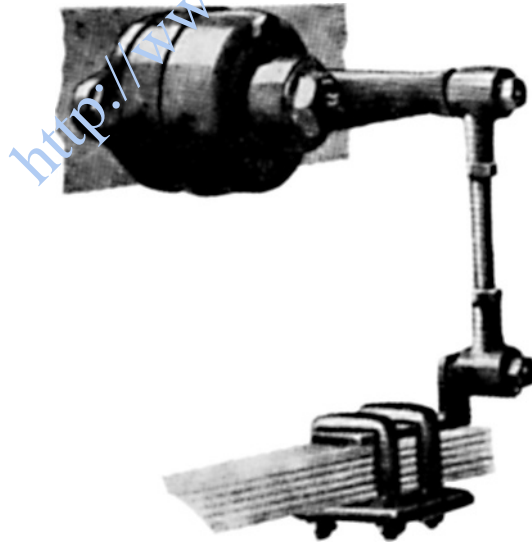


Figure 1.1.12 The Houdaille rotary vane damper, the first large quantity production hydraulic damper. This originated in 1909 and was double-acting from 1921.

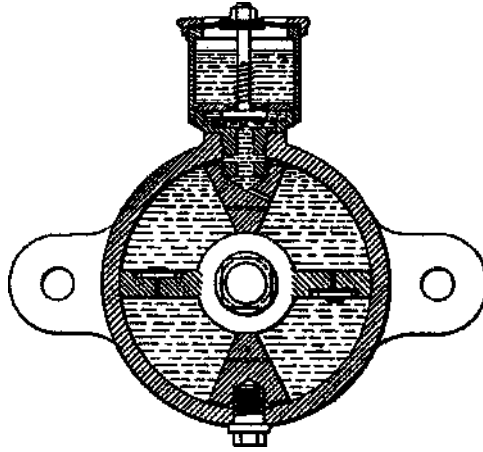


Figure 1.1.13 Cross-section of slightly different version of Houdaille rotary vane damper (from Simanaitis, 1976).

admittedly not perfect, and when travelling on rough roads there is the objectionable rebound of the body after it passes over a depression in the road, which it is desirable should be reduced as much as possible. The shock from this rebound is not only uncomfortable for the passengers, but it has a bad effect on the whole car. Hence these shock absorbers are applied as the best means available so far to check the rebound. They are made on various principles, generally employing a frictional effect such as is obtainable from two hardened steel surfaces in close contact. Another principle is that of using the fluid friction of oil, practically on the lines of any of the well-known dash-pot devices, viz., a piston moving in a cylinder against the resistance offered by the oil contained within it, the oil passing slowly through a small aperture into another chamber. This type of device is probably the best solution of the problem.

Up to 1920 hydraulic dampers were single acting, in droop only, but from 1921 a more complex valve system allowed some damping in bump too. At this point the operating characteristics of the

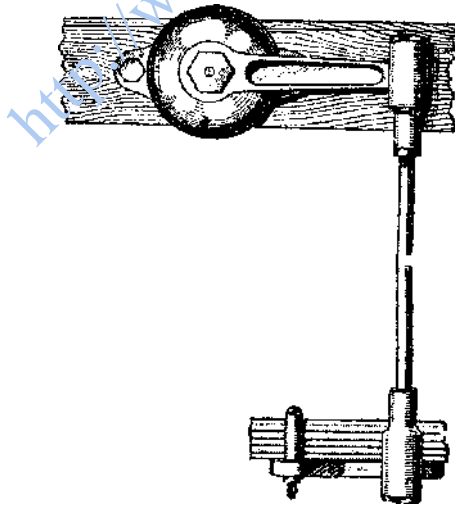


Figure 1.1.14 An early configuration of hydraulic damper, a rotary vane device with a drop arm to the axle. Note the wooden chassis rail (artist's impression, *The Motor Manual*, 1919).

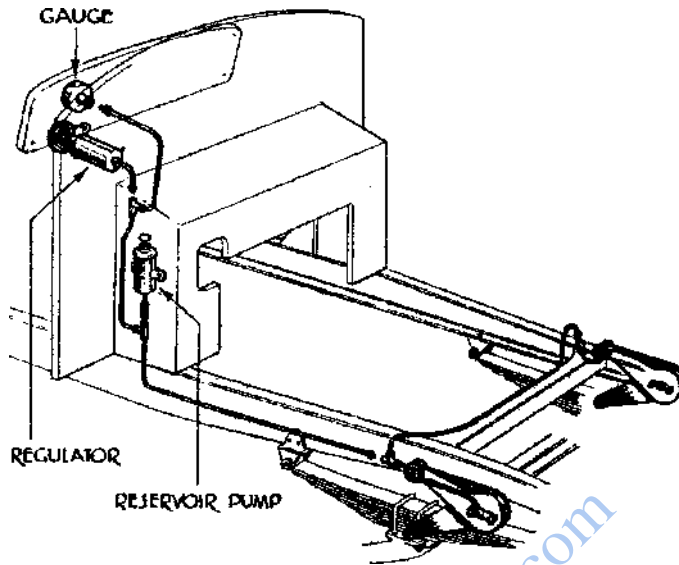


Figure 1.1.15 Layout of the hydraulically remote Andre Telecontrol damper, shown here on a front axle (The Motor Manual, 1939).

hydraulic damper had largely reached their modern form. More recent developments have had more to do with the general configuration, so that the lever-operated type has given way to the telescopic piston type which is cost-effective in manufacture, being less critical with regard to seal leakage, and has better air cooling, although lacking the conduction cooling of a body-mounted lever-arm damper. Most importantly perhaps, the telescopic type lends itself well to the modern form of suspension in terms of its mounting and ease of installation.

The 1939 Motor Manual (UK, 30th edition), devoted three pages to dampers, perhaps indicating the increased recognition of their importance for normal vehicles. An illustration was included of the Andre-Hartford dry friction scissor, and also one of the Luvax vane damper, shown later. There was also a diagram of the hydraulically adjusted, but dry action, version of the Andre Telecontrol system, as seen in Figure 1.1.15. That writer was moved to offer some additional explanation of damping and ‘shock absorbing’ in general, stating that:

Whatever form of springing is employed, it is always considered necessary to damp the suspension by auxiliaries, which have become known as shock absorbers. This term is unfortunate, because it is the function of the springs to absorb shocks, whereas the ‘shock absorbers’ serve the purpose of providing friction in a controlled form which prevents prolonged bouncing or pitching motions, by absorbing energy. A leaf spring is inherently damped by the friction between the leaves, and it may, therefore, seem strange that after lubricating these leaves friction should be put back into the system by the use of shock absorbers. The explanation is that leaf friction is not readily controllable, whereas the shock absorber imparts a definite and adjustable degree of damping to the system.

The most popular type of shock absorber is an hydraulic device which is bolted to the frame and is operated by an arm coupled to the axle. Four such devices are ordinarily fitted. When relative movement occurs between the axle and the frame, the arm on the shock absorber spindle is oscillated, and this motion is conveyed to a rotor, which fits within a circular casing. Oil in the casing is made to flow through valves from one side of the rotor to the other and so creates hydraulic resistance which damps the oscillations. In some cases the valves are arranged to give ‘double action’, the damping then being effective on both deflection and rebound. In other cases single-acting devices are used which can check rebound only. As a rule the action of the shock absorbers can be adjusted by means of a screw, which alters the tension of a spring and so varies the load on a ball valve.

The hydraulic shock absorber has the important merit of increasing its damping effect when subject to sudden movements, but suffers from the defect of providing very little resistance against slower motions, such as rolling. Consequently, for sports cars many users prefer frictional shock absorbers, of the scissor (constant resistance) type, of which the most famous is the Andre–Hartford.

The final comment above is significant in a modern context, regarding the preferred velocity–force relationship, which is a regressive shape with a ‘knee’, rather than simply linear.

The Lancia Lambda of 1925 had sliding pillar suspension, Figure 1.1.16, now almost extinct (except, e.g. Morgan) and regarded as primitive, but highly successful at the time. It was noted for the fact that its oil-filled cylinders required no maintenance, and was very reliable. This is an attractive option for a light vehicle, because it is such a compact and light system, although lacking the ability of modern suspensions to be adjusted to desired handling characteristics by detailed changes to the geometry.

Although dry friction snubbers remained in wide use through to the 1930s, hydraulic fluid-based dampers were in limited use from very early days and continued to grow in popularity. An early successful version in the USA was made by Lovejoy, Figure 1.1.17.

Difficulties with sealing and wear of vane lever arm types led to the lever arm parallel piston system as in the Lovejoy and in the Armstrong, Figure 1.1.18, in which the valve may also easily be made

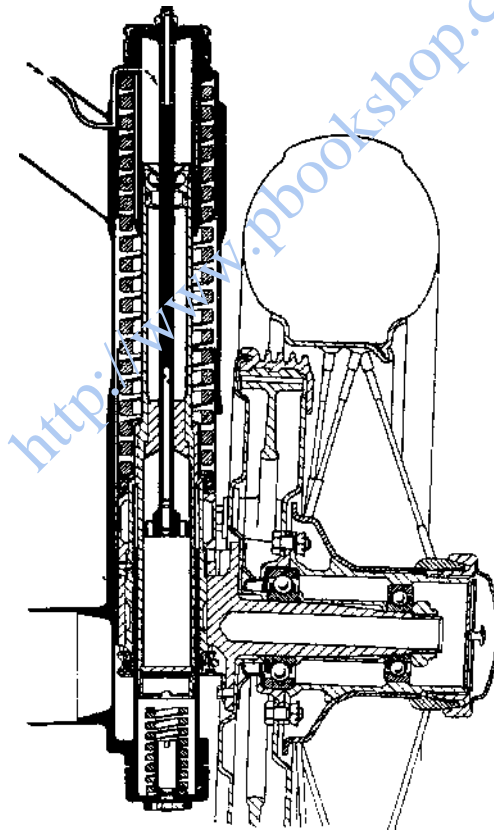


Figure 1.1.16 The Lancia Lambda sliding-pillar system had the spring and damper sealed into one unit (Lancia, 1925).

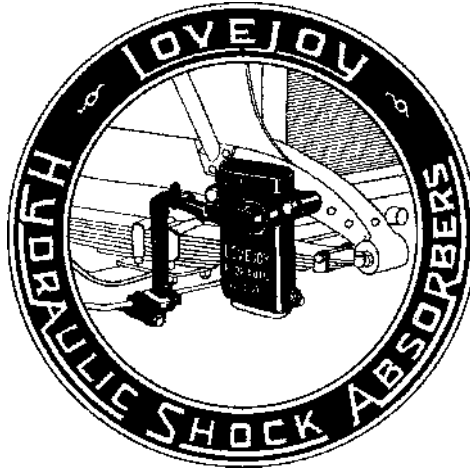


Figure 1.1.17 The Lovejoy lever-arm hydraulic damper, first produced in 1919.

interchangeable. This would still be a usable design today. Some economy of parts may be achieved by lengthening the bearing and using the lever as the load-carrying suspension arm, Figure 1.1.19. This can be taken further by putting the axle in double shear, so that the lever becomes an A-arm (wishbone), Figure 1.1.20.

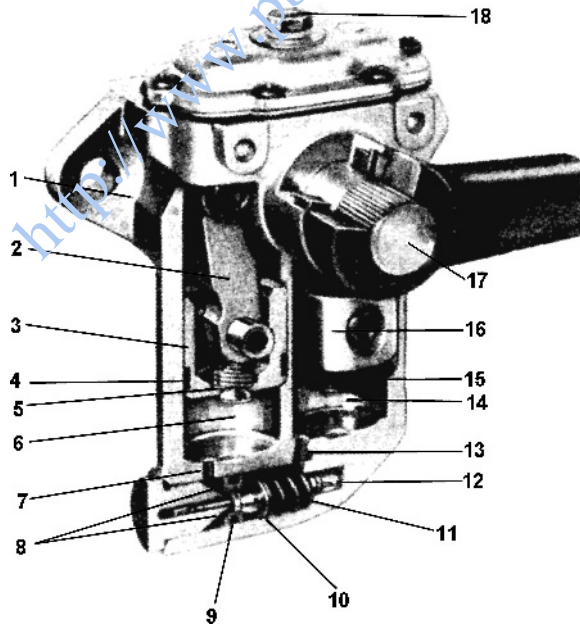


Figure 1.1.18 The double parallel-piston damper was the ultimate lever-arm configuration, overcoming the problems of the vane lever-arm type (Lucas) (see also Figure 1.3.7).



Figure 1.1.19 The simple lever-arm damper can be reinforced to carry suspension loads by lengthening the bearing rod.

However, despite the many creative innovations in lever arms, it seems that the telescopic is now almost universally preferred. At the front this has become the ubiquitous telescopic strut, partly because of the convenience of final assembly.



Figure 1.1.20 The A-arm (wishbone) suspension arm is lighter than a single arm when large loads are to be resisted, and adapts well to a double-shear connection to a lever-arm damper.

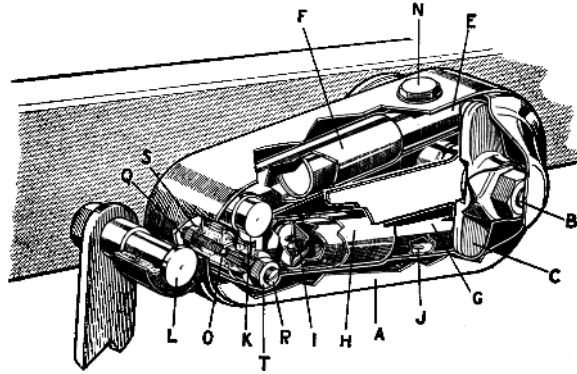


Figure 1.1.21 The Armstrong 'double telescopic lever arm' damper.

An interesting development was the Armstrong 'double telescopic lever arm', Figure 1.1.21, in which two telescopic dampers operate horizontally, fully immersed in an oil bath, with an external structure like a conventional lever arm type. Possibly this was done to combine the Armstrong-type telescopic into a unit that could be used interchangeably with its lever-arm competitors. An advantage of this layout over a plain telescopic is that any amount of damping is easily arranged in compression and rebound independently, with each damper of the pair acting in one direction only, without concern for oil cavitation.

As a final remark on the very early historical development, it may be noted that the dry friction scissor damper and the snubber were remarkably persistent. They were light in weight and low in cost, and perhaps more reliable than the early vane hydraulics which probably suffered from quality control problems and oil leakage. The parallel-piston lever-arm damper was functionally very good, and the fact that it has been superseded by the hydraulic telescopic, and the strut in particular at the front, is mainly due to the final assembly advantages of these, rather than any functional gain in the areas of ride and handling. In steering, the rack system has a better reputation than the old steering boxes, but it is hard, if not impossible, to tell the difference in practice. Similarly, the triumph of the telescopic damper system is not simply due to technical deficiencies of the older systems. The popular new direct acting telescopics that were ultimately to dominate were typified by the Woodhead–Monroe as in Figure 1.1.22.

1.2 Types of Friction

The purpose of a damper, or so-called 'shock absorber', is to introduce controlled friction into the suspension system. In this context, it is possible to identify three distinct types of friction:

- (1) dry solid friction;
- (2) fluid viscous friction;
- (3) fluid dynamic friction.

Any of these types may be used to give suspension damping, but their characteristics are totally different.

Dry solid friction between ordinary hard materials has a maximum shear friction force which is closely proportional to the normal force at the surface:

$$F_F \leq \mu_F F_N$$

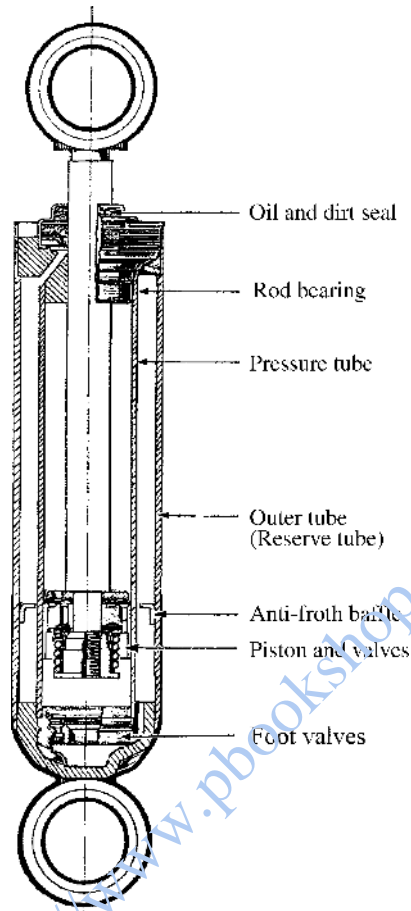


Figure 1.1.22 Cross-section of a typical telescopic damper showing the general features, shown without the dust shroud (Woodhead–Monroe).

where μ_F is the coefficient of limiting friction. For hard materials this is approximately constant over a good range of F_N , and relatively independent of the contact area. This is called Coulomb friction. However it is generally sensitive to temperature, reducing as this increases. Also it is sensitive to the sliding velocity in an undesirable way. For analysis it is common practice to consider there to be a static coefficient of friction μ_S available before any sliding occurs, and a dynamic value μ_D once there is relative motion. The dynamic value is lower, perhaps 70% of the static value.

Coulomb friction is undesirable in a suspension, provided that there is sufficient friction of desirable type, because it locks the suspension at small forces, and gives a poor ride on smooth surfaces, once known in the USA by the colourful term 'Boulevard Jerk'. Hence, nowadays, in order to optimise ride quality every effort is made to minimise the Coulomb friction, including the use of rubber bushes rather than sliding bushes at suspension pivot points.

Fluid friction is considered in detail in a later chapter, but basically viscous friction is proportional to the flow rate, and in this sense is an attractive option. Unfortunately, fluid viscosity is very sensitive to temperature. Fluid dynamic friction, arising with energy dissipation from turbulence, is proportional

to the flow rate squared, which is undesirable because it gives forces too high at high speed or too low at low speed. However it depends on the fluid density rather than the viscosity, so the temperature sensitivity, although not zero, is much less than for viscous damping.

Much of the subtlety of damper design therefore hinges around obtaining a desirable friction characteristic which is also consistent, i.e. not unduly sensitive to temperature. This is achieved by using the fluid-dynamic type of friction, with pressure-sensitive variable-area valves to give the desired variation with speed.

1.3 Damper Configurations

There have been numerous detailed variations of the hydraulic damper. The principal types may be classified as:

- (1) lever vane (e.g. Houdaille);
- (2) lever cam in-line pistons (e.g. Delco Lovejoy);
- (3) lever cam parallel pistons, (e.g. Delco);
- (4) lever rod piston (e.g. Armstrong);
- (5) telescopic.

These and some other types are further illustrated by the variety of diagrams in Figures 1.3.1–1.3.29.

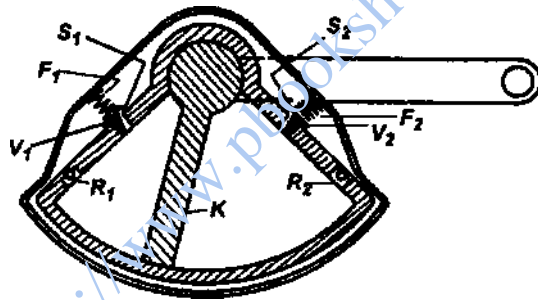


Figure 1.3.1 Double-acting vane type damper (Fuchs, 1933).

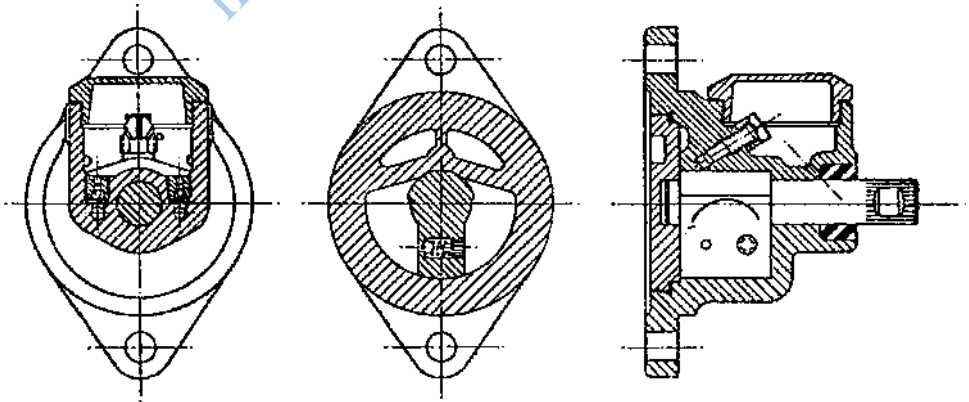


Figure 1.3.2 Early vane-type damper (Kinchin and Stock, 1951/1952).

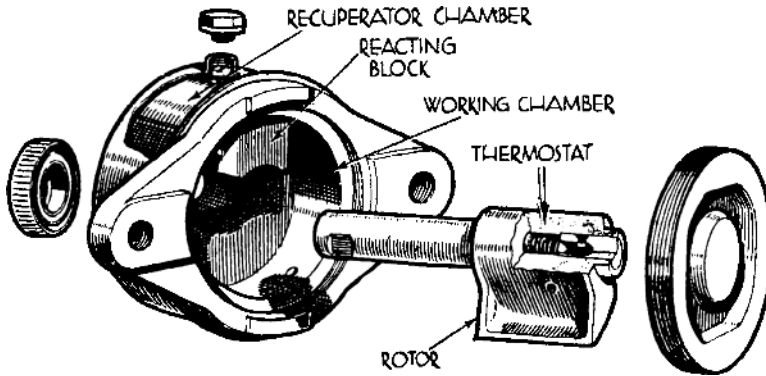


Figure 1.3.3 The Luvax rotary vane hydraulic damper, which featured thermostatic compensation of variation of oil properties. This was a genuine improvement on earlier vane types. The vane shape results in a radial force that takes up any freedom in the bearing in a way that minimises vane leakage (The Motor Manual, 1939).

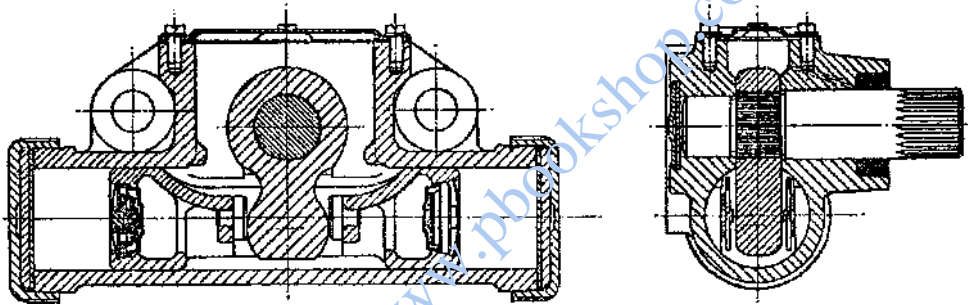


Figure 1.3.4 Lever-operated piston-type damper with discharge to recuperation space (Reproduced from Kinchin and Stock (1951) pp. 67–86 with permission).

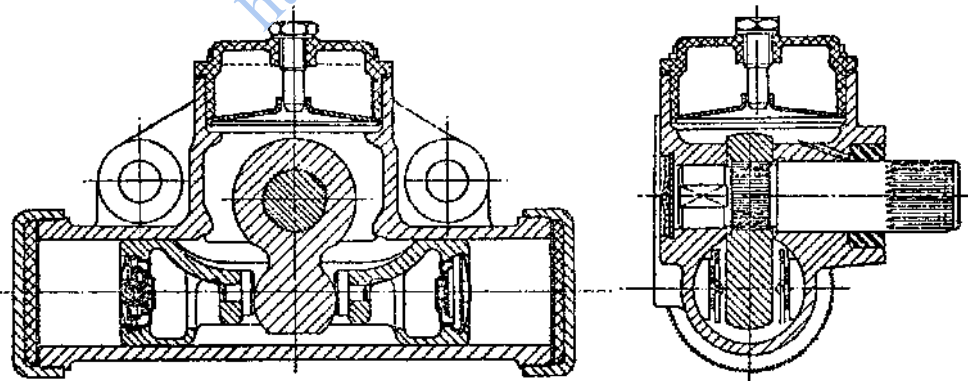


Figure 1.3.5 Lever-operated piston-type damper with pressure recuperation (Reproduced from Kinchin and Stock (1951) pp. 67–86 with permission).

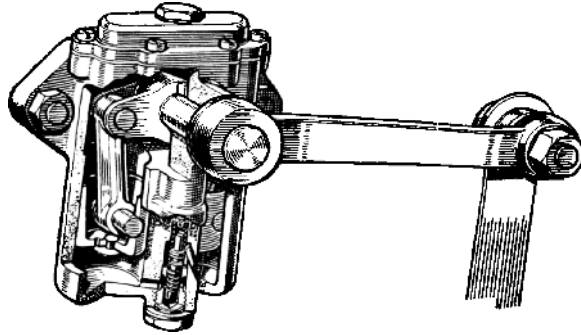


Figure 1.3.6 Double-piston lever-arm damper with removable valve (Armstrong).

Most passenger cars now have struts at the front. These combine the damping and structural functions, with an external spring. The main advantage, compared with double wishbones, is fast assembly line integration of pre-prepared assemblies. There are some disadvantages. The main rod must be of large diameter to give sufficient rigidity and bearing surface to accept running and cornering loads. The piston is subject to side loads, and must have a large rubbing area. These tend to add Coulomb friction. The top strut mounting must transmit the full vertical suspension force, so it is less easy to put a good compliance in series with the damper. The large dimensions mean larger oil flow rates and less critical valves, although wear may still be a problem in some cases.

Gas springing has been used for many years, two of the main exponents in passenger cars being Citroen and British Leyland/BMC/Austin/Morris. The gas is lighter than a metal spring, but requires containment. The damping function is then integrated with the spring units, as in Figure 1.3.22 *et seq.*

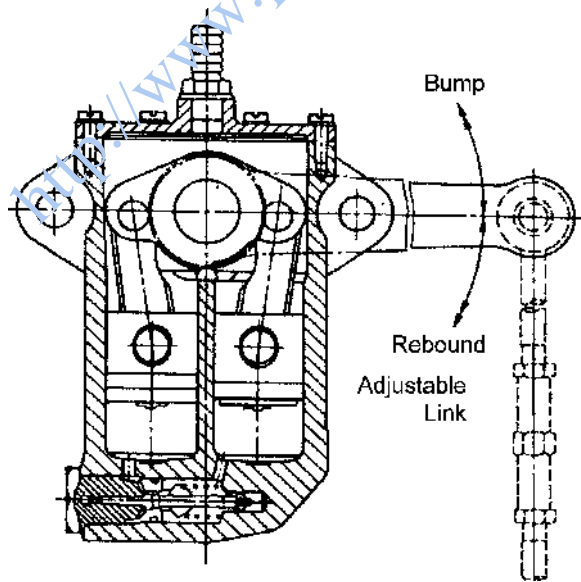


Figure 1.3.7 The classic lever-arm parallel piston type shown in engineering section, with different valve position. Reproduced from Komamura and Mizumukai (1987) *History of Shock Absorbers*, *JSAE*, 41(1), pp.126–131.

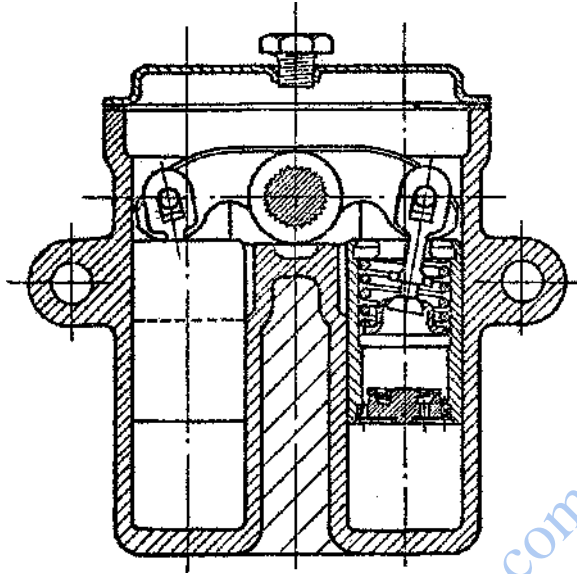


Figure 1.3.8 Lever-operated parallel-piston type damper with valves in the pistons (Reproduced from Kinchin and Stock (1951) pp. 67–86 with permission).

Front-to-rear interconnection allows reduction of the pitch frequency, which is particularly useful on small cars. BMC used simple rubber suspensions with separate dampers, and Hydrolastic and Hydragas with integrated damping.

The most common form of adjustable damper has a rotary valve with several positions each having a different orifice size. Some form of rotational position control, e.g. a stepper motor, is fixed to the top, controlling the piston valve through a shaft in the hollow rod, as seen in Figure 1.3.25. The more recent type uses magneto-rheological liquid, and is discussed separately.

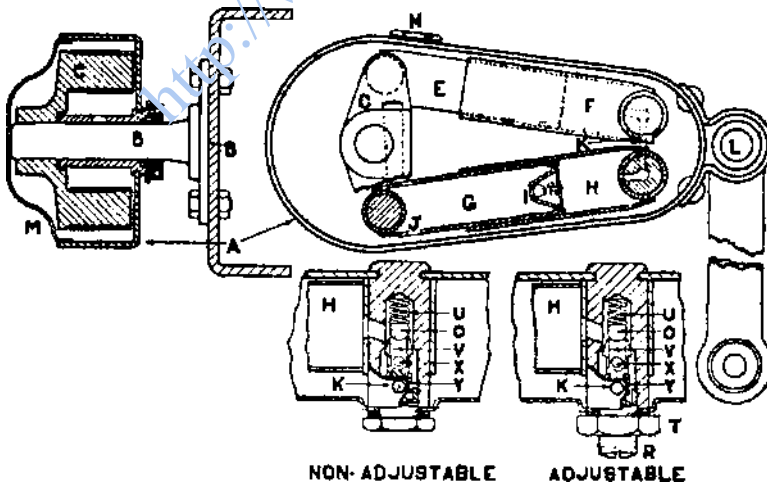


Figure 1.3.9 Double-telescopic lever-arm configuration showing details for standard fixed valve and for the manually adjustable *in situ* version (Armstrong).

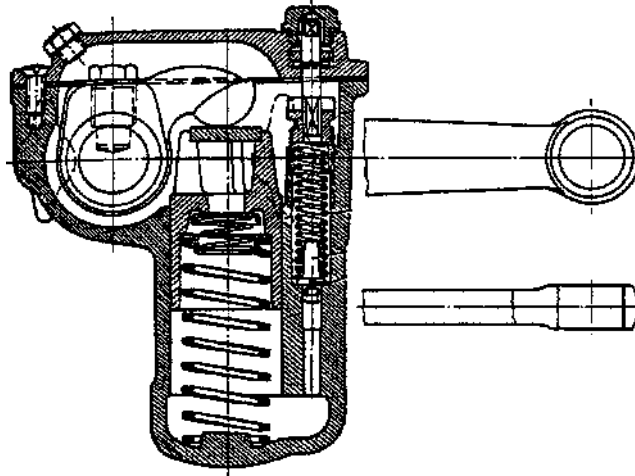


Figure 1.3.10 Single-acting lever-arm piston damper with easily changed valving (Fuchs, 1933).

Steering dampers are much smaller and lighter duty units, and usually operate in the horizontal position. Double tube dampers are not practical in this role. Figures 1.3.26 and 1.3.27 show two versions. In the first, the rod volume and oil thermal expansion are catered for by a spring-loaded free piston. In the second, there is an equalisation chamber having an elastic tube. This separates the oil and the gas, instead of a piston, reducing leakage problems.

In summary of vehicle damper types, then, the vane type is rarely used nowadays because the long seal length is prone to leakage and wear, and it therefore requires very viscous oil which increases the temperature sensitivity. The various lever and piston types are occasionally still used, but the construction implies use of a short piston stroke (in effect an extreme value of motion ratio),

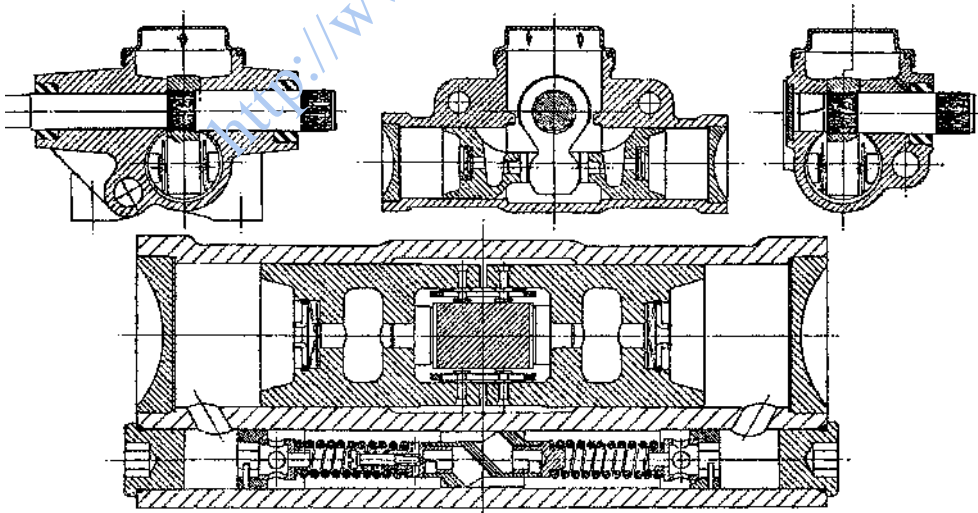


Figure 1.3.11 Lever-operated piston-type damper with end-to-end discharge (Reproduced from Kinchin and Stock (1951) pp. 67–86 with permission).

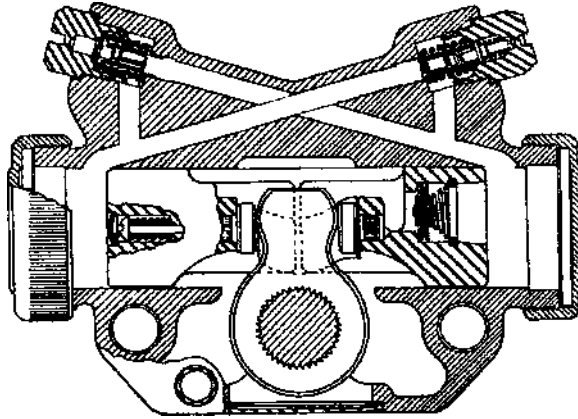


Figure 1.3.12 Double-acting lever-actuated damper with convenient alteration of characteristics by change of valve plugs, *ca* 1935 (Delco-Lovejoy).

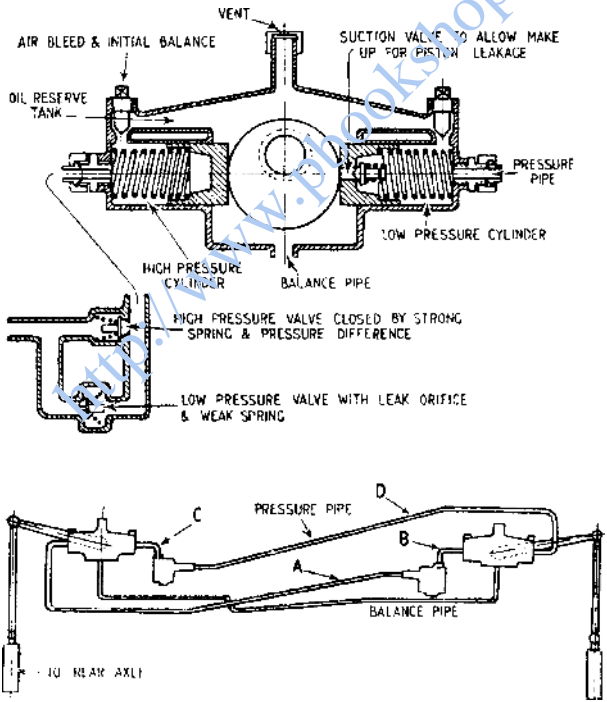


Figure 1.3.13 The German Stabilus damping system for commercial vehicles. Actuation was by the central eccentric circular cam, driven by a drop arm to each wheel. The two plugs at the top of each unit allow independent adjustment of bump and rebound forces. This forms a conventional independent system of unusual actuation. In addition, the two sides of an axle are interconnected through a balance pipe and by relief valves effective in roll only (asymmetrical action).

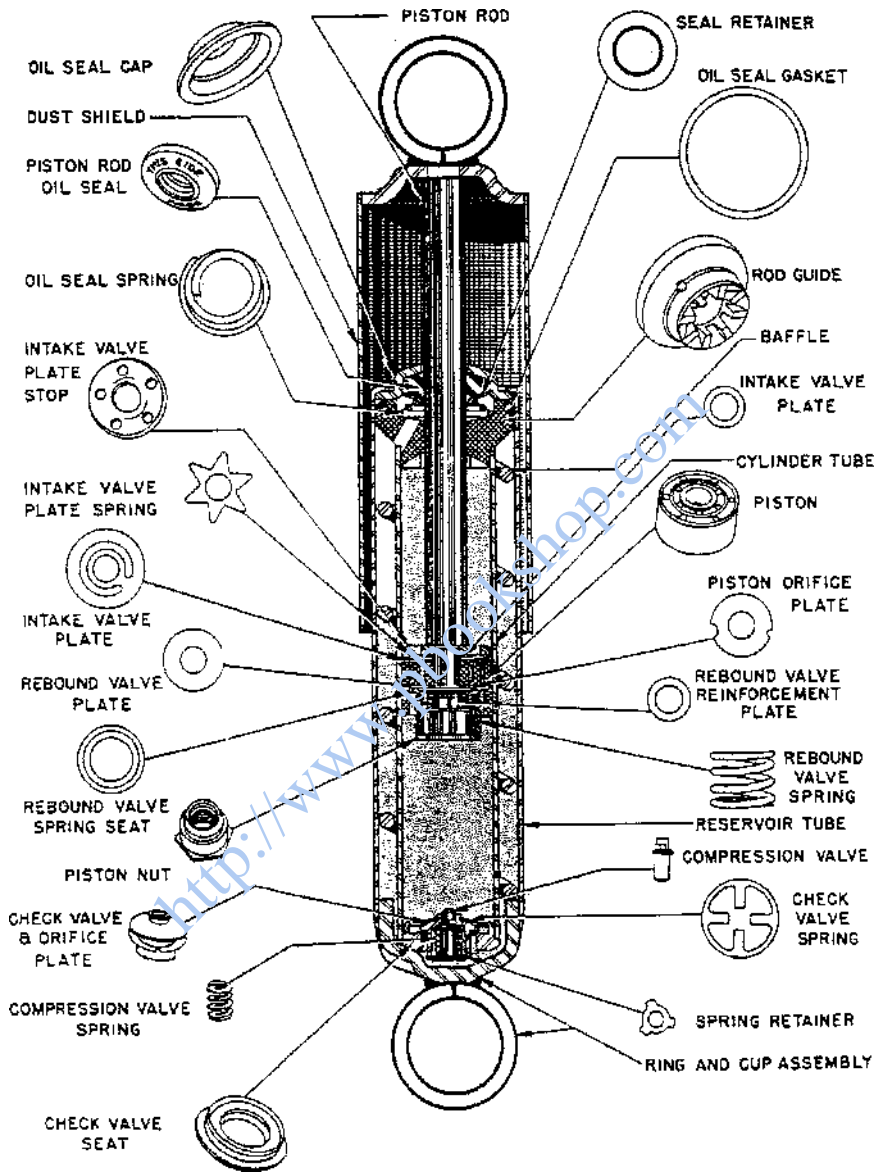


Figure 1.3.14 Parts of typical conventional telescopic damper of 1950. Note the four-coil air/oil separation rod in the reservoir to discourage the effects of agitation. Reproduced from Peterson (1953) *Proc. National Conference on Industrial Hydraulics*, 7, 23–43.

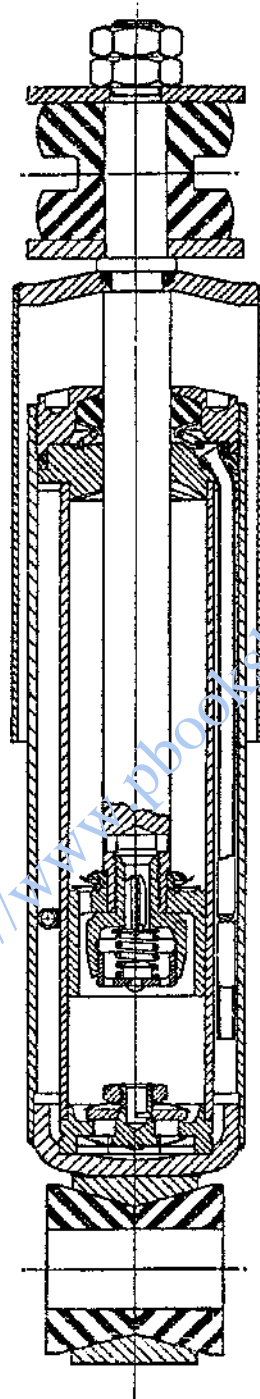


Figure 1.3.15 Standard form of direct-acting telescopic damper with double tubes (Reproduced from Kinchin and Stock (1951) pp. 67–86 with permission).

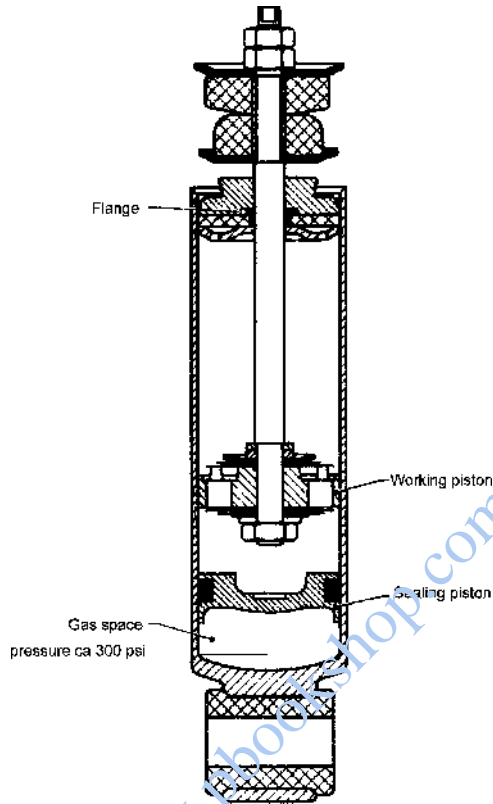


Figure 1.3.16 DeCarbon type of telescopic damper with floating secondary ('sealing') piston and high pressure in the gas chamber. The secondary piston must have sufficient free movement to accommodate the rod displacement volume and oil thermal expansion. A disadvantage is that the main rod seal is continuously subject to high pressure so good manufacturing quality is required to prevent long-term leakage. Also, in some applications (off road), the single tube is prone to damage.

so the forces and pressures need to be very high. Again this can create sensitivity to leakage. The lever types have the advantage that the damper body can be bolted firmly to the vehicle body, assisting with cooling. Another advantage is that there is no internal volume change due to the motion.

However, the lever type has now been almost entirely superseded by the telescopic type, which has numerous detail variations, and may be classified in several ways. The main classification concerns the method by which the insertion volume of the rod is accommodated. This is a major design problem because the oil itself is nowhere near compressible enough to accept the internal volume reduction of 10% or more associated with the full stroke insertion. Although this displacement volume seems to be a major disadvantage of the telescopic damper compared with the lever type, even the lever arm damper must allow for thermal expansion of the oil, which is significant, so the disadvantage in this respect is not great.

There are three basic telescopic types, as in Figure 1.3.28:

- (1) the through-rod telescopic;
- (2) the double-tube telescopic;
- (3) the single-tube telescopic.

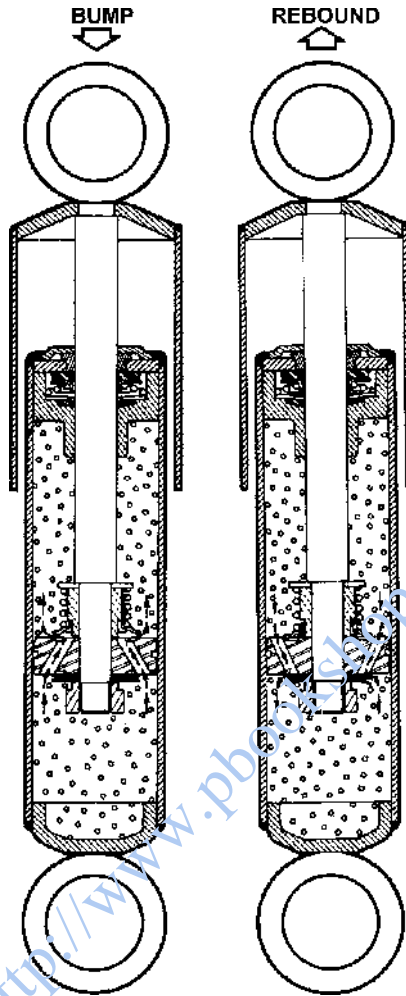


Figure 1.3.17 To eliminate the free piston, an emulsified oil may be used, distributing the expansion and rod-accommodation volume throughout the main oil volume. Overall length is reduced. On standing, the gas separates, but quickly re-emulsifies on action. The valves must be rated to allow for the passage of emulsion rather than liquid oil (Woodhead).

The through-rod telescopic avoids the displacement volume difficulty by passing the rod right through the cylinder. However this has several disadvantages; there are external seals at both ends subject to high pressures, the protruding free end may be inconvenient or dangerous, and there is still no provision for thermal expansion of the oil. However it is a simple solution which has the merit that it can be used in any orientation. This type has proved impractical for suspension damping, but is sometimes used for damping of the steering.

In the double-tube type of telescopic, a pair of concentric tubes are used, the exterior annulus containing some gas to accommodate the rod displacement volume. Hence it must be used the correct way up. In the single-tube type, some gas may be included, which normally forms an emulsion with the oil; alternatively the gas is separated by an independent floating piston (de Carbon type) as shown in

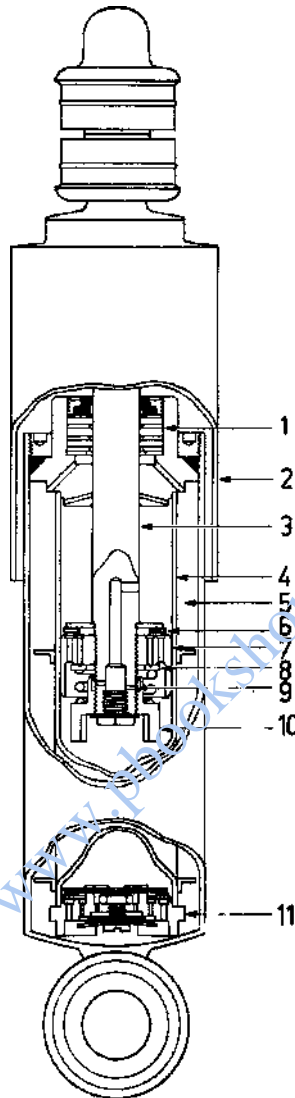


Figure 1.3.18 A double-tube damper showing various features. 1 seal; 2 shroud; 3 rod; 4 inner cylinder; 5 annular foot and gas chambers; 6 piston compression valve; 7 piston; 8 extension valve; 9 parallel hole feed; 10 adjuster; 11 foot valve. The adjuster screws along the rod compressing the valve preload spring. This is achieved by fully compressing the damper to engage the adjuster in the bottom of the cylinder, and rotating the two body parts (Koni).

Figure 1.3.28 (c). The rod is usually fitted with a shroud, of metal or plastic, or possibly a rubber boot, to reduce the amount of abrasive dirt depositing on the rod, which otherwise may cause premature seal wear.

Any internal pressure acts on the rod area to give a suspension force, normally lifting the vehicle. Such pressurisation is avoided as far as possible in the double-tube damper, which minimises leakage.

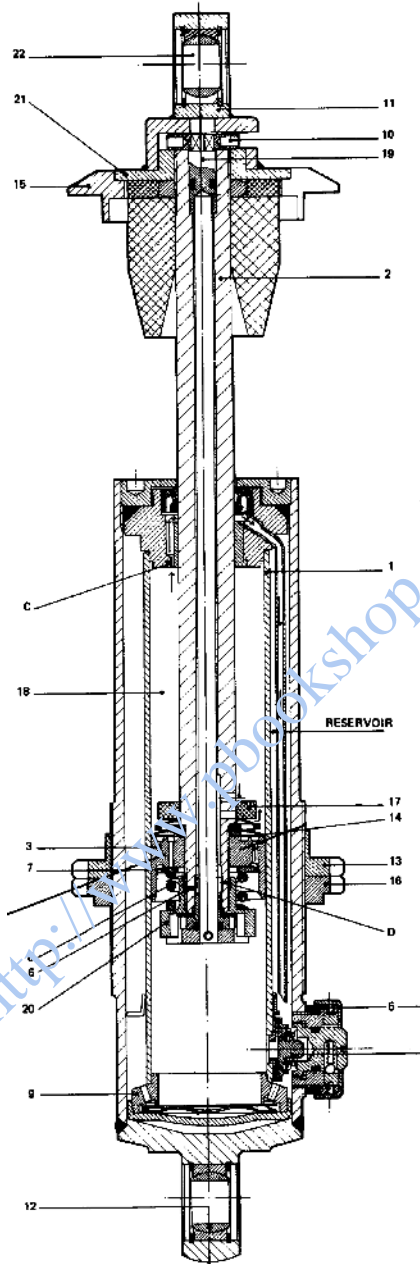


Figure 1.3.19 Detailed section of an adjustable double-tube racing damper. 1 inner cylinder; 2 hollow rod; 3 piston; 5 foot valve adjuster; 7 spring seat; 8 valve preload spring; 9 foot sintering with extension feed hole; 10 extension adjustment point; 11 mounting; 12 lower ball joint; 13 adjustable spring perch; 14 compression shim pack; 15 upper spring seat; 16 lower spring perch lock; 17 compression shim pack backing; 18 extension chamber; 19 extension adjustment member; 20 extension valve spring seat; 21 upper fixture; 22 upper ball joint (Koni).

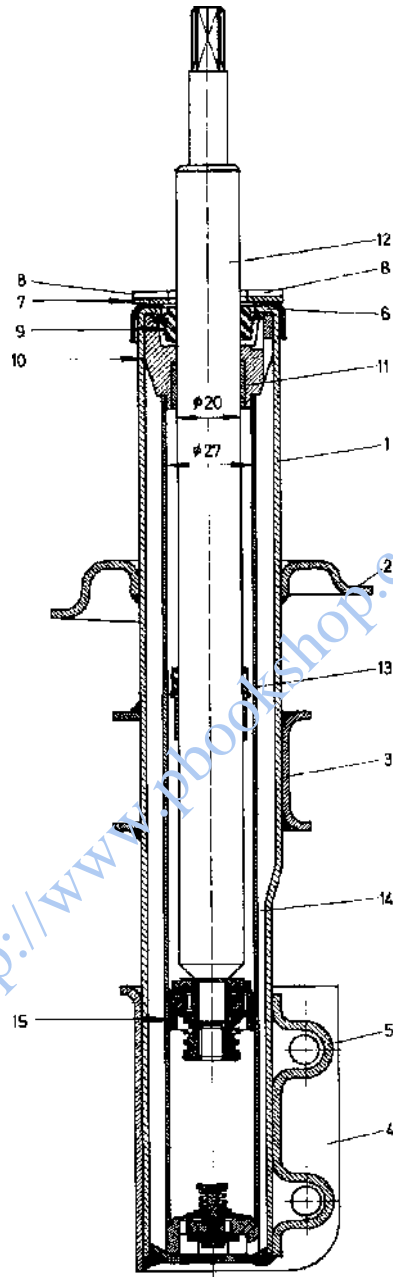


Figure 1.3.20 Sectional view of a front strut for a small car. Piston diameter 27 mm, rod diameter 20 mm. 1 outer cylinder; 2 spring seat; 3 guard; 4, 5 wheel hub fixture; 6 rolled closure; 7, 8 bump stop seat; 9 seal; 10 upper moulding; 11 bearing; 12 rod; 13 stroke limiter (?); 14 inner (working) cylinder; 15 piston (Fiat/Monroe).

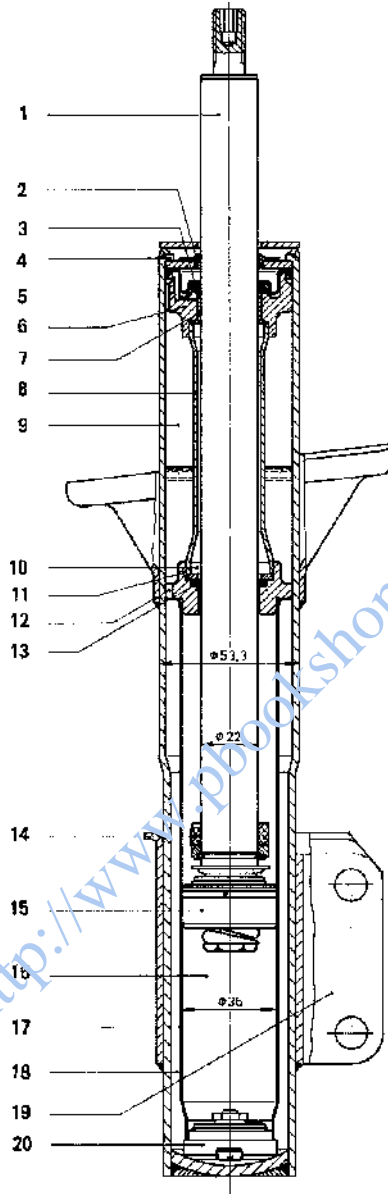


Figure 1.3.21 Sectional view of a front strut for a larger car. Piston diameter 36 mm, rod diameter 22 mm. 1 rod; 2 seal; 3 bush; 4 rolled closure; 5 bush; 6 top moulding; 7 bearing bush; 8 sleeve; 9 gas chamber; 11 centre moulding; 12 hole; 13 seal; 14 impact guard; 15 piston; 16 compression chamber; 17 inner (working) cylinder; 18 annular foot chamber; 19 wheel hub fixture; 20 foot valve (Boge).

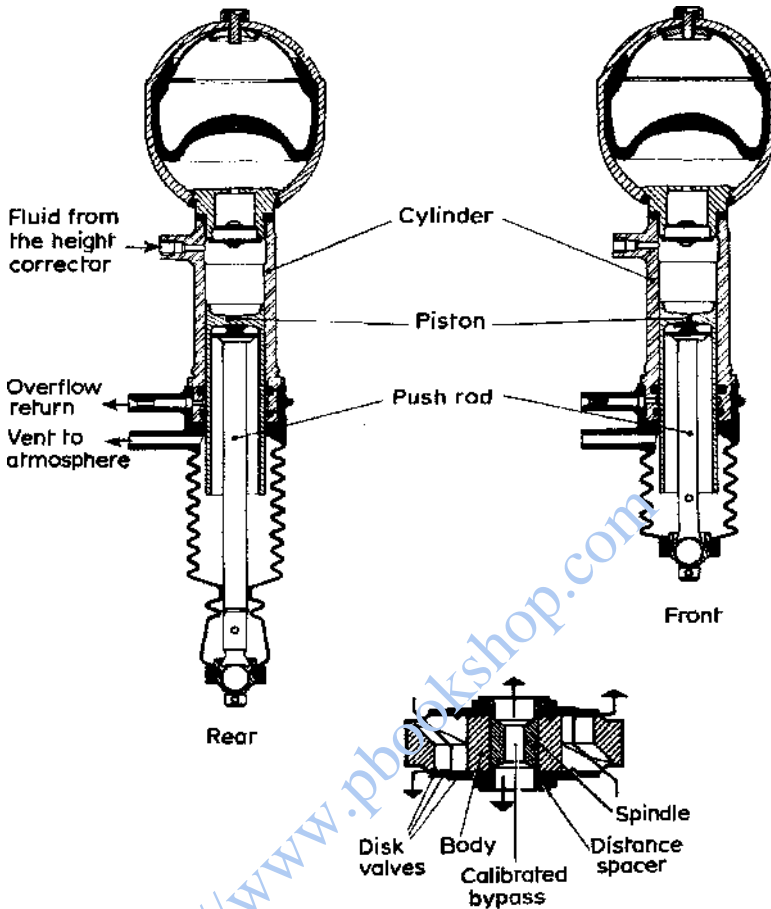


Figure 1.3.22 Citroen air suspension. The valve, with two shim pack valves, is fixed in position. It is not in the piston, which ideally would pass no oil. The gas is held in the elastomeric rolling seal bag. Nitrogen is used, reducing oxidation ageing of the rubber.

The pressurised single-tube type may suffer from loss of pressure with failure of correct function in compression due to cavitation behind the piston.

Suda *et al.* (2004) have proposed a nonhydraulic EM (electromagnetic) damper, of general configuration as in Figure 1.3.29. Actuation of the EM damper rotates the ball screw nut which drives an electrical generator through a planetary gearbox. An alternative arrangement uses a rack and pinion for the mechanical drive. The obvious advantage of an EM damper is controllability—the damper force depends on the generator and its electrical load. An external power supply is not needed, because the damper can generate its own electrical supply. A suggestion that energy from suspension motions can usefully be recovered to save fuel seems optimistic, as the average damper power dissipation is only a few watts for the whole vehicle. The Suda prototype successfully demonstrated appropriate characteristics with a damping coefficient around 1.6 kNs/m, and was tested with encouraging performance on the rear of a passenger car. The concept is an interesting alternative to ER and MR dampers, but it remains to demonstrate its life and manufacturing economics.

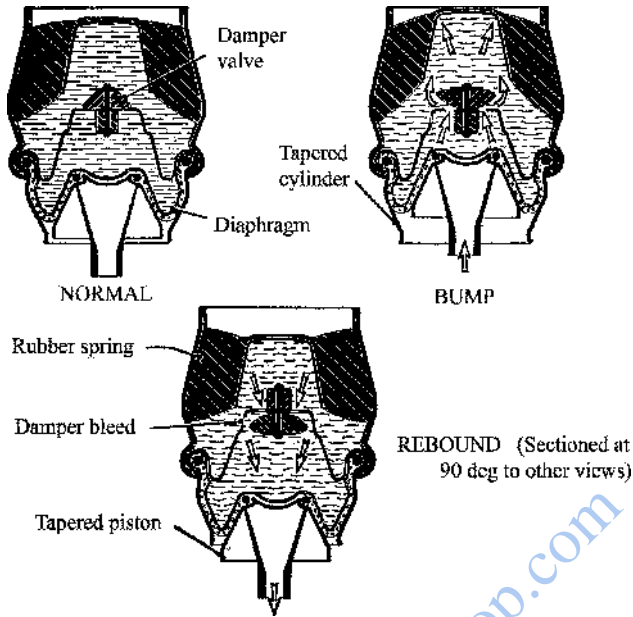


Figure 1.3.23 The BMC Hydrolastic system had a somewhat similar operation to the Citroen gas system, but replaced the gas spring by a rubber-in-shear spring already well proven. Based on Campbell, C., *Automobile Suspensions*, Chapman and Hall, 1981.

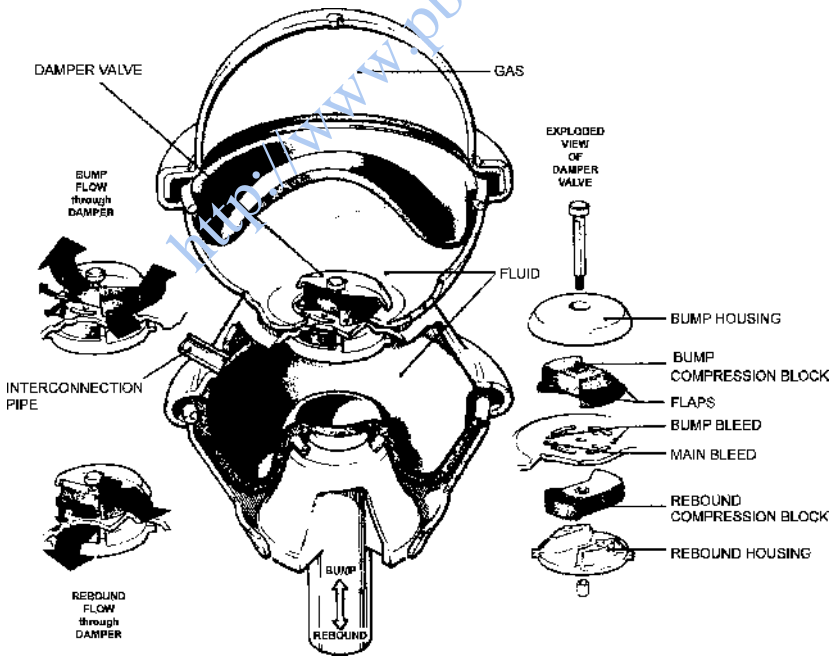


Figure 1.3.24 BMC Hydrolastic was replaced by Hydregas, a lighter system.

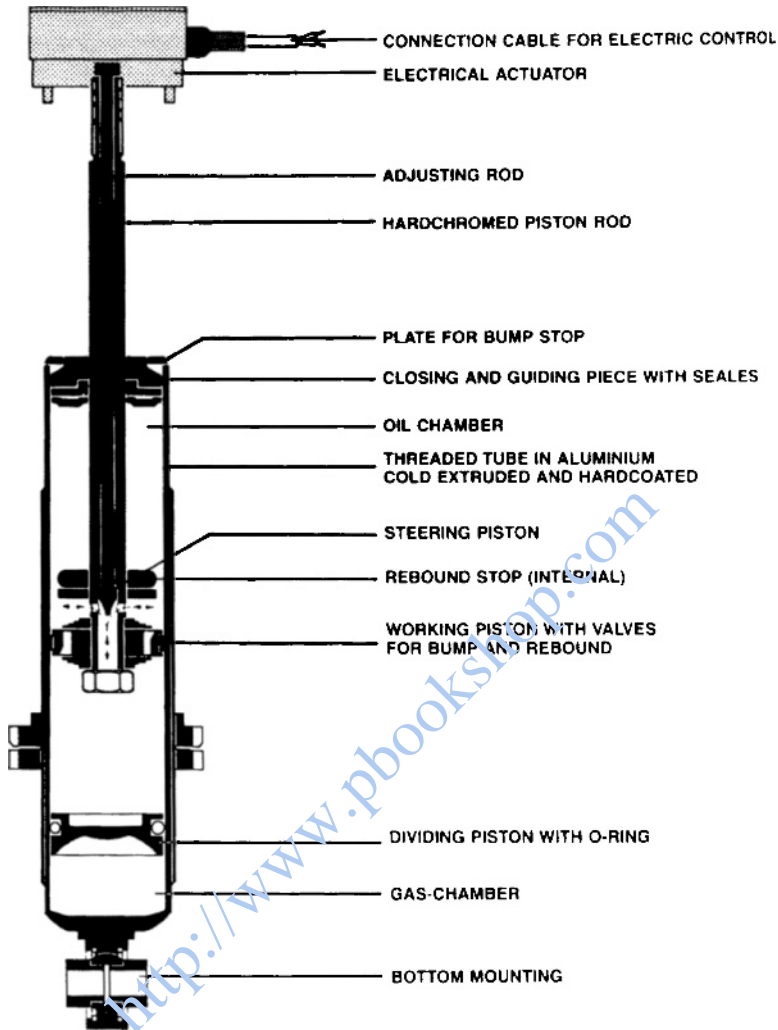


Figure 1.3.25 An electrically controlled adjustable damper. In this example, the basic construction is a single tube de Carbon configuration (Bilstein).

1.4 Ride-Levelling Dampers

One common problem with vehicles is that the load variation is a significant fraction of the kerb weight, perhaps 40%, particularly for small cars. This causes variation of the suspension performance with load condition. Many efforts have been made to overcome this. The most basic factor is the ride height, which varies, in particular at the rear. The telescopic damper offers the obvious possibility of making compensating adjustments to restore the ride height by simple pressurisation, Figures 1.4.1–1.4.5 illustrate some efforts along these lines. The operation of a self-levelling system can be very slow acting without detriment, so the pump may be very low power. It

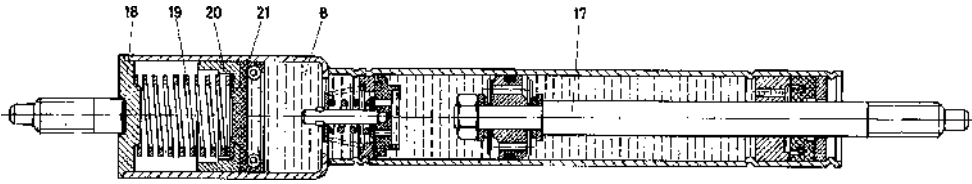


Figure 1.3.26 Steering damper of basic de Carbon layout, having a free piston separating the oil from the gas chamber, but with spring assistance. The piston has two shim packs. The foot valve has a coil spring blow-off valve (Stabilus).

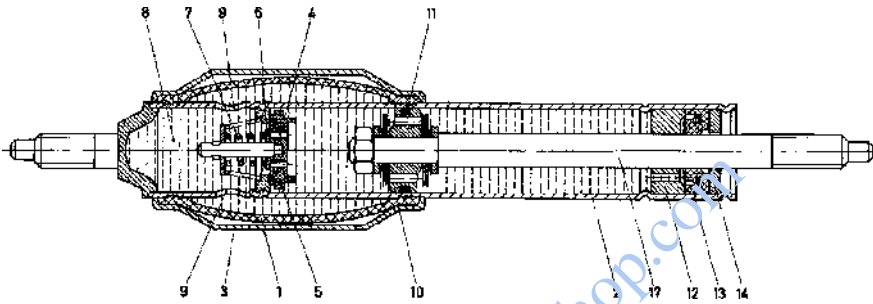


Figure 1.3.27 With similar internals to the last example, this steering damper uses a rubber oil/gas separator, achieving somewhat shorter overall length (Stabilus).

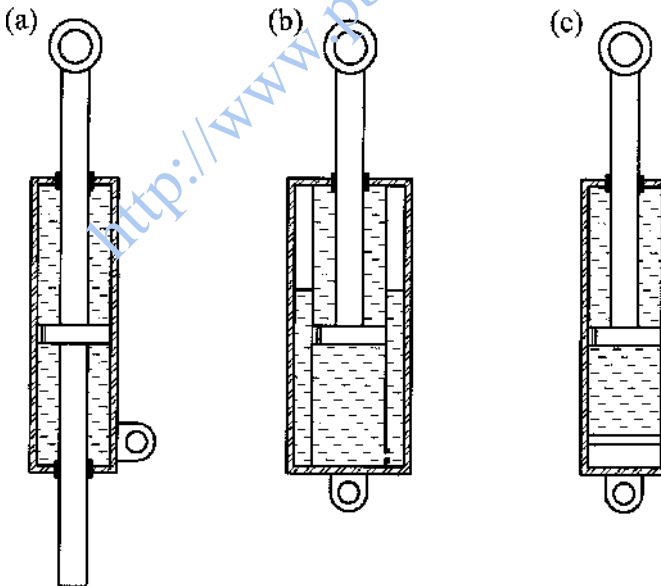


Figure 1.3.28 Basic types of telescopic damper: (a) through-rod; (b) double-tube; (c) single-tube (with floating piston).

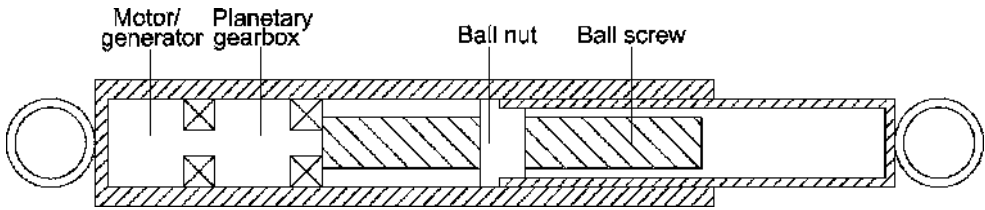


Figure 1.3.29 Electromagnetic damper configuration (after Suda *et al.*, 2004).

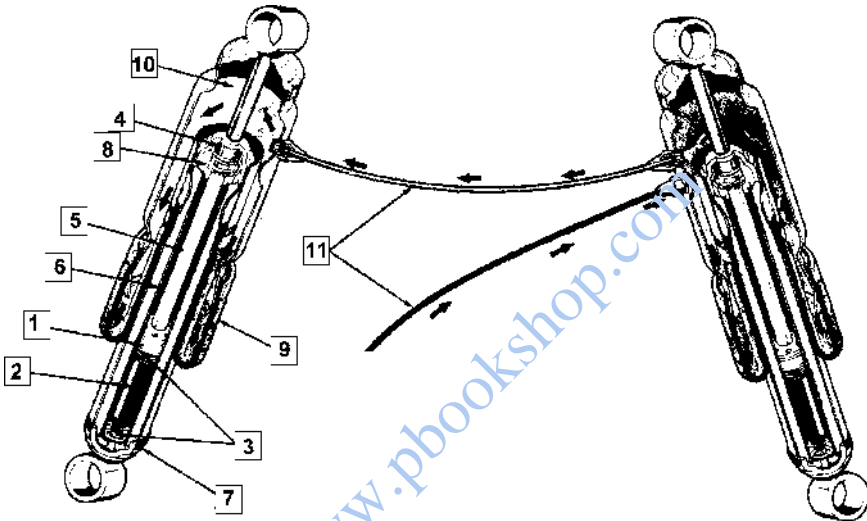


Figure 1.4.1 With an upper gas chamber, as in most dampers, the provision of a pump, a control system and a few pipes allows ride height compensation at modest cost (Delco).

is even possible to use the damper action when in motion to pump the damper up to a standard mean position.

1.5 Position-Dependent Dampers

Ordinary passenger cars have so far rarely used dampers with designed position dependence (other than indirectly, through the effect of the rubber mounting bushes), although they have been widely used on motorcycles and aircraft undercarriages. Figure 1.5.1 shows an example motorcycle front fork in which the sometimes problematic dive under braking is controlled by an internal bump stop which closes an orifice. This greatly softens the impact and allows weaker springing with improved ride quality. This basic method of position dependence by the sliding of a tapered needle in a hole to vary an orifice area has been used for aircraft undercarriages. In the Telefork the further feature is added that the rather short and blunt rubber 'needle' C entering orifice A can distort under pressure.

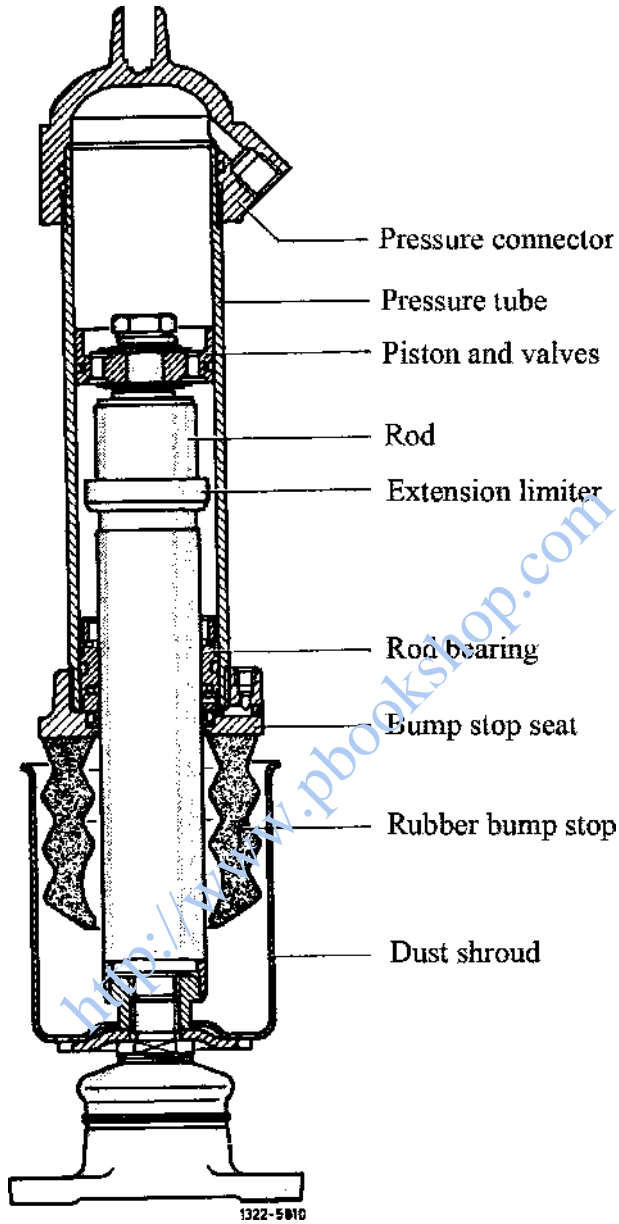


Figure 1.4.2 Ride height control can also be achieved on struts (DB 380).

Figure 1.5.2 shows a bus and truck damper in which extension is limited by the entry of the supplementary piston into the top cap, with hydraulic restraint.

Position or stroke dependence of damper force has recently become a development topic, and is discussed in detail later.

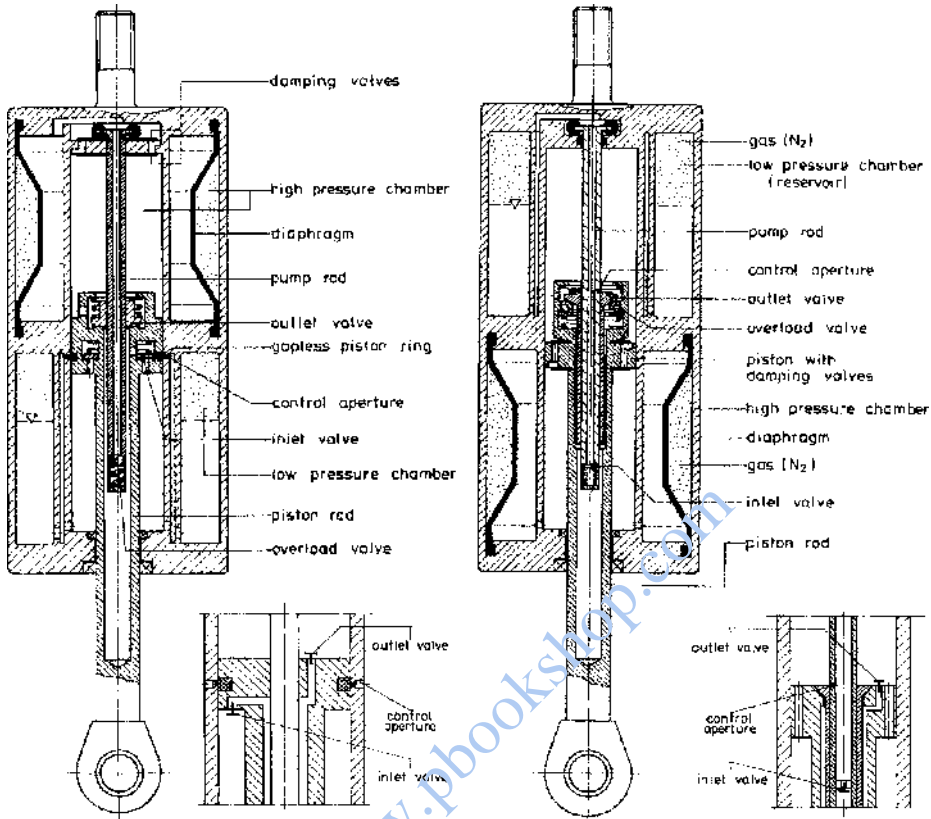


Figure 1.4.3 Boge Hydromat (left) and Nivomat (right) height-adjusting dampers.

1.6 General Form of the Telescopic Damper

A general form of the telescopic damper is shown in Figure 1.6.1, where there is a separate reservoir (chambers 0 and 1). Chamber 0 contains air, possibly pressurised, separated by a floating piston from chamber 1. Chamber 2 is called the compression chamber, at high pressure during compression, and chamber 3 is called the extension, expansion or rebound chamber, at high pressure during extension. During compression, fluid is displaced from the main cylinder (chamber 2 and 3) into the reservoir, through a restriction of given characteristics, the compression foot valve. By the Principle of Fluid Continuity, in normal noncavitating operation, neglecting compressibility, the quantity of fluid displaced through the foot valve is equal to the volume of the piston rod entering the main cylinder. During compression, fluid also passes through the piston from chamber 2 to chamber 3, through the piston compression valve. During damper extension, removal of the piston rod from the main cylinder requires a flow from the reservoir into the main cylinder, through the foot extension valve. Also, fluid passes through the piston extension valve, valve 4, from chamber 3 to chamber 2.

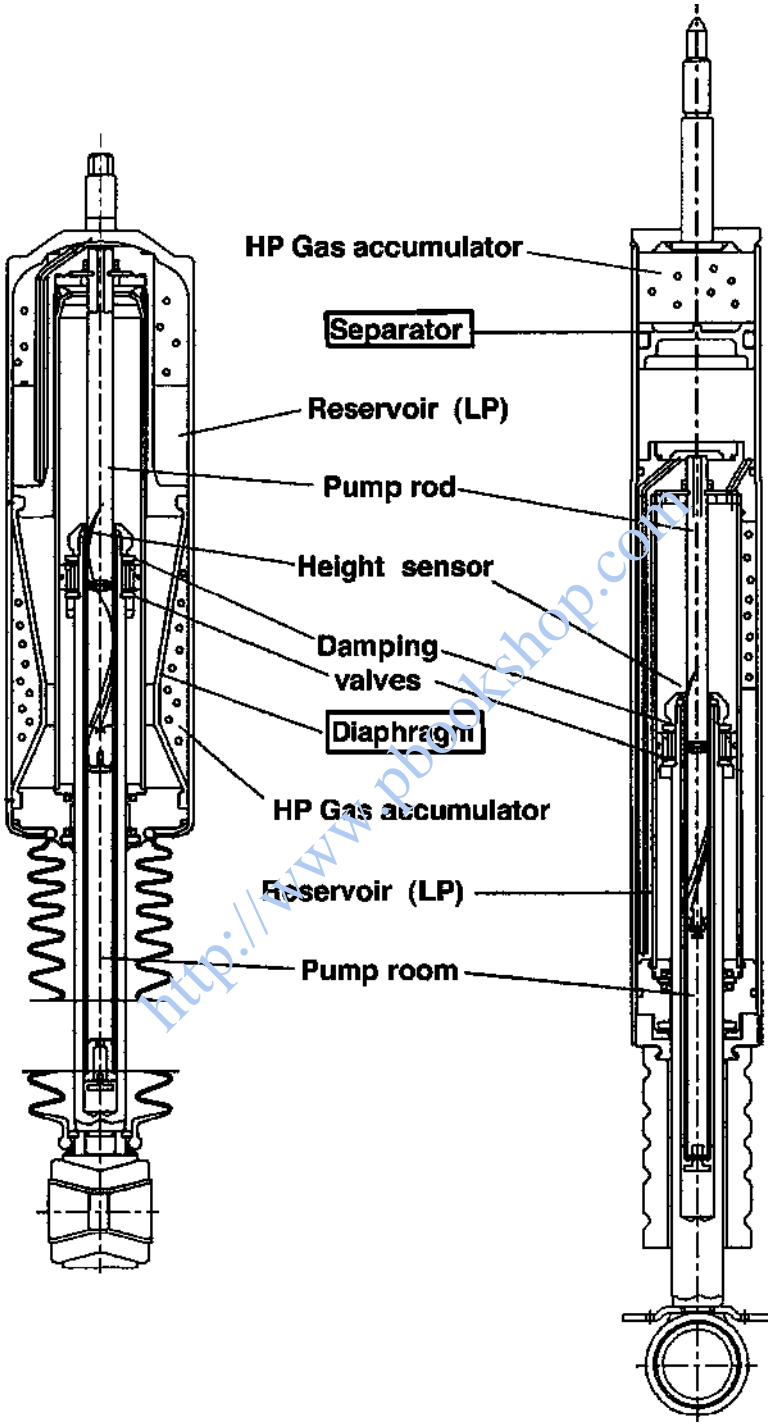


Figure 1.4.4 Mannesman Sachs height adjusting (type 1).

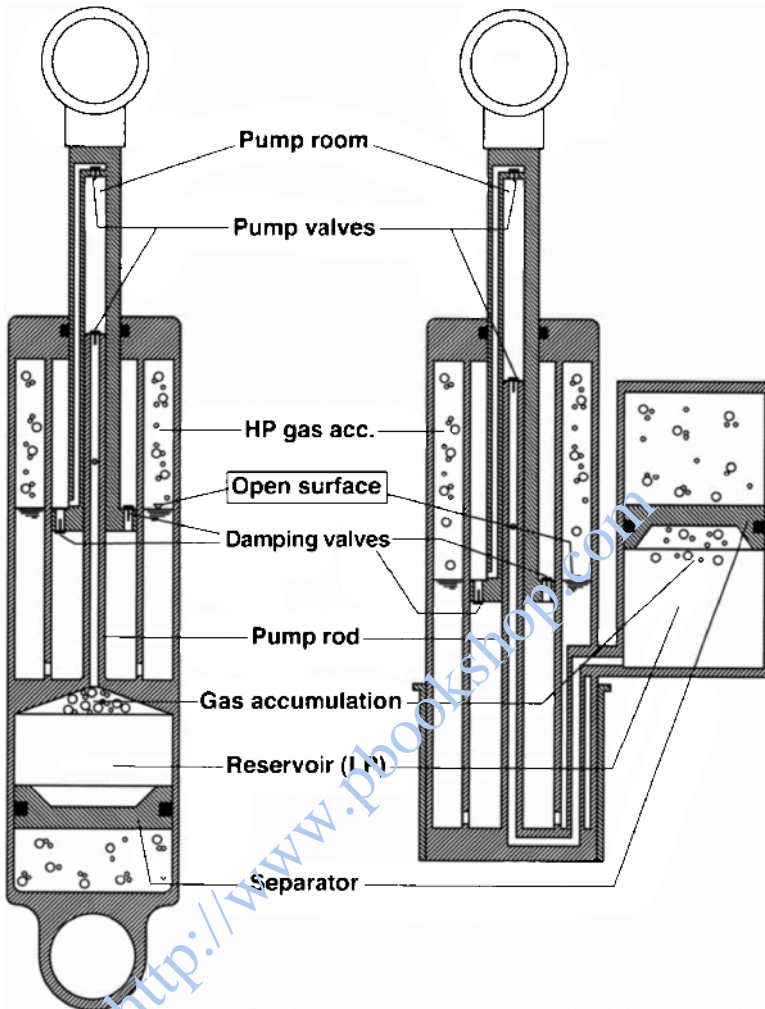


Figure 1.4.5 Mannesman Sachs ride-height adjustables (type 2).

The four valve flow rates are therefore:

- (1) Q_{PE} piston valve in extension,
- (2) Q_{PC} piston valve in compression,
- (3) Q_{FE} foot valve in extension,
- (4) Q_{FC} foot valve in compression.

These are denoted by the normal directions of flow. In extreme operation (with significant compressibility or cavitation), the actual flow direction through the valves may be momentarily abnormal, so, for example, there could be a positive flow Q_{PE} in the piston extension valve at a moment when the damper is actually compressing, immediately after reversing direction.

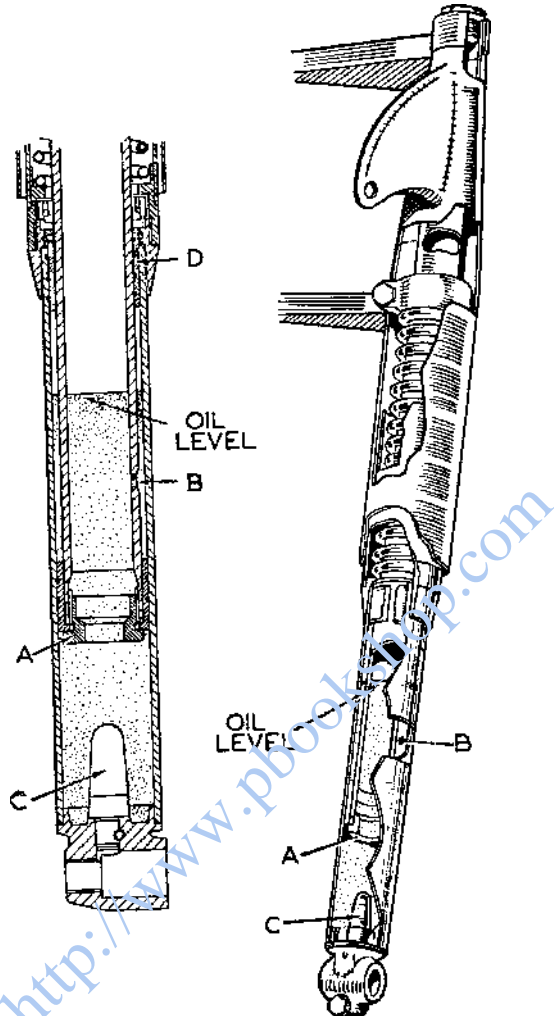


Figure 1.5.1 The BSA Telefork motorcycle spring-damper unit.

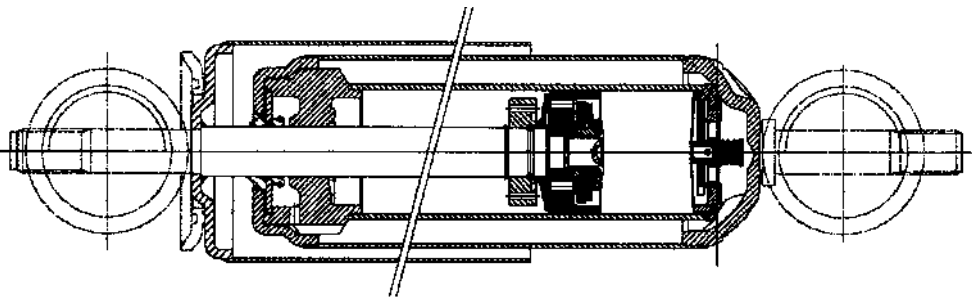


Figure 1.5.2 Truck damper with hydraulic extension limiter (Duym and Lauwerys, 2001).

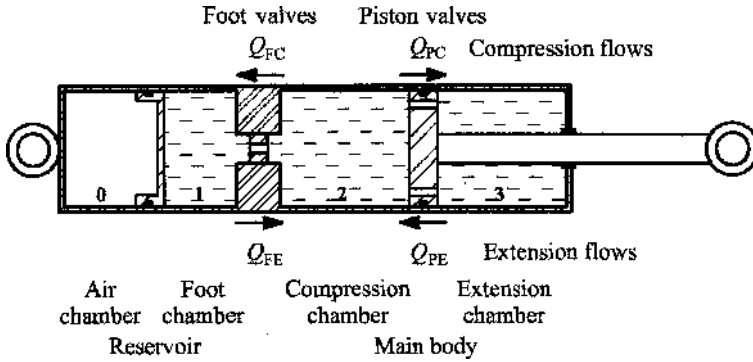


Figure 1.6.1 General form of telescopic damper.

The basic damper characteristics therefore depend on the piston and rod diameters, and on the characteristics of the four valves. Additional factors are the reservoir pressure (affecting cavitation) and the fluid properties: density, viscosity, temperature, vapour pressure, gas absorption, emulsification and compressibility, and so on. Further factors are friction at the rod bearing and seal, at the main piston and at the reservoir piston, and also leakage paths, e.g. from chamber 3 to the reservoir in many practical implementations (e.g. double-tube type).

In a real damper, the layout may actually be as shown in Figure 1.6.1, usually with the foot valve flow passing through a flexible pipe joining chambers 1 and 2; such remote reservoir dampers are used in racing, but are rare on passenger vehicles. With the layout of Figure 1.6.1, a pressurisation of 1.5 MPa might be used, with the foot compression valve providing extra pressure in the main cylinder to prevent cavitation during compression. (1 MPa, one megapascal, is about 10 atmospheres or 150 psi.) Alternatively, the reservoir may be incorporated directly into the main cylinder (in effect, the foot valves 1 and 2 then having zero resistance), it then being essential to have a high pressure, e.g. 3 MPa, to produce the required flow through valve 3 during compression without cavitation in chamber 3. In such cases a floating piston may still be used. Alternatively, this may be omitted, the gas being allowed to mix freely giving an emulsion, requiring appropriate valve calibration.

Another common configuration is to place the reservoir concentrically around the main cylinder giving the double-tube damper, usually unpressurised. This is less good for cooling, but the working inner cylinder carrying the piston is better protected against impact damage from flying stones. This type depends on gravity to separate the contained air and liquid, so it must be used with the cylinder underneath and the piston rod above, unless the gas is separated physically from the liquid, e.g. by an air bag in the reservoir. A spiral insert in the reservoir, or spiral rolling of the outer tube, can help to minimise mixing of air and oil. Extremely violent conditions can cause emulsification and loss of damping force, but this is not normally a problem for passenger cars. An advantage of this type is that it is possible to include a small annular chamber between the rod bearing (inside) and seal (outside), connected to the reservoir. The rod seal is then subject to the reservoir pressure only. The valving is arranged so that in normal operation, even in compression, the extension chamber pressure is always greater than the reservoir pressure, so the fluid leakage through the bearing (clearance 0.02–0.05 mm) is always outwards, circulating the fluid and hence improving cooling. This also prevents air being drawn back into the extension chamber (3), and ensures that any aeration gas brought through the piston valve collects at the top and is passed back to the outer reservoir.

Releasing the leaked fluid into the gas region greatly encourages emulsification, so a drain tube may be used to carry it down into the liquid at the bottom of the reservoir. This also ensures that, should there be a low extension chamber pressure causing reverse flow from the reservoir air chamber (0) into

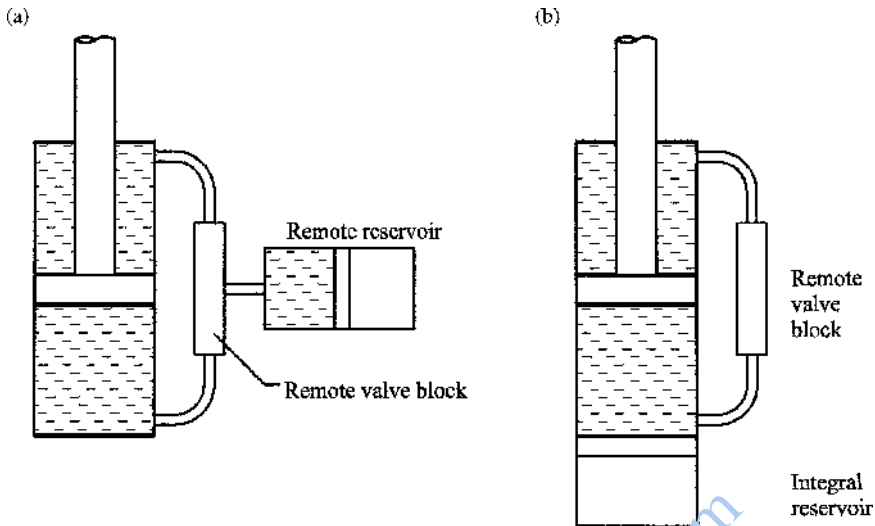


Figure 1.6.2 Two possible systems with remote valves: (a) with remote reservoir; (b) with integral floating piston.

the extension chamber (3), liquid rather than air is drawn back in. Otherwise, gas can pass very rapidly through the bearing bush, much more rapidly than liquid, with subsequent loss of damping function.

Numerous other configurations are possible, especially where one or two remote reservoirs are used. It is possible to use the piston for displacement only, with no piston valve, passing the liquid through an external circuit, giving excellent access for adjustability of the valves. This can be arranged, for example, with a remote reservoir, as in Figure 1.6.2(a), or with an integral secondary piston, as in Figure 1.6.2(b), or using emulsified gas. This concept can usefully be extended by adding oil take-off points near to the middle of the stroke, allowing position sensitivity with an extra flow and lower force when the piston is near to the centre position. This is known to offer a better combination of ride and handling. With external valves this can be controlled properly, instead of just having a bypass channel.

Adaptive dampers typically have rotary barrels with several holes of different sizes to adjust the resistance. These are of similar general configuration to the ones described above. The electrorheological variable damper uses a distinct configuration described later. The magnetorheological types are typically similar to the de Carbon floating secondary piston type, and, again, are considered in more detail later.

1.7 Mountings

To discourage the transmission of small-amplitude higher-frequency vibrations (noise, vibration and harshness) the ends of a damper are each mounted through a rubber bush (with the exception of a few very specialised cases such as racing cars). These bushes also permit some rotational motion and misalignment of the mounting points. The two principal configurations of end mounting are:

- (1) axially threaded rod, as in Figure 1.7.1 (and 1.3.10);
- (2) transverse eye, as in Figure 1.7.2 (and 1.3.10).

The former type is ideal for mounting directly into the top of a wheel arch. It is therefore frequently used on conventional passenger cars at the rear, and also for the top of front strut suspensions. In the

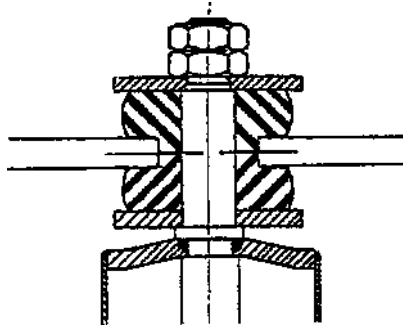


Figure 1.7.1 Axially threaded rod mounting (Reproduced from Kinchin and Stock (1951) pp. 67–86 with permission).

latter case, the bottom mounting of the strut is usually a ball joint, forming the lower defining point of the steering axis. The threaded rod is easily formed by machining the end of the damper rod or by inserting a stud.

The transverse eye of Figure 1.7.2. uses a concentric rubber bush with a bolt through. Preferably it is mounted in double shear, but frequently it simply uses a single bolt or stud into the side of the wheel upright or into the suspension arm. Figures 1.7.3–1.7.5. show some other common types of mounting.

The effect of the rubber mounting bushes is to put a nonlinear compliance in series with the damper, giving the complete unit a characteristic which depends upon the displacement amplitude, for a test at a given velocity amplitude. Small-amplitude motions with high frequencies are more readily met by bush compliance, hence reducing the transmission of such motions. More substantial movements relating to deflections of the suspension in handling movements (roll and pitch), or gross suspension movements in ride at the sprung mass natural frequency, are little affected by the bushes because their small compliance is effective for only a small deflection. The bushes are therefore important in introducing some stroke sensitivity to the transmitted forces, keeping the damping high for large amplitudes, as found in handling motions such as roll in corners, and high for large body motions on rough roads, but desirably reducing the damping at small amplitudes to improve the ride on smooth roads.

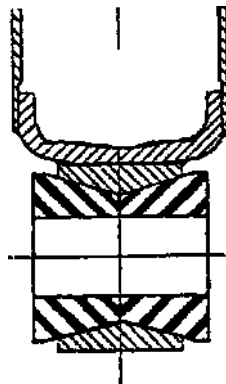


Figure 1.7.2 Transverse eye or integral sleeve mounting (Reproduced from Kinchin and Stock (1951) pp. 67–86 with permission).

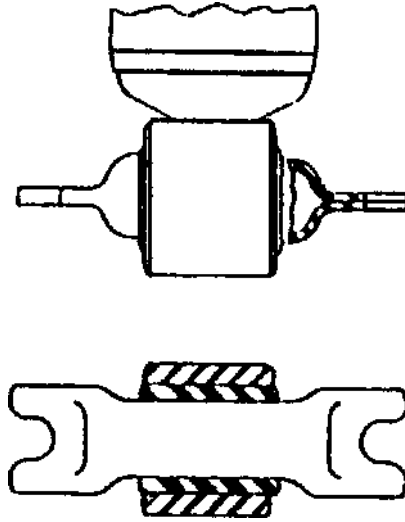


Figure 1.7.3 Integral bar mount (Jackson, 1959).

The axial-rod mounting lends itself to an axially asymmetrical form of compliance bushing, as in Figure 1.7.6 and 1.7.7. The asymmetry may be achieved by differing thickness, area or material properties, and differing axial preload distance. With a deflection such that one of the two bushes has expanded to reach axial freedom, that bush then contributes zero further stiffness. If the preload is small, the essential result is that the stiffness is different for damper compression and extension, a feature that can be turned to advantage, particularly on strut suspensions.

The basic damper characteristics are normally considered to be those when the mountings are rigid, not soft bushed, and that is how they are normally tested.

Struts require a more robust mounting than dampers alone, as shown by the examples in Figure 1.7.8 *et seq.* For front struts there must also be provision for steering action at this point.

It is advantageous to separate the seat force exerted by the spring and the damper. This is natural in many suspension designs, but not always automatic. Figure 1.7.9 above shows how the separation may be achieved. This is known as a 'dual-path mounting'. Figure 1.7.12 gives a further example.

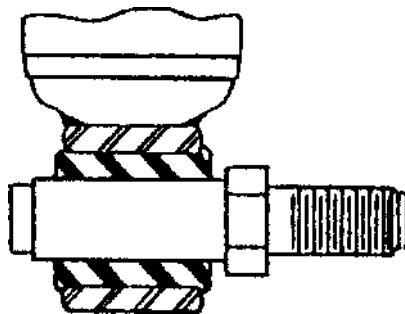


Figure 1.7.4 Integral stud mount (Jackson, 1959).

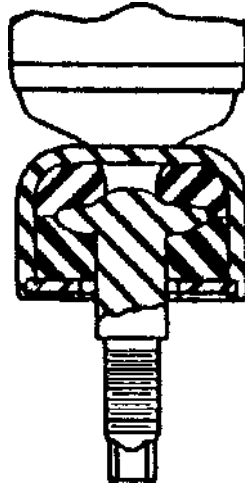


Figure 1.7.5 Integral bayonet stud mount (Jackson, 1953).

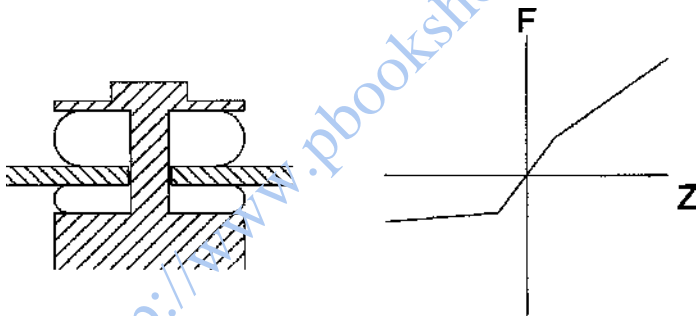


Figure 1.7.6 Asymmetrical type of damper or strut bushing.

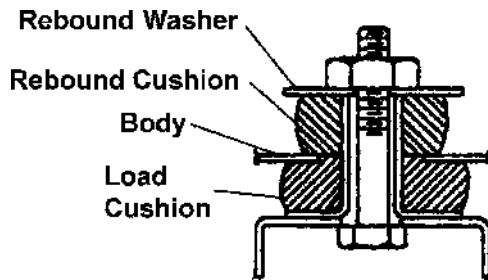


Figure 1.7.7 Asymmetrical damper bushing (Puydak and Auda, 1966).

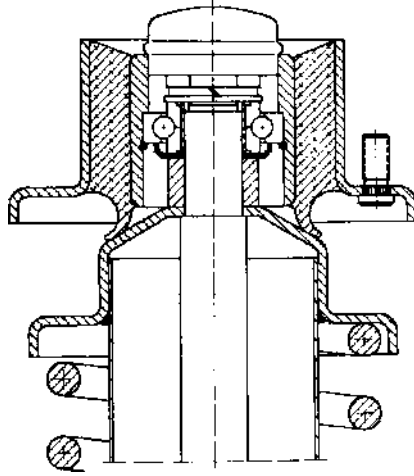


Figure 1.7.8 Strut top mounting (INA).

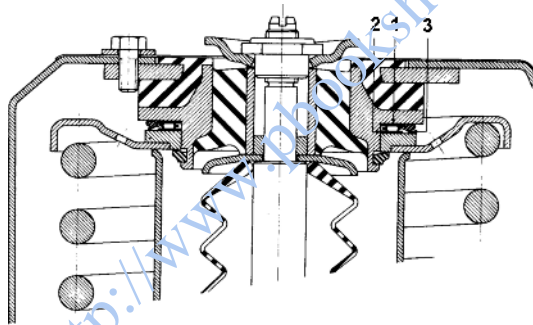


Figure 1.7.9 Strut top mounting (Peugeot).

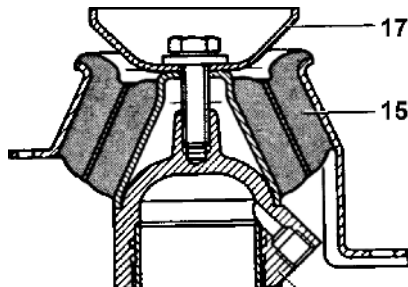


Figure 1.7.10 Strut top mounting (Daimler-Benz 380).

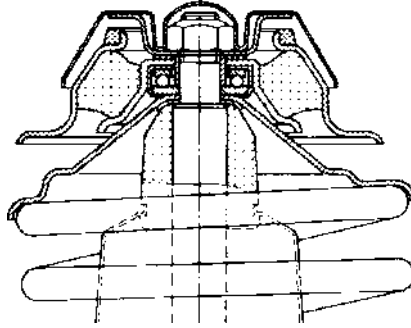


Figure 1.7.11 Strut top mounting (Audi).

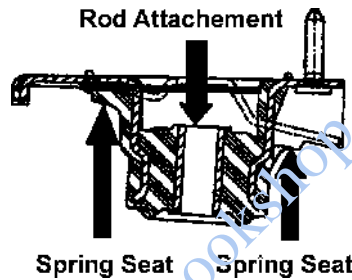


Figure 1.7.12 Dual-path damper/strut mounting (Lewitske and Lee, 2001).

1.8 Operating Speeds and Strokes

The suspension is brought into operation by:

- (1) dropping, e.g. falling off a jack, driving off a kerb, or in a drop test;
- (2) ride motions, in response to road roughness;
- (3) transient longitudinal motions giving pitch change in acceleration or braking;
- (4) transient handling response, mainly roll velocity during corner entry and exit.

The above are, in general, likely to be combined in real cases. The vehicle motion effectively defines the suspension wheel bump speed V_{SB} , but evaluation of the actual damper speed requires consideration of the particular installation geometry (Chapter 5) to determine the velocity ratio (motion ratio) V_D/V_{SB} .

The suspension bump velocities may be estimated under the following headings:

- (1) drop test;
- (2) ride motions;
- (3) longitudinal acceleration transients;
- (4) lateral acceleration transients;

- (5) combined effects;
- (6) damper failure speeds.

These will be dealt with in more detail as follows.

Drop Test

The vehicle is released to fall freely from height h_D above the position at which the wheels touch the ground. The wheels may initially be in the full droop position, simulating the situation where the vehicle leaves the ground, e.g. a rally car passing over a crest. Where it is intended to simulate a simple road step-up impact it is better to restrain the wheels to the normal ride position. When the vehicle is below that point at which contact of wheel and ground just occurs, the springs and dampers will act, and there will be relatively little further speed increase, unless the drop is from a very low initial position. This can be studied more accurately, either analytically or by time stepping, but for a simple high-drop analysis the impact speed V_1 is given by energy analysis of the fall as

$$\frac{1}{2}mV_1^2 = mgh_D$$

where m is the mass and g is the gravitational field strength, leading to

$$V_1 = \sqrt{2gh_D}$$

This is independent of the vehicle mass. Example values are given in Table 1.8.1, where it may be seen that even a small drop of 50 mm gives an impact speed of 1 m/s, which is quite high in damper terms.

A passenger car may experience a drop of 100 mm or so occasionally, for example by driving over a kerb, with a corresponding impact speed of 1.4 m/s. Rally cars may be expected to have drops of as much as one metre as a result of driving at high speed over a crest, with vertical-component impact speeds of 5 m/s or more in extreme cases. The tyre deflection will absorb some of this impact, but it is evident that very high damper speeds may occur in this way.

Free Droop Release

Considering the vehicle body to be temporarily fixed, a wheel may be suddenly released so that the suspension is forced into droop motion (jounce) under the action of the main suspension spring. This

Table 1.8.1 Impact speeds resulting from suspension drop test

h_D (m)	V_1 (m/s)
0.05	0.990
0.10	1.401
0.20	1.981
0.40	2.801
1.00	4.429
2.00	6.264

could occur in practical use if one wheel suddenly passes over a wide hole. If the suspension was previously in a normal position, the free force on the wheel equals the normal suspension force, about $mg/4$. This will give a free extension velocity of 2 m/s or more, large in damper terms. This is the speed at which the car would settle on its dampers if the springs could be instantly removed:

$$V_R = \frac{mg}{\sum C_D} = \frac{g}{2\zeta} \sqrt{\frac{m}{\sum K}} = \frac{g}{2\zeta\omega_{NH}}$$

where the spring stiffnesses K and the damping coefficients C_D are the effective values at the wheels. The natural heave frequency in rad/s is ω_{NH} .

In principle this is a feasible experiment, by removing the springs and holding the body, followed by a sudden release. If the wheel is in a compressed position at the moment of release then the potential speed is even greater. In practical conditions, of course, the wheel inertia plays a part.

Ride Motions

The vehicle is stimulated by a wide spectrum of frequencies from road roughness. The main suspension response occurs at the natural heave frequency f_{NH} of the vehicle, around 1.4 Hz for a passenger car. For a sinusoidal motion in displacement, the velocity and acceleration are also sinusoidal. The amplitudes of these are called the displacement amplitude, the velocity amplitude and the acceleration amplitude. The abbreviated term ‘amplitude’ means the displacement amplitude. Twice the displacement amplitude is the distance between extreme positions, and is called the stroke. The suspension bump velocity amplitude V_{SB} depends on the heave displacement amplitude Z_H according to

$$V_{SB} = \omega_{NH}Z_H = 2\pi f_{NH}Z_H$$

where ω_{NH} is the radian natural frequency (rad/s) of the body in heave.

Table 1.8.2 gives some example values. The total range of body motion, the stroke, is twice the amplitude, so at 1 Hz a total bump stroke of 0.16 m, essentially the full range of suspension movement corresponding to an amplitude of 80 mm, is needed to give a bump speed of 0.5 m/s. This is a very severe ride motion. Ride amplitudes of under 20 mm are normal, with corresponding suspensions bump speeds up to 0.15 m/s. Naturally, this depends upon the quality of road and the vehicle velocity, although for passenger cars these are compensating, as speed will be reduced by the driver if the ride motions become severe.

In the case of rally cars, which are often required to travel at high speed on rough roads, the suspension velocities are correspondingly higher. For racing cars, the natural heave frequency is higher, especially for ground effect vehicles where it may be around 5 Hz or more, even for very stiff suspensions, because of the tyre compliance. Logger data shows that the ride response is strong at this

Table 1.8.2 Bump speed amplitudes in suspension ride (at $f_{NH} = 1$ Hz)

Z_H (m)	V_{SB} (m/s)
0.005	0.031
0.010	0.063
0.020	0.126
0.040	0.251

natural frequency, but the amplitude is necessarily very small, so the actual suspension ride velocities are still quite modest, generally 0.1–0.2 m/s, although worse over particular bumps.

Longitudinal Acceleration Transients

Variations of longitudinal acceleration cause (angular) pitch motions, and occur by sudden application or removal of engine power or brake action, with a step change of A_X . This is most easily demonstrated when the brakes are kept firmly on as the vehicle actually comes to a halt, giving a distinct jerk with a subsequent damped pitching motion. The avoidance of this discomfort factor by ramping off the braking force at the end is called ‘feathering’ the brakes. This case is a fairly easy one to analyse. Consider a longitudinal deceleration A_X on a simple vehicle having wheelbase L with centre of mass at the mid point, centre of mass height H_G , and suspension stiffness wheel rate K_W at each wheel. An angular pitch angle θ radians gives a suspension bump displacement

$$z_{SB} = \frac{1}{2}L\theta$$

Hence, the restoring pitch moment M_P is

$$M_P = 2K_W z_{SB} L = K_W L^2 \theta$$

and the pitch angular stiffness K_P is

$$K_P = K_W L^2$$

A longitudinal acceleration A_X gives a longitudinal load transfer moment

$$M_{AX} = m A_X H_G$$

with an associated pitch angle

$$\theta = \frac{M_{AX}}{K_P} = \frac{m H_G}{K_W L^2} A_X$$

and a suspension deflection

$$z_S = \frac{1}{2}L\theta = \frac{m H_G}{2K_W L} A_X$$

The natural frequency f_{NP} of pitch motion depends upon the pitch stiffness and also on the sprung mass (body) second moment of mass in pitch, I_{PB} :

$$f_{NP} = \frac{1}{2\pi} \sqrt{\frac{K_P}{I_{PB}}}$$

with a value for most vehicles approximately equal to the heave frequency, at around 1 Hz for a passenger car. Hence the estimated suspension bump velocity for this amplitude and frequency $V_{SB,AX}$ is

$$\begin{aligned} V_{SB,AX} &= \omega_{NP} z_{SB} \\ &= 2\pi f_{NP} \left(\frac{m H_G}{2K_W L} \right) A_X \end{aligned}$$

where ω_{NP} is the natural frequency of the body in pitch and z_{SB} is the displacement of the suspension in bump.

Realistic physical values for a passenger car give a pitch angular deflection of about $0.4^\circ/\text{m s}^{-2}$ ($4^\circ/g$) with a suspension deflection/acceleration rate of $10 \text{ mm}/\text{m s}^{-2}$ ($100 \text{ mm}/g$). At a pitch natural frequency of 1 Hz the consequent suspension velocity is about $50 \text{ mm s}^{-1}/\text{m s}^{-2}$ ($0.5 \text{ m s}^{-1}/g$). Hence the associated suspension velocity for pitch motions is typically up to 0.2 m/s for normal braking transients and up to 0.5 m/s for extremes. The most severe case is a sudden switch from acceleration to braking. For a 5 m/s^2 ($0.5g$) change of A_x , the suspension stroke is about 50 mm with a velocity of 0.25 m/s .

A similar analysis may be performed for other kinds of vehicle. Racing cars, despite their high natural pitch and heave frequencies, tend to have smaller suspension travel velocities because of the limited suspension movements, plus the use of anti-dive suspension geometry, which reduces the pitch angles and velocities in proportion.

Lateral Transients (roll)

Sudden changes of lateral acceleration cause roll motions that can be analysed in a similar way to longitudinal motions. A lateral acceleration causes a suspension roll angle

$$\phi_S = k_{\phi S} A_y$$

where $k_{\phi S}$ is the suspension roll angle gradient, of about $0.014 \text{ rad}/\text{m s}^{-2}$ ($0.80^\circ/\text{m s}^{-2}$, $8^\circ/g$, $0.14 \text{ rad}/g$). The corresponding suspension deflection z_S is

$$z_S = \frac{1}{2} \phi_S T$$

with ϕ_S in radians and a track (tread) T of about 1.5 m . For a roll natural frequency f_{NR} the suspension velocity amplitude estimate becomes

$$V_{zS,Ay} = 2\pi f_{NR} z_S = \pi f_{NR} \phi_S T$$

Realistic values for a passenger car are a natural roll frequency of 1.5 Hz and a suspension deflection of $10 \text{ mm}/\text{m s}^{-2}$ ($100 \text{ mm}/g$). Entering a normal corner of lateral acceleration 3 m/s^2 ($0.3g$), the suspension deflection is about 30 mm and velocity 0.30 m/s . An extreme corner entry gives 80 mm stroke and 0.60 m/s , depending on the time for steering wheel movement.

It is essential to have good roll damping in this speed range. This is the basic reason why nonlinear $F(V)$ characteristics are used—to give adequate roll damping without excess forces at higher speed.

Combinations

The above suspension velocities are frequently combined. In general a handling motion is mixed with some ride motion, so the handling motion does not always occur in a progressive manner, there may even be velocity reversals.

Overall, for a passenger car the suspension bump velocities are up to 0.3 m/s in normal driving, up to 1 m/s in harder driving or poorer roads, with values in the range 1 to 2 m/s occurring relatively rarely.

Table 1.8.3 indicates a possible classification of bump velocity ranges for passenger cars. A racing car classification would generally be lower, especially for a ground effect vehicle such as Formula 1, or Indy. However, even ground effect racing cars are subject to driving over angled kerbs, to dropping from jacks and so on.

Table 1.8.3 Possible classification of suspension bump velocities (m/s)

	Passenger car	Racing car (F1, Indy)
Very low	Below 0.1	Below 0.025
Low	0.1 – 0.2	0.025 – 0.050
Medium	0.2 – 0.4	0.050 – 0.080
High	0.4 – 1.0	0.080 – 0.200
Very high	1.0 – 2.0	0.200 – 0.400
Extreme	Above 2.0	Above 0.400

Damper Failure Speeds

The damper will suffer permanent damage if subject to excessive velocities, although these limits are not widely advertised. The physical survival limit speed is that which will not quite cause any metal yielding or, more likely first, damage to the valves or seals. The limit speed is typically around 5 m/s.

Velocity and Stroke

Fukushima *et al.* (1983) suggested that dampers should have a stroke-dependent characteristic, such that for a given velocity a longer stroke would give a greater force. More information on this is given in Chapter 3. Table 1.8.4 shows the velocities and strokes found by Fukushima, with some extra analysis, where F/V is the desirable C_D , and F/S is the desirable ‘stiffness’ relationship if the force were produced by a spring. Slow steering is a ‘figure of 8’ course with large lateral acceleration, but gentle entry.

Perhaps the clearest form of analysis for this problem is to plot a diagram of damper velocity against amplitude or stroke, showing the active points. This is done in Figure 1.8.1. Points on one line radiating from the origin have the same frequency, since the velocity and displacement amplitudes of a sinusoidal motion are related by $V = 2\pi fX$. With suitable scales for a particular vehicle, the main diagonal is the natural frequency of the vehicle body in heave type motions, around 1.4 Hz. The basic handling manoeuvres, i.e. changes of A_X and A_Y , operate along this diagonal according to severity, or somewhat below it, according to the rate of application of the controls. Driving along a straight road gives heave motions along the main diagonal, with good-quality roads at the bottom left, and rough roads, or fast driving, further up. Bumps act on the bottom of the wheel and are transmitted through to the body at essentially the wheel hop frequency. Isolated bumps therefore lie on a steep line at about 12 Hz, on the left.

To avoid transmission of bumps, but to give good damping for handling evidently could be achieved by having frequency-dependent damping, within the meaning of this diagram, with low damping

Table 1.8.4 Suspension stroke, velocity and damper force

Manoeuvre	Stroke (mm)	V (m/s)	F (N)	F/V (kN s/m)	F/S (kN/m)
Slow steering	50	0.02	48	2.4	1.0
Rapid steering	45	0.12	288	2.4	6.4
Lane change	25	0.06	144	2.4	5.8
Single bumps	4	0.40	120	0.3	30.0
Brake/accelerate	50	0.20	400	2.0	8.0
Smooth road	3	0.08	8	0.1	2.7
Rough road	12	0.60	600	1.0	50.0

(Based on Fukushima *et al.*, 1983)

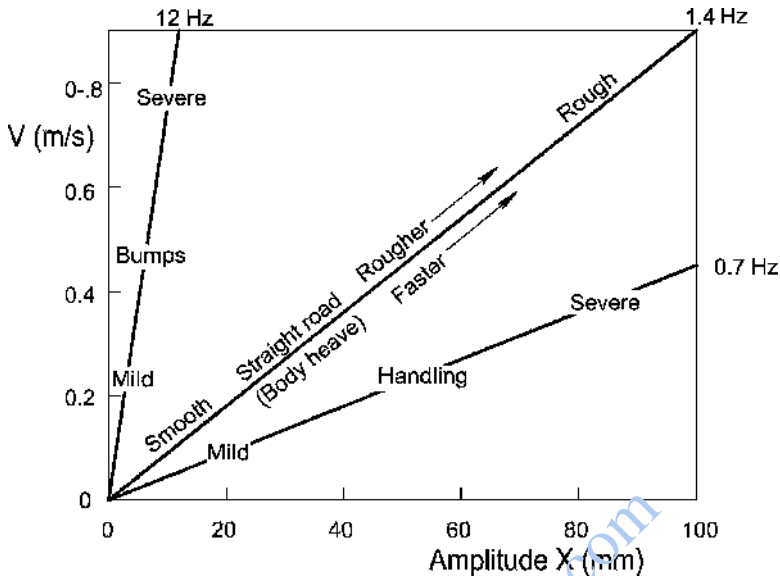


Figure 1.8.1 Damper speed versus damper amplitude for various ride and handling activities. Handling ‘frequency’ depends on the rate of control application.

above, say, 3 Hz. On the other hand, to also have low damping as desired on smooth roads, stroke-dependent damping with low damping for strokes below about 10 mm is desirable.

1.9 Manufacture

The complete manufacturing process involves:

- (1) material production;
- (2) individual component production;
- (3) assembly;
- (4) oil-filling;
- (5) pressurisation (in some cases);
- (6) insertion of bushes;
- (7) external finishing;
- (8) testing at various stages as appropriate.

Initial material production is of course not the province of the damper manufacturer, who will simply buy in materials, but may heat-treat them.

The production method of the tubes depends upon the price range of the damper. Cheaper ones may have tubes rolled from low-carbon steel with a resistance-welded seam. More expensive ones may avoid the seam by using extruded tubing, and in any case racing dampers, and possibly more passenger cars in the future, will have aluminium cylinders to reduce the weight. Steel tubes are annealed and then cold-drawn to final diameter, giving accurate sizing and cold-worked strength. The internal finish is important to give a good seal with the piston and also to give a low wear rate of the piston seal; the internal finishing process, e.g. honing, varies with manufacturer and budget.

The piston rod is normally made from high-carbon steel with cold-forged ends, often with some drilled passages, possibly threaded at the top, and case-hardened or chromed on rubbing areas. The life of the rod-bush bearing and of the main rod seal is sensitive to the rod surface finish, which should be $0.1 \mu\text{m}$ r.m.s. or better, achieved by grinding and lapping. It is now normal to use a hard-chrome-plated rod which enhances both wear resistance and corrosion resistance.

Pistons themselves may be forged or sintered. Piston seals vary considerably. Some have iron piston rings not unlike those in an engine, with a small end gap, necessary for thermal expansion. Plastic piston-type rings are also used, typically with stepped-overlapped ends, from Teflon (PTFE) or nylon, etc. Similar designs are used in Tufnol and other paper and cloth-reinforced phenolic materials. Square-section rubber rings and O-ring seals are common. Finally, some plastic seals are moulded solid to the piston with no apparent attempt to allow for self-adjustment to fit, possibly expected to run-in to a minimal correct working clearance, with a wide contact area. Racing pistons are usually machined aluminium, turned and drilled, with a plastic piston ring

Valve manufacture of course varies with design. At the low-cost high-volume end of the market, where possible, stamping and pressing are used, followed by automated or manual assembly of these parts plus coil springs, washers, *et cetera*.

For common double-tube dampers, the top and bottom cylinder ends are pressed from steel sheet, typically with a mounting bush ring spot-welded onto the bottom one. The lower cup is welded to the cylinder and foot valve inserted. The piston, piston valves, rod, rod-bush housing, top cup and its seal and bearing are assembled. The piston and oil are inserted and the top cup welded, spun or crimped into place. The rubber bushes are then pressed in, and, finally, paint is applied.

With a worldwide production of several hundred million dampers per year, there is obviously a considerable difference between the methods used for price-sensitive mass-production units and those for higher-price small-volume precision units produced for specialised applications such as racing and rallying.

1.10 Literature Review

Books on vehicle suspension or vehicle dynamics nearly always have some basic description of damper (shock absorber) design and operating characteristics, although this is invariably limited in scope e.g.:

- (1) Norbye (1980), 10 pages descriptive;
- (2) Campbell (1981), 8 pages descriptive, one equation;
- (3) Bastow (1987), 10 pages on the effect of damping, 19 pages of description of dampers;
- (4) Dixon (1991, 1996), 10 pages on history plus comments on effect on handling;
- (5) Gillespie (1992), 7 pages on dampers and their effect;
- (6) Milliken and Milliken (1995), 50 pages, almost all on vehicle behaviour rather than damper design itself;
- (7) Haney and Braun (1995), 32 pages, with the emphasis on racing car dampers;
- (8) Reimpell and Stoll (1996), 36 pages, on damper design, construction and characteristics.

Within the context of a complete book on ride, or on handling, or on vehicle dynamics as a whole, such limited treatment is understandable.

Although a literature search reveals several thousand papers referring to 'damper' or 'shock absorber' in the context of vehicles, rather few of these are about the design and characteristics of the damper itself. A full list of these is given in the references.

The earliest paper appears to be that by Hadley (1928), which deals with mechanical friction dampers, seeking to obtain a suitable characteristic by position dependence. The ability of a damper to control large amplitudes adequately whilst not overdamping small motions is called the 'range' of the

damper, and was a leading problem of that era. It seems surprising now that mechanical types were considered more promising than hydraulics.

Weaver (1929) investigated damper characteristics by applying them to a vibrating system, actually a 272 kg mass on a 31 kN/m spring giving 1.71 Hz undamped natural frequency, with initial deflections of typically 76 mm, and producing free vibration $F(X)$ plots, which are roughly spiral for less than critical damping. This is a simple and cheap method of investigation, of some value for comparative purposes, but not as useful as an $F(X)$ curve for a controlled displacement pattern, preferably sinusoidal, as obtained on a modern testing machine.

Nickelsen (1930) states that by 1930 nearly all cars had some kind of damper as standard equipment, and therefore, of course, by implication some still did not. He concentrates on the double-acting hydraulic type, recommending that no more than 20% of the resistance should be in bump. $F(X)$ curves are shown, and temperature fade considered, although with the apparent belief that resistance is simply proportional to viscosity.

James and Ullery (1932) discuss the problems of Coulomb friction against the advantages of hydraulic damping with a suitable force–speed relationship. Various force–displacement graphs ('card diagrams') are shown, illustrating various effects and faults, and the loss of damping effect with temperature rise is eliminated by a variable orifice comprising a plug valve regulated by a bimetallic strip. Oil viscosities of 20–200 centistokes were in use. They emphasise the need for a smooth characteristic, building up with speed, and the need for temperature compensation.

Fuchs (1933) gives $F(X)$ diagrams for single-acting and double-acting hydraulics, and for the old friction disc type, and also shows a characteristic in the phase plane ($V-X$) for a position–dependent and force–dependent damper.

Kindl (1933) proposed to regulate the damper by a valve controlled by body inertia, so that wheel-only movements are free, but the body motion is damped. Unfortunately his acceleration sensor and controller was a 680 g mass for each damper. However, with electrically controlled dampers, and modern sensors and control systems, this is an idea whose time may now be ripe.

Schilling and Fuchs (1941) show force–speed diagrams for year 1940 passenger car dampers, some of which are similar to modern dampers, showing a predominance of rebound force over bump.

Connor (1946) distinguishes various types of damping effect (Coulomb, real dry friction, viscous, degenerate viscous, 'hydraulic' damping proportional to V^2 , and solid material hysteresis). Typical damping ratios are stated to be 0.15–0.20, and the temperature rise in a telescopic damper is stated to be about 50°C on extremely rough roads, although only about 20°C for rotary and cam types because of conductivity into the vehicle body.

The related area of buffer design is considered by Brown (1948), Tatarinov (1948, 1949) and Brown (1950 a, b). For buffering, the resisting force is commonly made a function of position. This is not normally done for car dampers, although it has been used on motorcycle front forks, where it gives a bump stop effect, but is regarded as superior to an elastic bump stop because of reduced spring back.

Kinchin and Stock (1951/1952), in a comprehensive paper, discuss various details, such as the physical layout of various hydraulic types, fluids, testing, operating problems such as aeration, fade, noise problems, flow resistances, and force–displacement $F(X)$ diagram (work diagram) shapes.

Peterson (1953) describes the introduction of a damper designed to have a linear characteristic (in contrast to many of the previous ones which were almost Coulomb in character, as may also be seen in some of the characteristics shown by Schilling and Fuchs (1941)). Prototypes had tubes intended to give a simple viscous pressure drop, but actually giving force proportional to speed to the power 1.25. More conventional drilled and coined holes were used for production. The higher forces obtained at high speeds were considered a substantial improvement for rough roads.

Cline (1958) discusses various aspects of dampers in general terms, giving some example $F(X)$ curves, with brief comments on the effect of valve changes. He also shows a photograph of the then new Gabriel dampers with electrical remote adjustment.

Hoffman (1958) was amongst the first to do analytical work on the damper. He shows various experimental $F(V)$ and $F(X)$ curves for different types of damper, and used an analogue computer to study vehicle ride for various dampers.

Jackson (1959) discusses the basic principles of telescopic dampers and their manufacture, and the effect of valving, stating that control of unsprung mass vibrations had become the greater problem once independent suspension was introduced. The standard fluid is stated to be straight mineral oil with viscosity somewhat less than SAE 5, but this was considered unsatisfactory because of viscosity change with temperature causing difficulties at very low temperatures. High-viscosity-index oils were just coming into use. Curves of damping force against temperature are given. A freon-filled nylon cell for preventing aeration problems is described.

Eberan-Eberhorst and Willich (1962) show experimental $F(X)$ curves for various strokes, and $F(X)$ and $F(V)$ curves with lag and hysteresis. The experimentally observed temperature effect on force was about $-0.1\%/K$ for compression and $-0.3\%/K$ for extension. A Plexiglas damper was built, allowing cavitation to be observed directly.

Ellis and Karbowniczek (1962) deal mainly with buffers rather than vehicle dampers. However, they draw attention to possibilities for separation of the air from the fluid to prevent aeration, including a spring-loaded free piston, a diaphragm, a gas-filled bladder, or a compressible closed-cell solid foam.

Speckhart and Harrison (1968) return to the idea by Kindl (1933) of using an inertia-regulated valve, this time much lighter and built into the damper piston, the intention being to reduce vehicle ride jerk (rate of change of acceleration).

Polak and Burton (1971) discuss damper construction in general terms, giving attention to possible designs of amplitude sensitive dampers, to a multi-frequency 'seismically' actuated damper (mounted on the wheel only, not to the body), and propose a configuration with a conventional damper coupled between suspension and engine. They also mention the possibility of achieving adjustable damping by the use of certain electrostatically sensitive solid/oil mixtures which have variable viscosity, i.e. use of electrorheological liquids.

Wössner and Dantele (1971) compare the pressurised single-tube type (with and without a free separator piston) with the unpressurised dual-tube type, and give experimental results on the cooling effect of airstreams.

Mitschke and Riesenberg (1972) discuss damper temperature rise on various types of road, and the consequences for fluid viscosity and damper force.

Mayne (1973) gives an analysis of the effect of liquid and mechanical compliance on buffer performance.

Jennings (1974) gives 21 $F(X)$ loops for commercially available motorcycle front forks and rear dampers, which prove to be characterised by extreme compression/extension asymmetry.

Cline (1974) gives a simple review of the application of hydraulic dampers to recreational vehicles, considering briefly how to achieve some of the particular characteristics required.

Schubert and Racca (1974) describe an unconventional 'elastomeric-pneumatic isolator with orifice-type relaxation damping', proposing application to motor vehicles.

Simanaitis (1976) gives a brief history and a discussion of some of the operating principles and problems ('The Dutch call them schokdempers; the French, amortisseurs; and the English, dampers. Indeed, many early automotive designers called them unnecessary ...'). (As a matter of interest it is *Stossdämpfer* in German and *ammortizzatori* in Italian.) Oil viscosities are given as SAE 5 to 10. Aeration is stated to be the cause of compression lag (although actually this can also be cavitation or desorption of air). Manufacturing methods are outlined.

Dalibert (1977) considers the effect of some oil properties on damper performance. The sensitivity of peak force to temperature was found to be about $-0.3\%/^{\circ}C$, giving a reduction of 35% at $130^{\circ}C$,

which is stated to be the approximate maximum temperature encountered with hard driving on bad roads. Maximum safe viscosity is stated to be 4000 cs at the lowest temperature, and a minimum of 4 to 7 cs at 100 °C, depending on design. Noise problems are considered in some detail.

Segel and Lang (1981) report a detailed investigation (additional information being given in Lang's PhD thesis (1977)), using an 82 parameter analogue computer model that gave quite good agreement with the experimental data obtained. Compressibility of the liquid, plus a slight additional effect from cylinder compliance, was found to give hysteresis in the $F(V)$ curve at higher frequencies (above 1 Hz). Presumably series rubber bushes would markedly increase this effect, but were not mentioned. Gas compressibility and absorption were considered; this could not readily be handled by the analogue computer, but the use of an effective vapour pressure of up to 70 kPa (instead of the actual value of less than 2 kPa) was found to give realistic results. Orifice discharge coefficients were investigated, but for simplicity it was considered that a constant value of 0.7 was acceptable. Testing was actually done with a square acceleration wave, giving a triangular speed waveform and a piecewise parabolic displacement waveform.

Van Vliet and Sankar (1981) studied motorcycle forks and rear dampers, using analogue and digital simulations respectively to obtain good agreement with experimental $F(X)$ diagrams.

Arndt *et al.* (1981) consider seal design with tests of friction especially in the context of smaller, lighter vehicles where the problem seems more critical, advocating a lip design with a lubrication groove, allowing improved ride comfort and reduced pressurisation.

Ohtake *et al.* (1981) also consider seal design, with tests of friction and durability, and analysis of relevant parameters, considering variation of optimum design with details of the application.

Fukushima *et al.* (1983) advocate stroke-sensitive damping, and present the arguments in its favour, including analysis of ride and handling motions.

Fukushima *et al.* (1984) reiterate the points on stroke-sensitive damping, and consider a vortex valve type which has a resistance that depends on stroke as well as speed, being greater for larger stroke. They made successful tests of a demonstration unit.

Steeple *et al.* (1984) describe a damper testing facility, with the emphasis on durability testing.

Morman (1984) considers the mathematical modelling of dampers, expressed in terms of governing differential equations.

Holman (1984) considers a rotary-type damper configured to give better immunity to stone impact damage, intended for military and off-road vehicles.

Yukimasa *et al.* (1985) consider the design of oil and gas seals, sealing quality and frictional characteristics of damper seals.

Vannucci (1985) considers damper noise problems with special reference to the McPherson strut with integral damper.

Sugasawa *et al.* (1985) studied theoretically the optimum damping for ride and handling independently for a two-degrees-of-freedom system, and tested an automatically adjusting damper system using sensors for control inputs (accelerator, brakes, steering) and an ultrasonic ride height measurement to appraise the road quality.

Browne and Hamburg (1986) measured damper temperatures, and also the simultaneous forces and velocities of dampers on the vehicle, to obtain the energy dissipation rates. For passenger cars on normal roadways this was found to be 3–60 W, and about 12 W on average.

Karadayi and Masada (1986) consider factors such as directional asymmetry, dry friction, hysteresis, compressibility and backlash in a nonlinear model, aiming to create a simple damper model (i.e. fast computing) suitable for use in vehicle simulations. The total nonlinear compressibility is treated as piecewise linear, giving in effect a backlash plus a series stiffness, with Coulomb friction and asymmetric linear damping. The model gives quite good agreement with the general character of the real $F(V)$ curves, although the quantitative agreement is not particularly good, which presumably is the result of, rather oddly, omitting the most important nonlinearity of all, the resistance characteristics of the fluid valves. The usual end-fitting rubber bushes are not mentioned, but presumably could readily be incorporated in this model.

Young (1986) gives details of the internal configuration of various types of aircraft undercarriage dampers, which typically have free-piston gas separation and positional dependence of the damping force.

Hall and Gill (1986) describe a CSMP digital simulation of a dual-tube damper, including the effect of valve mass and valve damping. The foot compression valve (compression control) is a side exit spool valve, giving a fluid momentum force that gives Coulomb friction between the spool and its guide—another possible source of $F(V)$ curve hysteresis. Gas compressibility is dealt with by adopting a high effective vapour pressure of 41–85 kPa. In the simulation, a time step not exceeding 10 μ s was necessary to avoid instability, corresponding to 0.02° of phase angle. Leakages and temperature effects were neglected, and a constant discharge coefficient of 0.7 was used. With an empirically selected effective vapour pressure, the trend of results was correct, but the $F(V)$ behaviour was too oscillatory, either because of inadequate theoretical valve damping or inadequate experimental sensor frequency response.

Soltis (1987) describes the Ford PRC (Programmed Ride Control) automatic damper adjustment system, particularly considering the handling conditions when high damping is desirable, and the advantages of the use of steering wheel angle sensing.

Su *et al.* (1989) theoretically analysed an adaptive damper, concluding that significant improvements in ride performance could be achieved.

Gvineriya *et al.* (1989) investigated the extension of gas spring units to include gas suspension damping, considering it to be a viable option.

Lemme and Furrer (1990) describe a self-powered system for remotely adjusting dampers by hydraulic means.

Hennecke *et al.* (1990a, b) describe an in-production adaptive damping system, with three states independently for each axle according to road conditions and driver. This system uses body sensors for vertical acceleration above the axles, steering angle, and longitudinal speed and longitudinal acceleration (rather than brake line pressure and engine output), with electronic control of solenoid valves.

Hagele *et al.* (1990) tested variable dampers with fast-acting solenoid valves and electrorheological liquids, but found the latter as yet unsatisfactory.

Fan and Anderson (1990) tested a bus damper complete with its rubber mounting bushes, naturally obtaining a large hysteresis in the $F(V)$ curve. Modelling equations are presented, with the bushes represented as effective compressibility, and good correlation with experiments is shown, with frequency-dependent hysteresis.

Rakheja *et al.* (1990) studied a 'sequential' damper similar to a conventional positive damper with multi-stage asymmetrical valving proposed to be mounted externally to facilitate adjustment, concluding that it offered significant possible improvements in ride over a conventional damper.

Lemme (1990) considers the advantages of hydraulic control for variable dampers over electrical control, considering in detail the design of such a damper, with the idea that action of any one damper could produce a pressure to control all four damper settings, without other sensors.

Kumagai *et al.* (1991) studied the internal Coulomb friction of strut bearings relating to the transmission of NVH (noise, vibration and harshness).

Patten *et al.* (1991) described how fast semi-active dampers can take advantage of the phase relationship between front and rear axle disturbances to give a significant improvement in ride quality.

Shiozaki *et al.* (1991) propose a variable damper with piezoelectric sensor and actuator built into the damper rod, with rapid response (3 ms) and high actuation force for an axial valve, favouring a 'normally hard' setting with switching to soft when appropriate, e.g. hitting a bump. The short piezoelectric actuator motion is amplified by stacking the elements, and then amplifying the total motion with an inverted hydraulic jack method to give 2 mm of motion at the valve.

Tamura *et al.* (1992) consider the development of materials for the rod-guide bush of strut type suspensions, which carries a large side load with unreliable lubrication, but needs to have low friction

and long life, and is subject to other problems such as cavitation erosion. Various PTFE/Sn/Pb/Cu/PbF₂ sintered materials have been used.

Petek (1992a, b, c) describes the design principles of and tests on an experimental design of variable damper using ER (electrorheological) liquid, with durability tests and a retrofit vehicle road test. The ER effect is claimed to be fast in action (time constant about 5 ms), with good control power input efficiency (about 4 W), and giving lower cost and better reliability than mechanical valves, although the desired operating temperature range of -40 to $+120$ deg C may be a problem.

Nall and Steyn (1994) performed experimental evaluation of various control strategies for two-state dampers under practical conditions. It was concluded that such dampers offered significant ride improvements, and that the usual theoretical assessment of adaptive dampers did not adequately represent real conditions.

Pinkos *et al.* (1993) espouse ERM (electrorheological-magnetic, i.e. MR magnetorheological) liquid, which operates at low voltage. This was used in a lever-arm configuration test damper for road tests.

Baracat (1993) gives a theoretical analysis of damper forces and compares the predictions with experimental results.

Reybrouck (1994) considers modelling of conventional dampers to allow for force dependencies not just on velocity, but also on position, acceleration, temperature etc. for application in ride quality simulations, obtaining good agreement with experimental results.

Fash (1994) studied the application of a neural net computation model to data from dampers installed on vehicles. Correlation with the forces was superior to that of a simple linear model.

Besinger *et al.* (1995) describe a seven parameter damper model, particularly developed for use with studies of heavy vehicles. The model is compared with test damper data, and the effect of variation of the parameters on vehicle ride is investigated by simulation.

Sturk *et al.* (1995) developed and tested a high-voltage control unit for an electrorheological variable damper, and experimentally investigated the effect of control strategies on ride with a quarter-car rig.

Whelan (1995) investigated the use of triangular position waveform testing.

Cafferty *et al.* (1995) performed damper testing with excitation by a random waveform.

Audenino and Belingardi (1995) considered models of varying complexity for motorcycle dampers in comparison with experimental data.

Angermann (1995) investigated the use of aluminium bodies for dampers, and other means of weight reduction for passenger cars, already common on racing cars.

Petek *et al.* (1995) tested a complete electrorheological system on a vehicle, giving information on the support systems and control logic, concluding that such systems would be good if the working temperature range required for general use could be achieved.

La Joie (1996) discusses detailed damper modelling and comparison with experimental results, concentrating on racing dampers.

Haney (1996) made a bench comparison of several adjustable racing car dampers and presented their experimentally measured characteristics.

Ryan (1996) considers the merits of remote reservoir dampers with high shaft-displacement forces.

Warner and Rakheja (1996) investigated friction and gas spring characteristics, varying with temperature, with a particular emphasis on the significance of the latter for critical racing car ride heights.

Feigel and Romano (1996) describe a directly controlled electromagnetic valve for a damper.

Duym (1997) considers representation of damper characteristics to allow for the presence of hysteresis effects on the $F(V)$ curve, concluding that velocity and acceleration make a more satisfactory combination of variables than velocity and position.

Cafferty and Tomlinson (1997) discuss representation of damper characteristics by frequency domain techniques.

Tavner *et al.* (1997) consider test procedures for switchable dampers on the vehicle.

Kutsche *et al.* (1997) discuss pneumatically controlled variable truck dampers.

Lee (1997) analyses the double-acting shim valve monotube damper, using dimensional analysis and finite element analysis to deal with the nonlinear shim deflections.

Meller (1999) describes self-energising self-levelling systems and their incorporation into hydraulic dampers.

Herr *et al.* (1999) used a CFD (computational fluid dynamic) model to study flow in damper valves and to predict complete damper forces, obtaining good agreement with experimental results.

Els and Holman (1999) drew attention to the advantages of the lever-arm rotary damper in heavy duty applications.

Lion and Loose (2002) performed a thermomechanical analysis of dampers with experimental tests.

Choi (2003) tested a design of ER damper in which the ER effect was used to control the main valve rather than providing all the extra resistance itself.

Yamauchi *et al.* (2003) investigated a noisy strut suspension vibration arising from rod bending stimulated by damper piston friction against the tube, successfully obtaining a simple design criterion and testing a double-piston solution giving improved alignment.

Guglielmino and Edge (2004) investigated a dry friction telescopic damper using hydraulic control of the normal force and hence of the friction, claiming various advantages.

Smith and Wang (2004) investigate the possible application of 'inerters' in vehicle suspensions.

Suda *et al.* (2004) studied the possible use of an electromagnetic damper in telescopic form, with a high pitch shaft driving a ball nut which drives an electric motor/generator through a gearbox.

Lee and Moon (2005) reported on tests of a position-sensitive damper with a longitudinally grooved pressure cylinder to relax the damping around the central position.

Ramos *et al.* (2005) reported on a thermal model of double-tube dampers.

Yung and Cole (2005) described wavelet analysis of the high frequency (30–500 Hz) NVH characteristics of dampers.

Kasteel *et al.* (2005) described detailed modelling of the damper and its valves.

Alonso and Comas (2006) studied cavitation problems in dampers.