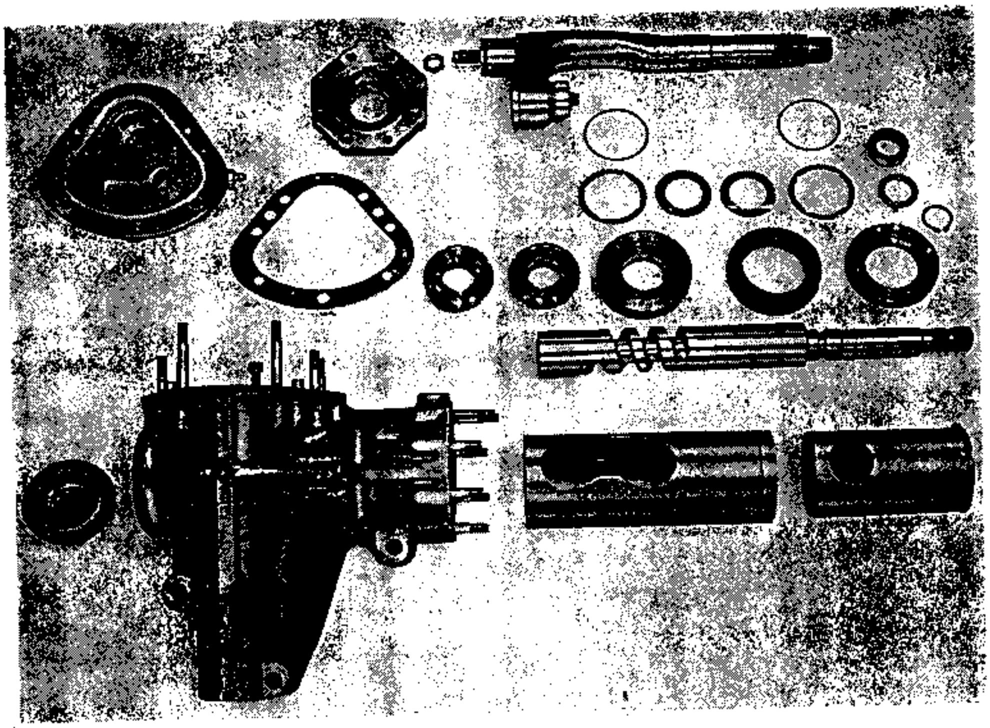


Components of the Hydrosteer fully integral, power-assisted steering gear. Just above the camshaft can be seen the parts of the valve, valve chamber and sealing assemblies. The casing and its top and end covers are LM22 aluminium alloy castings

HYDROSTEER POWERED STEERING

Integral System for Larger Cars; Cam-and-Peg Mechanism and Spool Type Valve; Marked Ratio Variation from Straight Ahead to Full Lock

UNLIKE previous power-assisted steering systems produced by Hydrosteer Ltd., of Luton, this company's latest unit is of the fully integral type: as in the case of the Burman and Saginaw designs, described in the February and October 1959 issues of *Automobile Engineer,* there are no external jacks. The cam-and-peg mechanism employed is of largely orthodox layout, except in that the cam is surrounded by the skirt of a piston working within the body of the steering box; the hydraulic power is transmitted directly from the piston to the peg carrier. A hydraulic valve, of spool type, moves axially in response to steering wheel movement to cause oil pressure to be applied to the appropriate side of the piston. The equipment has not yet been adopted for fitting to production cars, but it is being tested by several manufacturers.

From the outset, it was decided that the fail-safe type of design should be adopted: in other words, the driver should still be able to retain full control of the car in the event of a failure of the power system. Another essential was that jamming of the valve, either under very heavy loading or because of foreign matter in the oil, should be made virtually impossible. A third primary requirement was a high speed of response of the hydraulic assistance. In the design adopted, the valve can always keep pace with the speed of operation of the steering control, though an inadequate pump output can result in the full degree of servo not being available for exceptionally rapid wheel movements.

Rapid response is, of course, only one desirable feature of a powered steering gear: the response must also be sensitive and progressive if lack of "feel", a point of criticism of certain earlier designs, is to be avoided. The Hydrosteer technicians have aimed at giving the unit a feel comparable with that of a good unassisted system. They claim to have obtained this by careful attention to the degree of feedback, and to the design of the seals and the springs of the hydraulic valve. These springs are of disc type, with a high rate, and are balanced in their neutral position. Other important considerations in the design were that the size and weight should be the minimum consistent with adequate strength, and that the valve assembly should be of a simple type in which wear and its attendant backlash would be restricted.

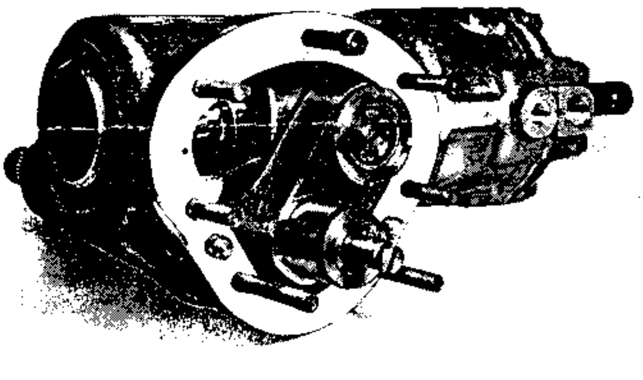
To obtain the maximum benefit from power-assisted steering, it was felt desirable that the ratio should vary appreciably between the straight-ahead position and full lock on each side. Obviously, a relatively high overall ratio was desirable to avoid excessive steering wheel movement during low-speed manoeuvring. However, in view of the low steering effort attainable, too direct an action in the middle of the range could result in the vehicle's being subjected to dangerously large lateral forces at high speeds. The cam selected therefore provides moderate gearing in the region of the straight-ahead position and a progressive numerical fall in the ratio towards the locks. On a typical example, the ratio diminution from the middle to full lock in each direction is about 39 per cent.

The nominal maximum working pressure is 1,000 lb/in^2, but there is usually sufficient power available at 700 lb/in^2. In the event of failure of the power supply, only 0·6 in movement of the steering wheel rim is needed, in either direction, before positive mechanical transmission occurs. Such a small amount of backlash should be undetectable under normal operating conditions. A typical Hydrosteer unit has length, breadth and height dimensions, as drawn, of 11$\frac{31}{32}$ in, 5$\frac{7}{16}$ in, and 14 in, though the last figure varies with the length of rocker shaft specified; an average dry weight is quoted by the manufacturers as about 20 lb.

Design and construction

The main housing and its three covers are substantial castings in LM22 aluminium alloy. A hole in the lower end of the body accommodates the spigot of the bottom cover, which is pushed home, from within, against a shoulder; oil sealing is effected by means of an O-ring. Contained in a counterbore in the cover are a p.t.f.e. oil seal, of castellated section, and the channel-section outer race of the lower of two uncaged ball bearings that support the camshaft. The upper end of the cover is relieved circumferentially to provide a spigot for the sleeve in which the piston operates. This sleeve is of cold-drawn seamless tubing, to B.S.S. 980/CDS2; it is a push fit in the bore of the housing and, since it is clamped by the components above it, prevents any upward movement of the bottom cover. The wall thickness of the sleeve is 0·109 in, and near the upper end is a kidney shape hole, which provides operating clearance for the peg assembly on the end of the rocker arm.

Thirty-two balls, $\frac{5}{32}$ in diameter, are used in each of the shaft bearings just mentioned. They run directly on the camshaft, which has a diameter of 1$\frac{3}{8}$ in for most of its



The inner castellated ring-nut on the rocker arm is for adjusting the cup-and-cone bearings of the peg. At the far end of the casing are two tapped holes, which are for the unions of the oil supply and return

length and is machined from En.34 steel; this component is case-hardened on the cam track and on the bearing tracks. The shaft and the piston, which is of En.32B steel, have a ground finish on their working surfaces, whereas the bore of the sleeve has a special fine finish. Although the piston is supported in the sleeve for its full length, the presence of the cut-out makes it necessary for the external piston-ring type seal to be at the upper end. To minimize friction, the bore of the piston does not bear on the shaft, the only contact being by the internal hydraulic seal. This seal is of unusual design: it comprises a D-section p.t.f.e. ring, disposed with the curved surface bearing on the shaft, lightly loaded inwards by a square-section, synthetic rubber ring installed around it in the same groove.

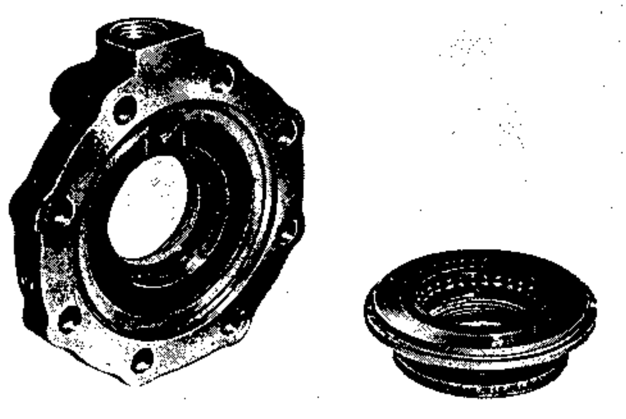
The cam track has a section somewhat similar to that of an Acme thread. Because of the geometrical considerations, the track is not a regular helix but varies in pitch, included angle and depth. The radius of swing of the follower peg is greater than the distance separating the axes of the rocker shaft and the camshaft. It follows that, as the rocker arm swings from the straight-ahead position towards either lock, the peg axis travels through an arc that successively approaches, intersects and diverges from the vertical plane containing the shaft axis. This lateral movement results in a continuous variation in the required depth of meshing of the peg. Also, since the peg axis is only radial, relative to the camshaft, at two points on the arc of travel, the included angle of the track flanks varies with the amount of offset.

In the example already mentioned, the numerical reduction in the ratio towards each lock is 15·6 : 1 at the straight-ahead position, falling to 9·56 : 1 at full lock. This variation is, of course, obtained by varying the pitch angle of the track. It will be appreciated, however, that the lateral movement of the peg affects this angle also: towards one lock it is in the same direction as the rotation of the shaft, whereas towards the other lock it is in opposition. In the former case, therefore, the pitch angle should be greater than in the latter to give the same effective ratio on each side of the neutral position. Because of these various factors, the true form of the track is a complex one, and it is produced by a generating process, using a form tool of the same section as the peg: the tool follows the movement of the rocker shaft while the camshaft is appropriately rotated.

A single forging of En.16T steel comprises the rocker arm and its shaft. The shaft is slightly cranked to clear the piston sleeve, and is carried in two phosphor-bronze plain bearings. Of these, the lower is a Vandervell wrapped, steel-backed component, and is fitted in a long boss projecting from the main housing. Below it is a synthetic rubber seal of X-section, retained by a washer and circlip. The upper bearing, a solid bush, is in the rocker arm cover; its oil seal, also of X-section, is housed in a groove at the upper end of the shaft. Both bearings are lubricated by the hydraulic oil.

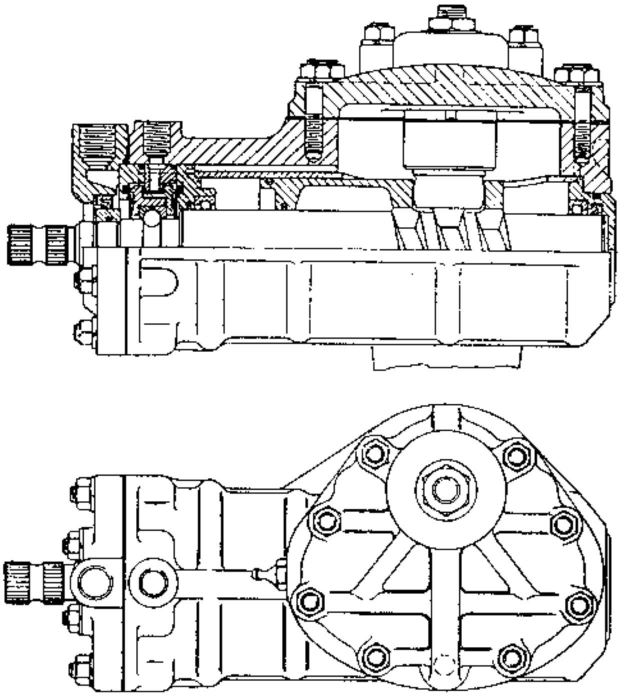
An unusual method has been adopted for supporting the rocker shaft, resisting cam thrust and adjusting the mesh of the peg: in effect, the shaft is suspended from the cover by means of the adjusting screw. Bored in the upper end of the rocker shaft is an axial, tapped blind hole. At the bottom of this hole is a smaller hole, into which is pressed

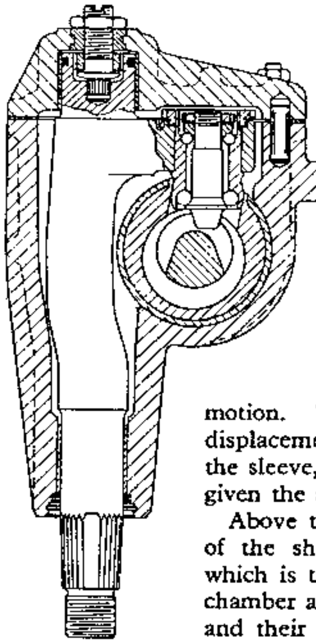
In the end covers, p.t.f.e. seals of castellated section are employed; the bottom cover carries the outer race of one of the camshaft bearings



a hardened plug. The adjusting screw has a button head at the lower end, and over its shank are slipped a Belleville washer and an externally threaded sleeve. This sleeve is screwed into the rocker shaft until the button is nipped between the plug and the washer. Then the cover is screwed on, and the assembly is offered up to the body.

After the nuts securing the cover have been tightened, the depth of engagement of the peg is set, by means of the adjusting screw, to give the minimum of backlash at the





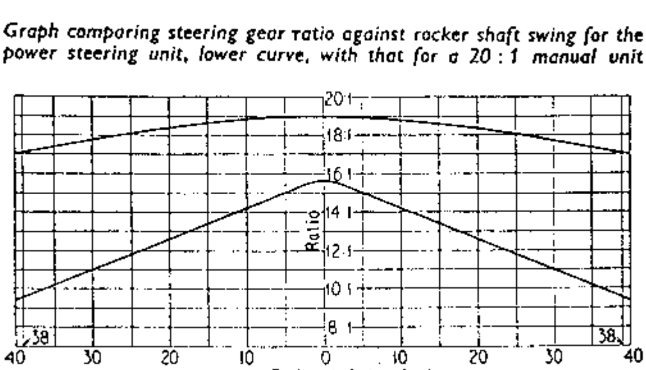
General arrangement of the Hydrosteer gear; the seals now used between the end covers and the camshaft are of a later type. The method of adjusting the depth of meshing of the peg, and of taking the thrust from the cam, is unusual. A vertical dowel peg prevents rotation of the piston sleeve in the casing

straight-ahead position. Finally, the screw is locked in position by an external nut. Dowels position the cover, which is secured by studs and a socket screw, all of $\frac{5}{16}$ in diameter. A metal gasket is fitted between the cover and the body, and it embraces a synthetic rubber sealing ring of square section.

The cam follower peg is made from En.34 steel, case-hardened, and its shape is a truncated taper of 30 deg included angle. Friction is minimized by permitting the peg to rotate, in two ball bearings of the cup-and-cone type. The two outer races are formed on a hardened, En.351 steel sleeve pressed into the rocker arm, against a shoulder, while the inner race of the lower, and larger, bearing is machined on the spindle of the peg. Because of the need for adjustment of these bearings, the cone of the upper bearing is a light push fit on the upper end of the spindle. A castellated adjusting nut is fitted above the cup, and is locked by a special cup washer, which is peened into two of the castellations. The sleeve that houses the bearings is secured in the rocker arm by another castellated nut and locking washer. Ten $\frac{1}{4}$ in diameter balls are used in the lower bearing and thirteen $\frac{3}{16}$ in diameter balls in the other.

A hole in the upper wall of the piston accepts the lower end of the sleeve, the section of the piston being thickened round the hole to provide an adequate bearing surface. It will readily be appreciated that, as the rocker arm swings in its horizontal arc, the piston receives a part-rotary motion. To accommodate the angular displacement of the axes of the hole and the sleeve, the lower end of the sleeve is given the appropriate spherical form.

Graph comparing steering gear ratio against rocker shaft swing for the power steering unit, lower curve, with that for a 20 : 1 manual unit



Above the piston, near the upper end of the shaft, is the valve chamber, in which is the spool type valve. Both the chamber and the spool are of En.1a steel, and their combined design is such that, for assembly purposes, divided construction has to be employed for each: the chamber is made in three parts, located relative to one another by dowels, while the spool comprises two parts, screwed together. The chamber components are contained in the housing, and the spool is mounted on the shaft.

The lower portion of the valve chamber houses the outer race for the upper of the two ball bearings carrying the camshaft. It is a push fit in the main casting and its lower end has a male spigot, which is pressed into the piston sleeve. In the bore, immediately above the race, is a groove containing an oil seal of similar type to that in the bore of the piston. On the upper face of this component is an annular groove, of stepped section, which houses the lower of the two valve seal assemblies. Each of these comprises an O-ring covered by a sealing plate—a thin steel annulus. The thickness of the O-ring is sufficient to hold the sealing plate just clear of its seating in the groove. Excessive spread of the ring is avoided by giving the bottom of the groove a 45 deg chamfer on the inner side. Encircled by the seal assembly is a shallow annular lip, which is the abutment for the lower of the two springs of the valve.

The intermediate portion of the valve chamber is of cylindrical form, with an inwardly projecting flange at its upper end. Machined in its periphery is an annular groove, which registers with the pressure connection from the pump. Four radial holes, equally spaced round the circumference, lead from the bottom of the groove to the inner face of the flange. The periphery is relieved at the lower end to accept the O-ring that forms the seal between the component and the body, in which it is a push fit.

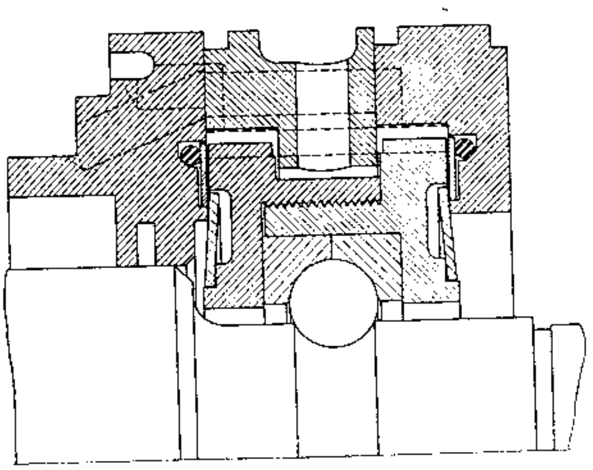
Completing the chamber is the portion forming the upper end-wall. It has a short cylindrical extension that seats against the upper face of the radial flange on the intermediate portion. As are the other two components of the chamber, it is a push fit in the body, against which it is sealed by an O-ring in the same manner as the intermediate portion. This third portion carries the second valve seal assembly, and it also has an annular lip for the upper spring abutment.

There is considerable radial clearance between the valve spool and the chamber, the insensitivity of the valve to foreign matter in the oil being attributable to this and to the end sealing arrangement. Mention has already been made of the divided construction of the spool. Between the halves is clamped the transversely split outer race of a deep-groove ball bearing. This bearing positions the spool on the shaft while permitting the latter to revolve. Since the length of the spool is critical, the seating face between the halves is accurately machined on the end of the lower, female portion, and the race is shimmed as necessary for its correct fit in the spool.

The spool is of double-flanged form, and the inwardly projecting flange of the intermediate part of the chamber is within the waisted portion. In addition to the radial clearance between the inner face of the chamber flange and the waisted portion of the spool, there is axial clearance between that flange and the end flanges of the spool, the purpose of which will be explained later. On each end of the spool is an annular lip, of the same radius as the sealing rings in the chamber, and a spigot, which is of smaller diameter than the lip. On each spigot fits one of the washer type control springs; the spigots are relieved at intervals to permit the passage of oil past the springs. As is shown by the accompanying graph, the combined rate of the springs varies a little but is in the region of 15,800 lb/in. In the neutral position of the valve, the springs are preloaded in opposition, between the lips on the chamber end walls, the deflection of each being about 0·01 in.

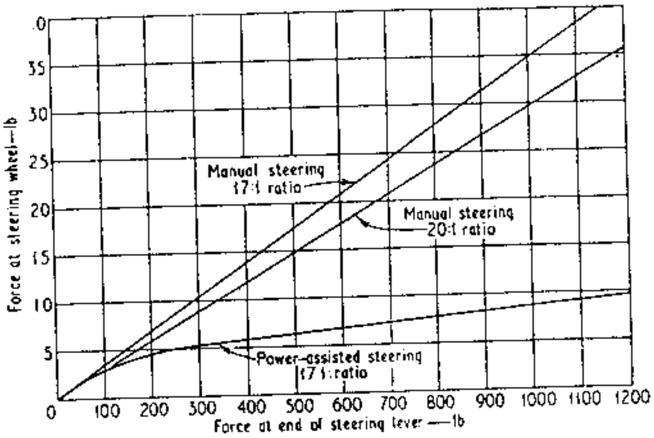
It can be seen from the appropriate illustration that there are four notches in the periphery of each of the two end flanges of the spool. In these notches sit steel rods, which pass through axial holes in the flange of the intermediate portion of the valve chamber. These rods fulfil the triple purpose of locking the halves of the spool together, of preventing the spool from rotating relative to the chamber, and of retaining the sealing plates in their grooves.

Some explanation is desirable here regarding the disposition of the various oilways. It has already been stated that the feed is to the intermediate portion of the valve chamber. Communication between the lower end of the chamber and the space below the piston is through an oblique drilling in the chamber wall to an axial groove cored in the body immediately above the piston sleeve. Passages formed in the three portions of the chamber lead from the upper end



of this chamber to the upper side of the piston. The return passage, on the low-pressure side, is through the reliefs in the valve spring spigots, and also through the clearances between the spool and the shaft, and between the shaft and the upper wall of the valve chamber.

A counterbore in the top cover fits closely over the upper portion of the valve chamber, so that the tightening of the cover clamps the complete internal assembly. Sealing between the cover and the body is effected by a square-section, synthetic rubber ring, and the cover is held down



In this graph, force at the steering wheel is plotted against force at the steering lever for typical manual units and for the powered system having a ratio of 17 : 1. The reduction in the effort is considerable

by eight $\frac{5}{16}$ in diameter studs and nuts. Since the cover is in contact only with the return side of the fluid circuit, high-pressure seals are not necessary between it and the shaft. However, the sealing arrangement is of interest since it makes possible the employment of the same size of seal as at the lower end, in spite of the smaller diameter of the shaft. The advantage here is that the longitudinal forces acting on the shaft are balanced, so there is no resultant load tending to move it in one direction or the other.

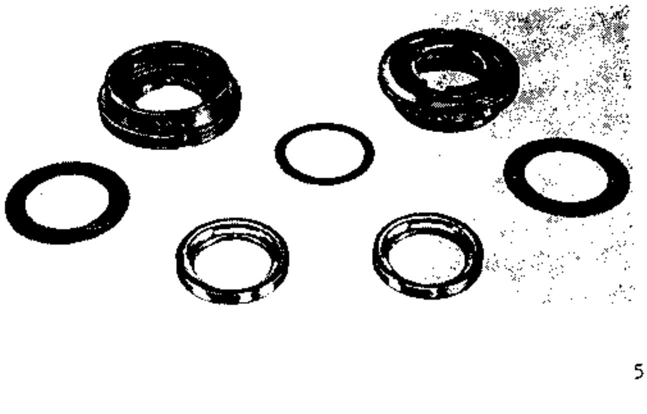
On the shaft, a short steel sleeve is held against a shoulder by a circlip fitting into a groove. It is a push fit on the shaft and its bore is grooved to take an O-ring. A second, smaller counterbore in the top cover houses a castellated p.t.f.e. seal identical with that at the other end of the shaft. This seal is retained by a washer and circlip, and it bears on the periphery of the sleeve, which is chromium plated to resist wear.

Principle of operation

Oil is supplied to the unit under pressure by a Hobourn-Eaton pump, of the latest type, in which hollow rollers are employed in place of vanes; this pump is belt driven from the engine. The hydraulic fluid, which is normally an engine oil of SAE 10W rating, is housed in a reservoir, of 2 pints capacity, which may be part of the pump assembly or be separately mounted. Flexible hoses connect the pump and reservoir to the steering box, and the couplings of the pump hose are of the high-pressure type.

From the valve chamber there are three possible paths for the oil: one is to the space above the piston, the second communicates with the lower side of the piston, while the third leads to the return to the oil reservoir. In the neutral

Left: Enlarged section of the valve and chamber assembly, showing the end seals, the preloading of the springs, and the divided construction of the valve and the bearing race. Below: The valve components; on the end of each half of the spool is a spigot that carries a spring

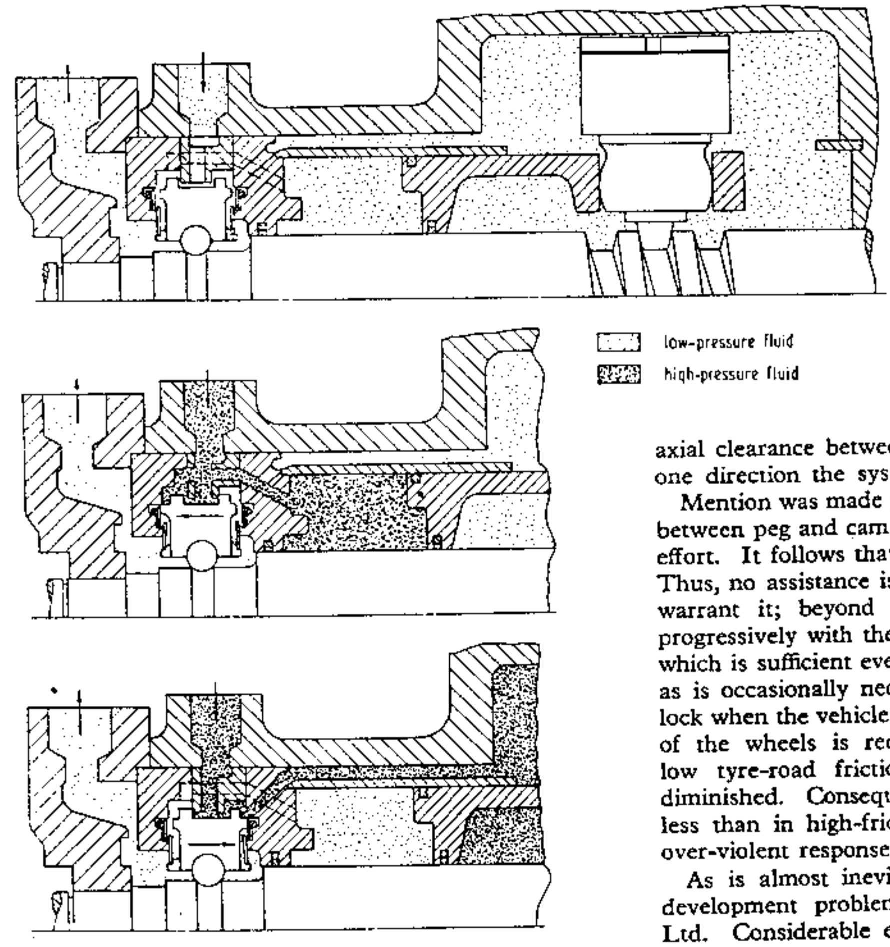


position, when no steering effort is being applied, the spool occupies a central position in the valve chamber. Neither ring seal is then in use, and fluid is free to circulate round the entire system, without resistance, and to return to the reservoir. This situation is illustrated in the first of the three accompanying operational diagrams.

Rotation of the steering wheel, and hence of the cam track, in either direction produces an axial reaction between the peg and the track. The magnitude of this reaction depends, of course, on the effort applied to the steering wheel and on the resistance to swivelling offered by the front wheels. Since the shaft has a limited amount of axial freedom, by virtue of the spool end clearances mentioned earlier, this reaction tends to cause the shaft and spool to move against the resistance of one of the springs.

If the reaction is strong enough, the load on the spool overcomes this resistance, and axial travel of the shaft occurs. This travel brings the lip on the appropriate end of the spool into contact with the adjacent seal, thereby closing the direct communication, at that end of the spool, between the oil supply and return passages. Direction of the oil pressure to the correct side of the piston is effected by introducing a flow restriction into one of the two remaining paths open to the oil.

The space between the end flanges of the spool and the flange in the chamber forms the communication between the two internal passages to the piston. It will therefore be apparent that, when the spool moves, and one of its flanges approaches the chamber flange, the entrance to one passage is restricted while that to the other is enlarged. The resulting pressure differential causes the piston to be moved in the required direction. Full powered assistance is attained on contact between the two approaching flanges, since the bleed from the high-pressure to the low-pressure side of the spool is then cut off. In the second and third operational diagrams, the conditions of maximum power assistance in the two directions are shown. Once the axial clearance between spool and chamber is taken up in one direction the system becomes mechanically positive.

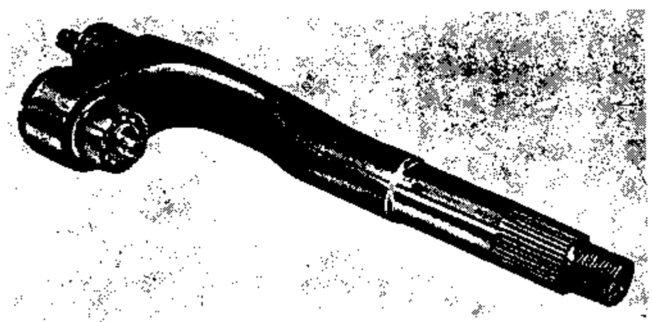


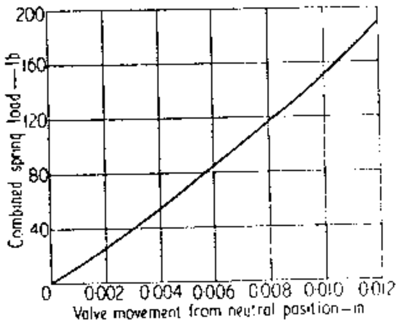
Diagrams indicating the passages for low- and high-pressure fluid within the Hydrosteer unit. In the top view no steering effort is being applied; in the second the piston is being moved to the right, and in the third its travel is reversed

Mention was made previously of the fact that the reaction between peg and cam track varies with the applied steering effort. It follows that the spool travel must vary similarly. Thus, no assistance is given until the effort is sufficient to warrant it; beyond that point, the assistance increases progressively with the effort up to the maximum available, which is sufficient even for violent manoeuvres at speed or, as is occasionally necessary, for putting the wheels on to lock when the vehicle is at rest. If the swivelling resistance of the wheels is reduced, however, as in conditions of low tyre-road friction, the reaction is correspondingly diminished. Consequently, the degree of power applied is less than in high-friction conditions, thereby avoiding an over-violent response.

As is almost inevitable with any new product, certain development problems were encountered by Hydrosteer Ltd. Considerable effort had to be devoted to obviating any stickiness in the action of the valve seals, and to eliminating oil leakage. Another difficulty experienced was the tendency of a pocket of air to form at the highest point

Below: The rocker arm and its cranked shaft are a single forging. On the end of the sleeve that carries the peg is a spherical seating face





Graph of the combined rate of the two valve springs, which work in opposition and are given a degree of preloading of the rocker shaft housing. Consequently, a bleed screw is now fitted at this point, in the cover, and bleeding is carried out in the same manner as on hydraulic braking systems. A more recent refinement is the embodiment of a hole in the piston wall, which at full lock permits a bleed to take place from the high-pressure to the low-pressure side, thus avoiding hydraulic overloading of the stops.

The design objective in respect of road behaviour was referred to initially. Its attainment was recently confirmed in a test run on a much used Super Snipe car. The route taken embraced town running, main roads and lanes, and included several acute corners, that had to be taken at low speeds, and more than one series of fast bends involving rapid changes of lock. Only in respect of the lightness of the steering was there any obvious indication of power assistance, and the feed-back was sufficient to avoid "deadness". The gearing finally chosen for this installation would be difficult to improve upon: only about $2\frac{1}{4}$ turns of the steering wheel were needed from lock to lock and, though control was positive—without being excessively sensitive—at high speeds, no substantial effort was needed near full lock when the car was crawling, or even when it was at rest. It was considered that low-speed manoeuvring involved no more work than it would with the average 1-litre car weighing about 15 cwt. The self-centring effect was mild, but adequate in view of the relatively high gearing.