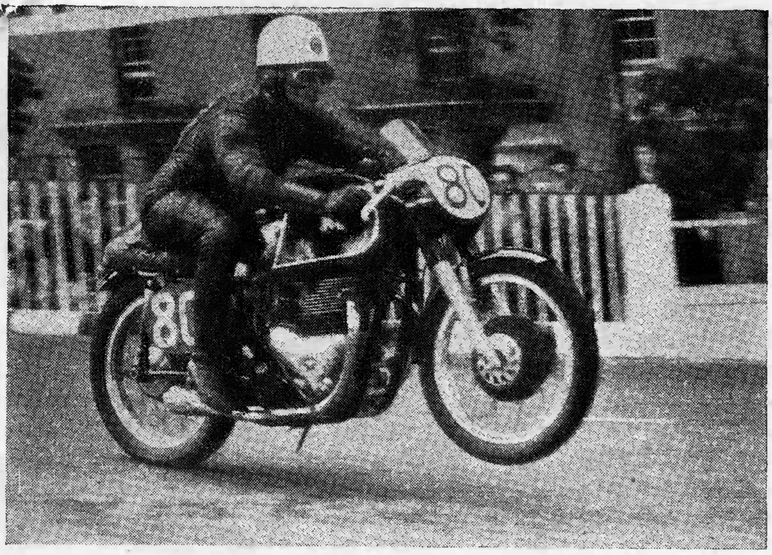


"Full bump." Derek Powell tests the rear suspension of his Matchless to the limit as he lands at Ballaugh in the Senior T.T. of 1957.

MOTORCYCLE ENGINEERING—10

WHAT GIVES?

Part Two—Detail design of dampers and compensation for varying loads

by PHIL IRVING

DAMPING, as the first part of this article explained, is an essential feature of any but the simplest type of lightweight suspension system. If the springing medium itself has no (or insufficient) internal friction, some damping device must be incorporated.

Dampers were originally termed "shock-absorbers" because their addition to an undamped system prevented bottoming, partly through artificially increasing the resistance to bump deflection and partly because they checked the build-up of rebound oscillations. Although the term is still in general use it is misleading, because it attributes a quality to these devices which they do not in fact possess. Their real function is to dissipate the energy stored in the spring through compression from some external source by converting it into heat, not necessarily in one stroke but in a succession of strokes in which the amplitude becomes progressively less.

Frictional Devices

Damping can be applied externally to the springs by either frictional or hydraulic methods. The former is intrinsically the less expensive, especially if existing components can be utilized or modified to accommodate friction-discs or pads. This method was adopted in the first girder forks to be made with in-built dampers and was later applied to the B. and D. rear suspension (illustrated in this series on October 22). An alternative is to fit a multi-plate damper of the Hartford pattern, as in some Guzzi designs. Contracting bands were used on the telescopic tubes enclosing the spring on some models of the Druid fork.

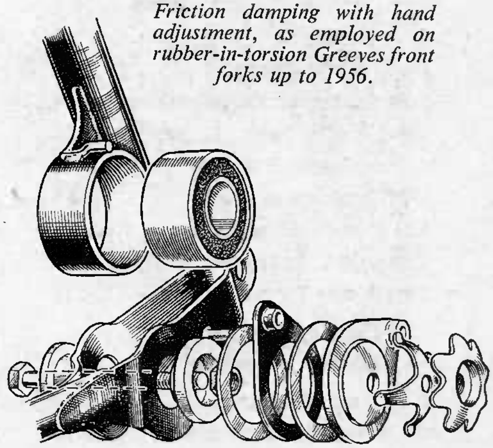
In all of these schemes, the amount of damping force supplied, which quantitatively is about 10% of the load on the spring, can be quickly and easily altered by turning a hand-wheel or by remote control through a Bowden cable. The damping effect is not seriously affected by temperature changes, but it is liable to be seriously reduced by the presence of water or oil.

A more serious shortcoming of friction-damping is the "stiction" which is present at the break-away point when movement is commencing, or is reversing in direction at the end of each stroke. This causes the resistance to be momentarily higher than it is when the movement has been established—which is just the reverse of what is needed under "cobblestoning" conditions.

To some extent, this bad effect can be reduced by restraining the fixed plates in a manner which allows a little flexibility—for example, by the use of soft rubber bushes surrounding the reaction pins—so that a small angular movement can take place before the discs commence to slide; but this is more of a palliative than a cure. Contrary to what one might expect, the effect is made worse by the presence of a lubricant; this is so because the difference between the coefficients of static and of sliding friction is less when the surfaces are dry than when they are oily.

Another feature of friction damping as normally applied is that it is equal in intensity in both directions, so that bump damping is equal to rebound damping. This is sometimes considered to be undesirable, but a great difference of opinion exists amongst suspension engineers as to the most suitable relationship between the two. Too much bump damping leads to a hard ride, while too much on rebound may result in lack of tyre-contact following a severe shock. It is also worth noting that it is not possible to compensate for insufficient spring strength by over-damping; this merely results in a state where the spring is overcompressed and less than the normal amount of wheel movement is available.

Hydraulic damping is achieved by forcing oil through restrictions. These may take the form of small holes, tapered slots or annular gaps, or may be valves containing balls or discs and closed by springs. By a suitable choice of these methods, hydraulic dampers may be made to give an almost infinite variety of resistance-patterns, with practically no damping in the centre of travel and stiffening up at each end, with different resistances on the bump and rebound strokes, or any other combination required. It is also possible to make them "velocity-conscious," with the damping automatically increasing according to the rate of vertical rise of the wheel, if this is considered to be desirable.



Friction damping with hand adjustment, as employed on rubber-in-torsion Greeves front forks up to 1956.

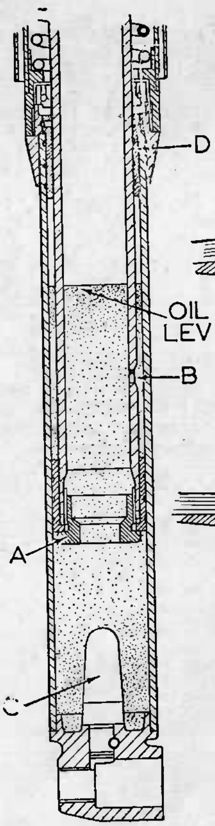
Another excellent feature is that it is possible to provide hydraulic stops at one or both ends of the travel, by trapping a small quantity of oil in spaces from which the avenue of escape diminishes to zero in a distance of, say, half an inch or so. This brings all relative motion of the moving parts to a stop, but introduces a time-factor into the operation that greatly reduces impact-loads in a manner which is absolutely silent and moreover does not give rise to the "throw-back" which may occur with rubber stops.

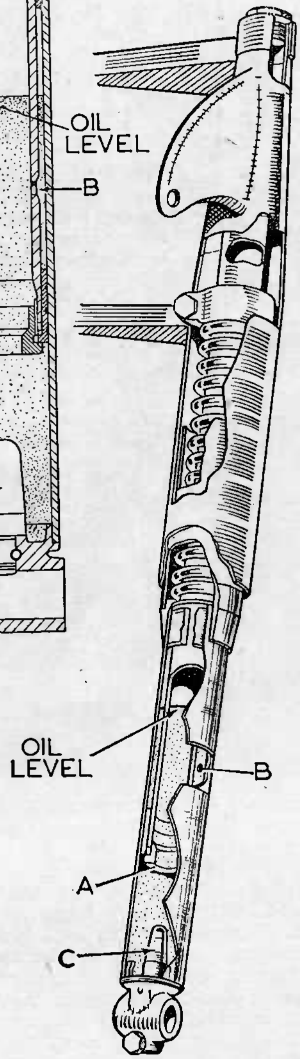
"Low-rate" Factor

When employed as a travel-limiting device on the rebound stroke, as in most telescopic forks, hydraulic stopping permits the use of low-rate springs which are partly compressed even when the forks are fully extended. This feature was in fact one of the main reasons for the superior characteristics of the telescopic fork as compared to the typical girder forks. In the latter, it was usual to anchor the springs at both ends, so that under full rebound the stress changed from compression to tension, in order to limit the fork travel—a practice automatically entailing the use of springs with a very high rate, giving a short, choppy action which was not nearly so comfortable as the long, soft movement obtainable from telescopics. Nevertheless, the "low-rate" characteristic of telescopics can be, and on occasion has been, over-emphasized to a point where the front end progresses with a slow pitching motion almost bad enough to make the rider "sea-sick" on a long journey.

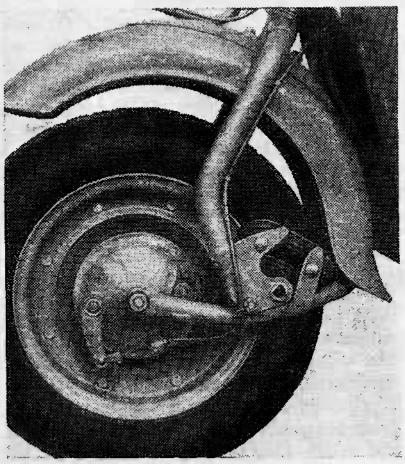
Hydraulic Troubles

Hydraulic dampers are not without their troubles. One is the difficulty of retaining the oil indefinitely, a mechanical problem which has largely been solved by the use of synthetic rubber seals and polished hard-chrome-plated spindles. Another bother is frothing or aeration of the oil, which makes the damper feel "spongy" and reduces the damping effect, because the aerated oil can pass more readily through the restrictions or valves.





The action of a hydraulically damped (and stopped) front fork—a B.S.A. telescopic of 1946. On compression, oil is forced through the centre of gland nut A into the upper sliding member and through hole B into the increasing annular space between the two tubes. On "full bump," tapered plug C enters A, leaving a decreasing escape route for the oil. On full rebound, maximum damping is secured when B is covered by bush D.



Simple rubber-in-tension springing on the forks of a Motobécane scooter.

A third difficulty is the change in oil viscosity with rising temperature. This rise may be due simply to a change in atmospheric temperature, but it is also brought about by the heating experienced by the oil as it is forced through the restrictions. If ordinary engine oil with a low viscosity index (i.e., one which becomes noticeably thinner when warmed) is used as damper fluid, the suspension may be over-damped on a frosty morning so much that it will scarcely move, and in the resulting absence of internally generated heat, the ride will remain hard until the ambient air-temperature rise, combined with a corresponding increase in movement of the suspension, thins out the oil sufficiently to reduce the damping effect until it is back to normal.

Proprietary dampers are therefore filled with a special grade of thin oil which has a high viscosity index. Air-bubbles entrained in such oil can "separate out" more freely than from thick oil, but in any case a trace of silicone anti-foaming agent will almost entirely suppress trouble with aeration or frothing.

Coil springs which are long in relation to their diameter must be provided with some form of guide, otherwise when compressed they will suddenly buckle into a boomerang shape. The simplest method is to fit the spring over a telescopic damper which maintains the whole unit in alignment, with additional external shroud-tubes to exclude grit and improve the appearance.

Rebound Check

The spring may be attached at each end coil by some form of claw, so that under extreme rebound it acts in tension as a travel-limiter, but it is easier and cheaper to locate the spring between collars on the damper mechanism and design the latter so that it cannot extend beyond a certain distance. Mechanically this is much the simpler method of restraining the wheel from dropping too far when unloaded.

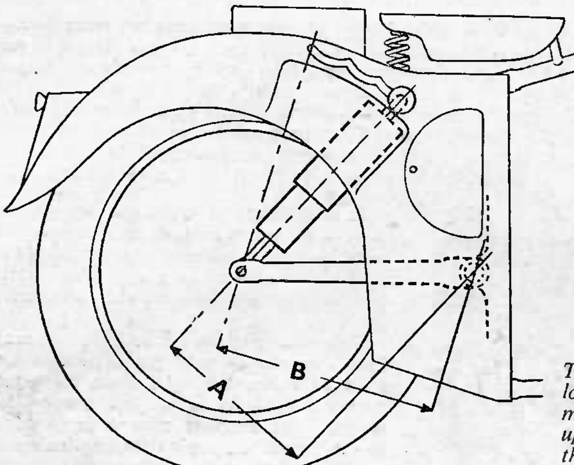
If desired, the damper can also form the bump-stop. However, the design of its internal parts may preclude this. In that case, either the spring can be permitted to close up solid or a rubber bump-stop can be incorporated, but care should be taken to avoid the occurrence of direct metal-to-metal contact on full-bump, otherwise destructive impact loads may be imposed. Even the provision of thick rubber bushes at top and bottom of the spring units will diminish the hammer-blow effect very markedly.

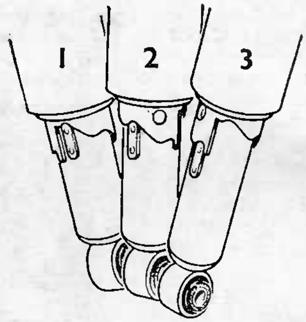
Whatever provision is made for limiting the travel—which, as already noted, should be in the region of 2½ to 3 in. above the normal load or mean position—the springs will last almost indefinitely if designed so that the maximum stress does not exceed 90,000 lb./sq. in. in the most heavily stressed coils.

Compensation for Varying Loads

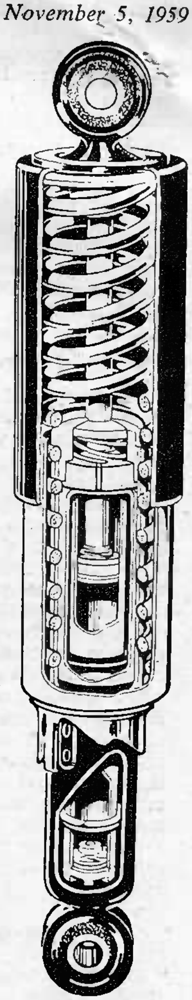
The rear suspension of a two-wheeler suffers more than any other suspension system, except perhaps that of a motor truck, from variation in the load carried, and this is a particularly serious matter in view of the limited amount of travel permissible. If no compensation can be made the springing is either too hard for a light solo rider or too soft, and too liable to bottom with a heavy crew of two.

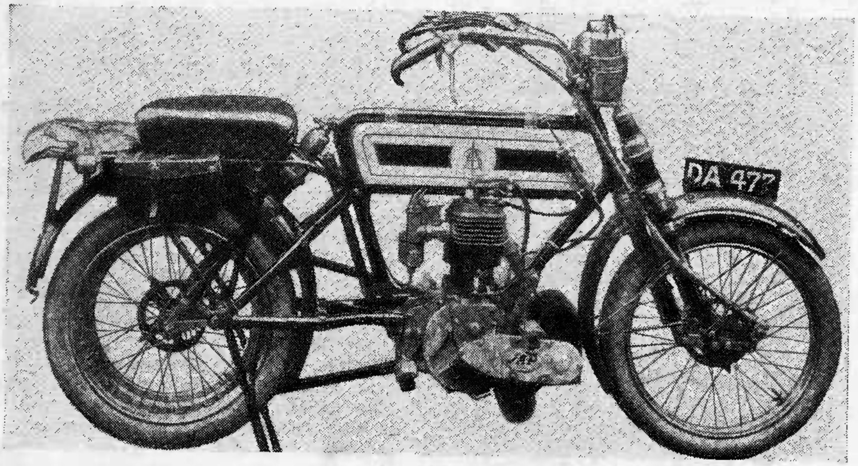
The table on the next page shows the effects of load variation upon travel and error in attitude (i.e., the amount by which the model is down or up at the stern from the designed mean position). The example used is a 385-lb. machine with different crews, allocating the weights carried on the





Two methods of providing compensation for varying loads, Fig. 1 (left) The early-1940s prototype of the modern Velocette system. Backward movement of the upper spring-unit anchorages along their slots increases the length of the lever-arm from A to B, increasing the effective spring-rate. Fig. 2 (above) Adjustment by notched cams, as used on the modern Girling unit shown on the right, varies the initial spring setting from soft (1) to medium (2) or hard (3).





(Left) Flash-back. This J.A.P.-engined A.S.L. (Air Springs Limited) of 1910 had pneumatic suspension at front and rear.

rear suspension in the proportions of 66% for the rider and 100% for the passenger. Figures are given for two sets of springs, rated at 225 lb./in. and 350 lb./in. respectively, measured *at the axle*.

It will be seen that with the maximum load and the lighter springs the machine is badly down at the rear and there is only 2 in. of bump-travel available. Under the same conditions, with the heavy springs, the attitude of the model and the travel available are virtually normal, so the head angle remains unaffected and the general "feel" of the rear suspension would be the same as with the single 12-stone load and the lighter springs. With the 8-stone rider and the heavier springs, however, the ride would be very hard and the rear end elevated nearly ¾ in. above normal. This illustration makes the necessity for load compensation quite clear.

A method of achieving partial compensation by relieving the frame of most of the additional passenger weight was described in the previous article, but a better solution, developed originally by Velocettes, is to alter the position of the springs in relation to the axle by making the locations of the upper anchorages, adjustable; see Fig. 1.

As the top ends are moved backward along the slots, the length of the effective lever-arm *A* is increased to *B* and the spring-rate *measured at the axle* is increased in the same proportion. The mounting points are not placed at a fixed radius from the axle-centre, but are arranged to be progressively lower towards the front in such fashion that the height of the saddle will remain almost constant when unloaded, irrespective of the spring setting. A rather similar scheme is used on a recent Guzzi road model, but in this case without rapid adjustability.

The second and more common solution is to use a hydraulically damped spring-unit of a fixed maximum length which controls the full-rebound position, and to provide a moveable abutment for the lower end of the spring, so that its initial compression can be varied merely by twisting the abutment, which has graduated stops bearing on fixed pins (Fig. 2). If the springs are of constant rate, this expedient corrects the attitude of the machine and leaves the full amount of travel available, but does not alter the rate, so that the suspension is still liable to bottom under heavy loads. Also the periodicity or frequency of vibration is reduced and the "feel" of the suspension is different.

If, however, double- or triple-rate springs are used, the effect will be to utilize all the coils and therefore bring the low rate into action at the light-load settings, but to push the close-wound coils up solid and thus increase the spring-rate in the heavy-load setting, thereby providing a very neat and relatively inexpensive solution to a problem which did much to undermine the value of rear springing in its early days.

Next week's article deals with rear frame construction.

EFFECT OF VARIOUS LOADS

State of machine and crew	Load on rear springs	225 lb./in. 4.25 in. total compression			350 lb./in. 4.4 in. total compression		
		Deflection	Available movement	Error in height	Deflection	Available movement	Error in height
	lb.	in.	in.	in.	in.	in.	in.
Solo, unladen	170	0.75	3.5	+0.5	0.49	3.91	+0.91
Solo, 8-stone rider ..	250	1.1	3.15	+0.15	0.71	3.69	+0.69
Solo, 12-stone rider ..	280	1.25	3.0	Nil	0.8	3.6	+0.6
Solo, 14-stone rider and 14-stone pillion passenger ..	500	2.22	2.08	−1.17	1.43	2.97	−0.03

Spring loadings, ratings and total compressions quoted are as measured at the rear axle, after allowing for any possible mechanical advantage conferred by the installation.