MOTORCYCLE ENGINEERING—7

REAR SPRINGING

A critical survey of past and present systems and their effect upon the basic frame design

by PHIL IRVING

THE desirability of rear springing was recognized 50 years ago and a number of makers experimented with it or even went into limited production. Many of these pioneer designs showed a disregard of essential principles and some merely took the form of additional mechanism attached to the fork-ends of existing unsprung frames—an expedient which has also been followed in more recent times.

Anyone who cares to delve into history will find that almost every type of springing system was tried in these early years (even to the use of air-sprung units), but the poor handling which many springers exhibited led to the whole conception of rear springing falling into disrepute, then to be almost abandoned for quite a while, despite sporadic attempts to reintroduce it.

This is a good example of a pattern of events noted in the introduction to this series—it is in effect a mechanical translation of the adage about "giving a dog a bad name." However, the picture has changed completely since then. Nowadays a motorcycle without rear springing would be almost as hard to sell as an unsprung motorcar, and the benefits in the way of better roadholding and braking are just as well appreciated as the more obvious improvement in bodily comfort.

To perform satisfactorily, a rear springing system must provide 3 in. of wheel movement (and preferably even more) and must ensure that the wheel remains strictly in the centre-plane of the frame at all times and under all stresses generated either by road-shocks or transmission forces. Further, it must not impair the transmission's efficiency or reliability and must be able to cope with wide variations of load—which may comprise one light rider or two heavyweights plus luggage. The effect of these requirements upon the detail design of the rear suspension system will come under review later; at this stage, it is desirable to concentrate on the structural problems involved in building a rear-sprung frame which will be mechanically rigid laterally and in torsion, without detrimental effects in other ways.

Broadly speaking there are three basic layouts: (a) the "plunger" system, with the axle carried by sliders moving inside housings attached to the fork ends; (b) the "short-link" system, with the axle attached to the ends of links pivoted at or near the fork ends; and (c) the "swinging fork" system, with the axle carried in a fork, pivoted on an axis lying between the tyre and the rear of the power unit. (See Fig. 1.)

Any conventional rigid frame can be adapted fairly easily to systems (a) or (b) without introducing a host of minor problems such as mudguard, silencer and rear stand attachments. Consequently they had considerable attraction in the days when alternative rigid and sprung models were listed.

System (b), however, has little to recommend it, because if chain drive is used the variation in distance between gearbox mainshaft and rear wheel centres is excessive, leading to great variation in chain tension. Further it is difficult to maintain torsional stiffness because the link-axle-link combination forms, in effect, a single crank, just as in the case of a bottom-link front fork, and this will twist easily unless the three components are so firmly attached that they form virtually a unit—a difficult requirement when the axle has to be made moveable for chain adjustment.

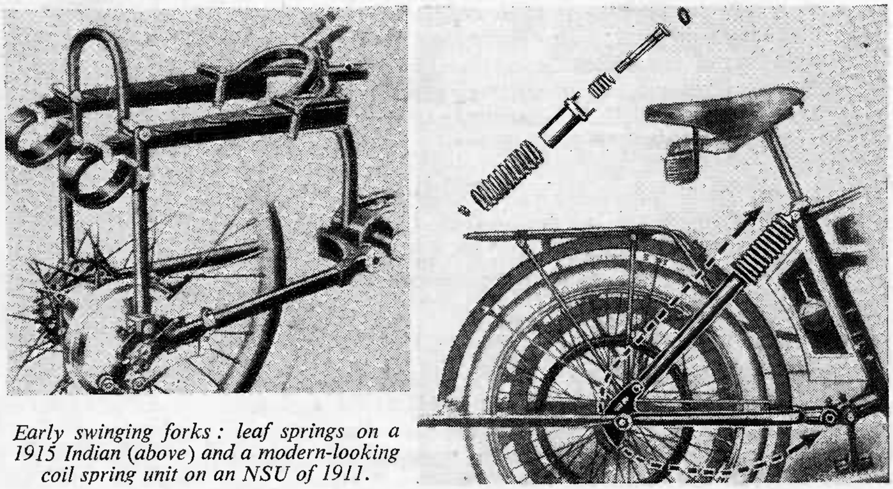
An interesting marriage of the short-link and plunger systems has been used for some time by Ariels, the linkage being so arranged that the axle moved in a curved path virtually centred on the gearbox mainshaft, thus obtaining almost constant chain tension at the expense of extra complication.

System (a) had a very good run for a number of years, largely because of its successful adoption on Norton racing machines. This public demonstration of the fact that a rear-sprung model could be made to handle did much to dispel popular distrust of the idea, although the system used is not, in principle, as good as the swinging-fork layout, represented at that time by several British and Continental makes.

The main trouble with any system involving parallel sliding members joined by a relatively flexible cross-member (the axle) is that under the action of loads applied more or less transversely to the tyre the axle will bend into an S-shape and one slider will move farther up than the other, thus permitting the wheel to deflect from the centre-plane (Fig. 1A). Any running clearance provided to permit free sliding movement will accentuate this bad effect. So will any lack of rigidity of the fixed portions in a vertical plane, and it is difficult to make these rigid because they are merely attached to the ends of tubes which are overhung several inches from the centre-section of the frame.

An additional source of weakness in some plunger systems was that the triangle previously formed on each side by the chain-stays and seat-stays was converted into a quadrilateral, dependent for vertical strength upon the strength of the tubes rather than upon the geometrical shape. Consequently fatigue failure of the tubes was not unknown, though its onset was largely a matter of how hard and over what surfaces the model was driven.

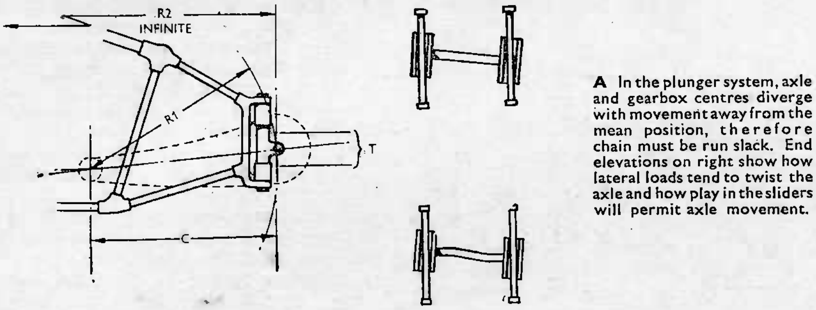
Plunger springing is the worst of the lot with regard to variation in chain tension,



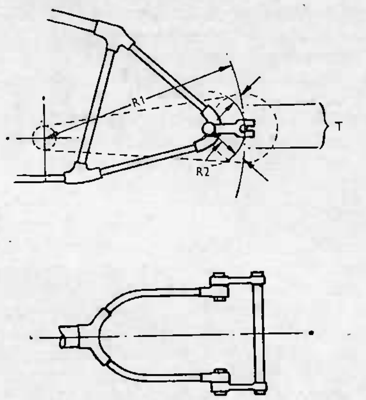
Early swinging forks: leaf springs on a 1915 Indian (above) and a modern-looking coil spring unit on an NSU of 1911.

FIG. 1—THE THREE PRINCIPAL SYSTEMS

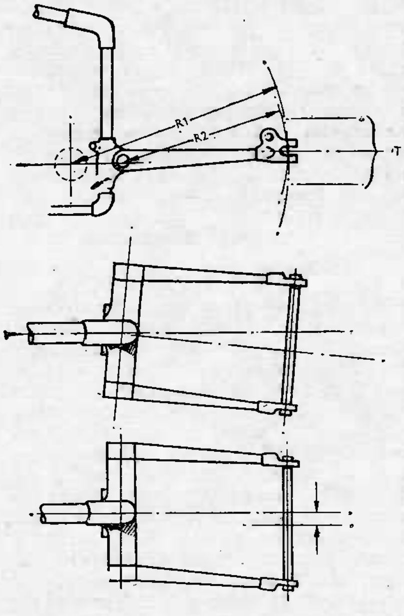
In all sketches, R_1 = distance between centres of gearbox shaft and axle, R_2 = radius of axle movement (in the plunger, movement is straight-line), and T = travel.



A In the plunger system, axle and gearbox centres diverge with movement away from the mean position, therefore chain must be run slack. End elevations on right show how lateral loads tend to twist the axle and how play in the sliders will permit axle movement.



B Weakness of the short-link system is the large divergence between R_1 and R_2 at extremes of movement. The "single crank" assembly, shown in plan view, is also weak in resistance to torsion.



C With the swinging fork, the difference between R_1 and R_2 remains small. Plan views show how poor designs may bend due to weakness of pivot mounting or be displaced laterally due to weakness in the fork itself.

CHAIN CENTRE VARIATION

Type of Frame	*Gearbox shaft to rear axle*	*Gearbox shaft to pivot centre*	*Rise of wheel above mean position*	*Centre variation*
	in.	in.	in.	in.
Pivoted fork ..	21	4.5	3	0.07 slack
Short-link ..	21	13	3	0.35 slack
Plunger.. ..	21	8	2	0.40 tight

because the chain tightens at the *extremes* of travel and so must be adjusted to be slack in the mean position. It therefore spends most of its time running slack and is liable to jump the sprockets, even if the chainguard is arranged to act as a retaining device —a somewhat barbarous expedient which has been resorted to at times.

On the other hand, if a swinging fork could be arranged to pivot about the centreline of the final drive, the chain tension would remain constant over the whole sprung movement. While it is not impossible to attain this result, in practice it is unnecessary, because if the pivot is placed fairly close to the gearbox the variation in tension over the permissible range of movement is so slight that it can be tolerated—especially as matters can be so arranged that the tension is correct in the normal position and is only slack at the extremes. This entails placing the pivot on a line drawn through the final-drive sprocket centres in the normal loaded position, or if anything slightly higher, as in general a wheel will spend more time above its normal position than below, owing to the action of bumps and the increase in effective weight on the suspension when cornering.

The table above shows the amount by which the centre-distances vary with the three systems and indicates clearly the superiority of the pivoted fork. When studying the table, remember that the variations shown are in linear distance; the effect on the chain as measured in the usual way by the amount of up-and-down slackness introduced is very much greater.

Various ideas, such as making the sliders curved, have been propounded for reducing the tension variation with the plunger system, but they inevitably introduce extra complication and the whole idea has ceased to be of much practical value except on some very light models where a limited amount of movement is all that is needed. The pivoted fork has, in fact, become almost universally adopted, though even this system introduces some complications which must be given close attention if a successful design is to be the outcome.

As the only positive connection between the fork and the frame is the pivot-bearing, it follows automatically that the duty of resisting dynamic and transmission loads devolves solely upon this component and its attachment to the main frame. Any inadequacy in stiffness of mounting, or looseness in the bearing, will permit the wheel to move out of line, irrespective of whether or not the fork itself is sufficiently robust.

Consequently the bearing—or rather bearings, as there is usually one at each side—should be as far apart as possible. But this is not sufficient in itself; if for instance, a wide lug carrying the bearing is brazed in the middle of a long, slender down-tube, this tube will be subjected to severe bending and torsional stresses and in time will fail by fatigue unless of adequate diameter.

The directions of the loads imposed depend upon the layout of the springing adopted. If, for instance, leaf springs extending backwards from the frame were fitted and connected to the fork ends by shackles—a layout which was used on several ancient models and one or two more recent experimental jobs—the weight is carried almost directly by the springs and the bearing has to withstand only transmission loads and those due to transverse forces on the wheel. This obsolete method, however, places very high local stresses on the spring anchorages.

Almost universal nowadays is the use of coil-spring units, usually inclined forwards at the top to some extent and thereby generating a horizontal component of the gravitational force, which tends to pull the bearing backwards by an amount depending upon the angle of the springs and their disposition with regard to the rear axle (Fig. 2). This force acts in the opposite direction to the pull of the rear chain, therefore if any slackness or flexibility is present in the bearing or its mounting the rear fork will not remain in a central position but will swing to one side or the other according to whether power is on or off.

(Continued overleaf)

An indication of whether a suspension system is lacking in lateral rigidity can be obtained by squeezing the top and bottom runs of the chain together with the fingers. If there is any flexibility, it will be possible to observe movement of the fork-ends in relation to the frame; it may also be possible to detect just where the flexure is occurring and, with this guidance, to reduce or even eliminate it entirely by adopting an appropriate course of action.

However effective the bearing itself, the result will not be good unless its mounting is sufficiently substantial; further, the more direct the path which transmission loads must follow, the better. The ideal is to mount the bearing actually on the engine/gearbox unit (big M.V.s and Ariel " Leader "), or by short, straight, widely spaced engine plates in direct compression. Satisfactory results, however, will be obtained from a frame with a single saddle down-tube provided this is of large diameter and the lug forming the pivot bearing attachment is bolted to the power unit with plates preferably spaced several inches apart.

The pivoted fork appears to be particularly suitable for shaft-drive machines because the propeller shaft can be concealed within one leg and only one universal joint will be required, whereas there must be two if a plunger system is employed. Further, the angular deflection is low enough to permit the use of an ordinary Hookes-type universal (that is, of the Hardy-Spicer pattern) though a constant-velocity joint would be preferable if the fork is very short and has a large range of movement.

Torque Reactions

The effects of power and brake reactions are, however, quite different in the cases of chain drive and shaft drive, and this alters the picture considerably. With a chain, there is no direct torque reaction, except the tendency under power to lift the front wheel and rotate the whole machine around the back axle—an effect which transfers some or all of the weight on the front wheel to the back, thus causing the rear springs to compress beyond their normal position and the pivot-bearing to sink. This action is augmented or diminished according to the height of the pivot bearing in relation to the rear axle; if the pivot is the lower of the two, a downward vertical component of the compression-load in the forks will be introduced tending to depress the pivot still further, whilst if it is the higher the reverse is the case. Usually both axes are at about the same level and the effect in practice is never very considerable.

With shaft drive (Fig. 3) the conditions are altered by the presence of strong torque reactions, one of which (*A*) tends to rotate the bevel-box backwards in relation to the wheel, and the other (*B*) tends to rotate the bevel-box in the same direction as the propeller shaft.

These effects can be easily envisaged. First, consider the crown wheel to be locked in position; the pinion will then try to " climb up " the crown-wheel teeth and attempt to take the casing with it. Next, if the pinion and crown wheel are envisaged as locked together, the whole assembly will try to rotate around the propeller shaft. If the bevel-box is bolted to one of a pair of plungers, reaction (*A*) is resisted locally by that plunger; but if the forks are pivoted they tend to rotate backwards, putting an upward force on the bearing which, on a powerful model, may be as much as 400 lb. This, fortunately, acts in opposition to the general tendency of the rear end to sink when accelerating, but may be the greater of the two forces if the forks are very short and the torque transmitted is high.

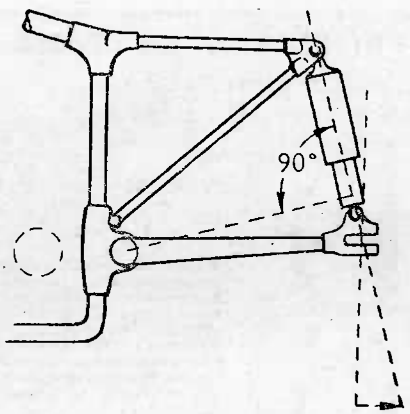


Fig. 2 *How inclination of the spring unit in a swinging-fork assembly produces a rearward-acting horizontal component of the reaction to the weight.*

On the other hand, under braking conditions this torque reaction is reversed and the pivot tends to be depressed, so that under alternating applications of power and brakes a " bucking " action can be initiated which does not improve the handling. This possibility was avoided in the early shaft-drive M.V.s by duplicating the forks and mounting the bevel-box between the ends, thus making the whole assembly a jointed parallelogram able to absorb braking and torque reactions in a vertical plane within itself, at the expense of much weight and complication.

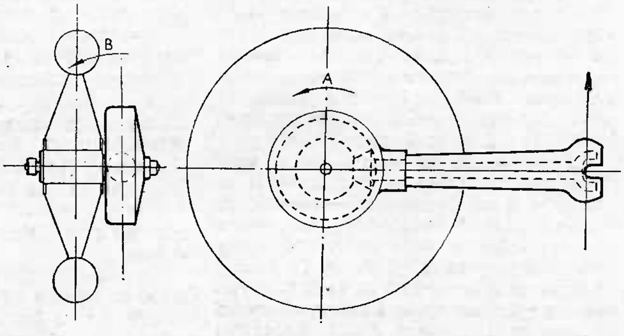


Fig. 3 *Torque reactions upon a shaft drive with power " on." In a pivoting fork system, force A places an upward load upon the pivot bearing. Force B tends to twist the assembly about the longitudinal axis of the machine.*

The transverse torque-reaction (*B*), tending to twist the axle out of the horizontal plane, would have still worse effects on steering if not guarded against by sufficient torsional stiffness of the forks, but if they are stiff enough to cope with normal road shocks they should be able to handle this reaction without a detrimental degree of deflection. The reaction as measured in foot-pounds varies with the gear ratio and engine power, but in round figures may be taken as 150 lb./ft. per litre of engine capacity in bottom gear. It can reach much higher momentary values when rapid down-changes are being made against the inertia of the engine. The reversal of torque may then cause the rear end to kick sideways if the rear fork momentarily twists.

Resistance to lateral loads is a major factor in the design of any suspension system. Since bumps are struck obliquely, alternating lateral forces are continually applied to the tyre, and if the suspension mechanism contains any slackness the wheel will oscillate from side to side even if the impulses are small. Heavier forces may cause elastic deformation or even a permanent set in the structure, as when cornering at extreme angles the lateral impulsive load on a 400-lb. machine may exceed 600 lb.

To resist forces of this order without detrimental effects, the stiffness of the wheel relative to the centre-section of the frame should be such that a load of 250 lb. applied sideways to the tyre near its contact point will not twist the wheel more than 1° (or a little less than $\frac{1}{4}$ in. by direct measurement), nor displace the whole wheel by more than $\frac{1}{8}$ in. It is of course essential that on removal of the load the wheel should return to its original position, which may not occur if the fork is built up from several components in such a manner that it relies for its rigidity on frictional grip rather than on precise mechanical location.

When cornering with a sidecar the lateral forces may be very much higher, but in this instance flexing of the structure will not have such serious effects as on a solo, provided that there is sufficient strength to prevent the structure taking a permanent set.

It will be appreciated, therefore, that the design of a frame with pivoted rear springing introduces many problems which do not exist in the rigid frame and obviously this has a profound effect on the whole layout. Provision has to be made for a crew of two, with the passenger's weight overhung a long distance from the main frame, and means must be found for attaching silencers, stands and other items which can simply be bolted to convenient places on a rigid or plunger-sprung design. The fitting of a sidecar also must be borne in mind.

These problems, and the detail design of the rear forks themselves, will be dealt with in next week's article. In the meantime, anyone who is deeply interested can find an analysis of the subject in the paper "Rear Suspension of Motor Cycles," by the present writer, in the proceedings of the Institution of Automobile Engineers, 1944-45, Vol. XXXIX.