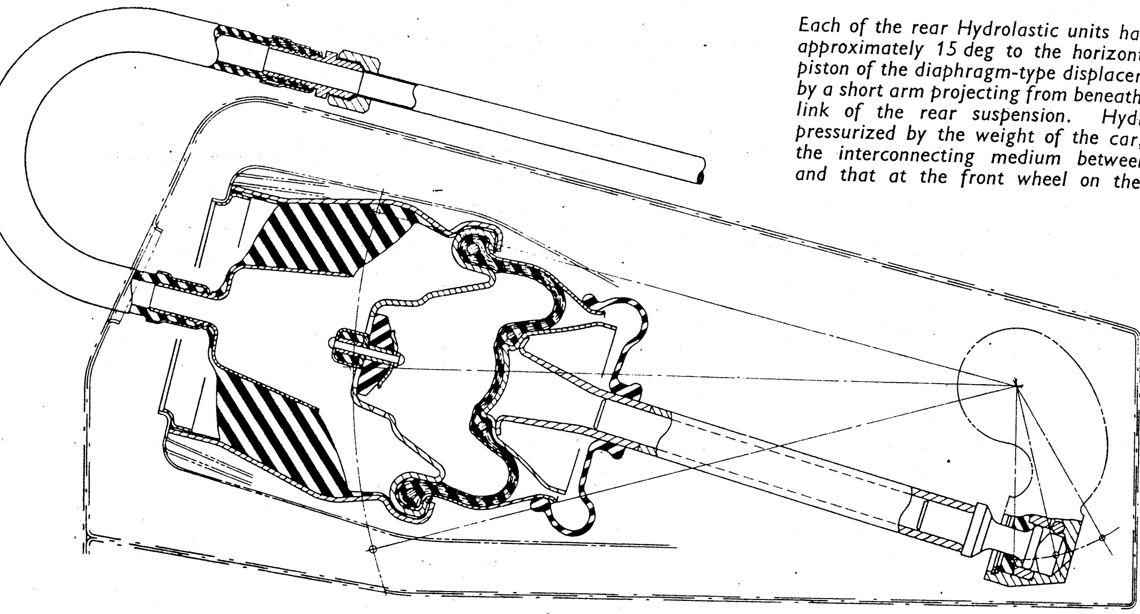


*Each of the rear Hydrolastic units has its axis at approximately 15 deg to the horizontal, and the piston of the diaphragm-type displacer is deflected by a short arm projecting from beneath the trailing link of the rear suspension. Hydraulic fluid, pressurized by the weight of the car, is used as the interconnecting medium between this unit and that at the front wheel on the same side*

# HYDROLASTIC SPRINGING

## *The Design and Development Story, as told by Alex Moulton, the Originator of this Hydraulically Interconnected Rubber Springing System Used on the Morris 1100 Car*

IN view of the considerable success of the ADO 15 vehicles, which have been introduced in the last three years by the British Motor Corporation, it has been obvious that a range of larger cars of similar layout, designed by the same team, headed by Alec Issigonis, would follow. This now-familiar arrangement — a transversely installed, in-line engine built integrally with the transmission unit; front-wheel drive and a long wheelbase; independent suspension, with rubber springs and small-diameter wheels—offers considerable benefits in terms of road-holding performance, fuel economy and the ratio of the interior to overall volume of the car. A factor that has contributed to both the excellent road-holding and the high efficiency in terms of space utilization has been the use of Moulton cone-type, rubber springs for the road wheels—they occupy a relatively small space by comparison with that required for orthodox steel springs, and their progressive-rate characteristics can be brought closer to the ideal than could those of the conventional springs.

*By virtue of the installation of the Hydrolastic system, the ride of the Morris 1100 is virtually free from pitching, and its roll-stiffness is high. This car is shown rounding a corner at speed over rough ground*

The larger car now introduced, the ADO 16, is to be known to the general public as the Morris 1100, and it is described later in this issue. From the technical viewpoint, its most interesting feature is the ingenious system of hydraulically interconnected rubber springs — also of Moulton design—which are hardly less compact than those of the ADO 15. The standards of road-holding and ride afforded by this layout have already been the subject of wide acclaim. The layout, termed the Hydrolastic suspension system, was developed through the collaboration of three organizations — British Motor Corporation Ltd, Moulton Developments Ltd, and Dunlop Rubber Co. Ltd. In each car there are two separate, identical systems, one for the left- and one for the right-hand pair of wheels: there are no levelling mechanisms, pumps or accumulators, and no need for adjusting headlamp angle to compensate for vehicle loading; the two systems are pressurized by the weight of the car. The cost is said to be comparable to that of an orthodox springing and damping system on a car with all-independent suspension.

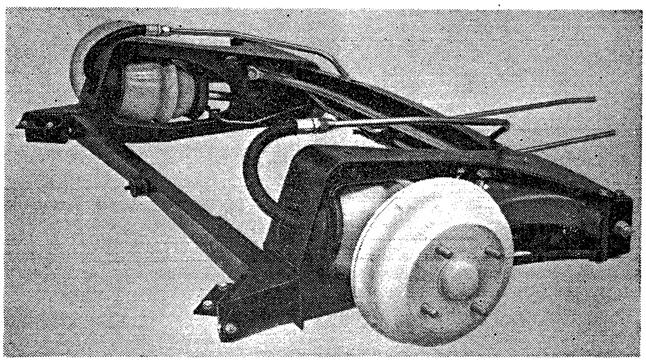
Judged in absolute terms, Hydrolastic springing is an innovation of the very highest technical merit. In the overall field of vehicle springing, its introduction on a small family car in volume production is an especially noteworthy aspect, and great credit is due to the British Motor Corporation for sponsoring the work. Experience with this new vehicle has already demonstrated convincingly that a highly efficient springing and damping system does not necessarily carry penalties in respect of bulk, complexity, weight, reliability or cost.

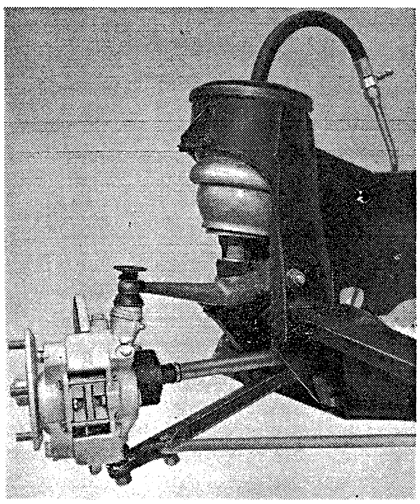
The fact that a totally original concept has been brought from the patent stage to production in six years is also creditable. For, although the characteristics of rubber as a springing medium are now well known, a completely new technology in respect of hydraulically deflected rubber elements has had to be developed. Furthermore, economy of production had to be borne in mind throughout every stage of the development programme, because this high quality springing system was planned, from its inception, for a low-cost vehicle destined for very high rates of production.

Since a prerequisite for obtaining the maximum space for the occupants is a long wheelbase relative to overall length, some form of interconnection of front and rear springs was considered essential for maintaining an acceptably high standard of ride. The need for adopting interconnection to reduce pitching frequency has, of course, been appreciated for many years and has been acknowledged in practice by the adoption of this principle—but with mechanical interconnection—in the Citroën 2CV and AMI 6 models. An article giving the theory of interconnected suspension was published in the January 1957 issue of *Automobile Engineer*.

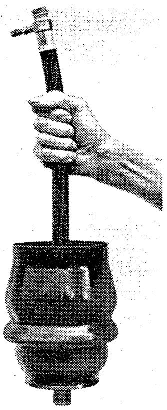
Orthodox springing layouts have the limitation that the front and rear suspension frequencies have to differ to minimize pitching, which would otherwise occur owing to coupling of the pitching and bouncing modes of vibration. As a result, the rear-seat ride is frequently inferior to that at the front. On the ADO 16, however, by virtue of the interconnection of the front and rear springs, pitching is minimized and the motion is harmonic and jerk-free. In practice, the permissible lowness of the frequency is limited by the need for avoiding excessive variations in the static attitude of the vehicle when the fore and aft distribution of its load is changed. The small Citroën car, for instance, has a low pitch frequency—of the order of 50 c/min—and since its attitude varies widely with load distribution, provision is made for adjusting the angle of the headlamp beams relative to the road.

Readers may be interested to know that a prototype car with an interconnected springing system was working as long ago as 1935. This was designed by Issigonis in collaboration with J. N. Morris, now Chief Engineer of S.U. Carburettor Co. Ltd. The car had double-wishbone suspension and torsion-bar springing at front and rear; on each side of the car, the pair of torsion bars extended forward or rearward to a differential gear mounted on the chassis midway between the wheels; the end of each bar was splined into one of the bevel gears.





*Above: In addition to the anti-roll bar between the trailing links of the rear suspension, there are small - diameter torsion bars to limit the static pitch attitude of the vehicle. Right: The front Hydrolastic unit is identical with that at the rear, but operates at a different leverage*



*The compactness of the unit, which weighs 9 lb, is demonstrated here. In service, the flexible hose allows for the deflections of the spring and for slight movements of the rubber-mounted sub-frame relative to the hull. Each unit is held firmly in place in its sub-frame by the pre-load on the spring. The damper is well protected*

### Description of the Hydrolastic system

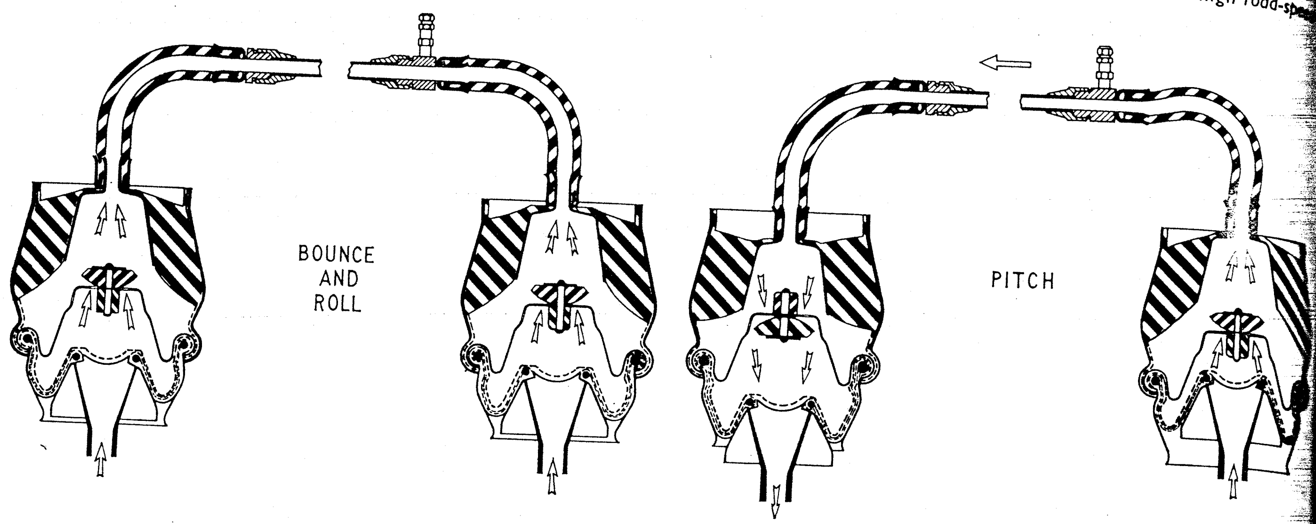
The spring element for each wheel comprises tapered inner and outer sleeves between which is bonded an annular, natural rubber spring of a highly resilient mix. Thus, when deflected axially, the spring is subjected to both compression and shear forces, and, like the cone springs used on the ADO 15 car, has a progressive-rate characteristic. Immediately beneath each spring, and forming part of a common assembly, is a hydraulic piston, diaphragm and cylinder unit, termed the *displacer unit*.

Both sleeves are steel pressings. The inner one is of cupped form, and the outer one is part of a fabricated canister serving as the hydraulic cylinder. The upper end of this canister is held, by the spring pre-load, firmly against the suspension sub-frame. Hydraulic fluid is contained between the lower face of the rubber spring and a diaphragm that closes the bottom end of the canister. Interposed between this diaphragm and the spring is the damper assembly, comprising a conical cup-shape pressing, termed the *port-plate,* on the centre of which are mounted the rubber flap type damper-valves. There is also a permanent bleed hole in this port-plate.

The outer edge of the diaphragm is, of course, clamped around the lower end of the canister, while a conical piston assembly, connected to the suspension arm, seats on the centre of the lower face. As the suspension moves up and down, the motion of the piston deflects the diaphragm, which rolls within a steel skirt pressing of approximately conical form, the upper end of which is also secured to the lower end of the canister. As can be seen in the accompanying illustrations, the taper of the skirt pressing is in the opposite sense to that of the piston.

Nylon-reinforced moulded rubber is employed for the diaphragm. Further reinforcement is afforded by two concentric beads of steel wire, to which the nylon cords are anchored. One of these beads reinforces the periphery and the other seats around the crown of the piston. A thin

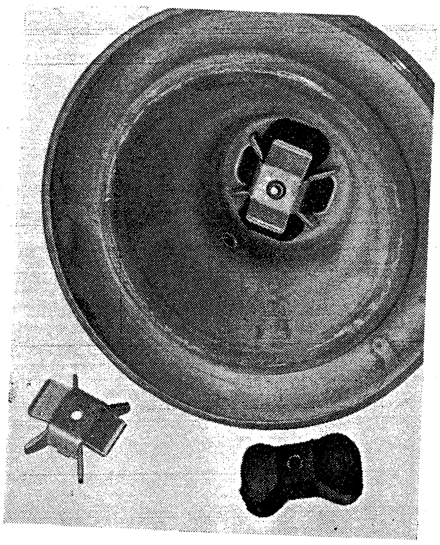
*There is one system on each side of the car. When both wheels are disturbed to the same degree, as in pure bounce—rarely experienced—or in roll, no interflow of fluid occurs. Consequently, the progressive-rate springs are deflected together and the ride is firm. If only one wheel is disturbed, fluid passes through the conduit to move the other wheel in the opposite sense. This reduces pitching and also, because the springs are only partially deflected, produces a softer ride. The throttling effect of the conduit is beneficial to the quality of the ride at high road-speeds*




liner of butyl rubber is fitted over the upper face of the diaphragm, to ensure impermeability.

There are two separate seals. One is between the periphery of the diaphragm and that of the port-plate, while the other is between the port-plate and the lower end of the canister. During assembly, the first is effected by clamping the peripheries of the butyl lining and the diaphragm between the flanged upper end of the skirt and the rolled-over periphery of the port-plate. Then the second seal is made by pressing the rubber lined lower end of the canister around the rolled-over portion of the port-plate. This bonded lining is an extension of the spring material within the canister. In this way, a remarkably sturdy unit, weighing approximately 9 lb, is formed.

The four damper ports punched in the port-plate are oval. They are identical and are equally spaced on a common pitch circle: one diametrically opposed pair works during bounce and the other during rebound strokes. Each of these two functions is controlled by a separate rubber flap component which, as viewed in plan, is of waisted form. There is one on the upper face of the pressing: this controls bounce. The other—disposed at 90 deg to



*The underside of a port-plate, showing the bleed-hole and the rebound valve in position. Above each end of the rubber component is a tongue that is pressed down to determine the blow-off pressure*

the first-mentioned one—is on the underside and controls rebound. Each flap component has a bonded-in ferrule, and a single rivet is passed through it and the centre of the port-plate to secure two small steel pressings, one straddling each flap. The ends of each of these two saddle pieces extend into the punched holes of the other damper valve, to afford angular location. Extending laterally from the central portion of each retainer saddle are two tongues, which during assembly are bent down, by means of special equipment, on to the projecting ends of the flap components, to set the blow-off pressure of the valve. It is claimed that rubber flap-valve dampers have longer working lives and are more reliable than orthodox metal ones, and that the mechanized setting process affords a much closer control over damping characteristics than has hitherto been possible. In addition to these pressure-controlled valves there is, as was already mentioned, a single fixed orifice that functions as a bleed control; this hole is situated on the tapered face of the port-plate.

On each side of the vehicle, Bundy tubing, $\frac{1}{2}$ in diameter, interconnects the two Hydrolastic units. It is accommodated in the floor-tunnel along which the exhaust pipe and brake line are taken. To allow for movements of both the spring and the suspension sub-frame, the ends of the conduit are connected to the spring units by a hose approximately 1 ft long. One end of each hose is fitted over a serrated extension-piece butt-welded to the inner pressing of the spring and, at the other end, the connection is a standard threaded union.

There is a Schrader charging valve in the pipeline: it is close to the front Hydrolastic unit and accessible through the engine compartment. Through this valve, the system is charged with hydraulic fluid—a mixture of 49 per cent demineralized water, 49 per cent alcohol, 1 per cent triethanolamine phosphate and 1 per cent sodium mercaptobenzthiazole. This fluid is, of course, an anti-freeze solution of constant viscosity, containing a rust inhibitor and an agent that is added to make the fluid distasteful — a legal requirement. Before the system is charged with fluid at a pressure of about 205 lb/in$^2$—the standardized pressure for an unladen vehicle—an 80 per cent vacuum condition is induced by an exhauster pump. During service, the pressure may rise to 450 lb/in$^2$ at full bump, and may drop to as low as 70 lb/in$^2$ on conditions of full rebound.

### Operation of the system

As was previously stated, the function of interconnection between front and rear springs is to reduce the pitching frequency of a vehicle equipped with road springs of a bounce frequency affording good handling characteristics. For the Morris 1100 car, the frequencies are 90-92 c/min bounce and 65 c/min pitch. A reduction in pitch frequency can be achieved by shortening the wheelbase, of course, but this, as already mentioned, is undesirable.

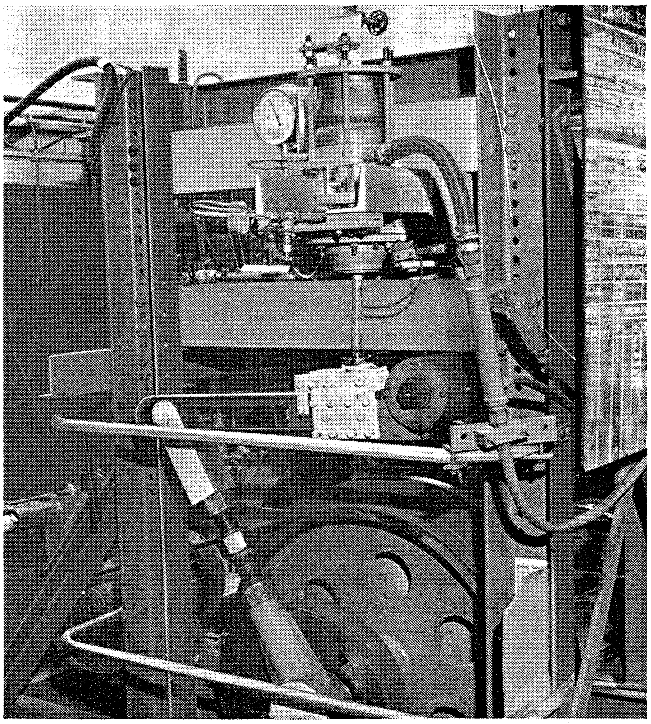
Clearly, the most frequent disturbances to a car travelling in a forward direction are those due to road irregularities that are contacted first by the front wheel and then by the rear wheel on the same side of the vehicle. In a conventionally sprung car, each of these movements causes angular acceleration in the pitch mode. When a front displacer unit of the Hydrolastic system is deflected by a movement of the road wheel, fluid is impelled through the conduit to the rear displacer, which raises the rear end of the car and thus reduces the angular acceleration. As a result, pitch is virtually eliminated and the car maintains a substantially level attitude. Because of the interconnection of the front and rear springs, the dynamic loads on any one wheel are shared by both, the actual proportions distributed between the two being determined by the throttling action of the conduit. Full wheel movement, therefore, is not normally accompanied by full spring movement, and a relatively soft ride is produced. *At higher road speeds the resistance of the pipe to fluid flow increases, as the cube of the fluid velocity, and this steadies the car in the pitch mode. The effective diameter of the conduit is critical. On the Morris 1100, the conduits are of larger bore than is necessary, but each incorporates an orifice plate which determines the ride characteristics.

If both displacer units are deflected simultaneously—as in pure bounce or roll—no interflow occurs, so the pressure of the fluid is conveyed entirely to the springs, which deflect simultaneously, giving an increased suspension rate. Since a condition of pure bounce is rarely experienced, the full effect of two springs acting together is normally felt only during roll, when the outer wheels deflect in concert. The figure quoted for the roll stiffness of the ADO 16 is 6,000 lb-in/deg, which is remarkably high for a saloon car.

The siting of the flap-valves adjacent to the diaphragms ensures that all wheel movements are damped. Small displacements of fluid are throttled only by the constant-diameter bleed hole. During more arduous conditions of operation, the main dampers take over; these are progressive in action, and an adequate level of damping is afforded for all types of terrain. Because of the volume of fluid available for dissipating heat—3·84 pints per system, or 7·68 pints per car—damper fade is not encountered.

Mention has been made of the taper configurations of the piston and the skirt against which the diaphragm rolls. This feature, in addition to being of value in respect of supporting a large area of the diaphragm, helps to stabilize the vehicle in the pitch mode and also contributes to the non-linear springing characteristics. As the piston deflects, the effective area of the diaphragm is increased. In pitch, the displaced fluid is conveyed to the other unit—in which the diaphragm's effective area progressively decreases—and the resistance to pitching therefore rises; in bounce, however, the rate of increase of pressure in the fluid falls, so the apparent spring rate rises.

To minimize changes in the static pitch-attitude, a small-diameter torsion bar is incorporated between each rear trailing arm and the chassis frame. This avoids both the unpleasant appearance of a car with an excessive nose-down or tail-down attitude, and the need of adjusting the head-lamp setting. Another anti-roll bar is fitted between the rear trailing arms, to counteract the inherent understeering



*Fatigue-testing of all components was carried out on rigs such as this. In the Hydrolastic unit on test here, the middle portion has been replaced by a Perspex section, so that operation of the valves can be observed. On the far side of the machine is an identical linkage, and interconnected units can be run simultaneously, either in or out of phase*

characteristic of the vehicle, which arises from the fact that 62 per cent of the unladen weight is carried by the front wheels. Since — to rationalize the production of primary equipment and spares — all four units on a vehicle are identical, different leverage ratios have been adopted at front and rear. That at the front is 3·95 : 1, and that at the rear 4·4 : 1; wheel rates are therefore greater at the front than at the rear, and the anti-roll bar reduces the resultant excessive understeer to an acceptable level.

The ends of the anti-roll bar are bent through 90 deg and drilled, so that they can be bolted in the usual manner to the inner faces of the trailing arms. Sandwiched between each drilled portion and the trailing arm is a forward-extending lever with a square hole near its end, to take the squared end of a pitch-control bar, which of course is installed laterally. The inner ends of the pitch bars are bent forward through 90 deg, and register in steel bushes carried in the cross-member of the sub-frame.

Three features therefore control the pitch of the vehicle. They are the taper configuration of the displacer, the geometry of the suspension linkage, and the pitch bars. Bounce is controlled by these three features and the road springs, and roll is controlled by all four, plus the anti-roll bar. An inherently good aspect of the vehicle design is that the pitching moment due to braking is counterbalanced by the torque reaction on the rear trailing arms, with the result that there is very little change of pitch attitude during braking. The manufacturers contend that, despite the absence of provision for control of ride or levelling, the qualities of ride and road-holding, even at their present level, are sufficiently in advance of current standards to justify the decision not to incorporate such features.

### Development history

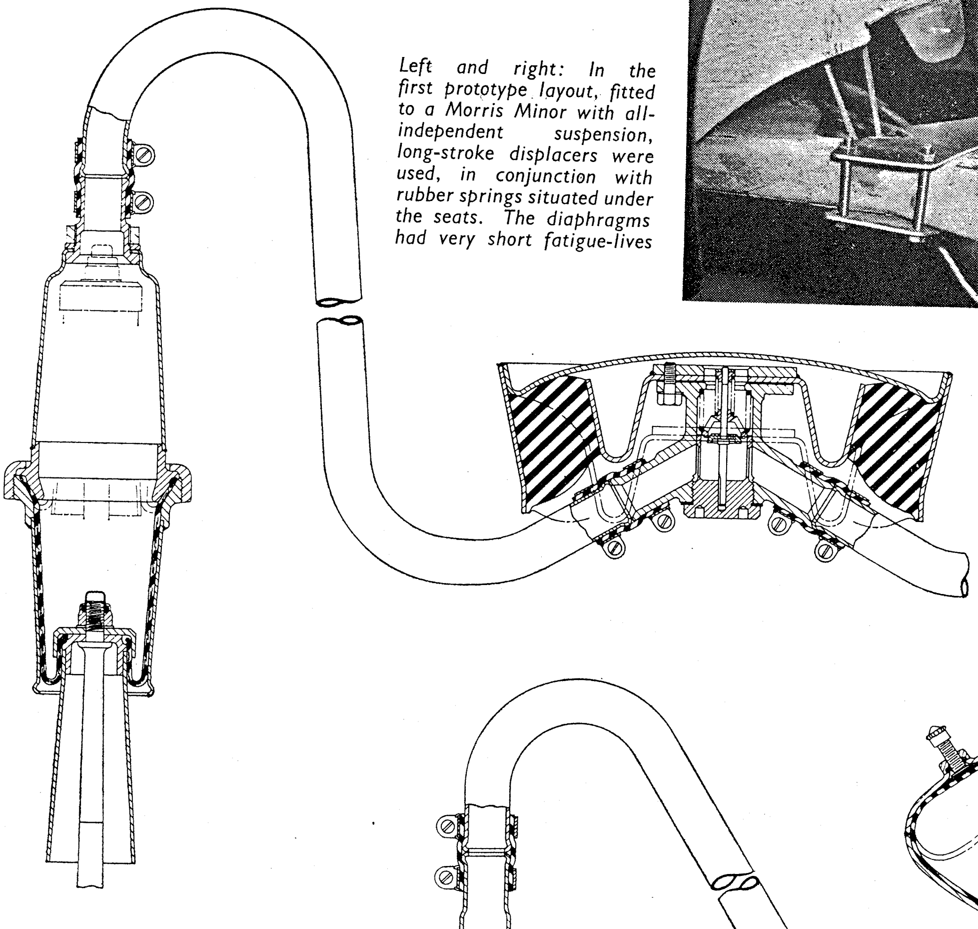
The first of many Moulton patents dealing with hydraulically interconnected rubber springs was taken out in 1955; in the specification, figures for bounce and pitch frequencies

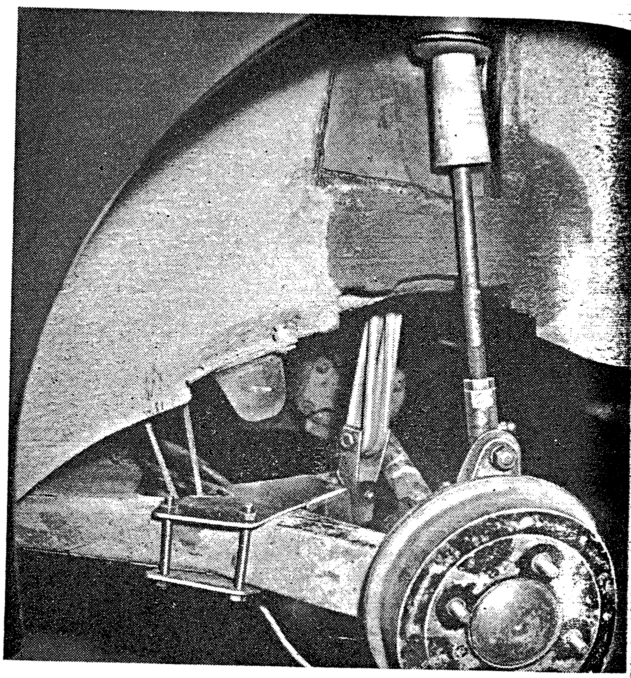
—85 and 60 c/min respectively—were quoted as being achieved on a rig arranged to simulate the inertia of a vehicle with hydraulically interconnected rubber springs. It was for the purpose of developing this and subsequent patents, in exclusive application to B.M.C. cars, that the firm of Moulton Developments Ltd.—an associate firm of B.M.C.—was incorporated.

The development history of the Hydrolastic system is remarkable for the close and harmonious collaboration between the three interested companies—Moulton Developments Ltd, the designers; British Motor Corporation, the users; and Dunlop Rubber Co. Ltd, the manufacturers. Throughout the development period, weekly visits were made by Alex Moulton to both the Dunlop factory, where the meetings were under the chairmanship of Mr. H. Trevaskis, the Technical Director, and also to the Suspension department at the Cowley works of B.M.C. The work of the Suspension department was entirely devoted to damping requirements and to problems connected with the application of the system to vehicles. Over 250,000 miles of road-running were carried out on Morris Minors converted to independent rear suspension. Concurrently, rig-testing took place at the Moulton establishment at Bradford-on-Avon.

In this way, the technical responsibility was vested in Moulton Developments Ltd, but Alec Issigonis — having overall technical control and approval — was kept informed of all changes and decisions. To keep the evaluation function separate from the manufacturing function, Moulton Developments Ltd. were responsible for the testing and evaluation of all components. During the whole of the development period, the rig shop at Bradford-on-Avon ran at a very high factor of utilization: the figure for one particular year—including nights, week-ends and holidays—was 82 per cent.

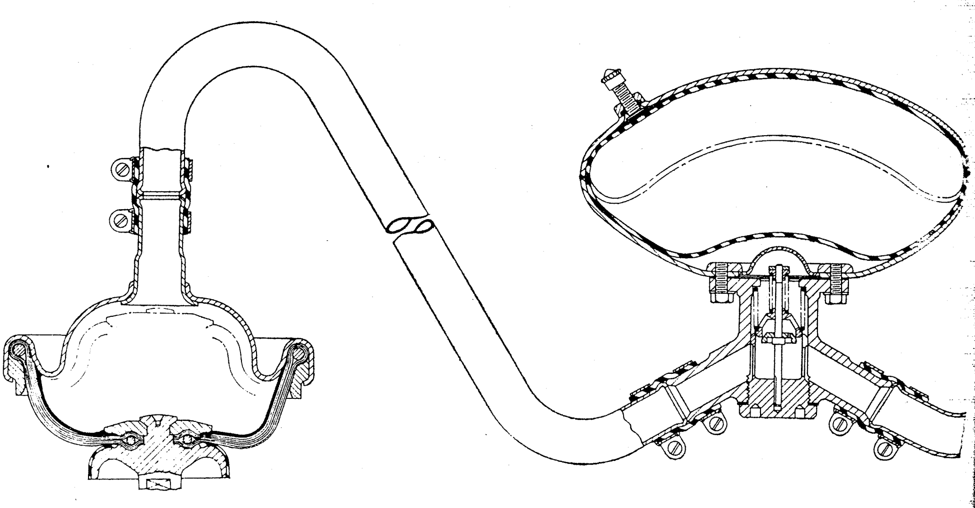
From the outset, there were three basic aims. First, the system was to be completely sealed, so that there should be no maintenance problems of the type associated with glands or pumps. Secondly, for simplicity, rubber was to be used as a springing medium in bounce. Thirdly, the damping





*Left and right: In the first prototype layout, fitted to a Morris Minor with all-independent suspension, long-stroke displacers were used, in conjunction with rubber springs situated under the seats. The diaphragms had very short fatigue-lives*

*Below: To improve the working conditions of the diaphragm, a lever-operated short-stroke displacer was introduced; the diaphragm incorporated two beading rings. The springing medium here was nitrogen contained in a butyl rubber sac*



*Above and right: In both of these systems, bounce-damping was afforded by the mechanical valves housed in the spring assemblies, and pitch-damping by the viscous flow of the hydraulic fluid in the conduits* mechanism was to be integral with the hydraulic interconnection. Most of the stages in the following account of the development programme are depicted in accompanying illustrations.

*Stage one, July* 1956. One of the first experiments to take place after the incorporation of Moulton Developments Ltd. involved a layout in which a 9 in diameter annular rubber spring — in which the rubber was stressed in combined compression and shear—was coupled hydraulically to a pair of $2\frac{1}{2}$ in diameter displacer units of the rolling diaphragm type; there was one displacer at each wheel, of course, and the springs for the two systems were installed beneath the front seats in the car. The stroke of the pistons was virtually the same as wheel movement, and static hydraulic pressure was 100 lb/in$^2$. Rig tests showed that the diaphragms had a fatigue life of only 0·25 million cycles; the target figure was 1 million cycles. Failures of these components—which were of only single ply construction, to accommodate stretching during rolling—were caused by inadequate burst strength, pinching due to inadequate radial clearance, and abrasion due to the high-velocity rolling action; hence, displacers of long stroke and small diameter were abandoned. In this layout, a single damper valve, serving two wheels, was contained within each spring casing, and was of the orthodox spring-loaded metal type.

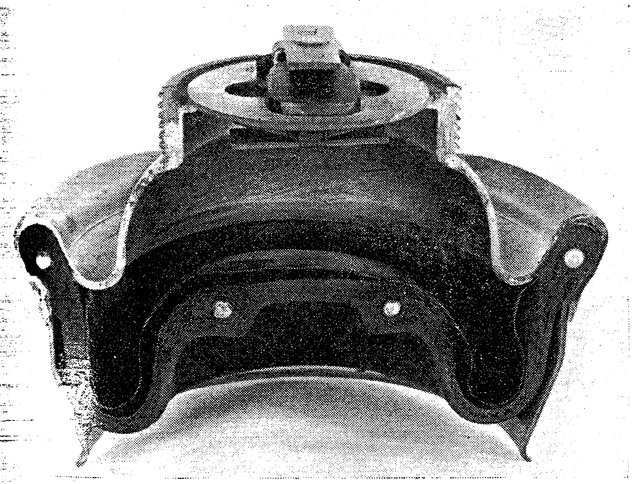
*Stage two, December* 1956. Operation of the next displacer, of $7\frac{1}{2}$ in diameter was based on flexing rather than rolling. The diaphragm was reinforced by six plies of nylon tyre cord, and withstood 2 million cycles on the rig. Lever operation of the displacer was adopted—and retained thereafter—to shorten the stroke of the unit. Despite its acceptable fatigue life, this diaphragm was found to be too cumbersome and stiff for production purposes. It incorporated two beads, however, the inner attached to the piston by a riveted-on collar, the outer clamped in an annular fold of the casing by a pressing operation. The sealing of the inner bead was not satisfactory, and subsequent diaphragms were therefore made continuous across the centre, where in each case support was afforded by the head of the piston.

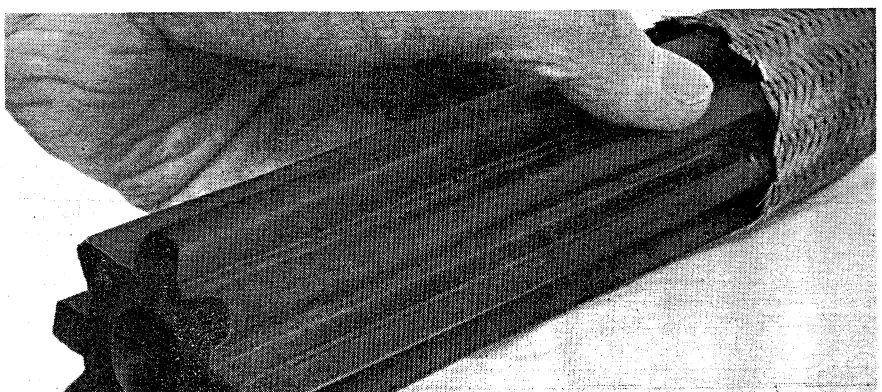
A development of this displacer was tried, in which the diaphragm was 6 in diameter and reinforced by four plies of flat nylon cord. This was subject to puncturing and ply separation, however, and had a fatigue life of only 0·6 million cycles. A modified reinforcement was therefore developed, in which three plies were wrapped into a cylinder, with their cords at a small angle to the axis. One end of the cylinder was expanded by a balloon, until the cords were almost radial, and the diaphragm was moulded in this condition. This diaphragm had a life of 2 million cycles.

*Stage three, March* 1957. With stronger diaphragms, a change was made to higher working pressures, and consequently to displacers of smaller diameter. It is interesting to reflect that the pursuit of higher working pressure was a central theme of the Moulton development work, with the result that, today, production diaphragms are subjected to unusually high pressures with complete safety. To give burst protection to the diaphragm, a short, tapered, skirt was incorporated in the displacer housing, extending downwards from the annular groove in which the bead was clamped; the piston was given a conical form.

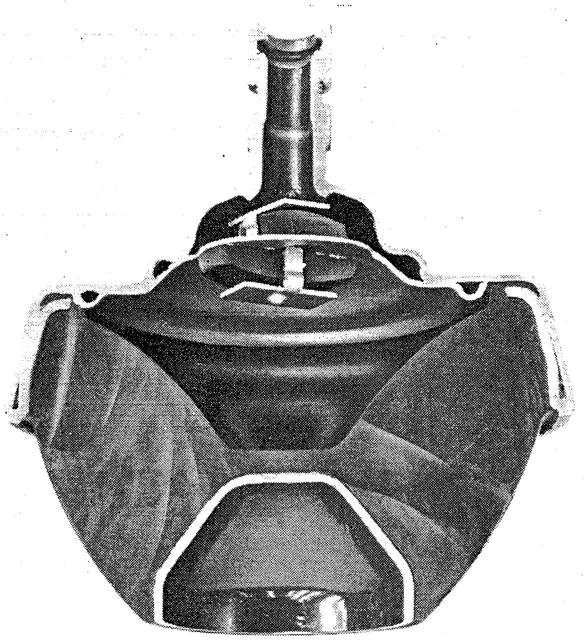
Concurrently, in an effort to make reductions in the space needed for the spring, a change was made to a central nitrogen spring contained in an 8 in diameter steel casing of oblate shape. Tests showed that this spring, although able to withstand 5 million working cycles, was unduly sensitive to changes of temperature; additionally, there was the feeling that a pressure vessel—particularly one situated close to the occupants of the vehicle—was a potential hazard in a collision, and the idea was abandoned.

*Stage four, September* 1957. Hitherto, damping in the pitch mode had been afforded by the viscous flow of the fluid in its interconnecting pipes, and damping in bounce and roll by mechanical valves in the spring unit. These valves were, of course, not progressive in action, and their opening and closing produced perceptible shocks in the test cars. Accordingly, rubber flap-valve dampers were developed, and sited within the displacer units. These had the virtues of being silent in operation, more reliable than orthodox valves, able to cope with the inherently high displacements of fluid—because of their large port areas—progressive in action, simple to make, able to operate

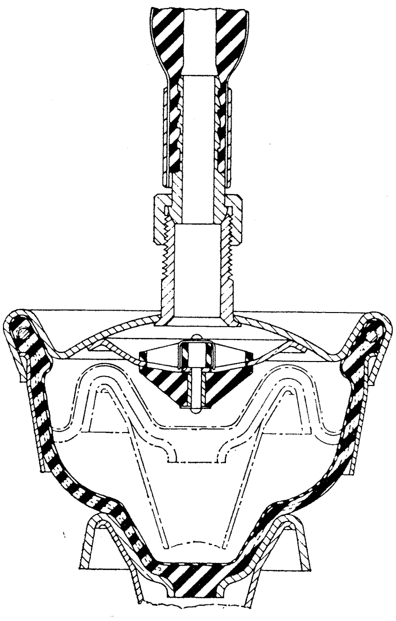


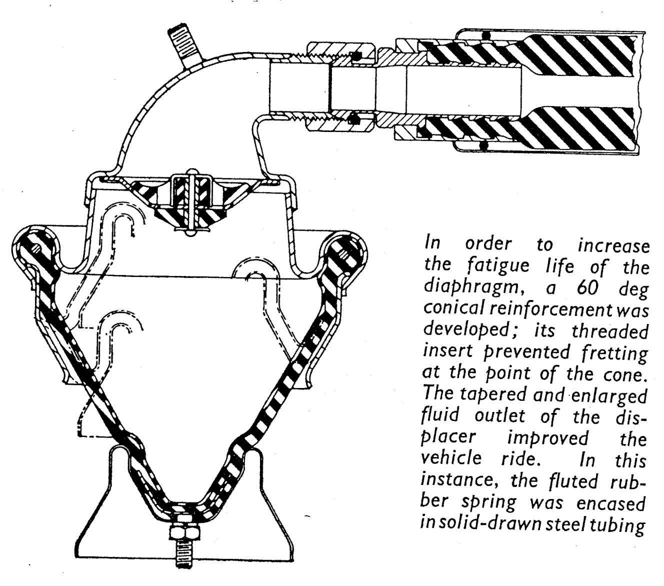


*Above: Ride was greatly improved when rubber flap-valve dampers—suitable for the inherently large displacements of fluid—were adopted, and re-sited at the displacer units. For burst protection for the diaphragm, the tapered skirt was incorporated; the piston was also tapered. Here, the butyl rubber lining for the diaphragm can be clearly seen. Right: This rubber tube served as both a spring and an interconnecting conduit. Increases of fluid pressure in the tube distended the flutes against the braided steel outer casing*



Above: A rubber spring, of the type used on the ADO 15, was adapted for use as a displacer, but was found to have inadequate hydraulic stiffness. Rubber damper-valves were used. Right: For compactness, the piston of this diaphragm-type displacer was dished to allow room for the damper-valves, and the port-plate was welded to the roof of the housing. The spring used was the fluted rubber one





In order to increase the fatigue life of the diaphragm, a 60 deg conical reinforcement was developed; its threaded insert prevented fretting at the point of the cone. The tapered and enlarged fluid outlet of the displacer improved the vehicle ride. In this instance, the fluted rubber spring was encased in solid-drawn steel tubing

without lubrication, and of low cost. Wheel-hop, which had occurred before the re-siting of the dampers, was obviated as a result of this modification.

To reduce space required for installation, the interconnection pipes were then used as tubular rubber springs: each pipe consisted of an outwardly fluted rubber tube surrounded by braided steel tubing $1\frac{7}{8}$ in diameter. As fluid pressure increased with wheel deflection, the rubber tubing distended and its flutes were elastically compressed against the casing. This spring, however, was subject to abrasion and kicking, which caused the braiding to break up.

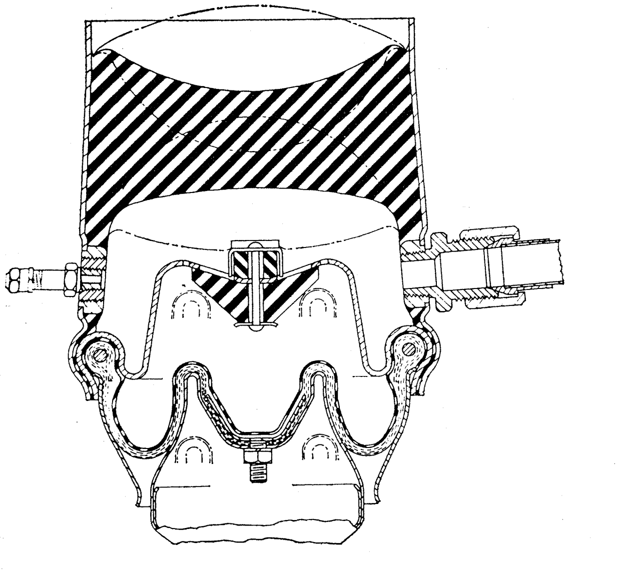
In the interests of perfect sealing, a diaphragm impermeable to the hydraulic fluid was essential. However, the production of an impermeable diaphragm was found to be unfeasible, so a separate liner of butyl rubber was incorporated in the displacer. It was moulded to the same shape as, but was not attached to, the diaphragm, and was similarly trapped at its periphery. Fluid pressure held it against the diaphragm. Owing to potential production difficulties with the type of diaphragm in use, a new construction was tried, in which the reinforcement consisted of two semi-circular flat pieces of tyre cord ply wrapped into 60 deg cones, and placed one inside the other with the joints on opposite sides. There was no central bead, so the head of the displacer piston was dished in the centre, to locate radially the apex of the cone. The displacer unit could now be made more compact, since the dishing made room for the damper valves, which were mounted on a dished pressing; the piston was now a simple pressing. In fact, at the end of 1957, all of the main components in the displacer were pressings. Hitherto, normal experimental practices had been employed, with consequent reliance on threaded, brazed and welded constructions.

*Stage five, February* 1958. A passing attempt was made to incorporate a simple Hydrolastic system in the smaller ADO 15 car, which was entering its final design stages. In this layout, a cone-type spring was used as a displacer, and a smaller version of the fluted spring was employed to augment the springing. Unfortunately, the cone spring had insufficient hydraulic stiffness, in that it became distended under hydraulic pressure; the rate of increase of fluid pressure bore no relation to the deflection of the displacer, and unreliable damping conditions were created. Consequently, the car went into production with mechanically operated rubber springs and dampers of the orthodox hydraulic type.

*Stage six, August* 1958. After experiments had been carried out with a development of the fluted spring—now encased by a solid-drawn steel tube—this line of investigation was abandoned. The extrusion of rubber tubing to the required dimensional accuracy was not feasible, and moulding was a forbidding problem. Additionally, accommodation of two long pipes of $1\frac{7}{8}$ in diameter presented difficulties on a small car.

Meanwhile, the type of hydraulic fluid was being decided on. Among others, paraffin and brake fluid were examined, but the first was ruled out for its incompatibility with rubber, and the second because of its unsuitable viscosity characteristics and cost. During this period, a degree of harshness in the ride — experienced mainly over rough roads — was cured by enlarging and tapering the outlet of the displacer unit, to eliminate choking of the fluid discharge when wheel velocities were high. At about this time, a screwed insert was incorporated in the rubber moulding, to stop the point of the cone-type diaphragm fretting in the dished piston.

*Stage seven, December* 1958. Experiments were carried out on the damper flaps, to assess their stiffness and the ability of the rubber to resist being forced through the punched holes. When these were completed, the finalized



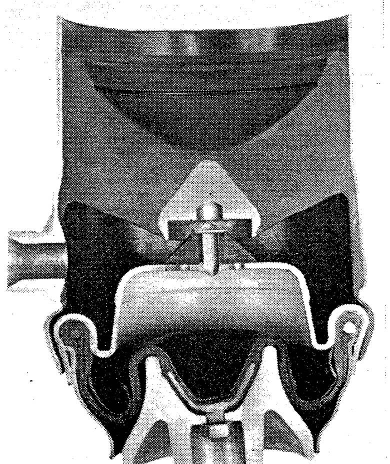
*In this version, recognizable as a forerunner of the production unit, the spring was incorporated in the displacer housing; a bending weakness in the central portion of the spring was later revealed by fatigue testing. The curved profiles of the piston and skirt were developed to increase the life of the diaphragm. At one side of the canister was the fluid outlet, and at the other the charging valve for the system*

dampers were calibrated against orthodox units at Cowley, and fatigue tests began at Bradford. Initially, the damping practice was conventional, in that more damping was applied in rebound than in bump, but the production settings are now similar in both directions.

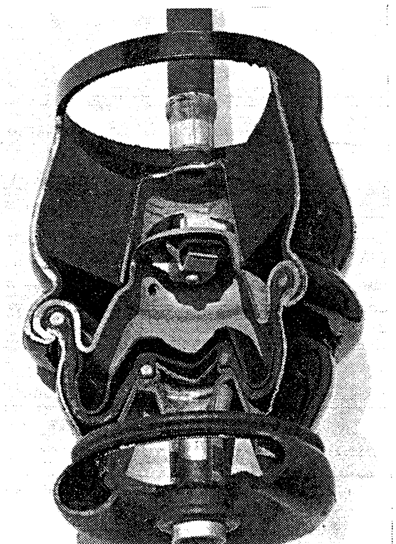
A rubber spring, of inverted dome shape, was incorporated in the upper part of the displacer unit, now tapered slightly, and the conduit tubing was changed to steel throughout, leading from a union in the side of the canister. By virtue of these changes, a slight harshness over small irregularities on the road was eliminated, for there was unrestricted flow of fluid to the spring. To help increase the life of the diaphragm, a study was instituted to develop the optimum profiles for the piston and skirt. The purely conical shapes gave rise to an unacceptably small radius of bending at the bottom of the stroke, and a large radius—with attendant circumferential strain—at the top. Eventually, profiles were developed to reduce the severity in these extreme conditions, and also to provide a smooth transition phase between the two.

*Final stage, June* 1959 *onwards.* When it was found that the inverted dome-spring was weak in bending, a metal insert—first a solid cone, later a cup-shape pressing—was bonded into the centre of the rubber element; this modification, coupled with an increase in the degree of taper of the canister surrounding the spring, modified the stress distribution and raised the fatigue life of the spring from under 1 million to over 2½ million cycles. The union for the conduit was transferred from the side of the canister to the end of the pressing in the centre of the spring. By virtue of this move, the hoop-strength of the canister was improved, the opportunity was taken of shortening the unit, and a flexible hose — attached to a serrated projection on the inner cone—was interposed between the unit and the end of the steel conduit, which was now re-routed—this last modification considerably eased the problem of installation. The two-ply cone type of diaphragm was meanwhile giving good results on rig-tests, but it was felt that, because of the type of construction of the reinforcement, there would be difficulty, in production, of maintaining consistency in the spacing of the calendered cords during the moulding process. However, the Dunlop Rubber Co, to obtain more uniform spacing of the cords than was afforded in calendered plies, developed a technique of constructing the reinforcement from a single nylon filament, and the resultant diaphragm was found to have the required standards of strength, flexibility and fatigue life—over 2½ million reversals—together with excellent dimensional consistency. In the production of this member, a tube of longitudinally disposed strands is created by threading the continuous filament around the teeth of two coaxial, toothed wheels placed about 5 in apart. When one wheel is rotated relative to the other, a hyperboloid of revolution is formed, and removal of the wheels leaves a hollow casing of waisted shape, with a short parallel skirt at each end.

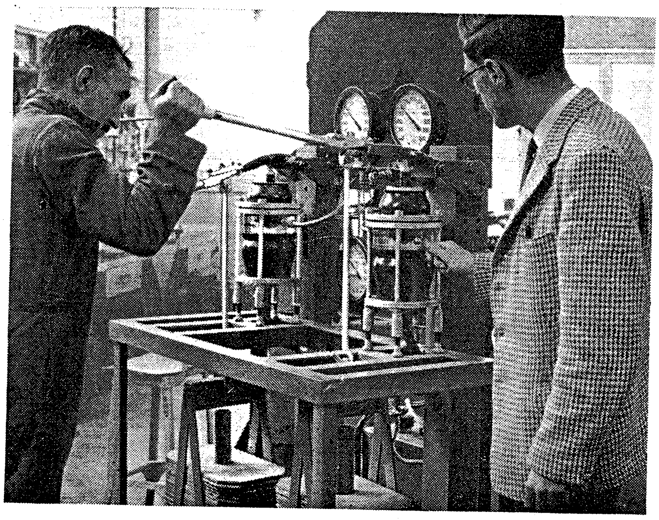
The inner bead ring is slipped over the end of the casing, to rest at the small-diameter portion, and one half of the casing is then folded back axially on the other, to enclose the ring and form a two-ply reinforcement of approximately conical shape, in which the spacing of the cords and the relative angle of the plies are extremely consistent. In the moulding of the diaphragm, the parallel skirt is wrapped round the outer bead ring, and a layer of rubber fills the central orifice. A combination of flexibility and unusually high strength in the diaphragm has been one of the most important factors in arriving at the final configuration of the Hydrolastic unit, for the remarkable and characteristic degree of compactness could be achieved only by using



*The fatigue life of the spring was improved considerably by the inclusion of a metal insert. Although the cone-type diaphragm was still being used, potential production problems caused its abandonment*



*The production unit. A cup-shape pressing replaces the conical insert in the spring, and the fluid outlet is in this pressing; the diaphragm incorporates two beading rings; to obviate metal-fatigue in the port-plate, the diameter of the plateau has been reduced*



*On this manually operated rig—built primarily for demonstration purposes, but found to be of use in development—the effective diameter of the bleed holes can be adjusted, and the operating behaviour of the valves can be watched. Bounce and pitch conditions in a complete Hydrolastic system are simulated by operating the levers in or out of phase, and fluid pressures at four points are registered on the dials*

much higher fluid pressures than were hitherto associated with diaphragm-operated mechanisms of comparable size. The production item withstands a static pressure of 2,000 lb/in$^2$. During the arduous rig- and road-testing that followed the finalization of the design, a fatigue weakness was revealed around the punched holes of the port-plate; the gauge of the metal was therefore increased from 0·080 in to 0·104 in, and the diameter of the plateau portion was reduced.

The later prototypes of the Morris 1100 were subjected to the standard acceptance tests imposed on all B.M.C. vehicles of new design. This rigorous programme includes sustained running over the various rough surfaces at the M.I.R.A. proving ground and the Long Valley course at Chobham. Because of the high standards set during the development of the suspension components, no failures were experienced during these tests.

All of the rigs used in the dynamic tests were constructed in a manner such as to make dynamic creep of the rubber spring measurable, with the consequence that enough information on this subject was amassed to allow an accurate design factor to be introduced into the calculations relevant to the ride height of the vehicle. Static creep of the rubber was also assessed, of course. In addition, the impermeability of the butyl rubber liner for the diaphragm was appraised on a separate static rig. Over a period of $2\frac{1}{2}$ years, it was found that any settling due to permeability could be ignored.

Clearly, in the interests of maintaining approximately equal ride height at all four wheel, the deflected height of Hydrolastic units must be held to close tolerances. During manufacture, the top of the canister is machined, to provide a tolerance of $\pm 0{\cdot}015$ in on the distance to the end of the piston rod. During this operation, the fluid pressure is held at 200 lb/in$^2$, and a load of 2,000 lb is applied to the piston. On the vehicle assembly line, a novel method—protected by patent, as are most of the manufacturing processes used for the Hydrolastic system—is used to eliminate primary creep in the rubber springs. The Hydrolastic systems are pressurized for 30 min to about 400 lb/in$^2$—which brings the suspension against its rebound stops—and then reduced to the standard static pressure, 205 lb/in$^2$. By virtue of this technique, the springs have no opportunity of regaining the free state, as the weight of the car is applied throughout.

**Design features**

Among the noteworthy features not already mentioned are the following:

(1) Of the main metal components, all are pressings except the piston rod—a forging. The characteristic taper of some pressings is, of course, of benefit in the metal-forming process. All metallic components in the system are cadmium plated, primarily to enhance shelf-life.

(2) Precision machining of the type associated with orthodox damping techniques is entirely unnecessary, and no lubrication is required during service.

(3) No gaskets or sealing compound are needed in the assembly of the unit.

(4) During the life of the vehicle, any settling attributable to secondary creep in the rubber can be compensated by restoration of the static hydraulic pressure.

(5) Angular movements of the actuating rod are catered for by the flexibility of the diaphragm.

(6) Because of the sturdy construction of the canister, the damper valves are extremely well protected; accidental dropping that might ruin an orthodox telescopic damper has no effect.

**Service considerations**

Arduous testing on rigs and on the road have shown that no noticeable drop in the hydraulic pressure can be expected during a twelve-month period. Tests carried out during development have shown that instantaneous dumping of the hydraulic fluid—simulating failure—from one of the systems in a car travelling in a straight line calls for no correction in steering; both springs are affected to the same degree, and the car merely sags to one side, with movement restricted as the bump-stops are contacted.

Similar tests on a car travelling round a corner at limiting tyre adhesion caused no alteration of cornering behaviour: in this instance the fluid in the system on the side of the car remote from the centre of turn was dumped. The hazard involved cannot be compared to that incurred by failure of brakes or tyres. Gradual loss of pressure has no noticeable effect on handling or steering; however, it would appear advisable for users to have the pressures checked at least annually. Each B.M.C. main dealer will keep a small manually-operated portable pumping unit for the purpose of charging the system when such a pressure drop occurs, or when work on a car is sufficiently drastic to justify uncoupling a spring unit. The pumping unit consists of an exhauster pump to evacuate the air, and a hydraulic pump to inject the Hydrolastic fluid and pressurize it. During this operation, the transition from evacuation to charging is effected instantaneously by turning a three-way valve. Thermal expansion due to changes of ambient temperature in service does not give rise to any significant changes in ride height. However, vehicles exported to territories where extreme temperature conditions are experienced will have their pressures normalized on arrival at the distributors.

## Larger Engine for Ford Consul Classic

At the end of last month, the Ford Motor Company began installing a re-designed, larger capacity engine in their Consul Classic and Capri models. Originally, the swept volume of the four-cylinder engines in these cars was 1,340 cm$^3$ and this has been increased to 1,499 cm$^3$ by lengthening the stroke from 65·07 mm to 72·7 mm. At the same time, a change has been made from a three bearing to a five bearing design for the new crankshaft which, as before, is an iron casting. A Zenith type 33 VN carburettor is fitted to the new engine: the diameter of its choke is nm greater than that of the instrument on the 1,340 cm³ rsion.

The reason for the change to a five journal shaft was to minate main bearing thump which the previous unit had hibited, and which was found to be accentuated on a small mber of 1,499 cm³, three-bearing engines that were built itially.

In the latest engine, since the loading at each bearing is wer than that on the three bearing version, it has been ssible to change from copper-lead to white metal main aring shells. The dimensions of each bearing have not en reduced and, in fact, the same components are used in the 997 cm³ Anglia engine, of the 105E model, with nsequent production economies.

Other changes to the engine are few. The cylinder block-ankcase casting is, of course, deeper to accommodate the nger stroke and the new connecting rods are of slightly heavier section. The oil capacity of the sump has been increased by 50 per cent to avoid an increase in oil running temperatures and the diameter of the inlet valve heads has been made a little larger. The compression ratio has been reduced from 8·5:1 to 8·3:1.

Maximum net power output of the 1,499 cm³ engine is 59·5 b.h.p. at 4,600 r.p.m, and the maximum net torque, of 81·5 lb-ft, occurs at 2,300 r.p.m. The corresponding figures for the superseded 1,340 cm³ unit are 54 b.h.p. at 4,900 r.p.m. and 74 lb-ft at 2,500 r.p.m.

A new gearbox having baulk-ring synchromesh on all forward gears is now fitted to these models. Previously the three upper ratios only had synchromesh. The overall ratio for first gear has been changed from 16·99:1 to 14·61:1. Sealed bearings and joints, which require no periodic lubrication, are now fitted at the front suspension assemblies, the steering tie rods and the propeller shaft universal joint.

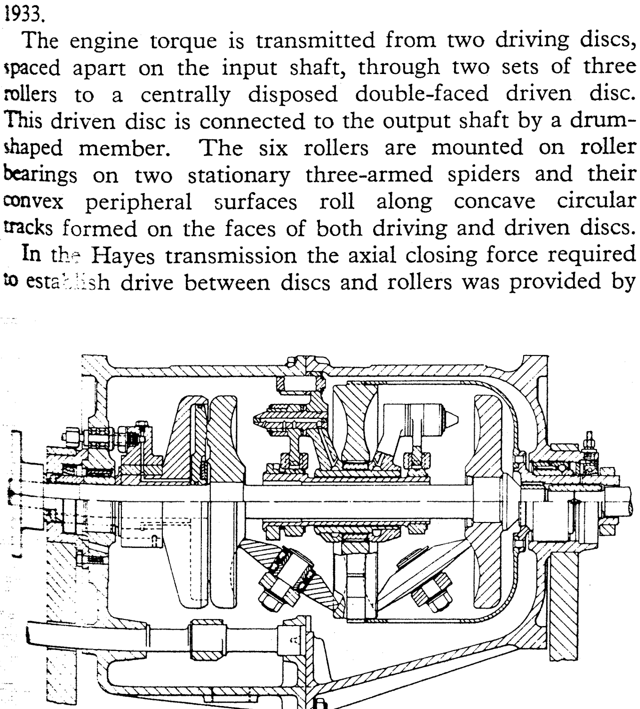
# Perbury Continuously Variable Gear

***Interim Report Issued by the National Research Development Corporation***

An interim report has been released on the Perbury gear, the development of which is being sponsored by the National Research Development Corporation. Basically the work of F. G. de B. Perry, this gear is, in principle, the Hayes continuously variable ratio transmission which was offered by Austin as optional equipment in 1933 on their 16 h.p. and 18 h.p. cars. The Hayes transmission was described in the *Automobile Engineer* of November 21st, 1933.

The engine torque is transmitted from two driving discs, spaced apart on the input shaft, through two sets of three rollers to a centrally disposed double-faced driven disc. This driven disc is connected to the output shaft by a drum-shaped member. The six rollers are mounted on roller bearings on two stationary three-armed spiders and their convex peripheral surfaces roll along concave circular tracks formed on the faces of both driving and driven discs.

In the Hayes transmission the axial closing force required to establish drive between discs and rollers was provided by the reaction from a series of steel balls acting on ramps at a coupling on the input shaft, the axial force being proportional to transmitted torque. For the experimental Perbury gear this force is provided by hydraulic means.



*Axial loading of the discs and rollers is effected by means of a hydraulic piston and cylinder adjacent to the left-hand driving disc. A special fluid has been developed for this transmission. For automotive applications, a centrifugal clutch and a reverse gear are used*

The speed ratio between input and output shaft, which can be varied between a 2·7:1 reduction and a 1·75:1 increase, is, of course, dependent upon the angle of the rollers relative to the driving and driven disc. To change ratio, the rollers are tilted simultaneously, by means of a hydraulically actuated linkage, so that the axes of the rollers do not intersect that of the input shaft, the resultant precession causing the rollers to be steered to a different attitude.

A study of the operation of oil film drives, of which the Perbury gear is an example, has been made by Perbury Engineering Ltd, and a gear installed in a Hillman Minx has covered 25,000 miles. A theoretical investigation into the conditions in the oil film is being carried out in the Department of Mechanical Engineering at Leeds University, and use is being made also of earlier investigations.

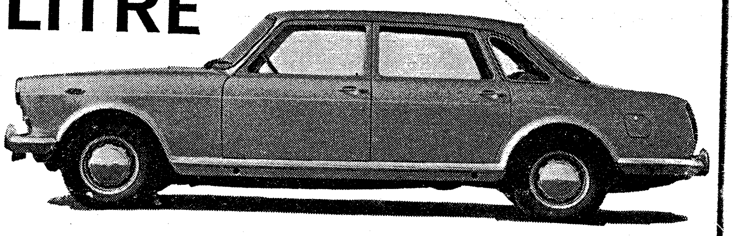
Perbury gears are being tested in pairs on a recirculating power test rig designed and built by Plint and Partners Ltd, of Wargrave. These tests have established that when 100 h.p. is being transmitted at an input speed of 5,000 r.p.m. the overall transmission efficiency is 94 per cent at a reduction ratio of 2:1. Losses at the oil film amount to approximately 3 per cent, the remainder being accounted for by bearings, seals and oil churning losses. Variations in overall efficiency are small throughout the range of ratios which can be obtained.

Endurance testing will begin shortly on a second test rig and it is expected that, as with the Hayes transmission, the limiting factor will be surface fatigue of the rollers. However, it is suggested that the normal roller bearing formulae relating life to stress will apply and that when 100 h.p. is being transmitted stresses are not high by industrial ball bearing standards.

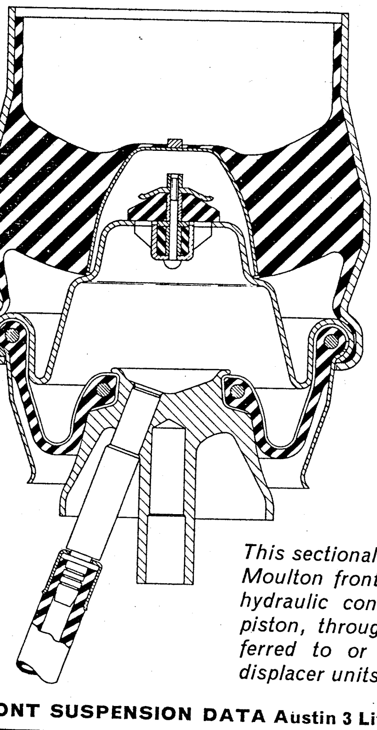
A design study to assess the automotive use of the Perbury gear has been made by Norris Brothers Ltd, Burgess Hill. There are also possibilities in the use of this type of transmission in machine tools, test houses and laboratories.

A DESIGN ANALYSIS OF A NEW LARGE CAR PRODUCED BY THE BRITISH LEYLAND MOTOR CORPORATION

# AUSTIN 3 LITRE



## Part 1: The front and rear suspension assemblies, interconnected Hydrolastic springs, and the self-levelling system



*This sectional illustration of one of the Moulton front spring units shows the hydraulic connector in the displacer piston, through which fluid is transferred to or from one of the rear displacer units*

**FRONT SUSPENSION DATA Austin 3 Litre**

| | |
|---|---|
| **Type** | Double transverse link |
| **Spring type** | Moulton; rubber in shear and compression. Interconnected hydraulically with rear springs |
| **Spring load per wheel** | 840 lbf |
| **Bump travel** | 3·5 in |
| **Rebound travel** | 3·5 in |
| **Wheel rate in bounce, static** | 205 lbf/in |
| **Periodicity, at kerb weight** | 82 c/min |
| **Wheel camber angle:** | |
| *Full bump* | −3 deg |
| *Static* | +1 deg |
| *Full rebound* | +3 deg |
| **Roll centre height** | 5·75 in |
| **Total roll stiffness** | 5 900 lbf in/deg |
| **Swivel axis inclination** | 7·5 deg |
| **Wheel offset at wheel axis** | 4·07 in |
| **Wheel offset at ground** | 2·35 in |
| **Castor angle** | 3 deg, power steering<br>1 deg, manual steering |
| **Toe-out** | 0·125 in |
| **Track** | 56 in |
| **Tyre size** | 185-14 in (D70SR14) |

SINCE THE INTRODUCTION of the Austin 3 Litre model last autumn, cars have been manufactured for appraisal, and production has now begun. A general description was published in the November 1967 issue of *Automobile Engineer*. This is the largest B.L.M.C. car, and the first with rear-wheel drive, to which interconnected Hydrolastic springing has been applied. Because the suspension system is of particular interest, it is described first in this series of articles on the design of the car.

Whereas the two Moulton Hydrolastic rubber spring and damper units for the suspension assemblies on the same side of the car are interconnected by means of fluid in a pipe, as on all the earlier B.M.C. front-drive cars, there is an important difference in the arrangement for the Austin 3 Litre. Transference of fluid is effected by connecting together chambers formed between the displacer pistons—actuated by the suspensions—and the pressings that carry the damper valves. Hence, fluid does not pass through the damper valve before it is displaced along the pipe to the other spring unit, as it does in other Hydrolastic systems.

One effect of this arrangement is to obviate suspension harshness at low speeds with radial ply tyres, which are fitted as standard. Another is to reduce the pitch frequency of the car, and this is approximately 25 per cent lower than that of the B.L.M.C. 1800 models. Damping of pitch movement is effected only by fluid friction in the interconnecting pipes, which are $\frac{3}{8}$ in bore as compared with $\frac{1}{2}$ in bore on the 1800 models.

A sectional illustration of one of the front spring units and its fluid displacer is shown. The external diameter of the canister is 6$\frac{5}{8}$ in; between the full rebound and full bump positions of the suspension, the travel of the displacer piston is 1·9 in, during which the rate of the spring increases progressively; the static mean rate is 205 lbf/in. Damping for the suspension during bounce is effected by the flow of fluid through ports—controlled by rubber flap-valves—in the bell shape steel pressing between the displacer and the spring.

To avoid encroachment of space in the boot, two duplex spring units, one for each rear suspension, are installed horizontally side-by-side beneath the pressing for the rear seat, on the right-hand side of the car. The canister for each is only 6$\frac{3}{8}$ in diameter, which has been made possible by the use of two opposed rubber springs. Vertical suspension-loads are transmitted hydraulically, through a $\frac{9}{16}$ in bore steel pipe and a flexible hose of the same size, to each spring by a displacer unit, actuated by one of the trailing arms of the rear suspension. It enters a coaxial duct in the rear spring and fills a chamber between this spring and the front one. Hence it applies equal thrusts in opposite directions to the springs.

In each displacer unit for the rear springs, a piston and rolling diaphragm, similar to that for the front springs, applies thrust to fluid in an inner chamber; this chamber is connected by means of the previously-mentioned $\frac{3}{8}$ in bore pipe to the equivalent chamber in the displacer unit for one of the front springs. Fluid to deflect a rear spring passes through ports controlled by rubber flap-valves for bump and rebound damping into an outer chamber, from which it flows to the spring unit. Between full bump and rebound positions of the suspension, the travel of the displacer piston is 1·6 in, and the static mean rate at the wheel of each duplex spring is 112 lbf/in.

As in other Hydrolastic systems, the fluid is a mixture of water and 33 per cent ethylene glycol, with a corrosion inhibitor, and the system is pressurized at 247 lbf/in², to raise the unladen car to the designed static height. The springs and the displacer units are manufactured by the Dunlop Co. Ltd, Coventry.

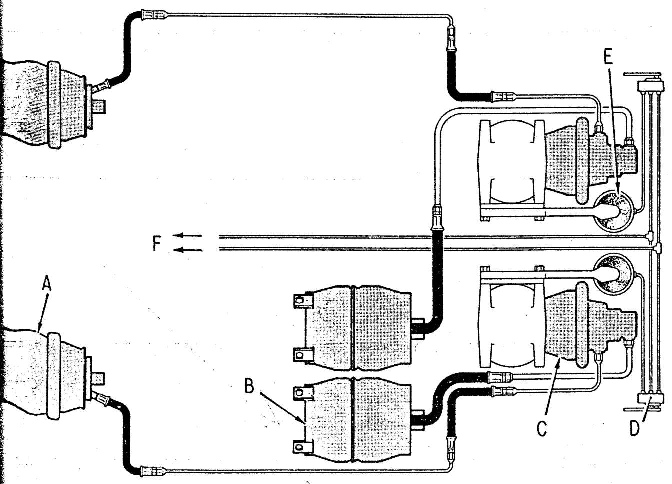
## Front suspension

As can be seen from accompanying illustrations, each front suspension assembly incorporates two transverse links, and vertical loads are transmitted by the lower link to the Hydrolastic spring. Ball-joints accommodate both suspension and steering movements. The effective lengths of the upper and lower links are $8\frac{11}{16}$ in and 12·95 in respectively. In the static laden position of the suspension, the axis of the inner pivot of the upper link is $18\frac{1}{4}$ in above the ground, and that of the lower link is $8\frac{3}{4}$ in. Between the centres of the two ball-joints the dimension is $11\frac{3}{8}$ in, and the centre of the lower ball-joint is $3\frac{1}{2}$ in below the axis of the wheel. Other relevant dimensions are given in an accompanying table.

For each suspension assembly, a BS 1490 : LM 22-W aluminium alloy casting carries the pivots for the inner ends of the links, and forms a mounting for the upper end of the spring unit. Bump and rebound limit-stops act upon

*A diagram of the Hydrolastic system, and the self-levelling arrangement for the rear suspension, as viewed from beneath the car*

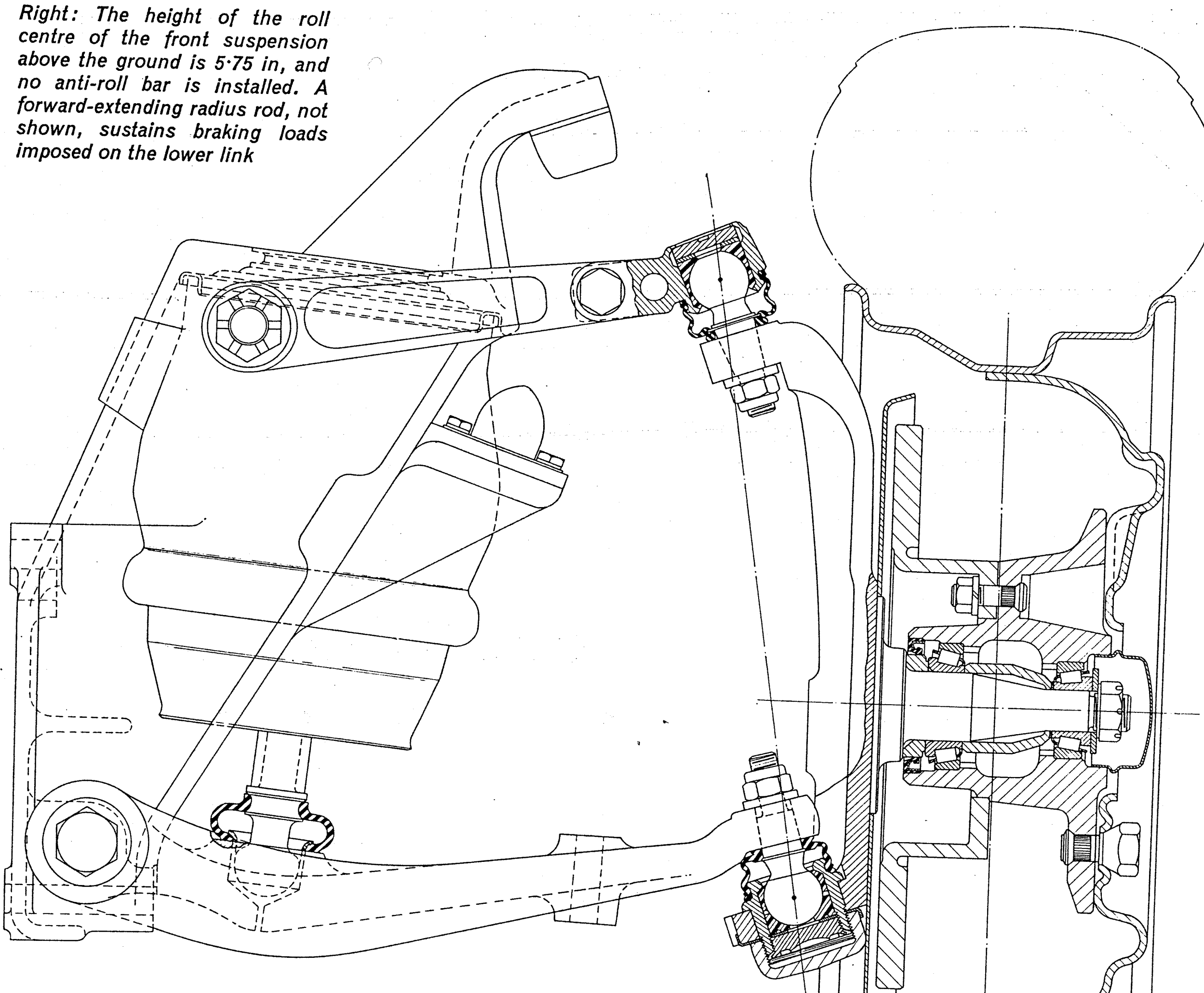
**A** front spring and displacer unit; **B** rear spring; **C** rear displacer unit; **D** levelling control unit; **E** levelling ram; **F** connections to hydraulic pump and reservoir

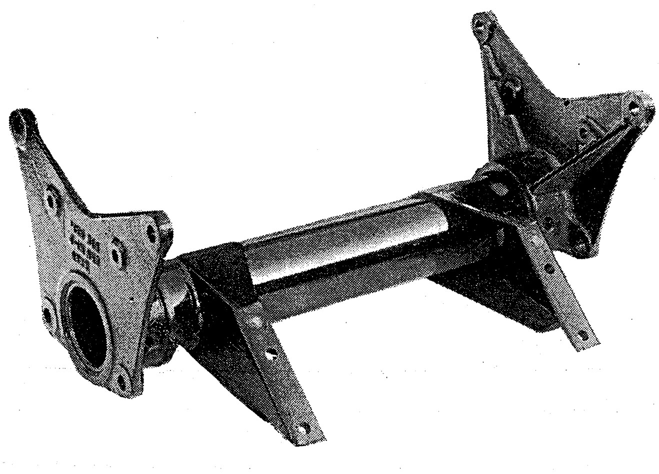


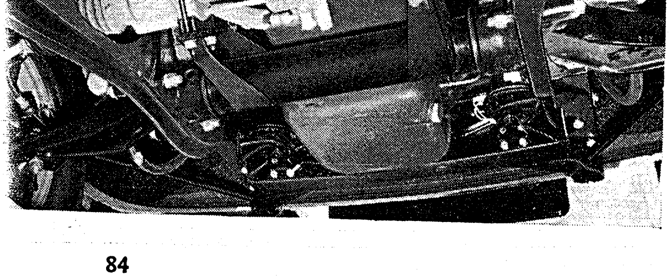

**SPECIFICATION DATA Austin 3 Litre**

| | |
|---|---|
| **Engine** | |
| *Number of cylinders* | 6 in line, longitudinal |
| *Firing order* | 1, 4, 2, 6, 3, 5 |
| *Bore* | 83·31 mm |
| *Stroke* | 88·90 mm |
| *Swept volume* | 2 914 cm³ |
| *Compression ratio* | 9·0:1 |
| *Maximum net b.h.p.* | 114 at 4 500 rev/min |
| *Maximum net b.m.e.p.* | 133 lbf/in² at 2 500 rev/min |
| *Maximum net torque* | 157 lbf ft at 2 500 rev/min |
| *Crankshaft* | Seven bearing, forged steel with integral balance weights |
| *Valves* | Vertical, in-line in head, pushrod operated; bath-tub shape combustion chambers |
| *Carburettors* | Two S.U. type HS.6 |
| *Fuel pump* | S.U. mechanical |
| *Oil filter* | Tecalemit full-flow |
| *Dry weight, with clutch* | 562 lb |
| **Transmission** | |
| *Clutch* | Borg and Beck 9 in diameter single dry plate, with diaphragm spring; hydraulic actuation |
| *Manual gearbox* | Four forward speed and reverse, synchromesh on all forward speeds, central control |
| *Gear ratios* | fourth 1:1<br>third 1·38:1<br>second 2·17:1<br>first 3·44:1<br>reverse 3·10:1 |
| *Overdrive* | Birfield type LH, ratio 0·82:1; operates on third and fourth gears |
| *Automatic transmission* | Borg-Warner type 35, with three-element torque converter and three-speed epicyclic gearbox |
| *Torque converter ratio* | 1·93:1 maximum |
| *Gear ratios* | third 1:1; second 1·45:1; first 2·39:1; reverse 2·09:1 |
| *Final drive* | Hypoid bevel gears, ratio 3·91:1 (Automatic transmission 3·55:1) |
| **Suspension** | |
| *Front* | Double transverse links |
| *Rear* | Independent, single semi-trailing arm; Armstrong hydraulic self-levelling |
| *Springs* | Moulton Hydrolastic with integral dampers; interconnected hydraulically right-hand front to right-hand rear, and left-hand front to left-hand rear |
| **Brakes** | |
| *Make* | Girling hydraulic, with direct acting vacuum servo |
| *Front* | Three-piston calipers and 10·4 in diameter discs; total swept area 224 in² |
| *Rear* | 9·0 × 2·25 in leading-and-trailing shoe type; total swept area 127·3 in² |
| *Front : rear braking ratio* | 69·6 per cent front; 30·4 per cent rear |
| *Handbrake* | Mechanical actuation of rear brakes |
| **Steering** | |
| *Manual* | Cam Gears rack-and-pinion, overall ratio 21:1; steering wheel diameter 16·5 in |
| *Turns lock-to-lock* | 4·29 |
| *Power assisted* | Hydrosteer rack-and-pinion, overall ratio 19:1; steering wheel diameter 16·5 in |
| *Turns lock-to-lock* | 3·93 |
| *Turning circle, between kerbs* | 40 ft |
| **Wheels and tyres** | |
| *Wheel type* | Pressed steel disc, five-stud attachment, 5 in wide rim |
| *Tyre size* | 185—14 in Dunlop radial-ply tubeless |
| *Pressures* | Front 28 lbf/in²; rear 25 lbf/in² |
| **Dimensions** | |
| *Wheelbase* | 9 ft 7·5 in |
| *Track, front* | 4 ft 8·0 in |
| *rear* | 4 ft 8·0 in |
| *Overall length* | 15 ft 5·75 in |
| *Overall width* | 5 ft 6·75 in |
| *Overall height (unladen)* | 4 ft 8·75 in |
| *Ground clearance (laden)* | 6·5 in |
| *Kerb weight* | 3 330 lb |
| *Weight distribution* | Front 56 per cent; rear 44 per cent |
| *Body shell weight, white* | 784 lb |
| *Frontal area* | 20·8 ft² |

*Right: The height of the roll centre of the front suspension above the ground is 5·75 in, and no anti-roll bar is installed. A forward-extending radius rod, not shown, sustains braking loads imposed on the lower link*

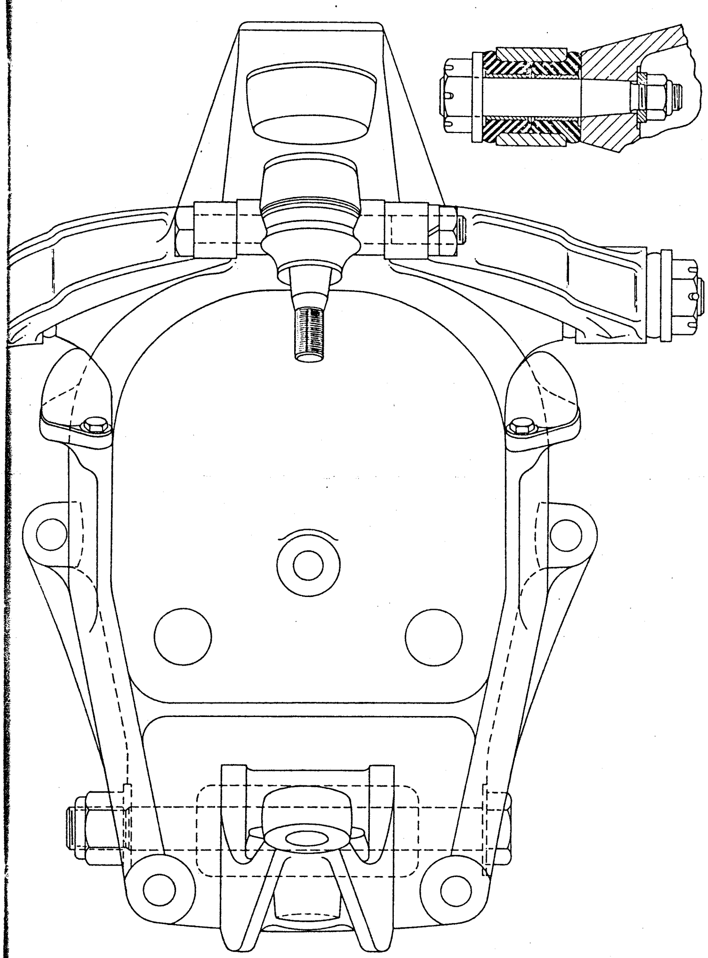




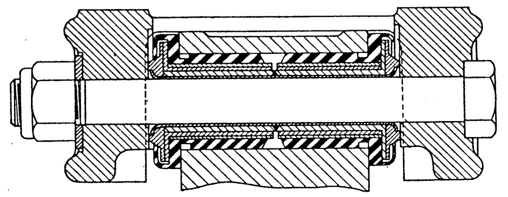


the upper link and are bolted to brackets integral with this casting. To reinforce the body structure for the attachment of the two castings, a transverse member braces together the two deep longitudinal members. Malleable iron brackets are assembled and welded to the ends of a $3\frac{1}{2}$ in external diameter, 0·212 in thick mild steel tube. Five $\frac{1}{2}$ in diameter bolts and nuts secure each of the aluminium alloy castings

*A tubular transverse member, top left, strengthens the body structure for mounting the front suspension assemblies, as shown in the lower illustration. Brackets welded to the rear of the tube carry the rack-and-pinion steering gear*



*An aluminium alloy casting carries the spring unit and the pivots for the inner ends of the links of the front suspension. Details of the Metalastik Slipflex bush for the lower pivot, and one of the rubber-bushed upper pivots are also shown*



to the outer face of one of the longitudinal members, and a bracket to the inner face.

A $\frac{3}{4}$ in diameter En.8R steel bolt, retained by a self-locking nut in two drilled lugs in the casting, forms the inner pivot for the lower link, which is an En.16T steel forging. A Metalastik Slipflex bush★, which does not increase the rate of the suspension, is assembled to this pivot. The bush is in two portions, and each is inserted into one end of a $1\frac{9}{16}$ in diameter hole in the link. Each portion comprises a flanged rubber bush bonded internally to a steel sleeve, the rubber preventing the transmission of road noise. Inside it is a steel bush, the internal surface of which is coated with Glacier Metal DX material, so that it can rotate freely about a chromium plated steel tube carried on the spindle. This bush is lubricated with Castrol 3 grease only during assembly. The diameter of the bearing surface is $\frac{13}{16}$ in, and its total length is $3\frac{1}{2}$ in.

★ *Circle reply card* **3**

Loads caused by braking, acting on the lower link, are sustained by a rod that extends forward to a pivot, incorporating two large rubber buffers, at the front end of the body structure. These buffers also afford longitudinal compliance of the suspension; at the central plane of the wheel, the rate of this compliance is 1 500 lbf/in. The rear end of the $\frac{5}{8}$ in diameter, En. 14A steel rod is welded to a forged BS.1449:3B, CS4 steel yoke, which is attached by means of a $\frac{1}{2}$ in diameter bolt and self-locking nut to the suspension arm, $9\frac{3}{16}$ in outboard of the inner pivot. The effective length of this rod is approximately 20 in.

A $1\frac{5}{16}$ in diameter blind hole is drilled in a boss on the upper surface of the lower link—$3\frac{9}{16}$ in from the axis of the inner pivot—in which a nylon socket for a ball-joint is pressed. The $1\frac{3}{16}$ in diameter spherical surface of the En.8D-R ball-pin is induction hardened; the neck is $\frac{11}{16}$ in diameter, and the shank is assembled in the bore of the stem of the displacer piston for the spring unit. Two shoulders machined on the ball-pin retain one end of a neoprene dust cover, and a lip on the socket retains the other end. This ball-joint is greased during assembly and does not require periodic lubrication.

Each upper link comprises two I-section En.16T steel forgings, between the outer ends of which the housing for the upper ball-joint is attached by means of two $\frac{1}{2}$ in diameter bolts and nuts. A separate $\frac{11}{16}$ in diameter En.16T steel inner pivot for each arm is provided. At one end of each pivot a tapered shank is machined; this is carried in a similarly tapered hole in the aluminium alloy casting, and is retained by a $\frac{7}{16}$ in diameter self-locking nut and steel washer. Two Metalastik flanged bonded rubber bushes are inserted, one into each end of a $1\frac{1}{4}$ in diameter hole in the arm. When they have been assembled to the pivot, and secured by an $\frac{11}{16}$ in diameter castellated nut—locked by a split-pin—and steel washer, their overall length is $1\frac{3}{4}$ in. Between the centres of the two rubber bushes for the upper link, the dimension is $10\frac{5}{8}$ in.

For the upper and lower ball-joints, the En.16T steel ball-pins are similar. The $1\frac{3}{32}$ in diameter spherical surface is induction hardened, and the diameter of the neck is $\frac{5}{8}$ in. Each has a shank with a 1 in 8 total taper, and is secured by a $\frac{1}{2}$ in diameter self-locking nut and steel washer. Moulded nylon, impregnated with molybdenum disulphide, is used for the sockets in both ball-joints; the joints are filled with Dexta grease during assembly, and require no subsequent lubrication. The two-piece socket for the upper ball-joint is retained by a screwed plug in the forged En.8 steel housing. The plug is locked by spinning over a lip on the housing. A neoprene dust cover is secured by garter springs to both the housing and the ball-pin.

Separate top and bottom sockets are assembled in the En.8 steel housing for the lower ball-joint. The top socket is a press fit in a $1\frac{7}{32}$ in diameter hole machined in the housing, whereas the bottom socket is a close clearance fit, and is retained axially by a screwed steel plug. The plug is screwed in sufficiently to provide zero axial clearance at the ball-pin, and a lip on the plug is peened into slots machined in the housing. The housing for the ball-joint is pressed into a $1\frac{5}{8}$ in diameter hole machined in the lower arm of the vertical link, and is retained by an En.3A steel cap-nut, which is locked by a tab-washer.

Each vertical link and integral stub-shaft is an En.16T steel forging. Two lugs on the front face are drilled and tapped for $\frac{3}{8}$ in diameter bolts that retain one of the Girling disc brake calipers; the rearward-extending En.16T steel steering arm is attached by two $\frac{1}{2}$ in diameter bolts and nuts to drilled bosses on the inner face of the forging.

Timken opposed taper roller bearings carry the BS.310: B20/10 malleable iron hub, and the dimension between their inner faces is $1\frac{5}{8}$ in. The nominal dimensions of these bearings are: inner $2\frac{5}{16} \times 1\frac{1}{4} \times \frac{5}{8}$ in; outer $1\frac{25}{32} \times \frac{3}{4} \times \frac{5}{8}$ in. Five $\frac{1}{2}$ in diameter En.18T steel studs for the wheel are pressed into holes in the flange, on a 5 in diameter pitch-circle. The $10\frac{5}{16}$ in diameter cast iron brake disc is spigoted to the hub, and is driven by five $\frac{3}{8}$ in diameter BS.1768, Code T, studs pressed into holes in bosses cast in the hub; their pitch-circle diameter is $4\frac{1}{8}$ in. Including the wheel and tyre, the unsprung weight of each front suspension assembly is 110 lb.

## Independent rear suspension

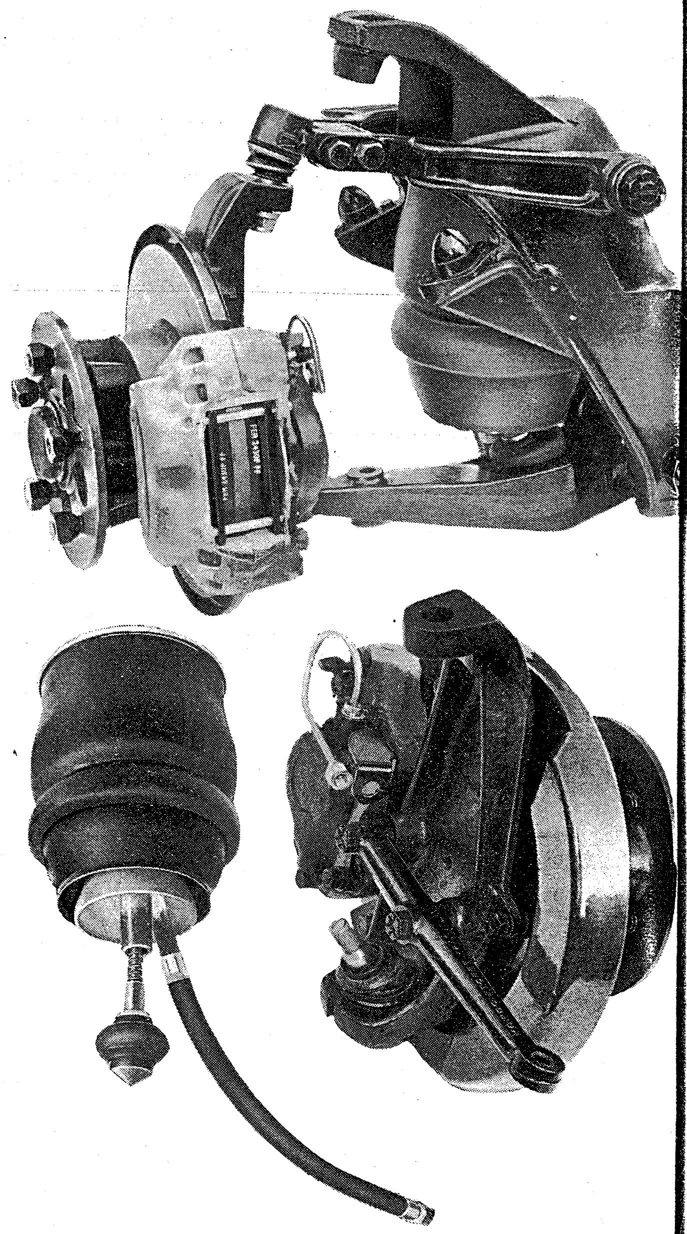
The Austin 3 Litre is the first B.M.C. rear-drive car with independent suspension for the rear wheels. A system incorporating trailing arms—similar to that on front-drive models—has been chosen. In respect of the last-mentioned models, because the front wheels are driven, and carry a relatively large proportion of the total weight, they have inherently strong understeer characteristics and rear suspension arms with full trail are satisfactory. On the Austin 3 Litre, however, the rear wheels are driven, and they carry a larger share of the total weight. For optimum tyre adhesion, particularly on wet surfaces, the axis of the pivot for each arm is at 75 deg to the longitudinal axis of the car. Also, in the static unladen condition, the wheels have a 1 deg negative camber. It is claimed that the contact of the tyre with the road remains fully effective up to 4 deg roll angle of the body. Furthermore, to reduce slip angles of the rear tyres and promote understeer during cornering, the toe-in in the static unladen condition is $\frac{1}{8}$ in.

The rates for the rear spring units on the Austin 3 Litre are lower than those for the other models with Hydrolastic systems. For example, as compared with the 1800 models they are: 3 litre 91 lbf/in; 1800 112 lbf/in. In the bounce mode of the suspension of the 3 Litre car, the periodicity is 82 c/min, and in the pitch mode it is 48 c/min; the corresponding figures for 1800 models are 94 c/min, and 64 c/min, respectively. A means of levelling the suspension, to maintain a constant static height with changing load, is therefore essential on the 3 Litre cars. Since the change in weight of the vehicle mainly affects the loads acting on the rear wheels, the hydraulic self-levelling system—described later in this article—is installed in the rear suspension only.

## Details of design

Each suspension arm is a BS.310:B20/10 malleable iron casting, the nominal wall-thickness of which is 0·26-0·28 in; when it is fully machined it weighs 32 lb. About the axis of the pivot, the effective length of the arm is 18 in. Between the centres of the two bearings that carry the pivot for the arm, the dimension is $18\frac{19}{32}$ in; the housing for each bearing is a BS.1490:LM22 aluminium alloy diecasting, which is attached by four $\frac{3}{8}$ in diameter bolts and nuts to the body structure.

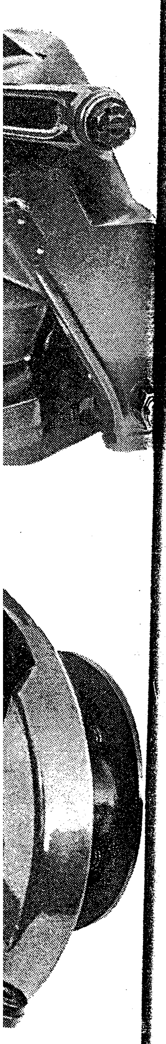
The outer bearing is a Metalastik cylindrical bonded rubber bush, and the inner bearing a Metalastik Slipflex bush. Natural rubber of 60-65 Durometer hardness is used for each outer bush, and the nominal radial thickness and length of the rubber element are $\frac{1}{2}$ in and $1\frac{13}{16}$ in respectively. As shown in a sectional illustration of this bush, slots in the rubber at the front and rear of the pivot afford greater resilience in the horizontal than in the vertical plane, to provide longitudinal compliance of the suspension. The graphs show that, for vertical deflections, the rate of the bush is 13 000 lbf/in, whereas for horizontal deflections it



*Top: A right-hand front suspension assembly; the unsprung weight, including the wheel and the tyre, is 110 lb. Left: In the push-rod for each front spring unit, a helical spring prevents disengagement of the ball-pin from the socket of the ball-joint when the hydraulic system is evacuated in preparation for charging it with fluid. Right: Each vertical link and stub shaft is an En.16T steel forging*

**REAR SUSPENSION DATA Austin 3 Litre**

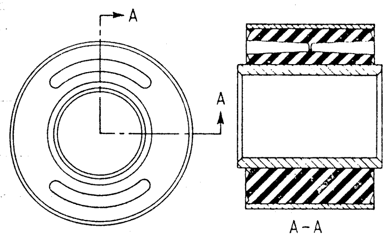
| | |
|---|---|
| **Type** | Single semi-trailing arm |
| **Spring type** | Moulton, rubber in shear and compression; interconnected hydraulically with front springs |
| **Spring load per wheel** | 610 lbf |
| **Bump travel** | 3·5 in |
| **Rebound travel** | 3·5 in |
| **Wheel rate in bounce, static** | 112 lbf/in |
| **Periodicity, at kerb weight** | 82 c/min |
| **Wheel camber angle:** | |
| *Full bump, laden* | −3·25 deg |
| *Static, unladen* | −1 deg |
| *Full rebound* | +1·5 deg |
| **Roll centre height** | 7·2 in |
| **Total roll-stiffness** | 2 500 lbf in/deg |
| **Toe-in** | $\frac{1}{8}$ in |
| **Track** | 56 in |
| **Tyre size** | 185-14 in (D70SR14) |

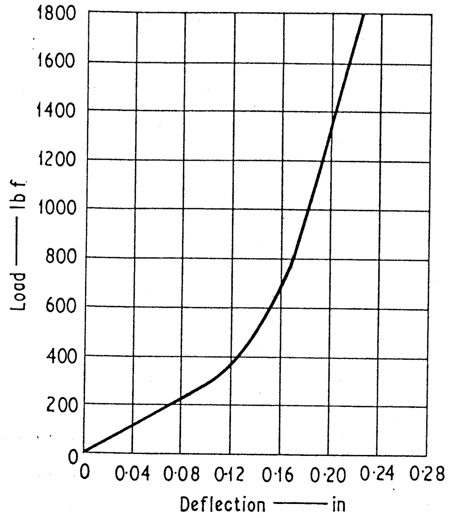


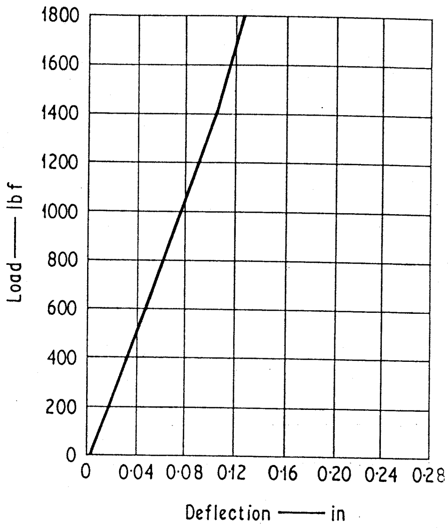
*sprung weight,*
*push-rod for*
*ngagement of*
*the hydraulic*
*t with fluid.*
*steel forging*

arm
shear and
connected
ont springs

*Compliance of the rear suspension is afforded by cavities moulded in the bonded rubber bush at the outer end of each pivot for the trailing arms. The right-hand graph shows the load-deflection curve for the bush in the vertical plane, and the left-hand graph that for the bush in the horizontal plane*




is 3 000 lbf/in, rising after $\frac{1}{8}$ in deflection, when the slots are closed, to 15 200 lbf/in.

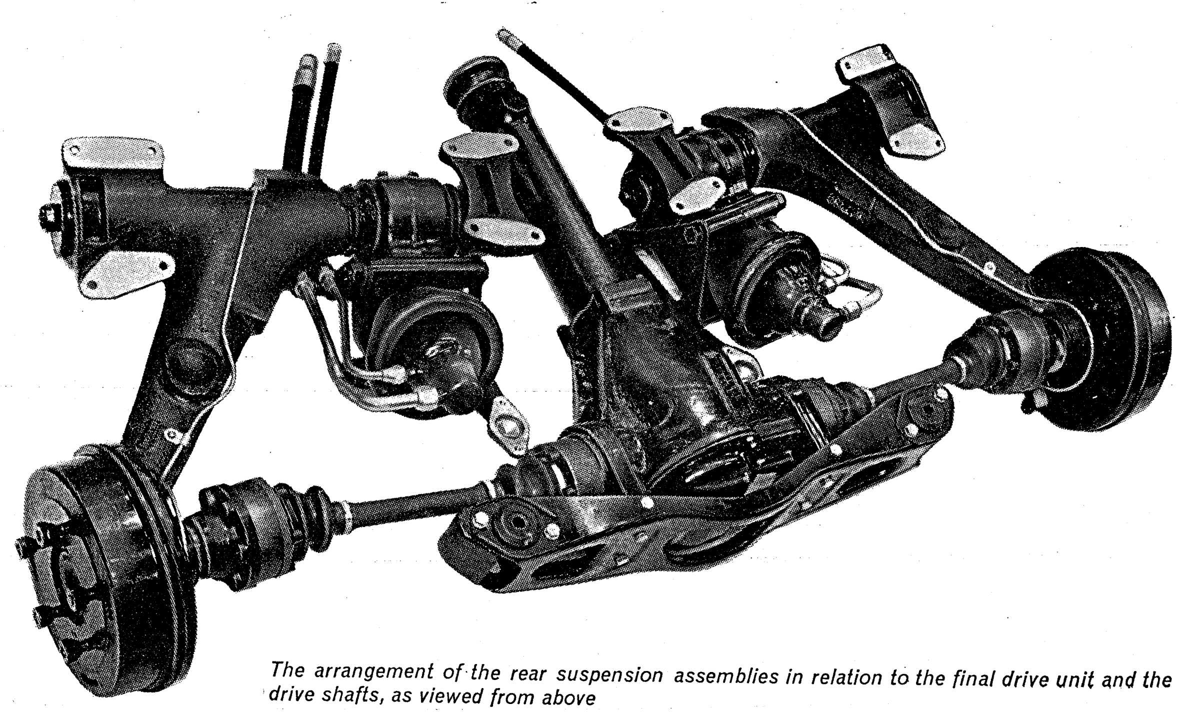
Axial as well as radial loads are sustained by the inner bearing. Nominally, the Glacier Metal DX bush in this bearing is $1\frac{1}{2}$ in diameter by $1\frac{7}{8}$ in long. Insulation against road noise is provided by the external flanged rubber element, which is $\frac{3}{16}$ in thick. Suspension loads are transmitted from the arm by $1\frac{9}{16}$ in diameter splines to an En.17V steel shaft on which the arm is located axially by a $\frac{3}{8}$ in diameter Roll-pin. The inner portion of the splines on the shaft also transmit the loads to the internally splined boss of a forged En.16T steel lever that actuates the displacer unit for the Hydrolastic spring.

A nylon socket for a ball-joint—similar to that in each lower link of the front suspension—is pressed into a hole drilled near the outer end of the lever, and suspension loads are transmitted by this ball-joint and a push-rod to the displacer piston. The effective length of the lever is 4 in; hence the leverage ratio of the linkage between the wheel and the displacer piston for the Hydrolastic spring is 4·5:1.

Each displacer unit is attached to a pair of castings pivoted coaxially with the shaft for the suspension arm. A lip pressed in the front end of the housing for the displacer unit is clamped between the two castings in an annular groove machined in the rear end of each. The two BS.1490 : LM22-W aluminium alloy castings are secured together by four lateral bolts and nuts. Three of these bolts are $\frac{5}{8}$ in diameter, and are assembled in reamed holes to provide an accurate register of one casting with the other; the fourth, in front of the pivot, is $\frac{7}{16}$ in diameter.

The pivot for the castings comprises two $\frac{3}{32}$ in thick, wrapped steel bushes, each of which can rotate on a $2\frac{1}{4}$ in diameter journal machined on one end of the internally splined boss of the lever. Each bush is $1\frac{1}{16}$ in long, and the internal surface is coated with Glacier Metal DX material★. Axial location is effected by two similarly coated steel thrust washers. A polyurethane dust cover at each end prevents the escape of grease inserted during initial assembly.

★ *Circle reply card* **4**



*The arrangement of the rear suspension assemblies in relation to the final drive unit and the drive shafts, as viewed from above*

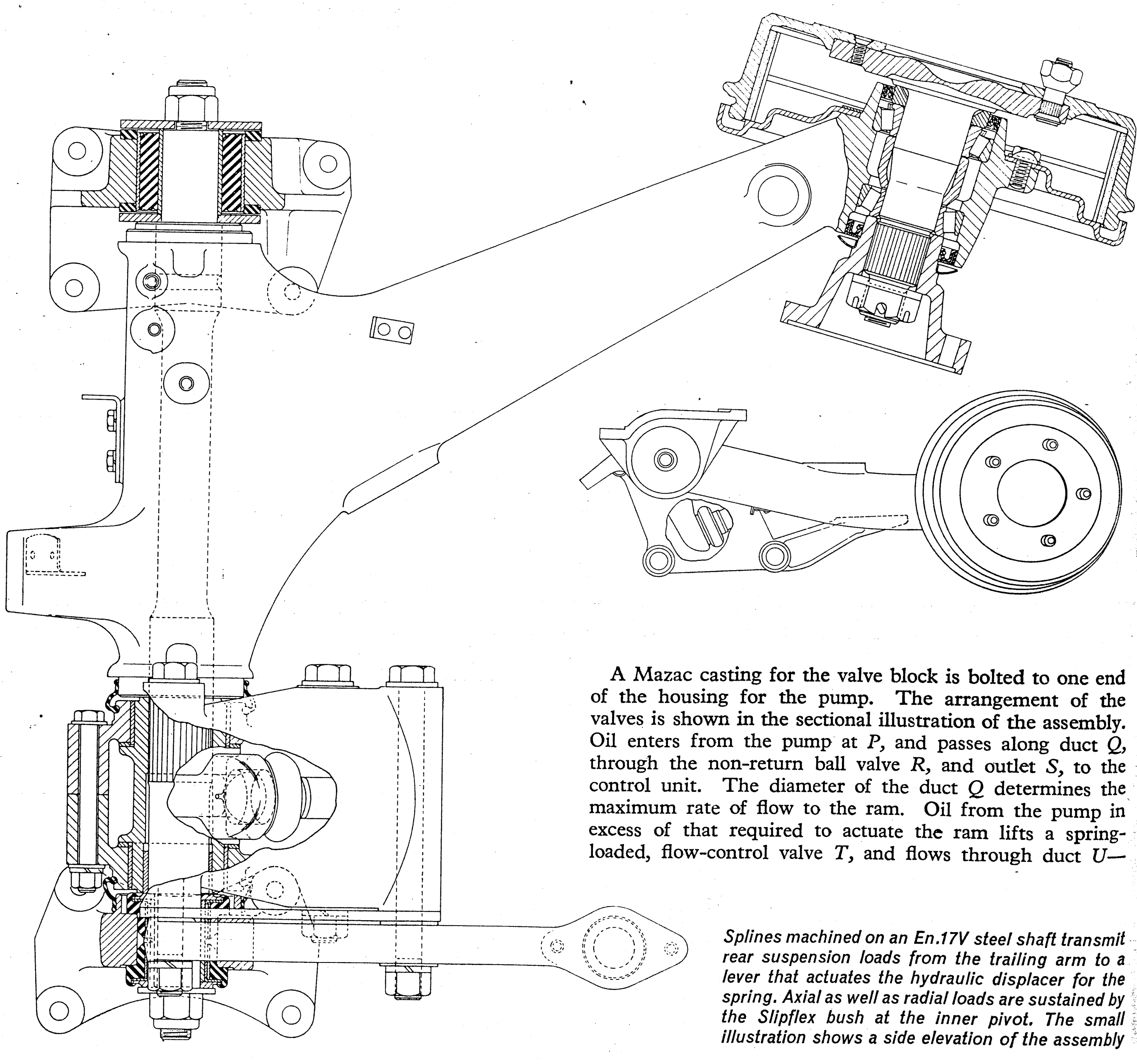
Suspension loads are sustained by a forged En.15S steel arm; it is attached to the inboard face of the inner casting by two of the $\frac{5}{8}$ in diameter bolts that secure the two castings together. This arm extends rearward, and a ball-joint with a nylon socket is assembled in the rear end. Loads are transmitted through this ball-joint to the piston-rod of a hydraulic ram, by means of which the angular position of the displacer unit is set, according to the load carried by the car, to maintain a constant standing height of the rear end of the car. The unsprung weight of each rear suspension assembly, including the wheel and the tyre, is 120 lb.

## Self-levelling system

All the components of the self-levelling system★ for the rear suspension, shown in sectional illustrations, are manufactured by Armstrong Patents Co. Ltd. They comprise a hydraulic pump, driven by the engine, a fluid reservoir, two control units, and two hydraulic rams. Each control unit, of course, is actuated by one of the rear suspension arms by means of a linkage. When the weight carried by the car increases, and the suspension springs are deflected, the control units admit fluid from the pump to the hydraulic rams, which extend to restore the car to its previous standing height. Conversely, a reduction in weight causes the control units to release fluid from the rams, and reduce the standing height. The control units and rams act independently to level the car laterally if the load is asymmetrical.



The spur gear type pump is driven at engine-speed by means of a pulley and V-belt assembly on the nose of the crankshaft. The two steel gears are $1\frac{1}{4}$ in diameter and $\frac{5}{32}$ in wide, and their integral $\frac{1}{2}$ in diameter shafts are carried in pairs of aluminium bushes in an aluminium alloy diecast housing. Oil that escapes between the outer bearings and the shafts returns through the hollow shaft of the idler gear, and $\frac{1}{16}$ in diameter longitudinal holes in the other two bushes, to the inlet for the pump. The maximum recommended speed for this pump is 5 000 rev/min.

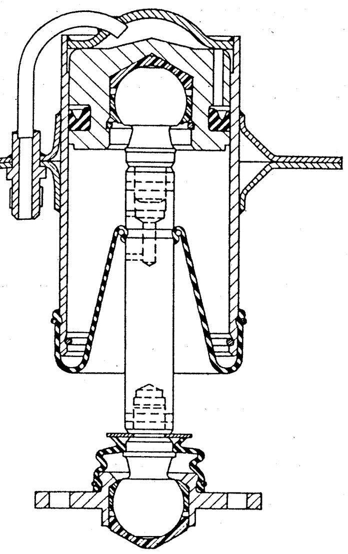


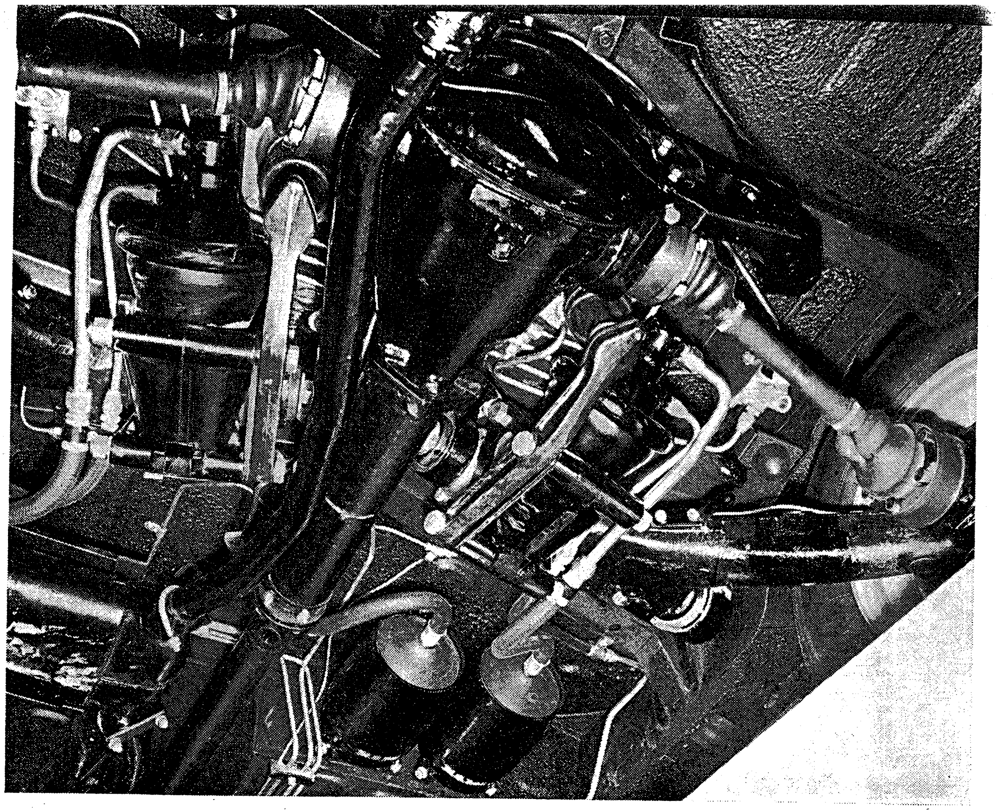
A Mazac casting for the valve block is bolted to one end of the housing for the pump. The arrangement of the valves is shown in the sectional illustration of the assembly. Oil enters from the pump at *P*, and passes along duct *Q*, through the non-return ball valve *R*, and outlet *S*, to the control unit. The diameter of the duct *Q* determines the maximum rate of flow to the ram. Oil from the pump in excess of that required to actuate the ram lifts a spring-loaded, flow-control valve *T*, and flows through duct *U*—

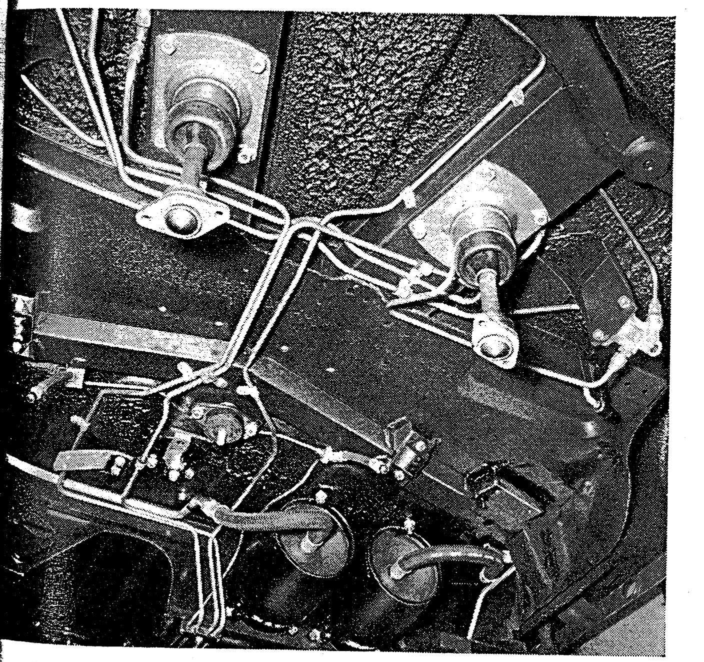
*Splines machined on an En.17V steel shaft transmit rear suspension loads from the trailing arm to a lever that actuates the hydraulic displacer for the spring. Axial as well as radial loads are sustained by the Slipflex bush at the inner pivot. The small illustration shows a side elevation of the assembly*

suspension
carried by
e deflected,
e hydraulic
s previous
ight causes
and reduce
rams act
ne load is

-speed by
nose of the
meter and
are carried
loy diecast
arings and
idler gear,
other two
m recom-







*Top left: A $2\frac{1}{2}$ in diameter aluminium alloy piston in the hydraulic ram is connected by a pushrod and two ball-joints with nylon sockets to the actuation lever on the rear suspension*

*Above: To minimize space occupied by the rear suspension assemblies, duplex spring units are mounted in a remote position under the floor of the body, and are connected by steel pipes and hoses to the hydraulic displacers. Pipes that connect the displacers to those on the front suspension can also be seen*

*Left: The two hydraulic rams, and the control units that are linked to the suspension arms, of the Armstrong self-levelling system for the rear suspension*

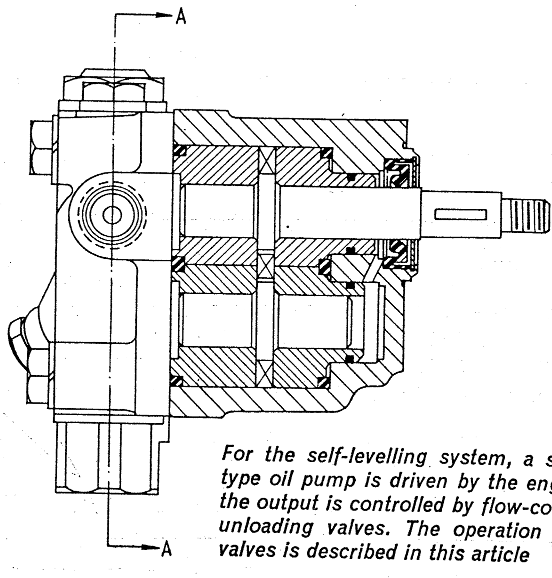
shown in section B-B—and outlet *V* to the reservoir. This last-mentioned valve limits the output from the pump to $\frac{1}{3}$ gal/min at 1 000 rev/min pump-speed, and 1 000 lbf/in$^2$ pressure. The maximum power required to drive the pump is 1/5 h.p.

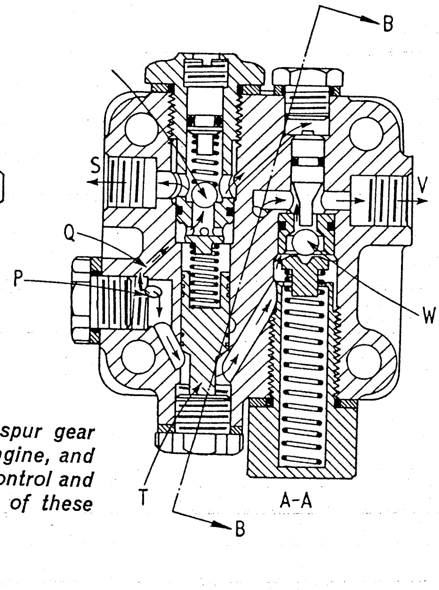
Oil flow from the pump, when none is required by the rams, is controlled by an unloading valve *W*. A $\frac{9}{32}$ in diameter steel ball is held on a seat by a helical spring and oil pressure. The same oil pressure also acts upon the upper end of a plunger in a bore above the ball-valve. The diameter of the plunger is greater than that of the seat for the ball; hence the resultant thrust is downward, and when

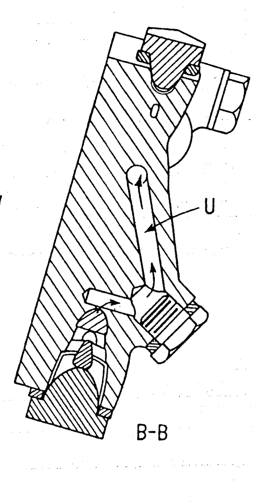
ne end
of the
sembly.
luct *Q*,
to the
nes the
mp in
spring-
t *U*—

*ansmit*
*m to a*
*for the*
*ned by*
*small*
*sembly*

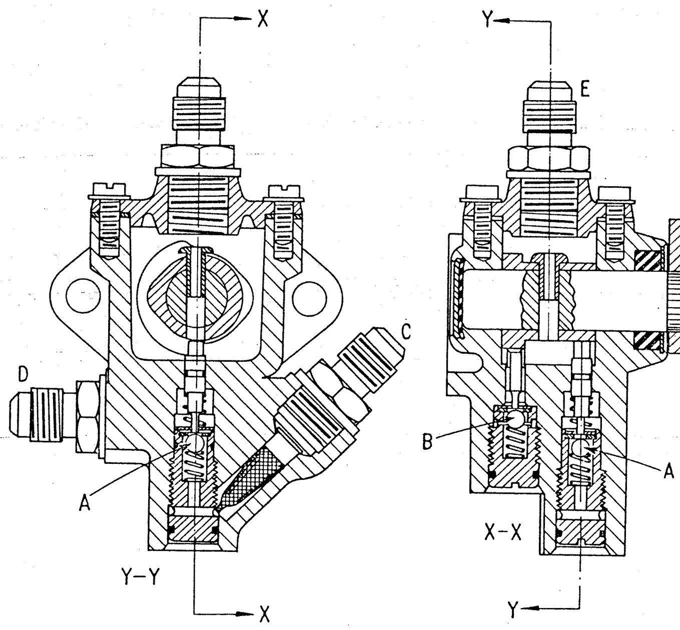
1968




*For the self-levelling system, a spur gear type oil pump is driven by the engine, and the output is controlled by flow-control and unloading valves. The operation of these valves is described in this article*




*Left: Two views of one of the control units. Valve A is opened by a cam to raise the car, and valve B by another cam to lower it. The inlet from the pump is at C, and the outlets to the ram and the reservoir are at D and E, respectively*

it is sufficient to overcome the force of the helical spring, the ball is unseated, and oil flows through outlet *V* to the reservoir. The helical spring limits oil pressure above the plunger to 900-950 $lbf/in^2$, and the non-return ball-valve *R* closes to maintain pressure in the supply pipe for the control unit.

The pressed steel cylindrical reservoir for the fluid is $3\frac{17}{32}$ in diameter and 7 in long, and is mounted within the engine compartment on the right-hand wheelarch. It contains two pints of a mineral base oil to Armstrong Specification 1709.

### Control units

Two spring-loaded ball-valves, each of which is actuated by a cam and a plunger, are incorporated in each control unit. Two hardened steel cams are machined on a sleeve, which is pressed on a $\frac{9}{16}$ in diameter shaft and retained by a $\frac{3}{16}$ in diameter Chobert snap-head rivet.

Movements of the rear suspension arm are transmitted to the shaft by a link, the length of which is adjustable, to one end of a lever, the other end of which is splined to the shaft. Rubber bushes are incorporated in the pivots for the link. The effective length of the lever is $3\frac{1}{4}$ in. Bearings for the shaft are machined in the Mazac diecast housing for the unit.

In the illustration, valve *A* is opened to admit oil to the ram, and raise the car, whereas valve *B* releases oil to the reservoir, to lower the car. When valve *A* is open, oil enters from the pump at *C*, flows through a fine mesh gauze filter and the valve, and leaves through outlet port *D* to the ram. This outlet port also communicates with valve *B* so, when this valve is open, and valve *A* is closed, oil escapes from the ram through valve *B*, along flutes in its actuating plunger to outlet *E*, and thence to the reservoir. For each valve, the area of the aperture of the steel-backed nylon seat is the same. The rate of lowering of the car is determined by the area of valve *B*, and the rate of raising by the area of the delivery duct *Q* in the pump unit. These rates are similar, and a car loaded with five 150 lb passengers and 100 lb of luggage in the boot is raised to the correct position in 18 sec. A typical time for lowering is 15 sec.

To facilitate the adjustment of each control unit when it has been installed, the cam that actuates valve *A* provides a more gradual lift of the valve than that afforded by the cam for valve *B*. The axial position of valve *A* is adjusted when the car is unladen, so that this valve is not unseated until 0·090 in travel of the end of the lever has occurred. This reduces unnecessary activity of the control unit. Another effect of this small free travel of the lever is that when a load is placed in the car, the rear end is raised to $\frac{1}{4}$ in below the correct height—a discrepancy of no practical importance. Whereas the levelling system is not applied to the front suspension, because of the arrangement of the rear suspension linkages, and the hydraulic interconnections with the front spring units, the change in load between the unladen and fully laden conditions only lowers the front end of the car approximately $\frac{1}{2}$ in.

The cylinder for each hydraulic ram is a $2\frac{1}{2}$ in internal diameter, $\frac{1}{8}$ in thick steel tube, to one end of which a domed steel pressing is welded. Two other steel pressings, each 0·080 in thick, are welded to the external surface of the tube to form a mounting bracket. A rubber seal, pressurized by the oil, is assembled in an annular groove in the aluminium alloy piston. Nominally, the maximum stroke of the piston is 3 in. The piston-rod comprises a $\frac{3}{4}$ in diameter steel bar, each end of which is drilled to receive the shank of a ball-pin; each is retained by an $\frac{1}{8}$ in diameter Mills Roll-pin. Nylon sockets for the upper ball-pin are assembled in a machined housing in the piston, and retained by a circlip. Similar sockets for the lower ball-pin are housed in a Mazac casting bolted to the arm that extends rearward from the displacer housing on the suspension assembly. The spherical diameter of each ball-pin is $1\frac{1}{4}$ in.

A neoprene dust cover, retained by garter springs, seals the lower end of the cylinder, and the chamber between it and the piston is vented to atmosphere through axial and radial holes drilled in the piston-rod. The total weight of the six components of the system is 18 lb. *To be continued*

*This illustration of a right-hand rear suspension assembly shows the aluminium alloy casting that carries the displacer unit, and the lever bolted to it for the levelling ram*