Austin 1800

A BRITISH MEDIUM-SIZE,

FRONT-WHEEL DRIVE CAR

OF OUTSTANDING

TECHNICAL IMPORTANCE

Part I: The Power Unit

IN respect of space-saving, the fundamental advantage of a transversely mounted engine and transmission unit is obvious. However, there are, of course, limitations to the length of an in-line engine that can be accommodated in this way. For their A.D.O. 17, Austin 1800 model, the British Motor Corporation found that four in-line cylinders were the maximum, and, because there would be a gain in smoothness by having more than four, investigated other configurations, including V-six units. Whereas, undoubtedly a narrow angle V-six engine offers the greatest compactness, the complexities of the induction and exhaust ports are such that volumetric efficiency and specific power output suffer. Alternatively, if a wide-angle, V-six layout is chosen, and the engine is installed transversely, the overall fore-and-aft dimension of the engine is large enough to obviate the advantage of this type of installation.

Therefore, it was decided that a version of the existing B series engine would be employed for the A.D.O. 17 model, and that chosen was the largest of the group, the 1 789 cm³ unit used in the M.G. MGB. However, to improve the smoothness of running of this four-cylinder engine, a new crankshaft, carried in five instead of three bearings, was designed. Also, the three flexible mountings for this engine were arranged to provide considerable freedom of movement of the power unit in the vertical sense, to isolate, as thoroughly as is possible, out-of-balance forces from the body structure.

The scantlings of the iron casting of the existing cylinder block have been modified to incorporate housings for two intermediate main bearings, between the axes of cylinders 1 and 2, and 3 and 4. These intermediate bearings, and the other three, are Vandervell steel backed, indium flashed copper-lead. The two intermediate bearings are 0·765 in wide, whereas the remainder are 1⅛ in wide; all have a diameter of 2¼ in. Each cast iron bearing cap is retained by self-locking nuts on two ½ in diameter studs.

One of the few differences between this cylinder block and that of the three-bearing engine is that the housing for the lubrication pump, adjacent to the rear intermediate main bearing, is machined ½ in lower in the block. This is to avoid the cutters fouling the bearing housing.

The new crankshaft is machined from an En.16 forging, and the journals and crankpins are not hardened. Integral counterweights are incorporated in the web on each side of the central journal, and in the web adjacent to each outer journal. To accommodate the two intermediate journals
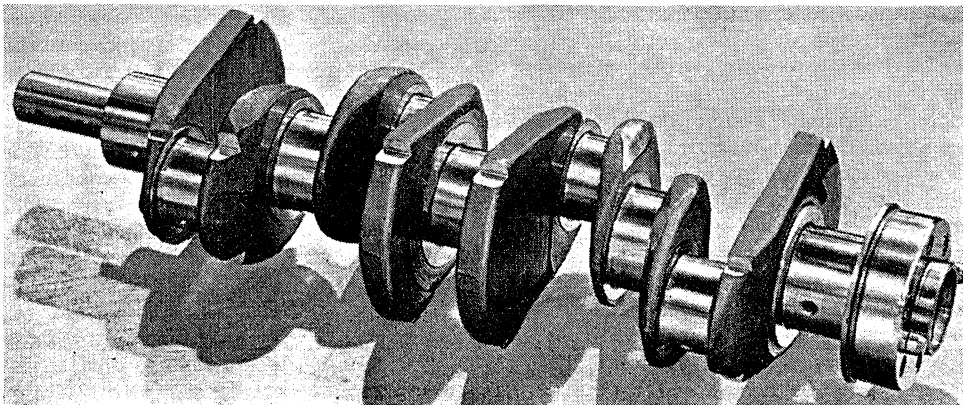
The new forged steel crankshaft for the engine of the Austin 1800 is carried in five bearings

and their webs, it has been necessary to reduce the width of the crankpins from $1\frac{1}{32}$ in to 1 in. The width of each Vandervell steel backed, indium-flashed copper-lead big-end bearing is $\frac{7}{8}$ in. Whereas, in the engine in which the crankshaft is carried in three bearings, the tranverse axes of the big-end bearings of the connecting rods are offset longitudinally 0·109 in from the axes of the cylinders, they are coaxial with the cylinders in the engines that have a five-bearing crankshaft.

To prevent oil escaping into the clutch housing, a silicone rubber lip-type seal has been incorporated between the rear main bearing and the flywheel. On other B series engines, a return-thread, scroll type seal is employed. A warning lamp on the facia, actuated by a switch in the lubrication system, warns the driver when the cartridge of the Tecalemit full-flow filter requires changing.

Bonded rubber, block type mountings carry the power unit in the unitary body-chassis structure. Two of these mountings, one in front and the other behind the crankcase, at the left hand end of the engine, are disposed at 10 deg to the vertical, and their horizontal axes are 3 in above the axis of the crankshaft. The third mounting is attached vertically to the right hand end of the cylinder block, and the horizontal axis of this mounting is $9\frac{1}{2}$ in above the axis of the crankshaft. Clearly, these mountings allow considerable freedom of movement of the power unit in a vertical plane.

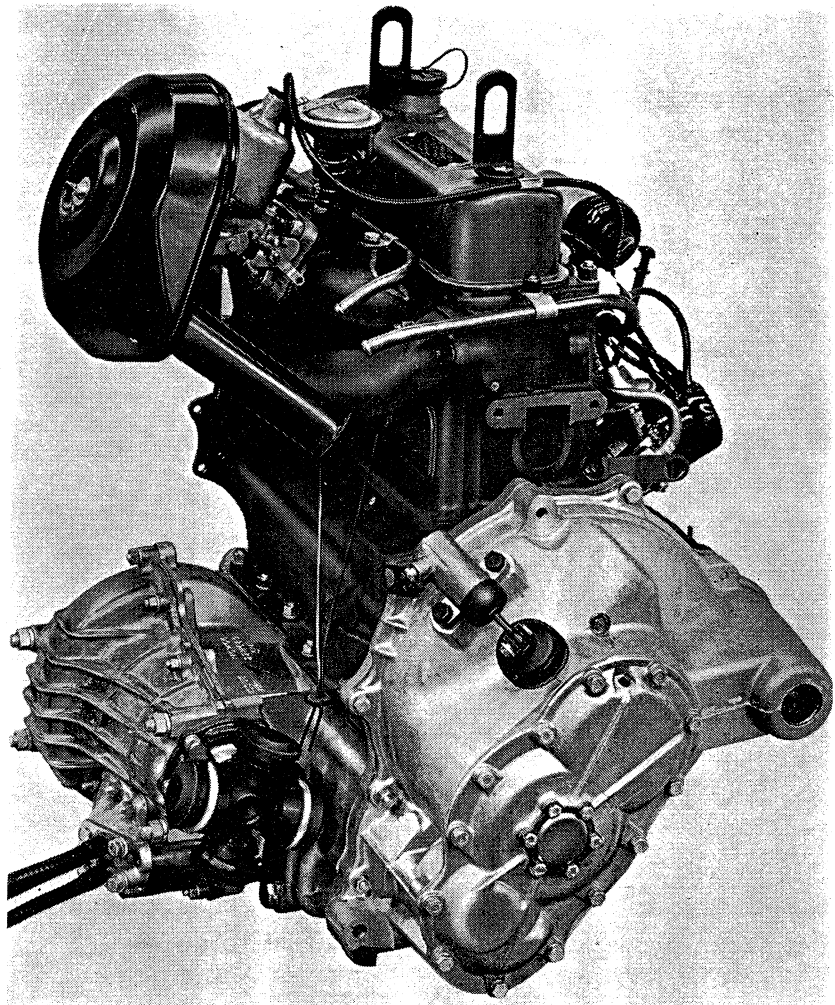The outer steel member of the upper mounting is bolted to a longitudinal, channel-section pressing, the ends of which are bolted to the body structure. A bolt in a bracket on this pressing forms the upper pivot for a Girling telescopic hydraulic damper, and the lower pivot is a bolt in a lug at the top of the clutch housing. This damper has a bore of $1\frac{1}{8}$ in and the valves are of orifice type. It prevents resonance—at the natural frequency of the mountings—excited by movements of the front suspension at high road speeds.
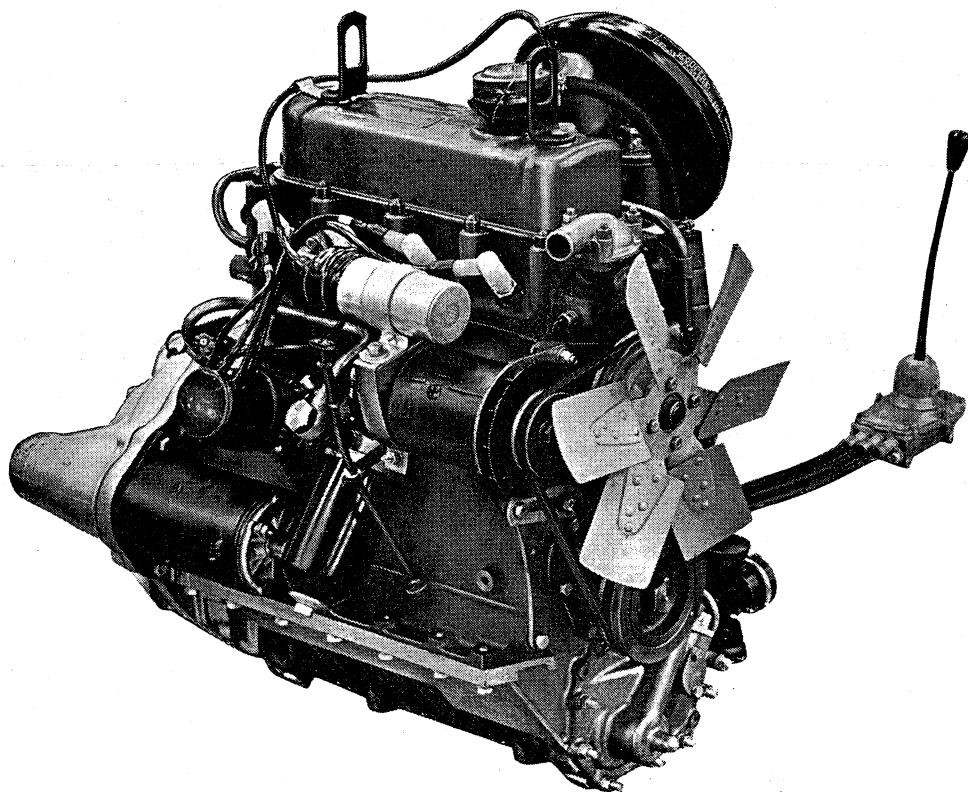
Movements of the power unit on its mountings, caused by torque reaction, are limited by a pivoted link. It is disposed longitudinally, and extends rearwards from a bolt at the base of the clutch housing, with its rear end pivoting on a bolt in the body structure. The axis of this link is 8 in below that of the crankshaft, and the dimension between the axes of its pivots is $6\frac{1}{16}$ in.

The cast iron cylinder head for the engine of the A.D.O. 17 is the same as that of the M.G. MGB model However, whereas two S.U. HS4 carburettors are installed on the M.G. MGB version, the A.D.O. 17 model has one S.U. HS6 carburettor; a Coopers air filter having a single paper element is mounted on its intake..

A positively controlled crankcase ventilation system is incorporated. The crankcase breather, and a drilled hole in the induction manifold, are connected by flexible pipes to a Smiths FVP 2001/03 one-way valve assembly. This valve regulates the flow of crankcase gases to the manifold, from whence they enter the combustion chambers and are burnt; a baffle at the outlet from the crankcase breather prevents oil being drawn into the manifold. Air can enter
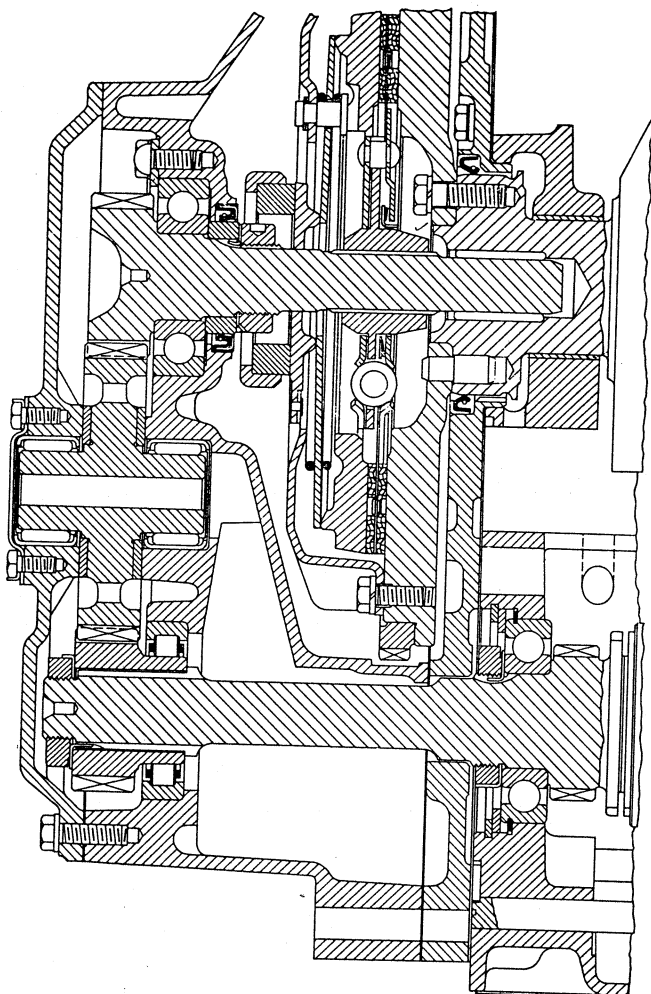
Aluminium alloy castings that house the flywheel and clutch, the primary drive gears, and the final drive assembly, can be seen in this illustration of the power unit. A Smiths control valve for the crankcase ventilation system is on the right of the carburettor. The upper engine-mounting is bolted to the machined face on the end of the cylinder block

the crankcase only through two $\frac{1}{4}$ in $\times \frac{1}{8}$ in slots and gauze filter in the oil filler cap on the rocker cover. It is prevented from entering the crankcase, at the opening for the dipstick, by a moulded plastics sleeve assembled on the dipstick. This sleeve registers with a reamed bore in the upper part of the tube in which the dipstick is carried. Hence, when the engine is running, air circulates inside the valve cover and prevents condensation of water vapour on the valve gear. Because the coolant radiator is mounted at one side of the engine compartment, temperatures within the compartment are lower than if the radiator were placed in the conventional position, ahead of the engine. Hence, in this installation, thorough ventilation of the valve gear is important.

Coolant water circulation is assisted by a centrifugal impeller, driven by a $\frac{3}{8}$ in wide V-belt from a pulley on the nose of the crankshaft. Bolted to a flange on the front of the shaft of this pump, which is driven at 0·9 crankshaft speed, is a six-blade steel fan; the diameter of the blades is 13 in. This fan draws air from the engine compartment and directs it through the radiator to outlets in the left-hand front wheel valance. The 2·038 in thick matrix of the radiator, which comprises brass tubes and copper gills, is $14\frac{1}{2}$ in deep and 14 in wide.

The cooling system is filled with a mixture of 67 per cent water and 33 per cent anti-freeze, and has a capacity of $9\frac{1}{2}$ pints, including the interior heater. A flexible pipe connects a pipe in the neck of the filler cap with a $\frac{1}{2}$ pint pressed steel tank, mounted on the dash. This tank is fitted with a filler cap containing a valve which controls the pressure in the system to 13 lb/in². This tank also accommodates coolant expelled from the header tank when the coolant expands. The manufacturers state that neither topping up, nor draining and refilling the system, is required between intervals of two years. A Western-Thomson wax-element thermostat is installed.

The transmission

In principle, the arrangement of the transmission is similar to that of the A.D.O. 15 and A.D.O. 16 models, but there are two important differences: first, the flywheel and clutch are inboard of the primary drive gears, whereas, of course, in the A.D.O. 15 and A.D.O. 16 models, they

Left: In this end elevation of the transmission casing and clutch housing, X-X is the oil level in the transmission casing. The higher oil level, Y-Y, for the primary drive gears, is maintained by a weir in their casing. The axes of the shafts are: A crankshaft; B primary drive idler gear; C first and third motion shafts; D layshaft; E reverse pinion. Below: Splined, sliding couplings are incorporated in the output shafts of the final drive assembly. On the right-hand side of the gauze suction filter in the transmission casing, is a ceramic magnetic filter
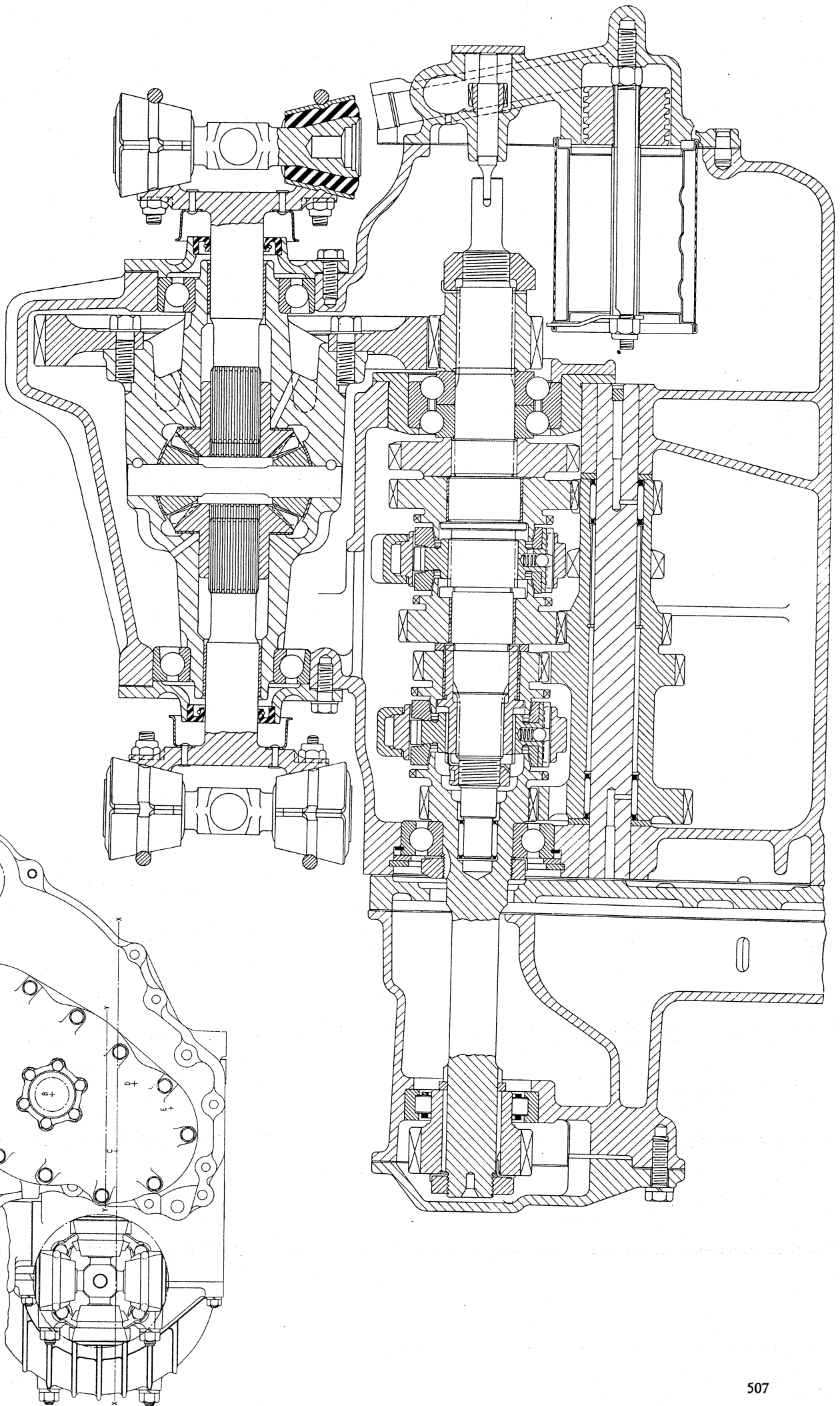
The mainshaft and layshaft gears are housed in the aluminium alloy transmission case, bolted to the lower face of the crankcase. Also in the casing is a suction filter for the oil that lubricates these gears, the final drive gears and the engine
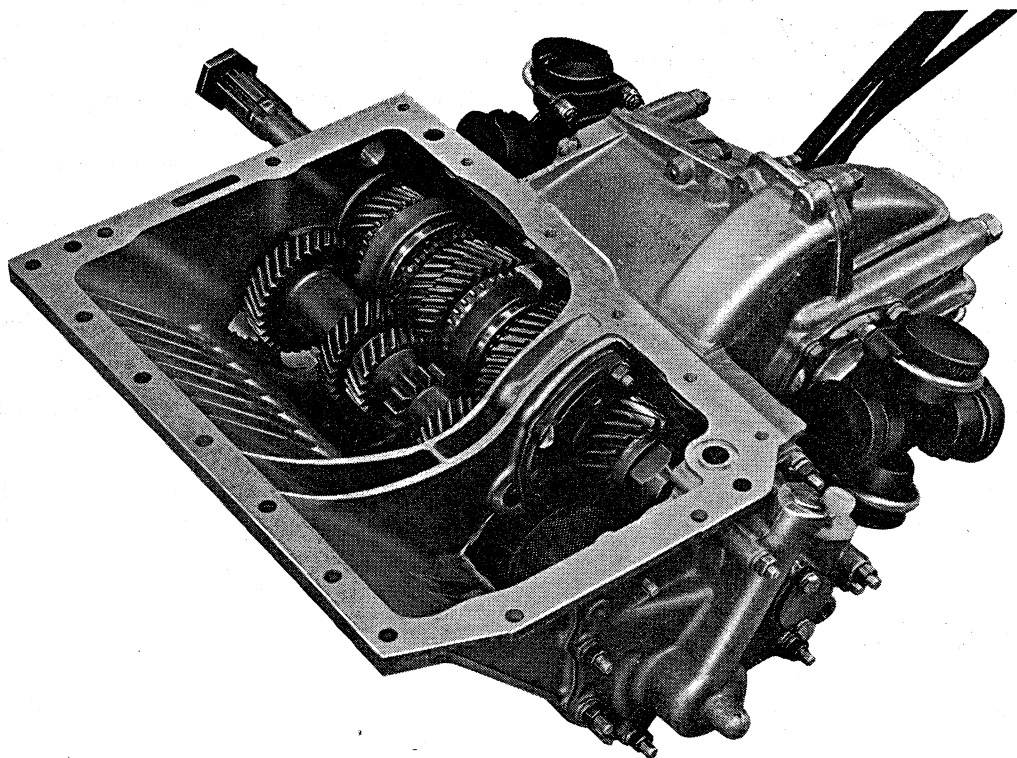
AUSTIN 1800 SPECIFICATION DATA

Engine

Number of cylinders 4, in line

Bore 80·26 mm

Stroke 88·90 mm

Swept volume 1 798 cm³

Firing order 1, 3, 4, 2

Maximum b.h.p. 85 net at 5 300 r.p.m.

Maximum b.m.e.p. 136 lb/in² at 2 100 r.p.m.

Maximum torque 99 lb-ft at 2 100 r.p.m.

Compression ratio 8·2:1 (6·8:1 optional)

Crankshaft Five-bearing, forged steel, counter-balanced

Valves Vertical, in-line in cylinder head, pushrod operated

Carburettor S.U. type HS6

Combustion chamber type approximately bath-tub shape

Fuel lift pump S.U. electric

Transmission

Clutch Borg and Beck 8 in diameter single dry plate, with diaphragm spring; hydraulic actuation

Gearbox Four forward ratios, all with baulk-ring synchromesh

Gear ratios:

fourth	1:1
third	1·384:1
second	2·217:1
first	3·292:1
reverse	3·075:1

Final drive Helical spur gears, ratio 4·188:1 (3·882:1 optional)

Suspension

Front Transverse upper arm; transverse lower arm and radius rod

Rear Independent, single trailing arm, auxiliary torsion bars and anti-roll bar

*Springing—front and rear—*Moulton Hydrolastic units, comprising shear-compression rubber springs, and integral hydraulic damping. Units hydraulically interconnected left front to left rear, and right front to right rear

Steering

Type Rack and pinion

Turns, lock to lock 4·4

Turning circle 34·5 ft (between kerbs)

Brakes

Make and type Girling hydraulic with vacuum servo, and skid control valve

Front 9$\frac{9}{32}$ in diameter discs; total swept area 183 in²

Rear Leading and trailing shoes, 9 in diameter drums and 1$\frac{3}{4}$ in wide linings; total swept area 99 in²

Wheels and tyres

Wheel type Pressed steel disc, 4·5 in wide rim, 5-stud fixing

Tyre size 175-13 in, Dunlop SP41

Pressure

front	30 lb/in²
rear	20 lb/in²

Dimensions

Wheelbase 8 ft 10 in

Front track 4 ft 8$\frac{1}{4}$ in

Rear track 4 ft 7$\frac{1}{2}$ in

Ground clearance 6$\frac{1}{2}$ in (laden)

Overall length 13 ft 8$\frac{5}{16}$ in

Overall width 5 ft 7 in

Overall height 4 ft 7$\frac{1}{2}$ in (unladen)

Frontal area 20·7 ft²

Kerb weight with 8 gallons of petrol 2 644 lb

Weight distribution

front	63 per cent
rear	37 per cent

Engine, gearbox and final drive, dry weight 555 lb

Body shell weight, white 765 lb

are outboard of these gears; secondly, a synchronizer has been incorporated in the first speed gear.

Because the flywheel of the A.D.O. 17 is spigoted and bolted directly to a flange on the end of the crankshaft, torsional loads on the crankshaft are avoided. The cast iron flywheel is 11$\frac{3}{8}$ in diameter, weighs 21$\frac{1}{2}$ lb, and has an inertia of 585 lb-in². A Borg and Beck, 8 in diameter, diaphragm-spring clutch has been chosen. Its spring exerts a clamping load of 800 lb minimum on the single driven plate, the Mintex H.22 friction faces of which have a total area of 45 in². A carbon thrust withdrawal bearing is installed, and the hydraulic components of the withdrawal
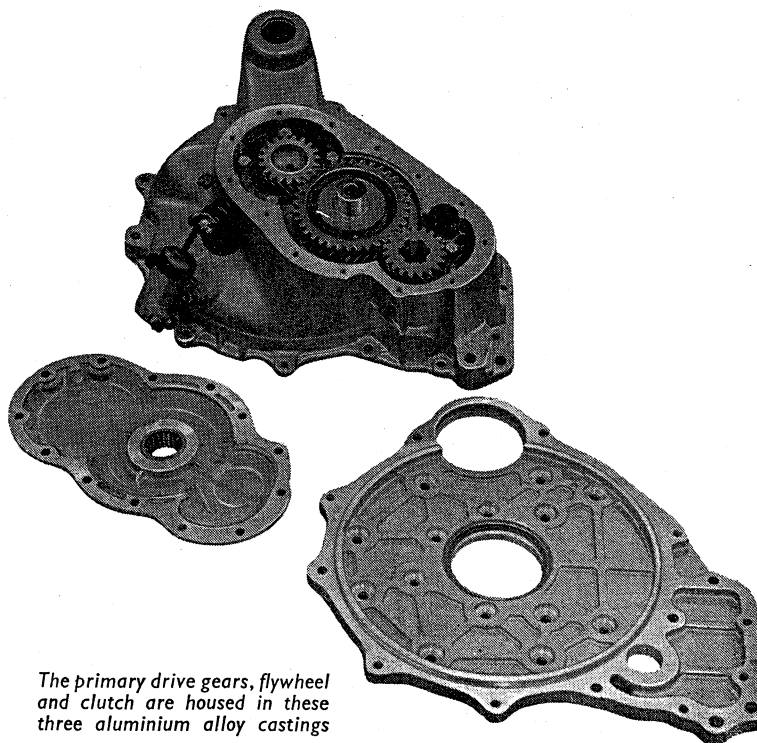
The primary drive gears, flywheel and clutch are housed in these three aluminium alloy castings

mechanism comprise a $\frac{5}{8}$ in diameter master cylinder, and a 1 in diameter slave cylinder. The flywheel-clutch unit and primary drive gears are housed in a BS.1490 LM24M aluminium alloy, high pressure diecasting. A wall in this casting, and a silicon rubber oil seal, separates the space occupied by the clutch from that in which the primary drive gears are housed; hence lubricant is excluded from the clutch. Two pints of oil, in a reservoir formed in the base of the housing, serve to lubricate the primary drive gears.
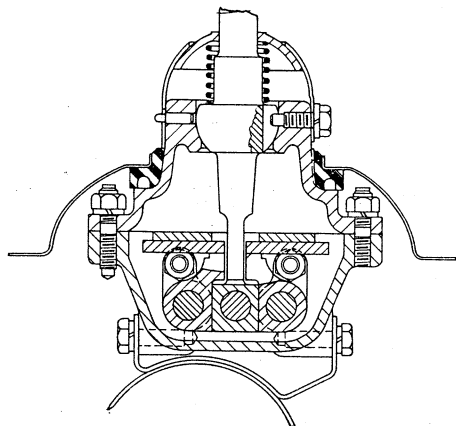
The first of the three primary drive gears, shown in an accompanying sectional illustration, is machined integral with the outer end of the En.352 steel output shaft of the clutch. A Ransome and Marles MJ 35 mm ball journal adjacent to the gear, and a porous bronze bush at the inner end, carry this shaft; the ball bearing is in the flywheel-housing, and the bush is in a hole machined coaxially in the end of the crankshaft. An idler gear, which meshes with the gears on the output shaft of the clutch and the outer end of the gearbox first-motion shaft, is machined from En.352 steel, and has a hollow, integral spindle. This spindle is carried in two caged needle roller bearings; one of these bearings is pressed into a hole machined in the flywheel housing and the other in a hole in the wall of the aluminium alloy diecast cover for the housing. An En.18C thrust washer on each side of the gear locates it axially. The gear on the outer end of the gearbox first-motion shaft is carried by a Ransome and Marles LLRJ 35 mm roller bearing. This gear is splined to the shaft and is retained axially by a ring-nut and a collar.

All the primary drive gears are 1 in wide, and have single helical teeth. The diametral pitch of the teeth is 8, the helix angle is 30 deg, and the pressure angle is 20 deg. A Ransome and Marles LJ $1\frac{1}{2}$ in diameter ball bearing carries the inner end of the En.352 steel first-motion shaft. This bearing is housed in a wall at one end of the transmission case. At present, this case, which houses the forward and reverse gears and forms also the oil sump for the

pints of oil in the transmission case also is shown. This oil, which lubricates also the final drive gears, circulates in the lubrication system of the engine. As was mentioned previously, the inner end of the first-motion shaft is carried in a ball bearing in the wall at the right-hand end of the transmission case. In the plane of this bearing, and coaxial with the first motion shaft is a caged needle-roller spigot bearing that carries one end of the third-motion shaft. The other end of this shaft is carried in a Ransome and Marles MJ 83 mm double-row ball bearing; this bearing is adjacent to a helical spur gear—splined to an overhung part of the shaft—that meshes with the input gear of the final drive assembly, and it therefore sustains axial and radial loads imposed by these gears. The dimension between the axis of this double-thrust bearing and that of the needle roller bearing that carries the other end of the shaft is $9\frac{1}{16}$ in.
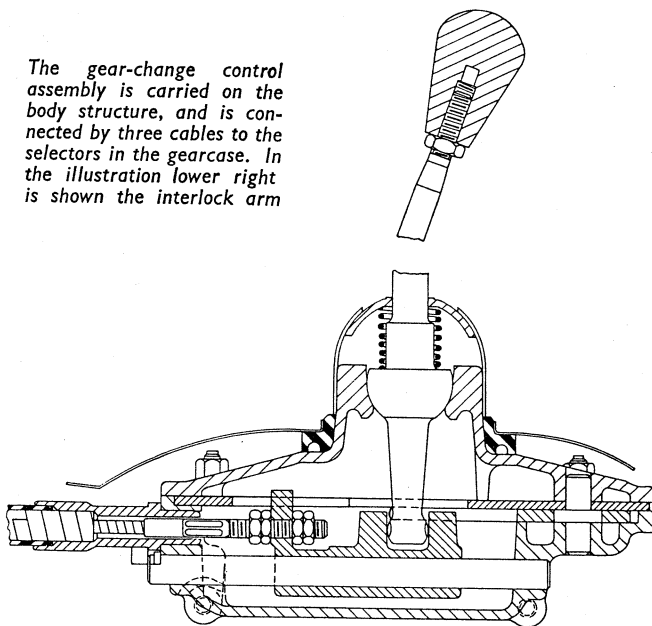
En.352 steel has been chosen for the third-motion shaft, and bi-metal bearings, pressed into the bores of the constant-mesh first and second gears, carry these gears on journals machined on the shaft. However, the similarly bushed, third gear operates on the outer surface of an En.352 steel sleeve splined to the shaft. Splines extending for part of the length of the external surface of this sleeve engage with others machined in the inner member of the synchromesh assembly for third and fourth speeds. A ring-nut, locked by a tab washer, is screwed on the shaft to retain this sleeve.

Baulk-ring synchronizers of the type installed in other

The gear-change control assembly is carried on the body structure, and is connected by three cables to the selectors in the gearcase. In the illustration lower right is shown the interlock arm
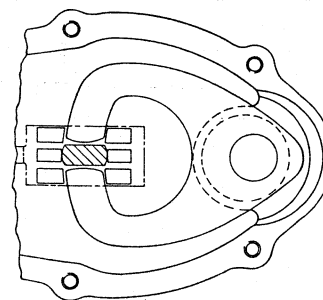
engine and gearbox, is a BS.1490 LM24M aluminium alloy gravity diecasting produced by Stirling Metals Ltd. Eventually, it will be replaced by a high pressure diecasting of the same material, produced by Birmingham Aluminium Co. Ltd, and will be one of the largest castings of this type in the motor industry. The nominal wall thickness of the gravity diecasting is $\frac{3}{16}$ in. Cast integrally with the rear wall of the case is the front half of the housing for the final drive assembly.
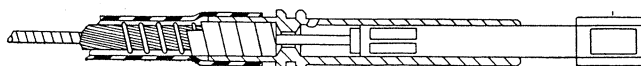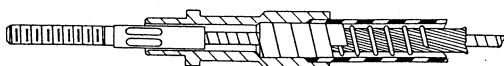
An accompanying illustration shows the relative positions of the axes of the mainshafts, the layshaft, the reverse speed pinion, and the final drive assembly; the level of the $13\frac{3}{4}$

Below: Steel strip, spirally wound round the inner member of each cable, allows it to transmit compression loads

B.M.C. models, are employed for all forward gears of the A.D.O. 17; the two synchronizer assemblies are similar. Sintered steel, induction hardened to 60 Rockwell, C scale, has been chosen for the baulk rings, and the included angle of their conical faces is 12 deg. The conical faces on the gears, with which the baulk rings engage, are sprayed with molybdenum to a depth of 0·010-0·014 in. This method allows steel baulk rings with induction hardened cam faces to be used, yet avoids the risk of seizure of the steel cone surfaces. It is employed also in the gearboxes of A-series power units. Full details were published in the May 1964 issue of *Automobile Engineer*.

All the gears on the third-motion shaft are machined from En.352 steel, and their surface hardness is 60 Rockwell, C scale. All forward gears have single helical teeth. The gear ratios, diametral pitches and helix angles are as follows:

Gear	Ratio	Diametral pitch	Helix angle
4th	1:1	10·0231	33 deg 21 min 26 sec
3rd	1·3841:1	10·0231	33 deg 21 min 26 sec
2nd	2·217:1	10	30 deg
1st	3·292:1	10	30 deg

The reverse gears have straight spur teeth, of 8 diametral pitch, and the ratio is 3·075:1.

All the gears on the layshaft are machined integrally with this shaft, which is carried by two needle roller bearings on a $\frac{7}{8}$ in diameter En.352 steel spindle. An En.18C washer at each end of the shaft sustains axial thrust. The layshaft is machined from En.352 steel, and the surface hardness of the gears is 60 Rockwell, C scale.

Gear-change control

The problem of the isolation of the motions of the power unit from the gear lever is more acute when the engine is installed transversely than when it is in the conventional, longitudinal position. Whereas the gear lever of the A.D.O. 15 and A.D.O. 16 models is carried at the rear end of a long casting bolted to the power unit, and the rocking of the engine on its mountings, caused by torque reaction, is amplified at the gear lever, the base of the

gear lever of the A.D.O. 17 is carried in a housing mounted on the body structure, and longitudinal movements of the lever are transmitted by three enclosed cables to the selector levers in the transmission case.

Sectional illustrations show details of the design of the gear-change control assembly and of the cables. The housing of the gear-change control assembly comprises two BS.1490 LM24M aluminium alloy diecastings. A hemisphere machined near the base of the gear lever is loaded by a helical spring against a spherical seat in the upper casting. Extending down from the hemisphere on the gear lever is a striker-arm which moves to engage with a fork in any of three members that slide on spindles pressed into holes in the front and rear walls of the lower casting. At the front end of each of the sliding members is a lug in which there is a tapped hole to receive the screwed end of one of the cables, retained by lock nuts. One of these cables actuates the selector lever for first and second gears, another third and fourth gears, and the other reverse gear.

To prevent the simultaneous selection of two gears, an interlock arm is incorporated. This is shown in the illustration on page 509. The assembly is lubricated by grease. Four setscrews in the base of the lower casting attach it to a 16 s.w.g. steel pressing. This pressing is spot-welded to an 18 s.w.g. pressing, of channel section, which extends forwards towards the dash. Two bobbin type rubber mountings at the rear end of this pressing and one at the front attach it to the floor pressing of the body structure. The aperture in the floor pressing, through which the upper casting of the gear change control assembly projects, is sealed by a moulded rubber ring; this ring also forms an additional support for the assembly.

British Wire Products Ltd, Stourport, manufacture the gear change cables. The inner member of each cable has steel strip spirally wrapped round it to allow it to transmit compression loads; the nominal diameter is $\frac{5}{32}$ in. Each slides in a steel wire case, lined with pvc. This material reduces friction and seals the lubricant in the cable. A $\frac{9}{16}$ in outside diameter rubber tube, $\frac{1}{8}$ in thick protects each cable. The overall length of the two cables for the forward gears is 27 in, and that of the cable for reverse gear is 27½ in.

Final drive

For the final drive, the gear reduction of 4·188:1 is effected by two helical gears. One of these gears has 16
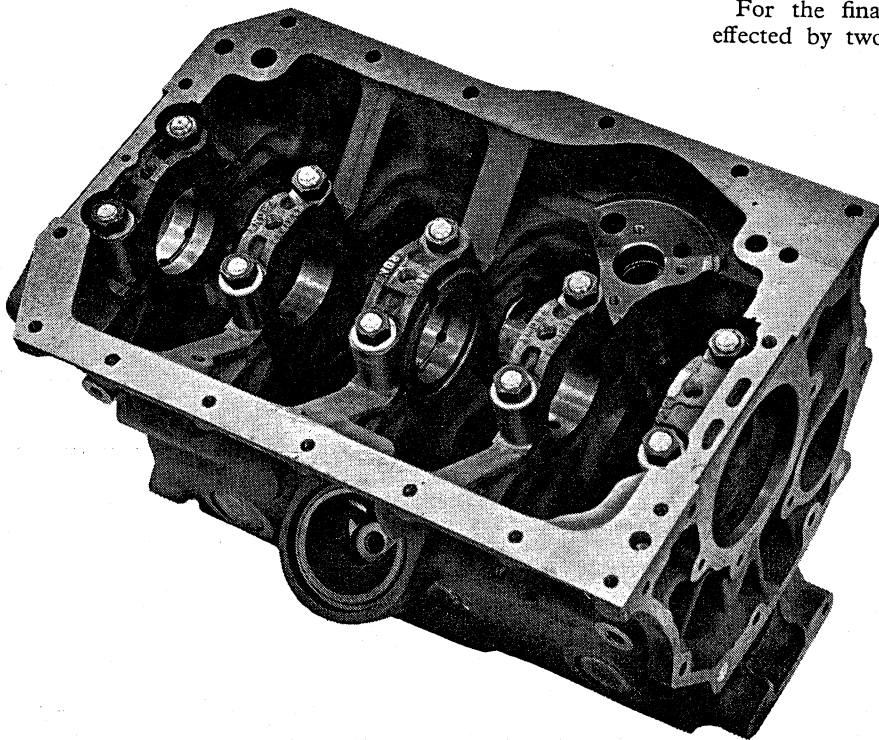
In this view of an inverted crankcase can be seen, beyond an intermediate main bearing, the machined face for the oil pump; on the outside of the crankcase is the mounting for the full-flow filter of the lubrication system

teeth, and is splined to one end of the third motion shaft, and the other, a 67 tooth gear, is spigoted and bolted to one end of the differential housing. These gears are machined from En.352 steel, and their surface hardness is 60 Rockwell C scale. The diametral pitch of the teeth is 8, the helix

At present, the transmission case is an aluminium alloy gravity diecasting, but in due course it will be replaced by a high-pressure diecasting
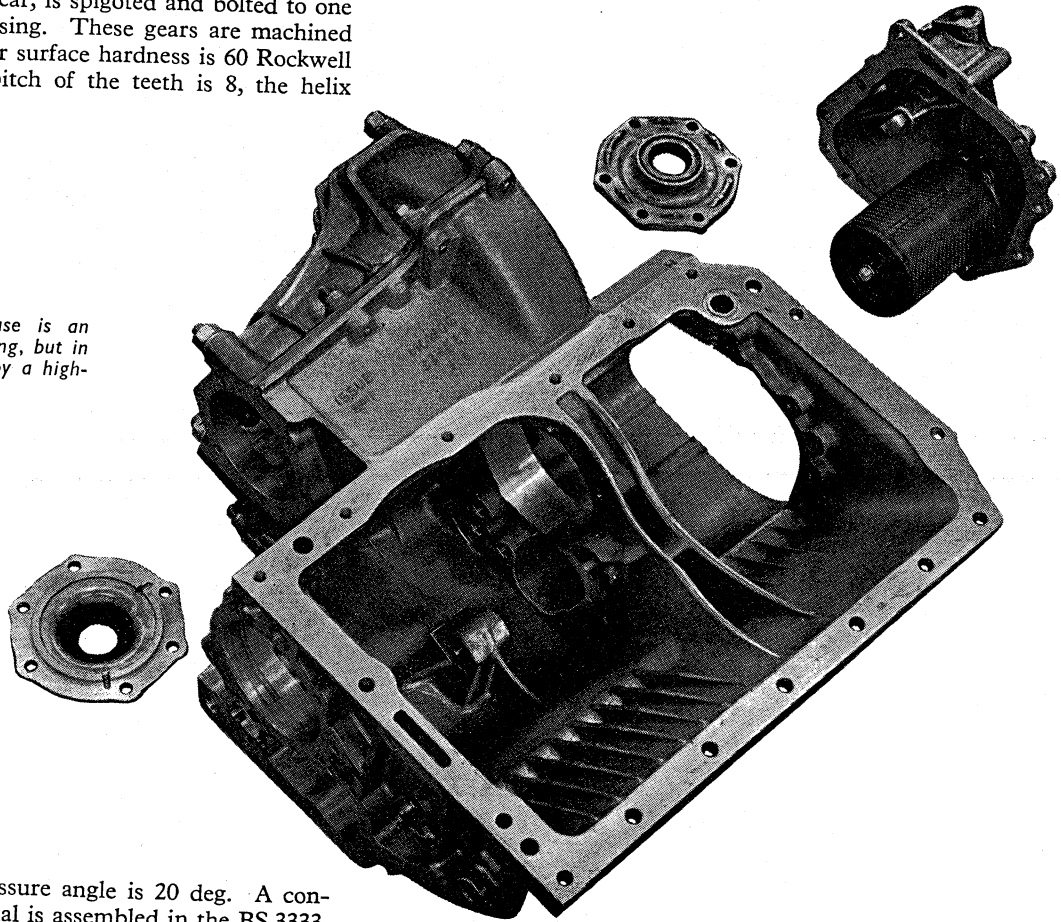
angle is 30 deg, and the pressure angle is 20 deg. A conventional bevel gear differential is assembled in the BS.3333, P33/4 malleable iron differential housing, each end of which is carried in a Ransome and Marles LJT 45 mm ball bearing.

A splined, sliding coupling is incorporated between each bevel gear in the differential and its output shaft. These couplings, of course, accommodate axial movements of the drive shafts, attached by Hookes joints to the output shafts. Clearly, since the splined couplings are within the final drive assembly, and are lubricated by the oil in this assembly, sliding friction is reduced to a minimum, and consistent action is obtained. The total axial travel of each output shaft, from its position when the suspension is in the static laden condition, is $\frac{5}{16}$ in. For the output shafts, En.12B-R steel has been chosen, and the outside diameter of the splines of the coupling is $1\frac{1}{16}$ in. A $1\frac{1}{8}$ in inside diameter bi-metal bush in each end of the differential housing carries the outer ends of the output shafts.

Drive shafts

The Hookes joint at the inner end of each of the drive shafts is of the same type as that installed in the other B.M.C. front-wheel-drive models. In place of the conventional needle roller bearings, this joint—a Moulton flexible coupling—has rubber bushes of conical shape to prevent harshness in the take-up of the drive to the shafts. The solid drive shafts, which have a nominal diameter of 1 in, are of En.19C steel. Splined on to the inner end of each shaft is the En.8M steel forged yoke for the Hookes joint. A lip at the outer end of this yoke is rolled over a shoulder on the shaft, to retain it axially.

Birfield Industries Ltd. manufacture the drive shafts and the Rzeppa constant-velocity universal joints at their outer ends. The maximum angle at which these joints are required to operate is 42 deg. From the accompanying illustration of a drive shaft, it can be seen that the outer member of the universal joint is integral with the stub shaft for the hub of the road wheel. This stub shaft is machined from an En.207 forging; splines of $1\frac{3}{8}$ in outside diameter and $1\frac{1}{8}$ in long are machined on the shaft to receive the hub of the wheel.

To be continued

The Birfield-Rzeppa constant velocity universal joints, at the outer ends of the drive shafts, operate at a maximum angle of 42 deg
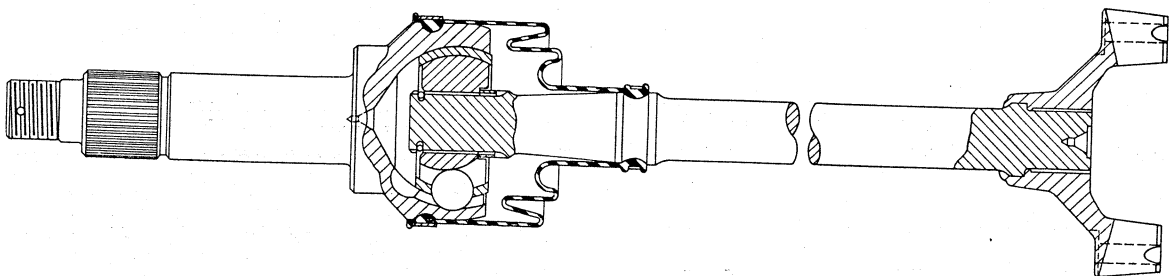
AUSTIN 1800 part two

Front and rear suspension

Hydrolastic interconnected springs

Rack and pinion steering

Girling braking system

N ITS OVERALL CONCEPTION, the design of the suspension system for the A.D.O.17 is the same as that for the A.D.O.16, 1100 models introduced in 1962. Briefly, a long wheelbase, relative to the overall length of the car, has been chosen to ensure good directional stability and rapid steering-response. To avoid a high frequency of pitch and to give good ride qualities, the suspension springs of the A.D.O.17 are interconnected, left front to left rear, and right front to right rear, and the effect is to produce a virtual wheelbase that is shorter than the actual. Hence, a lower pitch frequency and a better ride are obtained.

As is well known, the Moulton Hydrolastic spring units, one of which is installed in each suspension assembly, comprises a rubber spring, which is loaded in compression and shear by fluid pressure. A fluid damper is incorporated in the unit, and loads imposed on one of the springs are transferred by fluid in a small bore pipe to the other. The characteristics of the rubber springs are such that they offer a progressively increasing resistance to the loads imposed on them; also the volume of fluid displaced through the damper valves increases more than linearly with the deflection of the spring. Because the fluid is water—with antifreeze added—its viscosity remains virtually constant at all ambient temperatures; moreover, by virtue of its large volume, the fluid in the system can absorb and dissipate heat readily. When the rates of movement of the suspension are high, the fluid flows more rapidly in the interconnecting pipes and damping is therefore greater than at lower rates of movement.

It is evident that, when the car is cornering, the hydraulic interconnection between the spring units for the outer, heavily loaded wheels effects a dynamic balance between the reactions to the vertical loads at these wheels, which promotes good adhesion. Clearly, when roll occurs, there is little or no flow of fluid from one outer spring unit to the other, and the combined action of both springs produces a high resistance to roll.

Front suspension assemblies

Accompanying illustrations show the general arrangement of a front suspension assembly. Unlike the layout on the other B.M.C. front-wheel-drive cars, the Hydrolastic spring units are opposed, and are carried inside a transverse tubular member of the body structure. The piston of each spring unit is actuated by the upper arm of the suspension, through the medium of a short push-rod and a ball-joint embodied in the arm. Between the axis of the inner pivot of this arm and that of the ball-joint at its outer end, the dimension is $8\frac{5}{32}$ in. In the static laden condition, the leverage ratio is $3.75 : 1$. The dimension betwen the axes of the inner pivot and the ball-joint on the lower arm of the suspension is $10\frac{5}{32}$ in, and the height above the ground of the roll centre of the suspension is $4\frac{5}{16}$ in.

Other dimensions relevant to the front suspension are: track 4 ft $8\frac{1}{2}$ in; wheel camber angle 1 deg; angle of swivel axis in a transverse plane $12\frac{1}{2}$ deg; offset at the road between the swivel axis and the central plane of the wheel $\frac{15}{16}$ in; castor angle 1 deg. Dunlop SP41 173-13 in radial ply tyres have been chosen, and they are fitted to $4\frac{1}{2}$ in wide rims.

The static laden weight on each front wheel is 792 lb. From the static laden position, the maximum deflections of the suspension, at the central plane of the wheel, are: bump $3\frac{1}{16}$ in; rebound $3\frac{3}{8}$ in. No anti-roll bar is installed.

Details of design

The inner pivots of the upper and lower arms for each suspension assembly are carried in a BS 1490 LM 22W aluminium alloy casting. A machined face at the upper end of each casting registers with a forged steel flange welded to one end of the transverse steel tube in which the

spring units are carried. Two $\frac{1}{2}$ in diameter and two $\frac{3}{8}$ in diameter studs in each flange, and self-locking nuts, retain the castings; the lower end of each casting is attached by a $\frac{1}{2}$ in diameter bolt and self-locking nut to a deep gusset-pressing welded to the outer portion of the tube. The manufacturers maintain that the mass of metal in these castings and in the substantial forged steel suspension arms helps to damp road-excited vibrations and prevent their transmission to the body structure. A rubber bump stop is bonded to a steel bracket and attached by screws to the upper edge of an opening in the casting, through which the upper arm of the suspension protrudes; a similar rebound stop is attached to the lower edge of this opening. From the static laden position, the road-wheel rises $2\frac{3}{8}$ in before the arm begins to compress the bump stop; the corresponding travel to contact the rebound stop is $2\frac{1}{16}$ in.

The bearing for the inner pivot of each upper arm comprises Timken $1\frac{63}{64}$ in outside diameter opposed taper roller bearings; between the central plane of these two bearings the dimension is $4\frac{3}{16}$ in. Their inner races are carried on a 1 in diameter cold drawn steel tube, which is a loose fit on the En.8R steel pivot spindle. This spindle is a $\frac{5}{8}$ in diameter bolt in two drilled bosses in the aluminium alloy casting, and is retained by a self-locking nut tightened to 60 lb-ft torque. A moulded rubber lip type seal incorporated in each roller bearing prevents the escape of grease from the assembly; no periodic lubrication is required.

Rubber bushes have been provided for the inner pivot bearings of the lower arms. For each bearing, there are two Harrisflex half bushes, shown in an illustration on page 4. The $\frac{3}{4}$ in diameter pivot is machined from En.16T steel, and a $\frac{7}{16}$ in. nut secures a tapered portion of the pivot in a tapered hole in the arm.

Housings for two ball-joints supplied by Engineering Productions (Clevedon) Ltd. are machined in the En.16T forging of the upper arm of the suspension. Near the inner end of the arm, the ball-joint[o] that actuates the spring unit comprises an En.8D ball-pin, a neoprene dust cover and a socket of nylon impregnated with molybdenum disulphide pressed into a hole in the forging. Because the ball-pin is subjected only to compression loads, a second socket is

not required. The diameter of the spherical head of the ball-pin, which is induction hardened to 52 Rockwell C, is $1\frac{1}{16}$ in. During assembly, the joint is filled with Dexta grease containing 10 per cent Dextrol. It is effectively sealed by the dust cover, and requires no periodic lubrication.

The upper and lower ball-joints[*] are identical. A steel-housing for the upper one is screwed into a tapped hole in the suspension arm, and is locked by a tab washer. The blind end of the hole in the arm is conical, to form a seat for the nylon upper socket; nylon is used also for the lower socket, and both are impregnated with molybdenum disulphide.

The upper end of the neoprene dust cover is retained in a groove on the housing by a circlip; and a steel ring is bonded into its lower end, which fits between a shoulder on the ball-pin and the machined upper face of the vertical link. Neither this joint, nor the lower one, require periodic lubrication, since they are filled with Dexta grease, containing 10 per cent Dextrol, when they are assembled.

The lower transverse arm of the suspension is an I-section En.15R forging. To sustain horizontal loads imposed by traction and braking, it is braced by a $\frac{5}{8}$ in diameter En.14A radius rod that extends forwards to a pivot on the body structure. The effective length of this rod is $18\frac{5}{8}$ in; its rear end is bolted to the transverse arm $7\frac{1}{4}$ in from the axis of its inner pivot. At the rear end of this radius rod, an En.2D-1 steel yoke is arc welded, with a low-hydrogen welding rod. A $\frac{7}{16}$ in diameter bolt and self-locking nut secure the yoke to the transverse arm. Near the forward end of the rod, a steel collar is welded to form one abutment for a rubber bush: the other abutment is the face of a bracket welded to a steel pressing bolted to the body structure. A similar rubber bush is pressed against the opposite face of the bracket by a self-locking nut and a steel washer on the threaded end of the rod. These rubber bushes, of course, accommodate articulation of the rod, and provide insulation against the transmission of road-excited noise to the body.
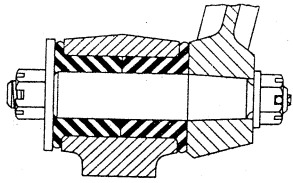
The vertical links are BS 310, B 20/10 malleable iron castings. Each carries the drive-shaft in a pair of Timken,
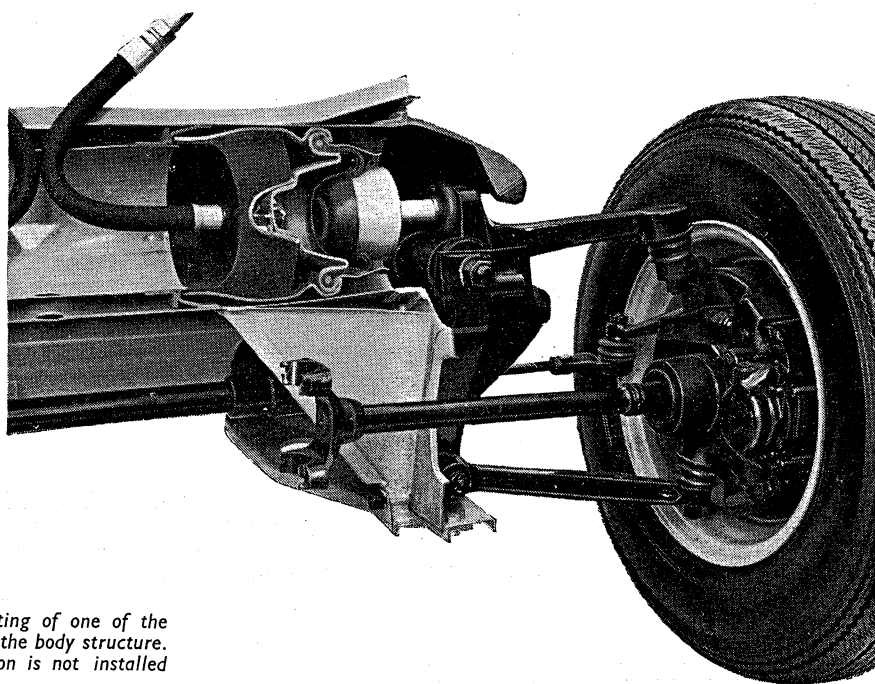
A partly sectioned assembly that shows the mounting of one of the Hydrolastic units in a transverse tubular member of the body structure. The radius rod for the lower arm of the suspension is not installed

The bearing for the pivot of the lower arm, above, for the front suspension, has rubber bushes, whereas that for the upper arm, below, which sustains vertical loads, comprises opposed taper roller bearings
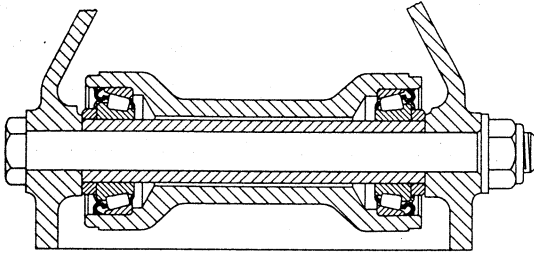
$1\frac{3}{8}$ in internal diameter, opposed taper roller bearings. The preload of these bearings is set by the selection of a spacing tube of suitable length, for assembly between their inner races. An En.15R forged steel hub is splined to the drive-shaft. It is pulled up against the inner race of the outer bearing by a 1 in diameter castellated nut on the threaded end of the shaft. A conical face machined at an included angle of 60 deg on an En.15R steel collar under this nut, seats on a similar face on the outer end of the hub, to centralize the hub on the splines.

Five $\frac{3}{8}$ in diameter setscrews retain the BS 1452, grade 17, cast iron brake disc on a spigot on the inner face of the hub. The diameter and thickness of this disc are $9\frac{9}{32}$ in and $\frac{1}{2}$ in respectively. Five $\frac{1}{2}$ in diameter En.18T steel studs in the hub carry the pressed steel wheel, which is retained by En.3A steel conical seating nuts.

Steering gear

Cam Gears Ltd. manufacture the rack-and-pinion steering gear, illustrations of which appear on page 6. This is bolted to a box-section member under the toeboard on the body, and its axis is $7\frac{5}{16}$ in behind the axis of the front wheels. The housing for the rack is a BS 980, ER W1 steel tube of $1\frac{1}{16}$ in outside diameter and 0·064 in wall thickness. One end of this tube is cast into the LM 4M aluminium alloy housing for the pinion; the other end is similarly attached to a casting of the same material, housing
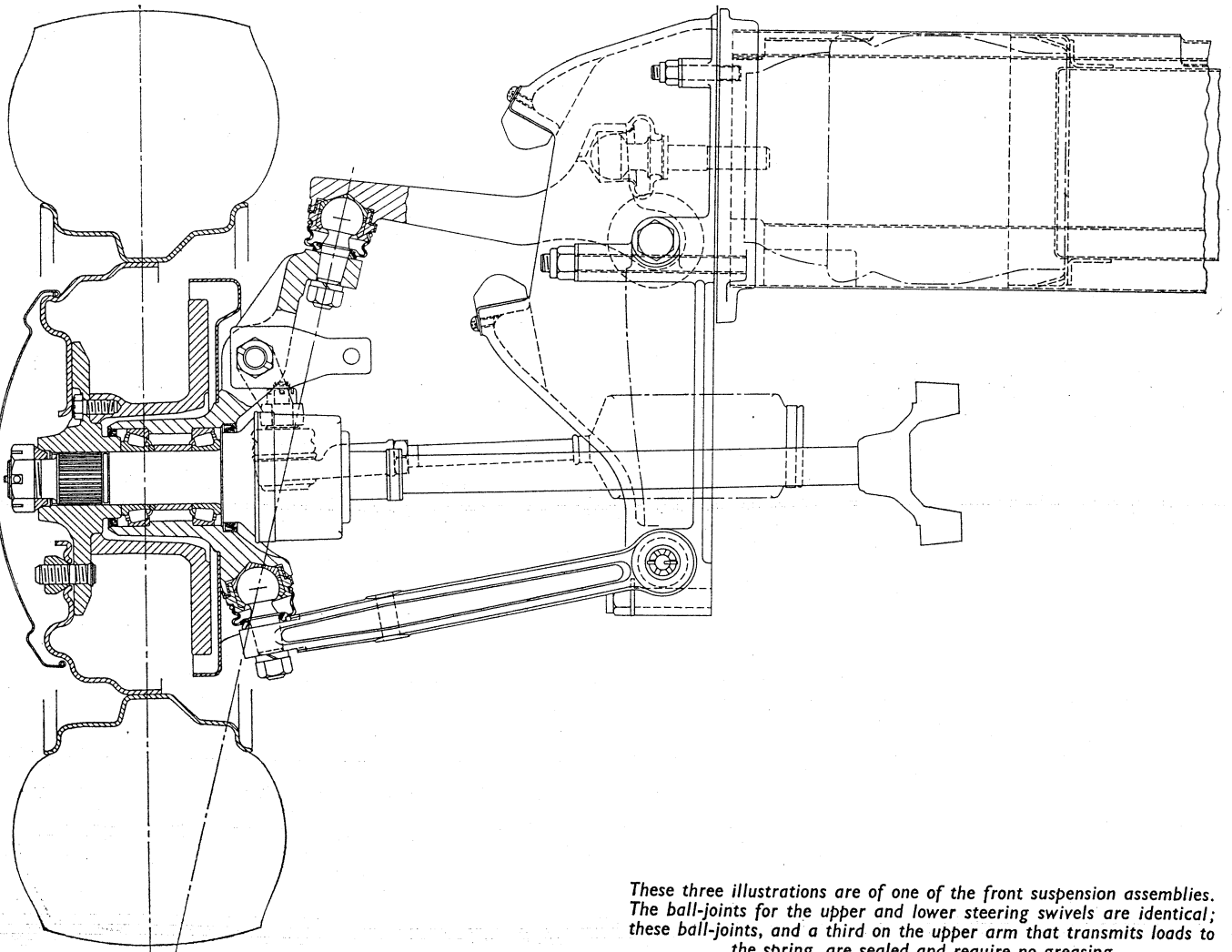
These three illustrations are of one of the front suspension assemblies. The ball-joints for the upper and lower steering swivels are identical; these ball-joints, and a third on the upper arm that transmits loads to the spring, are sealed and require no greasing

the felt bush for the rack. Two $\frac{3}{8}$ in diameter bolts in holes in a bracket integral with each casting attach the assembly to the toe-board; the holes in one of these brackets are elongated transversely to facilitate installation.

En.8C-Q steel has been chosen for the rack, which has a nominal diameter of 1 in. Twenty-five teeth, induction hardened to 54 Rockwell C scale, are machined on it. The diametral pitch of the five teeth on the pinion that engages with the rack teeth is 12. Each end of the rack is machined to receive the inner ball-joint for one of the tie-rods. A locking collar and an En.8CM-Q housing, having an integral induction hardened spherical seat for the ball-pin, are assembled on a 1 in diameter thread on each end of the rack.
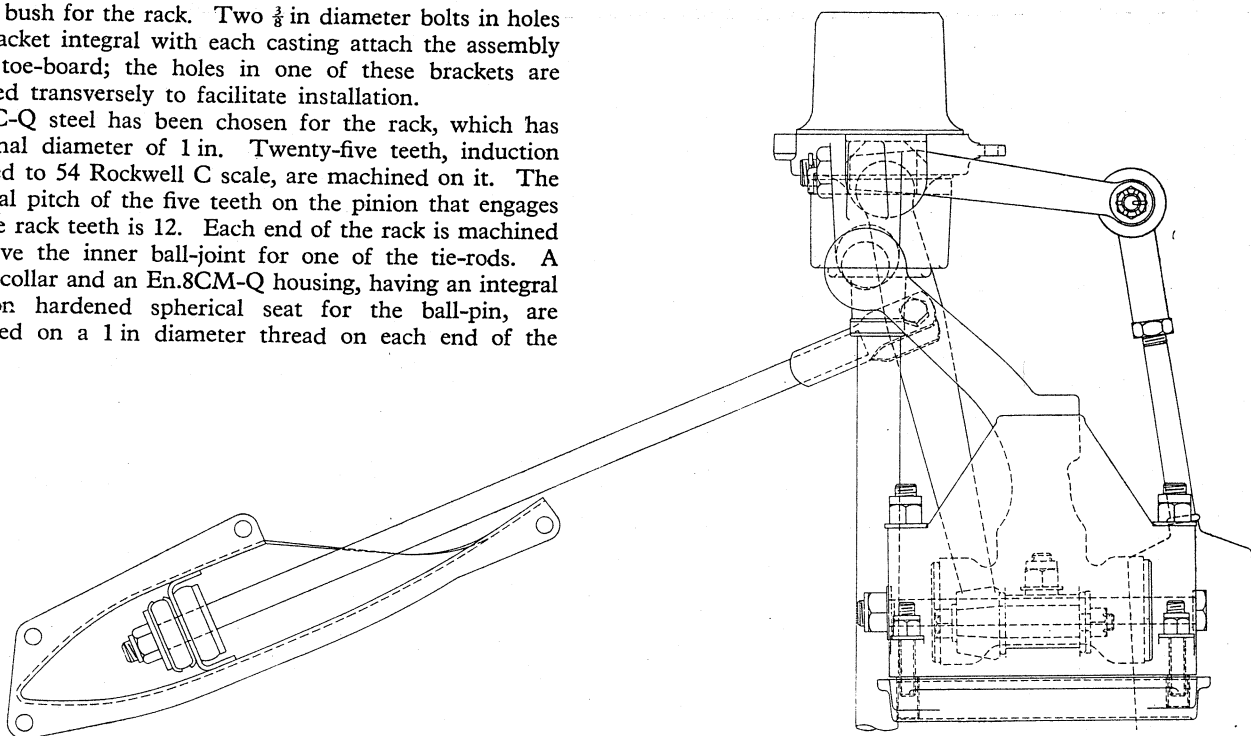
The En.32B inner socket of a ball-joint and its helical preload-spring are assembled into a counterbore in a coaxial hole in each end of the rack. Each ball-end has two spherical surfaces, one $1\frac{5}{16}$ in diameter, to seat in the outer socket, and the other $\frac{9}{16}$ in diameter to seat in the inner socket. The shank of the ball-pin is extended outwards to form the tie-rod; it is 7·66 in long and its outer end is threaded to receive the outer ball-joint assembly. The pre-load of each inner ball-joint is such that a torque of 32 to 52 lb-in is required to cause angular movement of the tie-rod.

Axial travel of the rack in both directions is limited to 3·28 in by a shoulder on each ball-joint housing, this shoulder forms a stop, coming into contact with one end of the rack housing. Full travel of the rack between locks is equivalent to 4·4 turns of the steering wheel.

Two Hoffman angular contact ball bearings carry the En.36A steel pinion. The outer race of the lower bearing is retained by a steel plate bolted to a machined face on the pinion-housing; the selection of shims between the plate and the housing preloads the bearings. A machined hole in the pinion-housing, perpendicular to the axis of the pinion, houses a $1\frac{1}{4}$ in diameter, sintered iron plunger which bears on the rack. Shims between a machined face on the
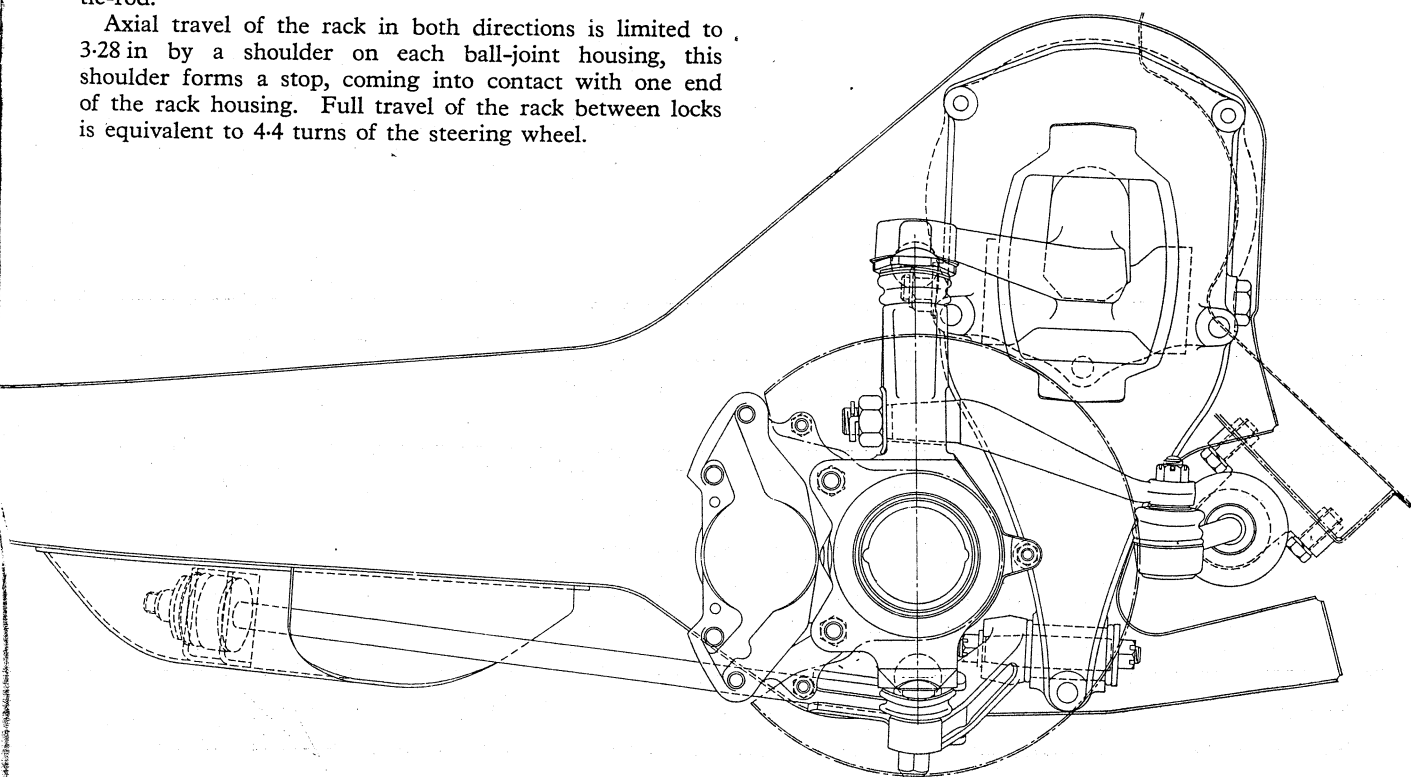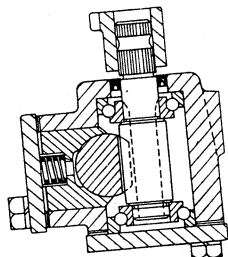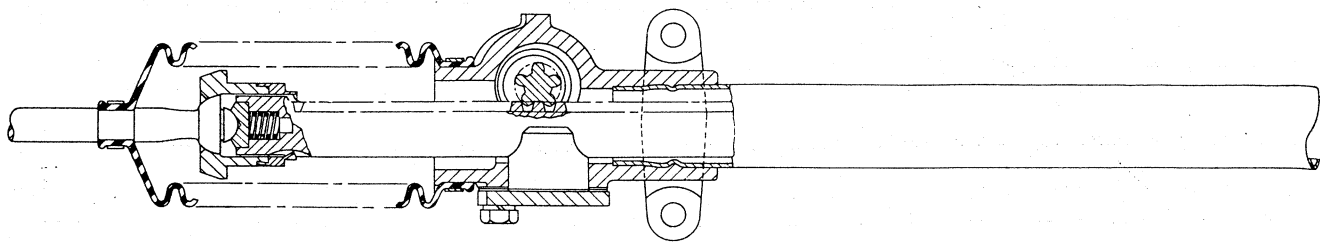
The rack-and-pinion steering gear is bolted rigidly to the body structure. Shown at section A-A is the method of retaining in its housing the felt bush that carries one end of the rack

housing and a steel cover bolted to it limit the axial travel of the plunger to 0·001-0·006 in. A low rate helical spring in a counterbore in the outer end of the plunger causes it to load the rack against the pinion. It is stipulated that the maximum torque on the pinion to initiate movement of the rack shall be 25 lb-in.

When the steering gear has been assembled, it contains $\frac{1}{3}$ pint of S.A.E. 140 hypoid gear oil. Convoluted rubber dust covers prevent oil leaking from the rack housing, and the entry of dirt. One end of each dust cover is clamped to the rack-housing, and the other end is clamped to the tie-rod; hence the inner ball joints of the tie-rods also are lubricated by this oil. Provided no leakage occurs, oil does not have to be added during service.

Between a flange on the upper end of the pinion and the steering shaft is a flexible bonded rubber coupling. This coupling accommodates any small misalignment between the axis of the pinion and that of the steering shaft.

Hydrolastic units

In the original design of the suspension system for the A.D.O.17 model, Hydrolastic suspension units of $6\frac{1}{2}$ in diameter were installed at both the front and the rear of the car. However, to match front-rear weight distribution of the car in the unladen condition, the leverage ratio of the linkage that actuates the rear springs was made numerically higher than that for the front springs. Later, because the 6 in diameter Hydrolastic unit was used in the smaller A.D.O.16 model, and was readily available, it was installed also in the rear suspension of the A.D.O.17. This made possible the use of a numerically lower leverage ratio for the linkage of the rear spring units, and, consequently, the loads on the bearings at the pivots of the trailing arms were reduced.

The $6\frac{1}{2}$ in diameter units are, in effect, enlarged versions of the 6 in diameter components, the design and development of which were described in the September 1962 issue of *Automobile Engineer*. Both types were designed and developed by Moulton Developments Ltd, in co-operation with the British Motor Corporation and the Dunlop Rubber Co. Ltd, who manufacture them.

Few difficulties were encountered during the development of the larger unit; this was aided by the large amount of data that had already been collected on the performance of the smaller unit, and by the fact that larger units present fewer problems than do smaller ones.

Bundy tubes of $\frac{7}{16}$ in bore connect together the hoses of the front and rear spring units. The bore of the pipes is the same as that for the A.D.O.16, but the orifice plates installed in this model have been omitted. In the bounce mode of the suspension, the frequency is 85 c/min; in the pitch mode it is 55 c/min. For bounce or roll motions of the car, the combined rate of front and rear units is 2 400 lb/in; the corresponding figure for the A.D.O.16 is 2 170 lb/in. One effect of using larger spring units at the front than at the rear was that the auxiliary torsion springs on the rear suspension could be omitted.

Before the systems are charged with fluid, an 80 per cent vacuum condition is induced by an exhauster pump through Schrader valves. Subsequently, the systems are charged through those valves with fluid at 230 lb/in²—the standard pressure for an unladen car.

Independent rear suspension

Each rear wheel is carried by a trailing arm; the axis of the pivot of each arm is parallel with the ground and perpendicular to the longitudinal axis of the car. The roll-centre of the suspension, of course, is at ground level. Between the axis of each pivot and that of the wheel the dimension is $11\frac{7}{8}$ in. In the static, unladen position of the suspension, the axis of each pivot is $11\frac{3}{16}$ in above the ground, or $\frac{1}{8}$ in above the axis of the wheel. The hydrolastic spring unit for each suspension assembly is disposed at 7 deg to the horizontal, ahead of the trailing arm, and is carried in a 10 s.w.g. steel pressing bolted to the floor pressing of the body structure, immediately below the ends of the rear seat pan. In these positions, the spring units do not encroach on space for the accommodation of passengers, nor on that in the boot. The space between the two trailing arms is utilized for mounting the fuel tank and the spare wheel.

The displacer in each spring unit is actuated by a push-rod and ball-joint in a lug projecting upwards, above the pivot, from the trailing arm. Hence the trailing arm is, in effect, a bell-crank lever, and its leverage ratio is 4·56 : 1.

An anti-roll bar connects together the two trailing arms.

A-A
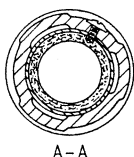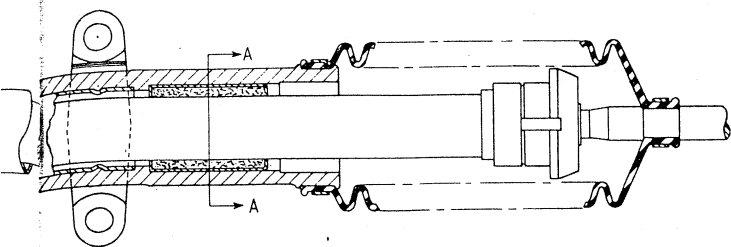
The diameter of this bar is $\frac{11}{16}$ in, and its effective length is 36 in. The material is En.45A steel, and each of its ends —of rectangular cross section—are attached by two set-screws to a lug on the inner face of each trailing arm. This anti-roll bar reduces the strong inherent understeer characteristics of the car.

Each trailing arm is a BS 310/B20/10 malleable iron casting, the nominal wall-thickness of which is $\frac{1}{4}$ in; when fully machined it weighs $16\frac{1}{2}$ lb. The En.24W or En.25W steel spindle that carries the hub of the road wheel has a parallel shank, which is a press fit in a hole in a boss at the rear end of the arm. Opposed taper roller bearings carry the BS 310 grade 3 malleable iron hub on the spindle. Both bearings are manufactured by British Timken Ltd; the dimensions of the inner bearing are $2\frac{21}{64} \times 1\frac{1}{4} \times \frac{3}{4}$ in, and those of the outer bearing are $1\frac{63}{64} \times 1\frac{1}{16} \times \frac{9}{16}$ in. A spigot on the periphery of the flange of the hub locates the BS 1452 grade 17 cast iron brake drum, which is retained by two $\frac{1}{4}$ in diameter screws. Two $\frac{3}{8}$ in diameter setscrews and one $\frac{3}{8}$ in diameter bolt in lugs on the boss of the trailing arm retain the backplate of the brake; this backplate is located radially by a spigot on the hub spindle.

Two British Timken $1\frac{63}{64} \times 1 \times \frac{5}{8}$ in opposed taper roller bearings carry the front end of each arm on a 1 in diameter steel spindle, machined from BS 980 cold drawn steel tube. This spindle is assembled on to a $\frac{5}{8}$ in diameter En.8R steel bolt, each end of which is carried in a U-shape steel bracket, disposed transversely beneath the pivot, and bolted at its extremities to the body structure. The dimension between the central plane of the two roller bearings is $4\frac{3}{16}$ in. These bearings are sealed and do not require periodic lubrication.

Housed in a hole in a lug projecting upwards from the trailing arm is the ball-joint that permits articulation between the arm and the pushrod for the spring unit. This ball-joint, manufactured by Engineering Productions (Clevedon) Ltd, is of the same design as that incorporated in the upper arms of the front suspension assemblies. The $1\frac{3}{16}$ in diameter, induction hardened spherical end of the ball-pin seats in a moulded nylon socket, in the hole in the trailing arm. The bearing surface of the socket is impregnated with molybdenum disulphide. During assembly the joint is packed with grease and requires no subsequent lubrication; it is sealed by a neoprene dust cover. The shank

of the ball-pin is in a coaxial hole in one end of the push-rod; thrust is applied through a shoulder on the shank to the end of the push-rod. A helical compression spring is assembled between the inner end of the hole in the push-rod and the end of the shank. This spring maintains the ball in contact with its socket when the Hydrolastic unit is evacuated, before it is charged. A spring in the push-rod of each spring unit for the front suspension performs the same purpose. An arm that projects upwards from the housing of the ball-joint compresses rubber stops that limit the bump and rebound movement of the suspension; these stops are attached to the body structure. The total travel to full bump at the wheel is $5\frac{7}{16}$ in, and on rebound it is $1\frac{7}{16}$ in; in the unladen condition, the corresponding movements before the arm makes contact with the stops are $3\frac{13}{16}$ in and $\frac{5}{8}$ in respectively.

Braking system

Girling Ltd. supply all the components of the braking system, which incorporates $9\frac{9}{32}$ in diameter disc brakes for the front wheels and $9 \times 1\frac{3}{4}$ in drum brakes for the rear wheels. A vacuum servo unit and a skid control valve, which was described in the October 1964 issue of *Automobile Engineer*, are installed in the hydraulic actuation system.

Each of the $\frac{1}{2}$ in thick discs for the front brakes is machined from a grey-iron casting. The cast iron housing of the caliper assembly is forward of the axis of the road wheel; it contains two rectangular shape pads of Ferodo DA.6 friction material, actuated by opposed $2\frac{1}{8}$ in diameter

In design and construction, the Moulton Hydrolastic spring-and-damper unit installed in the front suspension of the A.D.O.17 is a larger version of that incorporated in the A.D.O.16 model; this last-mentioned, smaller unit is installed also in the rear suspension of the A.D.O.17. The rubber spring and the displacer are shown in the free condition and, by chain-dot lines, in the static laden condition
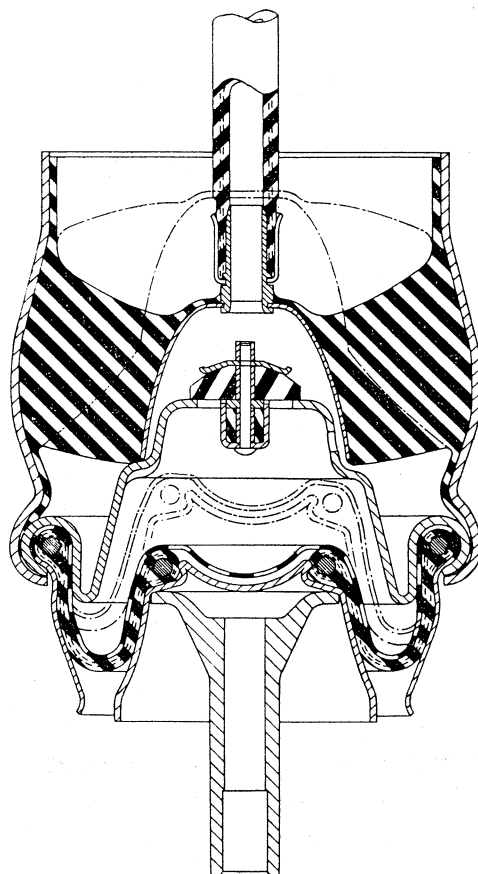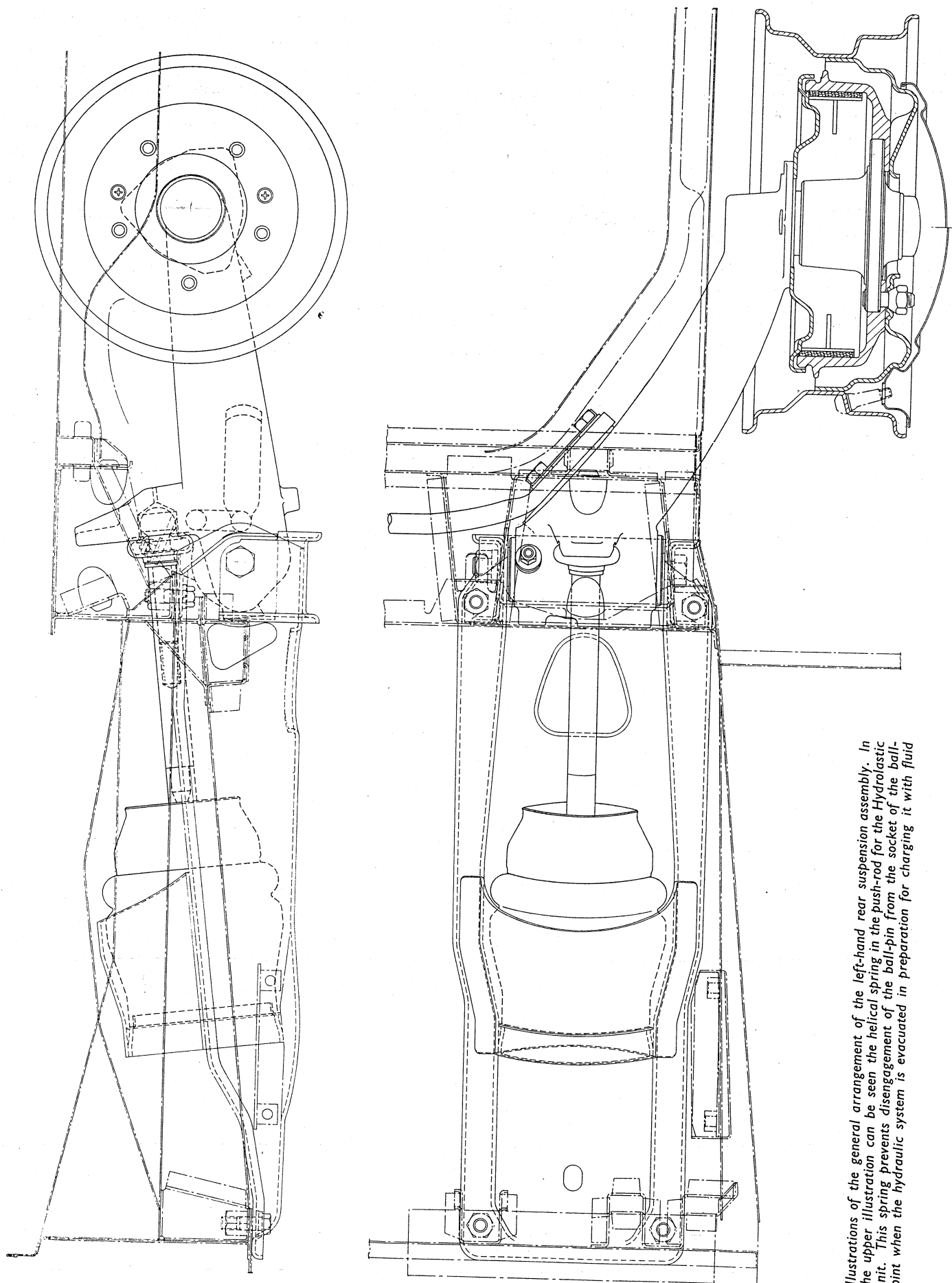
Illustrations of the general arrangement of the left-hand rear suspension assembly. In the upper illustration can be seen the helical spring in the push-rod for the Hydrolastic unit. This spring prevents disengagement of the ball-pin from the socket of the ball-joint when the hydraulic system is evacuated in preparation for charging it with fluid
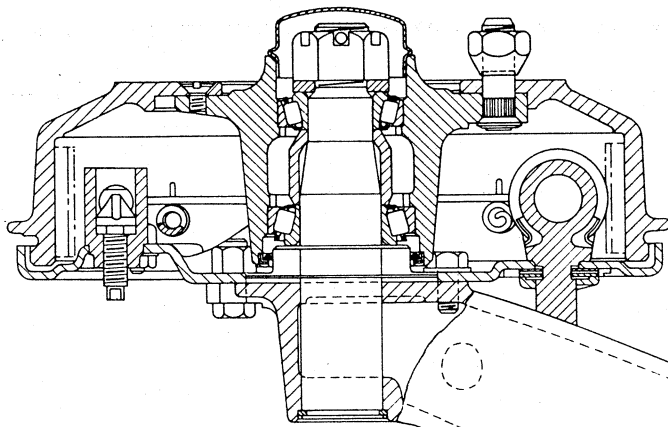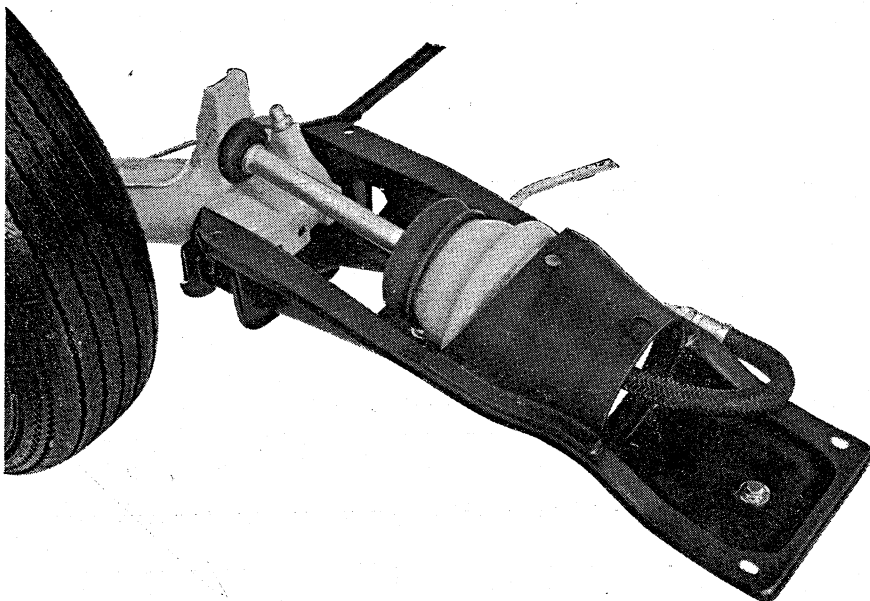
pistons. The friction area of each pad is 5·25 in², its effective radius of action is 3·58 in, and the total swept area of each disc is 91·5 in²; a pressed steel shield protects the inner face of each disc.

In the rear brakes, the shoes are of leading-and-trailing type, and each pair is actuated by a $\frac{3}{4}$ in diameter cylinder. A manually operated adjuster for the drum-to-shoe clearance is incorporated in the anchorage for the shoes. The handbrake is connected by cables to a lever pivoted on each wheel-cylinder; these levers actuate the trailing shoe of each brake, and the reaction forces at the pivots on the floating wheel-cylinders actuate the leading shoes.

The leverage ratio of the pendant brake pedal is 4 : 1 and the diameter of the master cylinder is $\frac{3}{4}$ in. A type AHY, 6·89 in diameter vacuum-servo unit is installed in the hydraulic system, between the master cylinder and the brakes; its input-output ratio is 2·78 : 1, up to a maximum effort on the pedal of 61 lb. When the car carries four passengers, and 100 lb of luggage, an effort of 61 lb on the brake pedal produces a deceleration of 0·87 g, from 30 m.p.h. For a maximum retardation of 0·98 g, an effort of 90 lb on the pedal is required.

At a deceleration of 0·87 g, the braking effort is divided in the proportion of 81·6 per cent on the front brakes, and 18·4 per cent on the rear brakes; the weight distribution, unladen, is 63 per cent front, and 37 per cent rear. The proportioning of braking effort, of course, is modified by the action of the Girling skid control valve, installed in the hydraulic system for the rear brakes. Its action is progressive with the degree of deceleration above 0·3 g; below this deceleration, the front-rear braking ratio is 68·8 per cent front, and 31·2 per cent rear. Above it, the increase in the fluid-pressure in the pipes for the rear brakes is regulated by a pressure transducer. The ratio of the increase in pressure, front to rear, is 3·65 : 1.   *To be concluded*
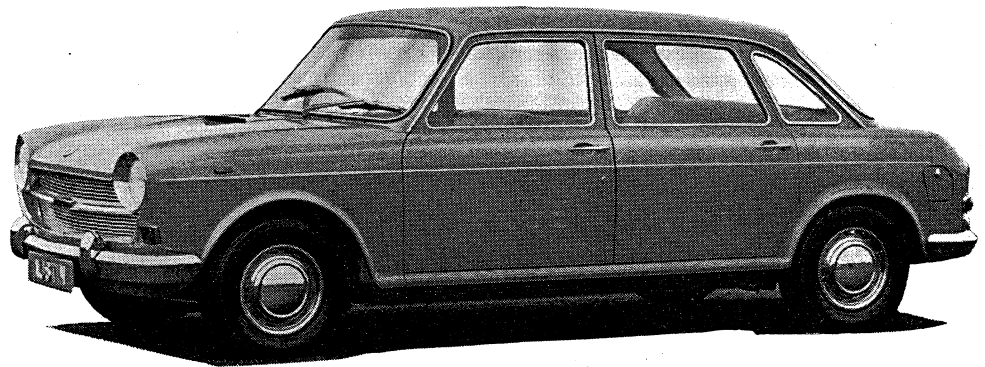
AUSTIN 1800 part three

Unitary body-chassis structure
## Electrical equipment and components

A repeater lamp for the direction indicators is installed, at a level above the wheel arch, in each front wing. In spite of the very small rear overhang of the body, and the unusual shape of the tail, the boot is large, and has a volume of 17 ft³

BECAUSE OF ITS very high torsional stiffness, the chassis-body structure of the Austin 1800 is of particular interest. The figure quoted by the manufacturers is 13 300 lb-ft/deg, measured between the planes of the front and rear wheels. This is approximately three times higher than the average for structures of comparable size. The beam stiffness can be assessed from the following test: a body was supported in the planes of the front and rear wheels, and a vertical load of 1 000 lb was applied mid-way between them. In the plane in which the load was applied the maximum deflection was 0·019 in. The weight of the structure, in the white, is 765 lb.

Neither the power unit and front suspension, nor the rear suspension is mounted on a sub-frame—this of course is in contradistinction to the arrangement on the A.D.O. 15 and A.D.O. 16 models. It has been reasoned that, with this larger vehicle, insufficient flexibility could have been allowed at the mountings to insulate effectively the body from road-excited noise. Also, because a high degree of rigidity would have been required in the attachment of the
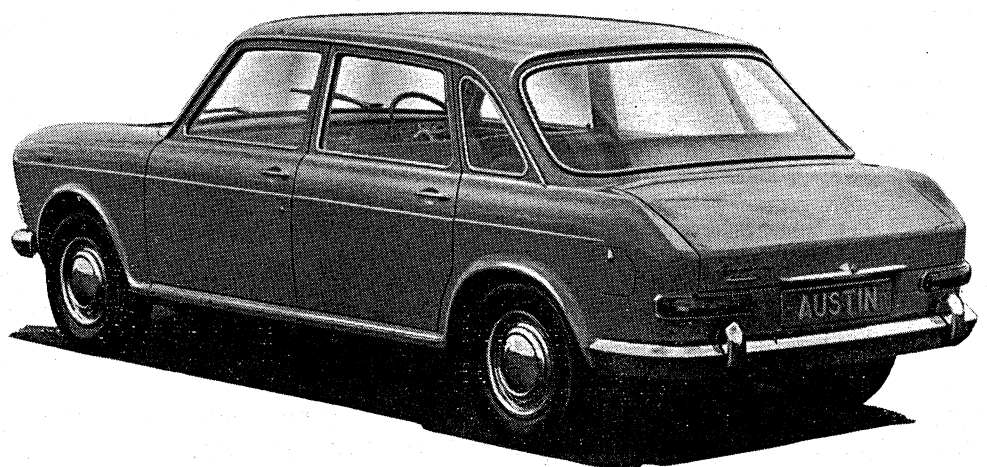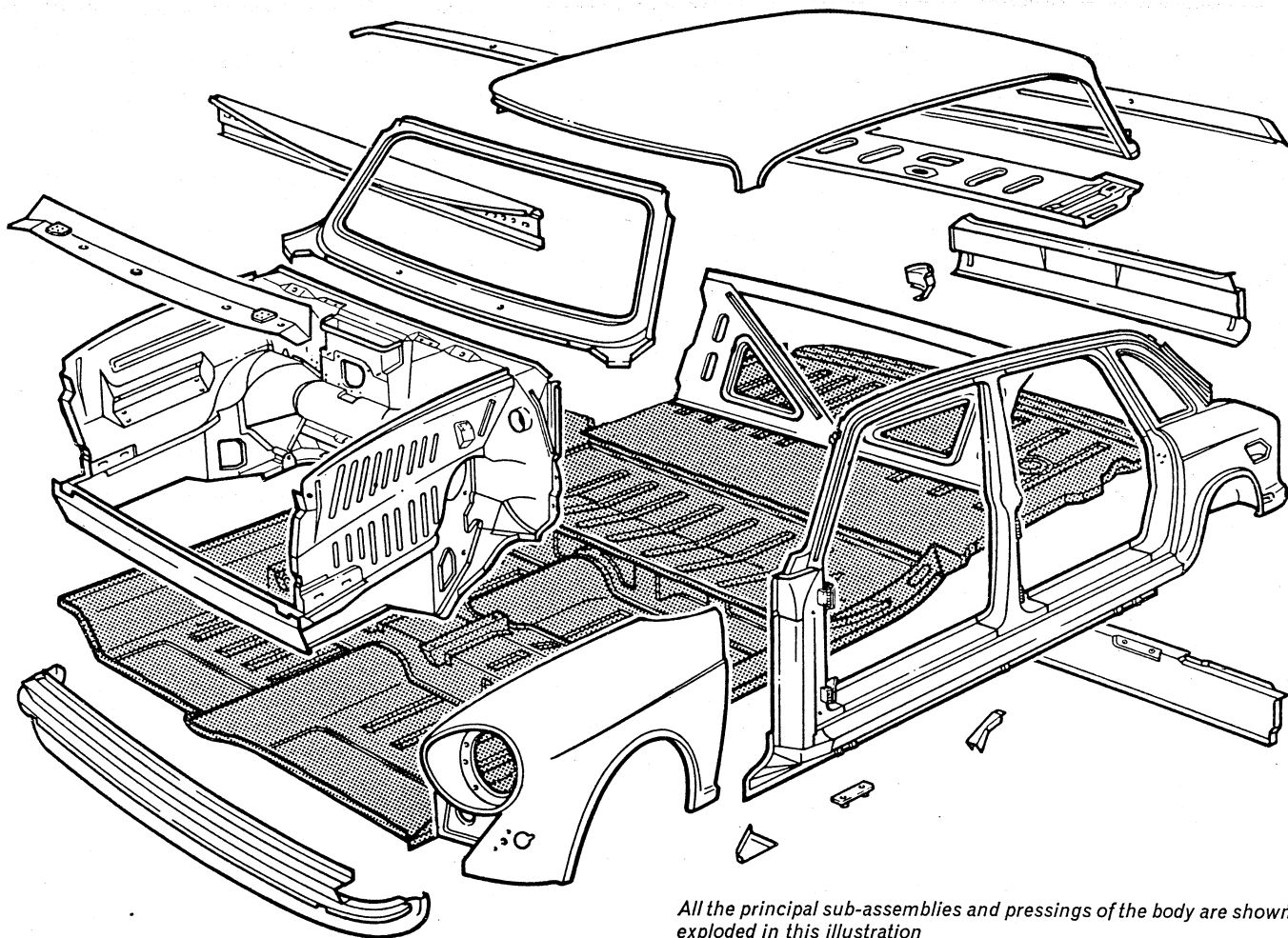
*All the principal sub-assemblies and pressings of the body are shown exploded in this illustration*

A bulge in each end of the rear seat pan provides clearance for the spring-damper units mounted underneath


main structure to the sub-frames, the result would have been uneconomical use of material.

In the design of the A.D.O. 17, a different approach has been made. First, as mentioned in part I of this article, the mountings for the power unit allow an unusual amount of flexibility in the vertical plane. Secondly, the use of large masses of material in the castings on which the inner ends of the arms of the front suspension assemblies are pivoted and in the substantial malleable iron trailing arms of the rear suspension helps to absorb road-noise. Additionally, sound deadening sheets are applied to the whole of the inner surface of the floor, the dash and the roof of the body; these measures will be described in a later part of this article.

An important factor in the design of the A.D.O. 17 is that loads from the suspension assemblies are applied directly to the front and rear ends of the main part of the structure that carries the passengers. In comparison with a design in which these loads are applied towards the extreme ends of the structure, the deflections produced by the loads, of course, are smaller. A transverse tubular member integrated with the dash is loaded axially by the front suspension; thus, of the lateral loads from the suspension, only the asymmetrical ones are transferred from this tube to the dash structure. On the other hand the pivots of the trailing arms of the rear suspension are mounted conventionally on a transverse member behind the cushion of the rear seat.

All pressings and sub-assemblies for the body structure
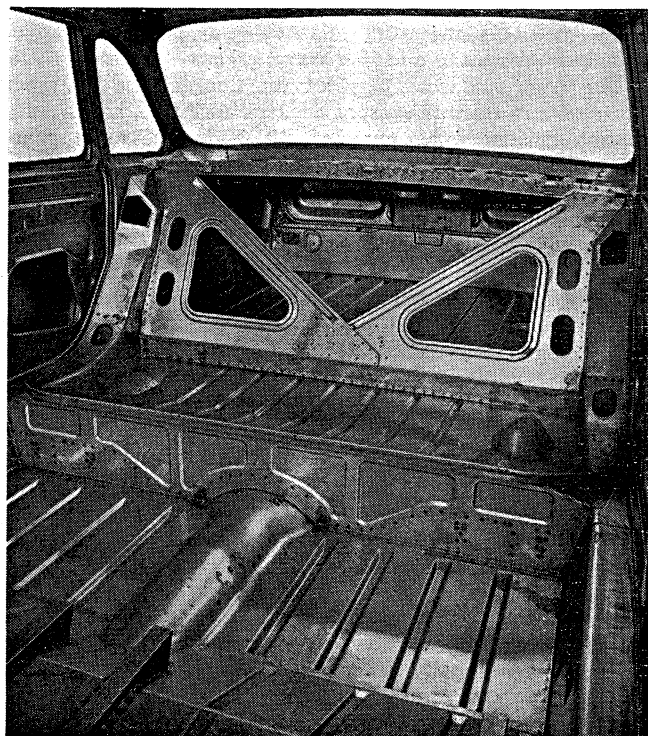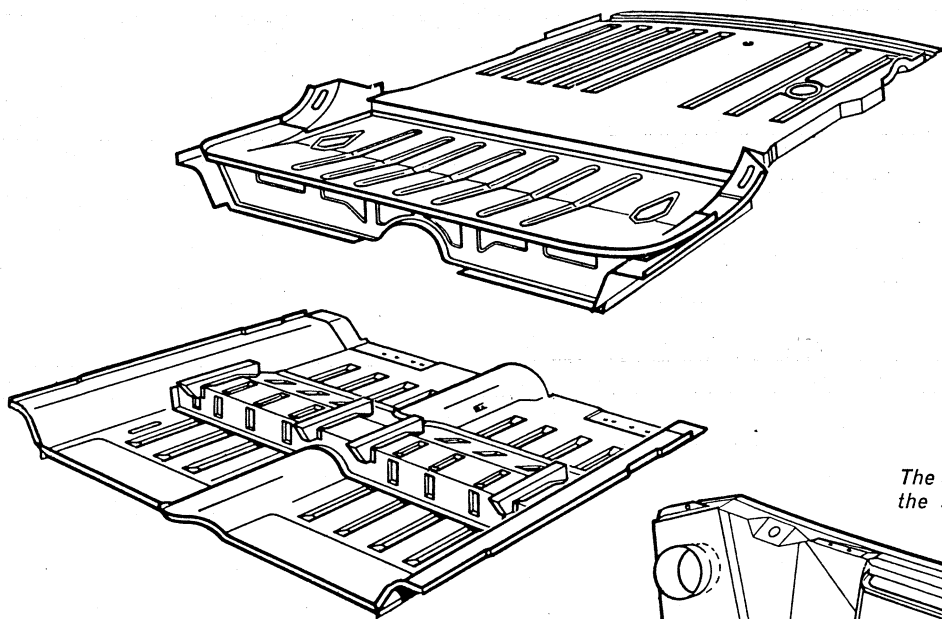
These two parts of the floor are welded together on the assembly line
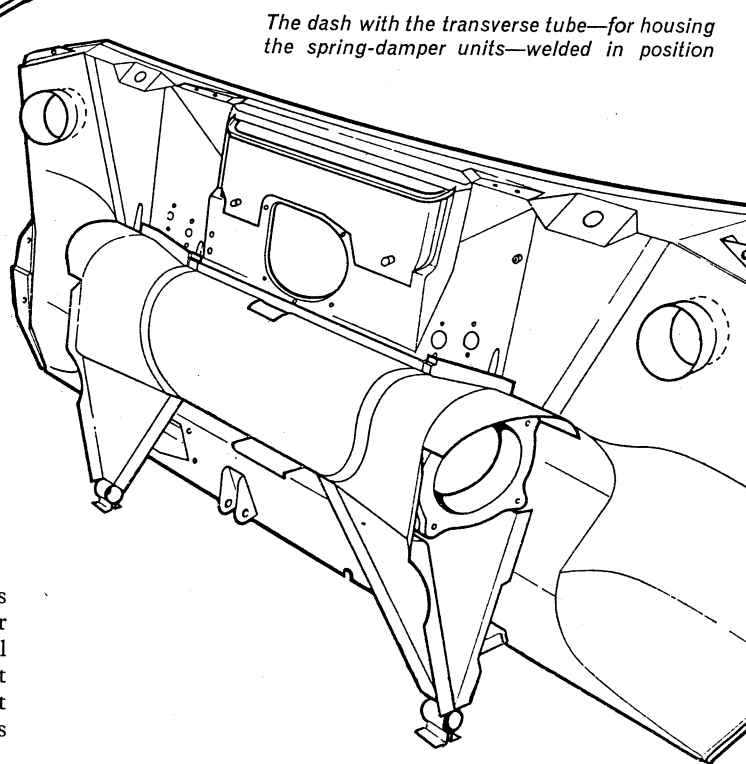
The dash with the transverse tube—for housing the spring-damper units—welded in position

are manufactured by Fisher and Ludlow Ltd, but the bodies are assembled at the Longbridge works of the Austin Motor Co. Ltd. An accompanying illustration shows the principal units of the structure. These are: floor, seat pan and boot floor assembly, dash assembly—including side and front frame members and wing valances—side panel assemblies —including tonneau and rear wheel arches—and roof.

The front and rear parts of the floor of the body are separate sub-assemblies spot welded together. As a precaution against water entering the body, the main panel of the front part of the floor is a single 20 s.w.g. (0·036 in) pressing, the outer edges of which form the inner panels of the body sills. The outer component of each sill is a 20 s.w.g. (0·036 in) pressing formed along the lower edge of each side-panel assembly; an 18 s.w.g. (0·049 in) flat plate is welded vertically between the inner and outer members of each sill, to contribute to the beam stiffness of the structure. A cross section of a sill, the maximum depth of which is 6 in, excluding the flanges, is shown in an accompanying illustration. These sills extend from the dash to the rear edge of the rear seat pan on the other unit of the floor. The pressings that comprise the rear unit of the floor assembly, are an 18 s.w.g. (0.049 in) heel board, the rear seat pan, a floor pressing beneath the seat pan, and a transverse, channel section pressing at the front of the floor of the boot. These three last mentioned pressings are 20 s.w.g. (0·036 in) thick. A heavily swaged 20 s.w.g. (0·036 in) pressing forms the floor of the boot; it is supported in a central vertical plane by a 2 in deep channel section pressing secured longitudinally beneath it.

The principal member of the dash assembly is a 20 s.w.g. (0·036 in) pressing in which a semi-circular concavity is

In the lower illustration are transverse sections of a body sill and a cantrail
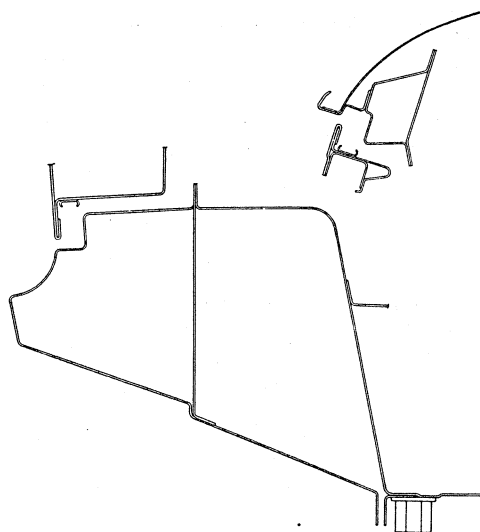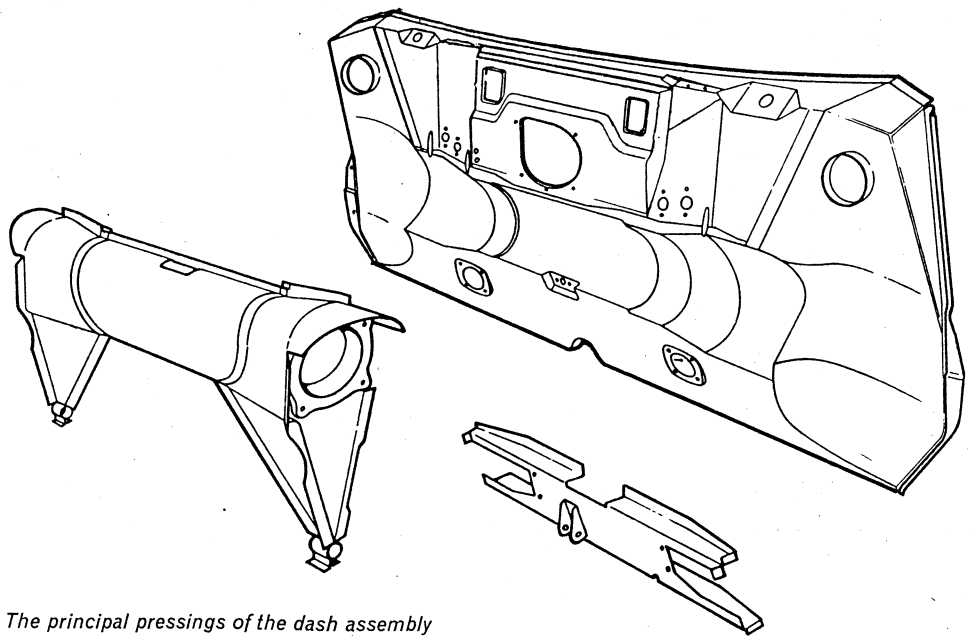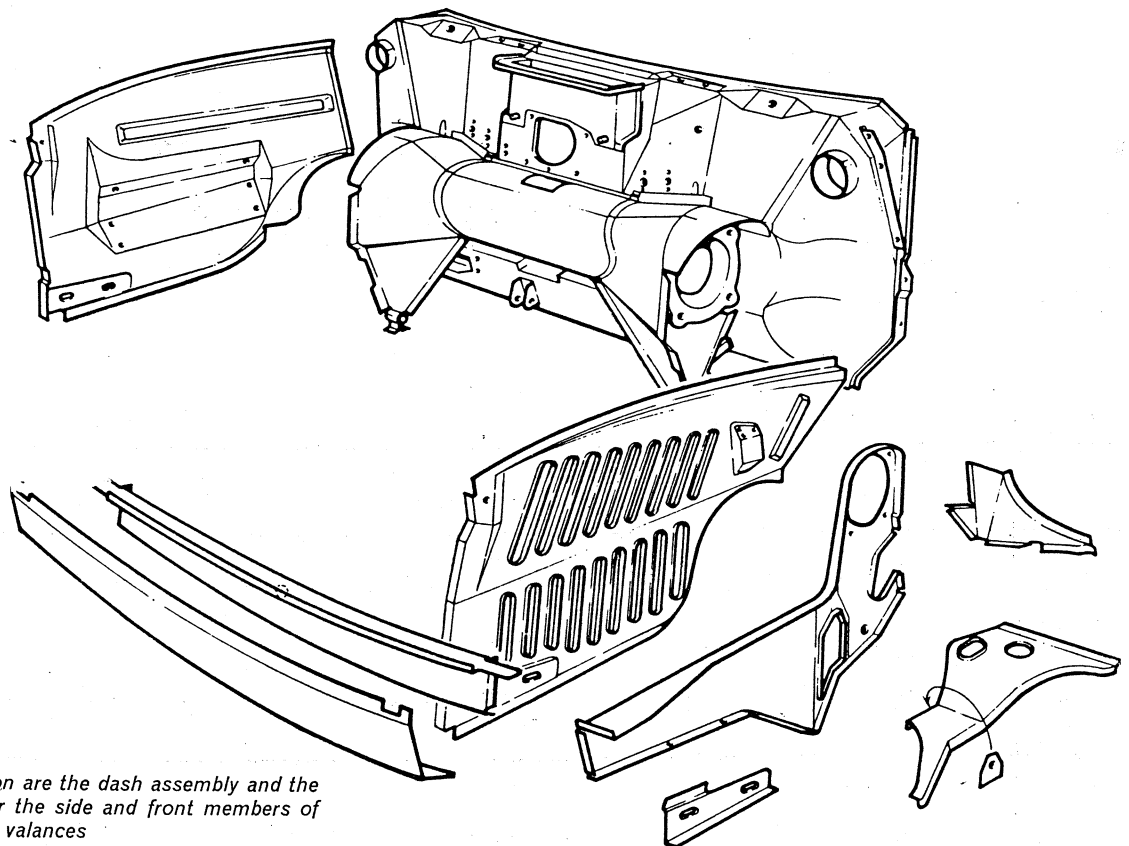
The principal pressings of the dash assembly

formed transversely. The transverse tubular assembly, and its associated pressings, to carry the opposed Hydrolastic spring-damper units, seats in this concavity, and is welded to the dash. Two coaxial tubes comprise the housing for the spring-damper units; the outer, $6\frac{5}{8}$ in inside diameter tube is $\frac{1}{16}$ in thick, and the inner $5\frac{13}{16}$ in outside diameter tube is $\frac{1}{16}$ in thick. Each end of the inner tube is attached to the outer tube by means of two pressings—all four components being welded together. The pressings form also an abutment for the inner ends of the spring units. Welded to each end of the outer tube is a forged En.4 steel flange, to which the two castings that carry the inner pivots of the

suspension assemblies are bolted. After the flange has been welded to the tube, its outer face is machined to a specific dimension from the abutment on the inner tube. Since variations in this dimension affect the standing height of the car, the tolerance is 0·010 in. Equal axial loads applied by each suspension assembly to the spring units are sustained by the inner tube alone; when the load applied to one spring is greater than that applied to the other, the resultant unbalanced load is sustained by the outer tube.

Pressings of 18 s.w.g. (0·049 in) thickness, welded to the ends of the outer tube, form triangular shape downward projections, braced by gussets; at the apex of each projection

Shown in this illustration are the dash assembly and the associated pressings for the side and front members of the frame, and the wing valances
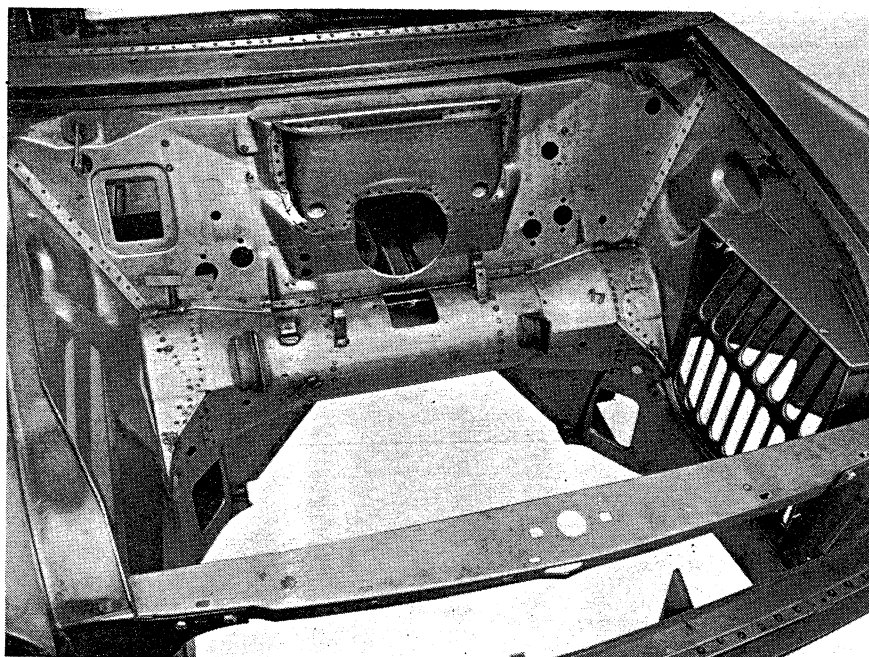
The dash assembly incorporates the transverse tubular member that houses the spring-damper units for the front suspension. Air from the coolant radiator leaves the engine compartment through apertures in the left-hand wing valance

is welded a fabricated eye for a transverse $\frac{1}{2}$ in diameter bolt. This bolt secures to the pressings the base of one of the aluminium castings that carry the inner pivots of the suspension; another longitudinal pressing braces each apex to the dash. The vertical arm of a 14 s.w.g. (0·080 in) L-shape pressing—named the frame side member—is bolted between the flange on the outer transverse tube, and the casting. The end of the horizontal arm of the frame side member is welded to one end of a transverse pressing—the frame front member—in the nose of the structure; the upper edge of the arm is welded to the lower edge of the wing valance. A projection at the rear of the pressing of each frame side member is welded to the dash.

Beneath the lower part of the dash, which forms the toe-board, a channel section 18 s.w.g. (0·049 in) pressing is welded to form a substantial base to which the rack-and-pinion steering gear is bolted. A claim made by the manufacturers is that the large diameter transverse tubes in the dash, and the transverse disposition of the power unit ahead of it, would help to absorb the energy of a collision, and prevent penetration of the front of the passenger compartment.

The frame side-members—one of which is shown in this illustration—are bolted to a flange on each end of the transverse tube for the spring-damper units, and welded to the wing valance and dash
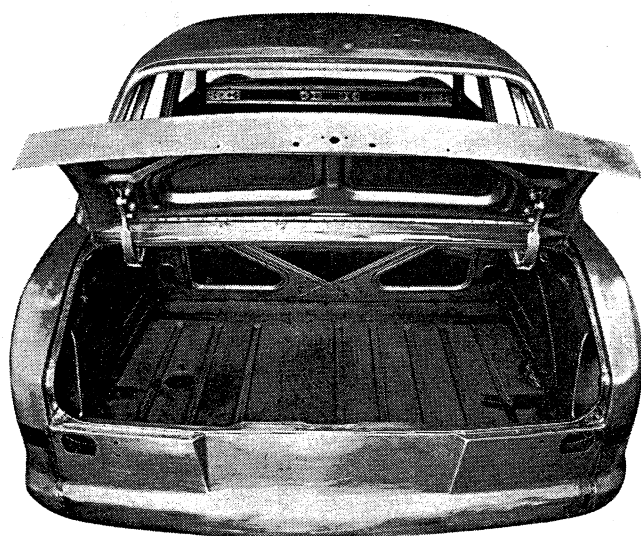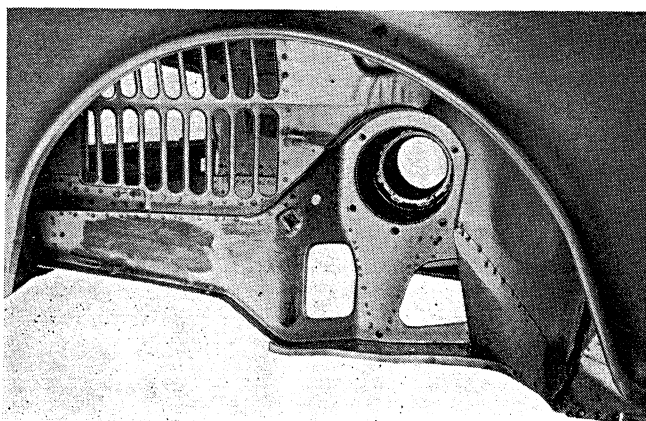

A deep transverse pressing in the tail of the body, braces the wing pressings to the rear edge of the floor of the boot

The remainder of the body structure is of conventional design and construction. However, the method of manufacture of the bonnet lid and boot lid is of interest. Flanges around the edges of the inner and outer panel of each lid are welded together, and a metal-to-metal adhesive forms a structural connection between the inner and outer member and also helps to damp out drumming.

Sound-deadening material is applied to the principal internal surfaces of the passenger compartment. A $\frac{3}{32}$ in thick perforated bitumastic sheet, coated with adhesive, is laid on the floor panel. Subsequently, during the painting process, when the body passes through the ovens, the adhesive becomes active, and the sheet is stuck to the floor. A moulded glass-fibre panel, $\frac{1}{2}$ in thick, is secured by adhesive to the inner surface of the dash, which it covers completely. Similarly, another $\frac{3}{8}$ in thick moulded glass-

48

Details of the rear face of the pressing for the dash, the front of the assembly of the floor, and part of the body sills, can be seen in this illustration. The facia is an integral, stressed member of the structure
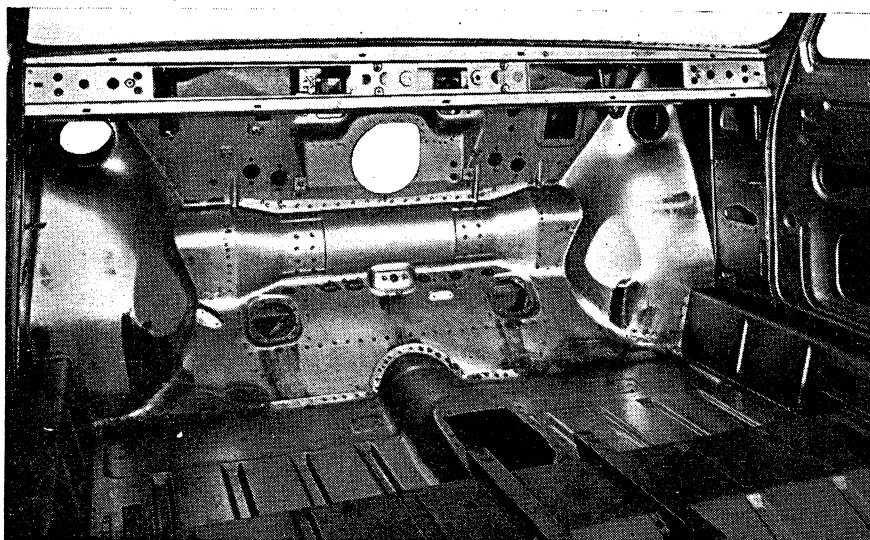
fibre panel, to the lower surface of which the pvc roof-lining is secured by an adhesive, is applied to the inside of the roof pressing.

In the construction of the four doors, the internal and external panels are spot welded together and clinched. The hinges are welded to the body structure and bolted to the doors. In the lower part of each door, a large

locker is formed partly by the cavity between the inner and outer panels, and partly by a pocket moulded in the trim panel. Wilmot Breeden zero-torque locks are used, and the plungers of the locking catches protrude upwards through the waist-rails. As regards windows, this is a six-light layout. The $\frac{3}{16}$ in thick toughened glasses in the doors are curved in the vertical plane. To ensure maximum

Details of the lower members of the dash sub-assembly, and of the front section of the floor, can be seen here. Inside the front wing on the left of the illustration is one of the two ducts for interior ventilation
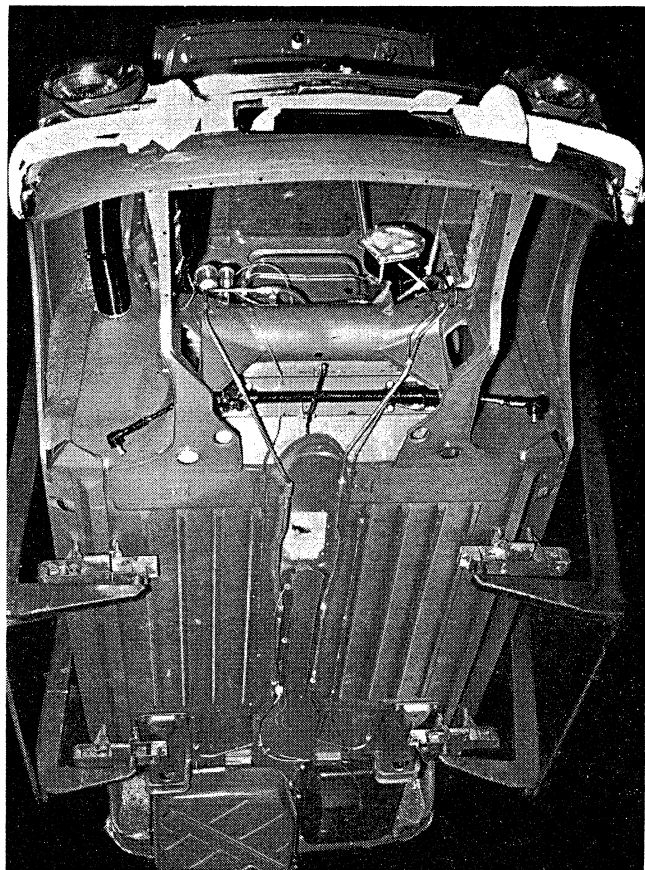

The two pressings that carry the spring-damper units for the rear suspension are bolted to the floor and the rear seat pan. Beneath the floor of the boot are the 10¾ gallon fuel tank and the spare whee
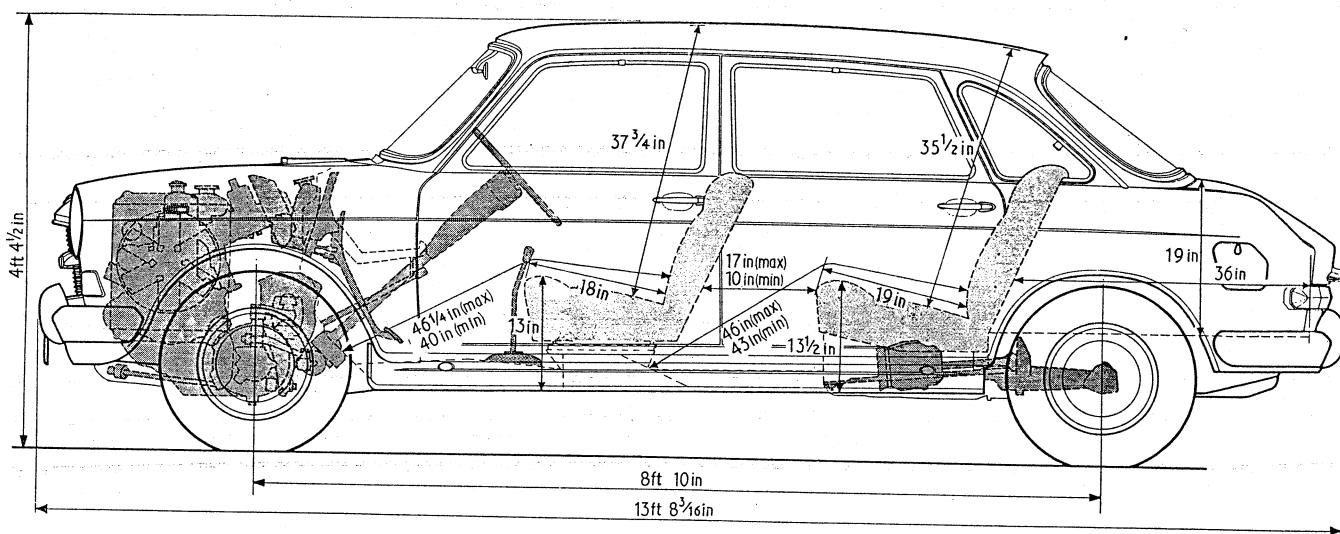
From these illustrations, it is evident that a very large proportion of the envelope of the Austin 1800 is allocated to passengers and luggage. Contributory factors are the transverse disposition of the power unit, front wheel drive, the large wheelbase relative to the overall length of the car, and the choice of a trailing arm rear suspension. Because of this last-mentioned feature, the fuel tank and the spare wheel can be accommodated beneath the floor of the boot, between the rear wheels. The overall height quoted is that for a laden car
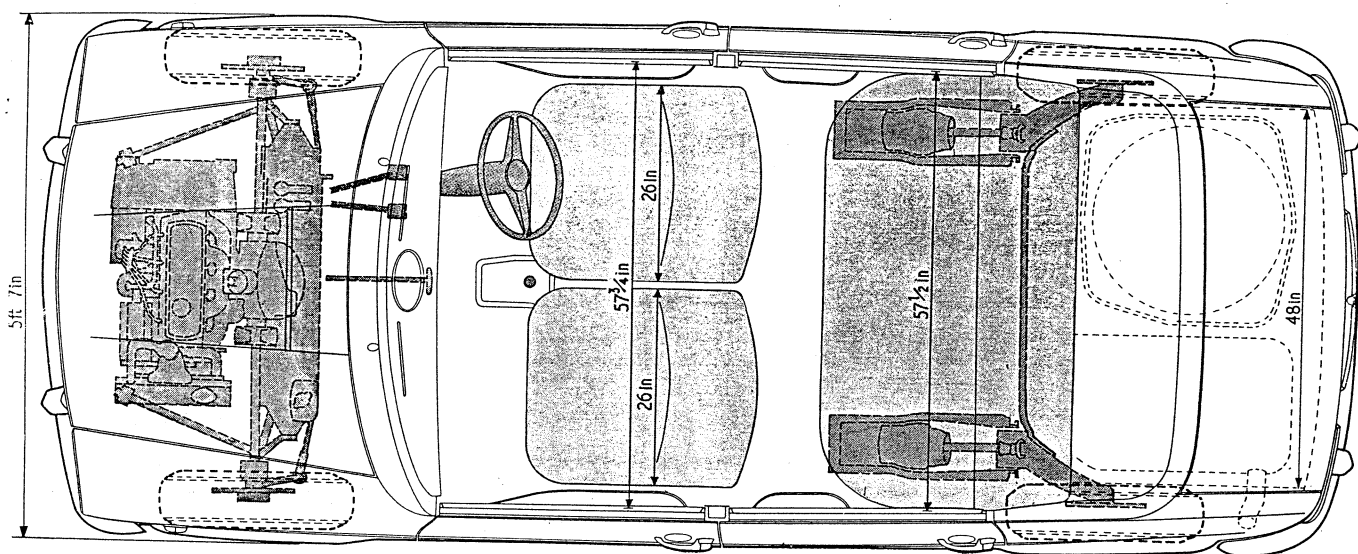
The assembled, unpainted body structure weighs 765 lb, and has a very high torsional stiffness of 13 300 lb-ft/deg
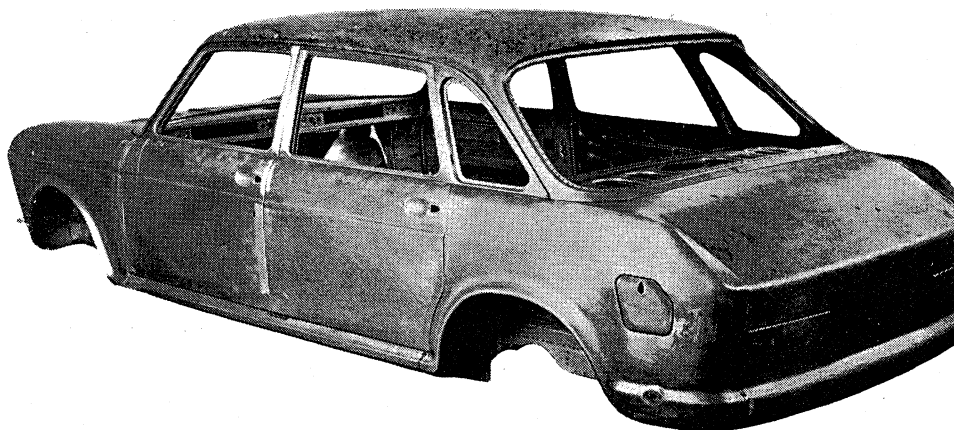
The deep body sills are formed partly by the pressings at the bottoms of the side frames, and partly by the edges of the floor

Unheated air enters the interior of the car through this assembly of louvres, the angles of which can be adjusted by the passengers

range of vision through the front lights there are no triangular vent panels: instead the whole of the glass on each side can be wound down, as also can those in the rear doors. But the glasses of the rear lights are pivoted at their leading edges and can be opened to promote ventilation.

A Smiths 5 kW interior heater unit is installed. It has been designed especially for this car and is mounted on the rear face of the dash. An illustration of the installation appears on page 52. When the engine is running, hot water is supplied continuously to the matrix of the heater, but the flow can be stopped by closing a cock in the feed pipe. A single air intake, from a region of high air-pressure above the rear end of the bonnet, directs air into the heater. A transversely mounted electric motor drives two fans, one on each end of the motor. With this arrangement, the ducts between the outlets from these fans and the slots for
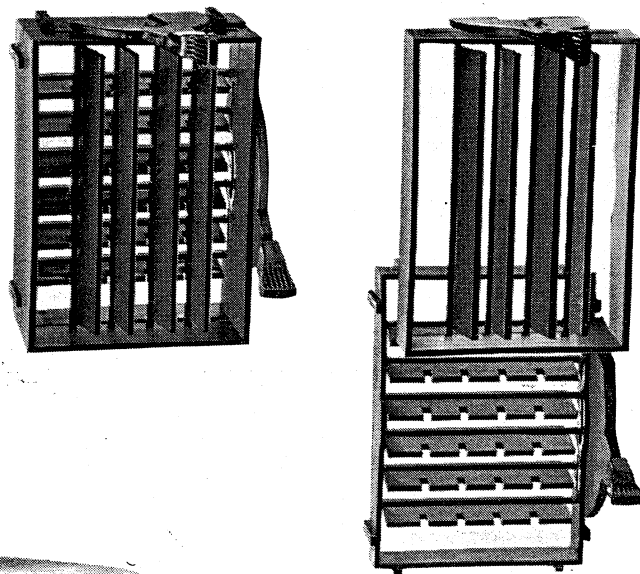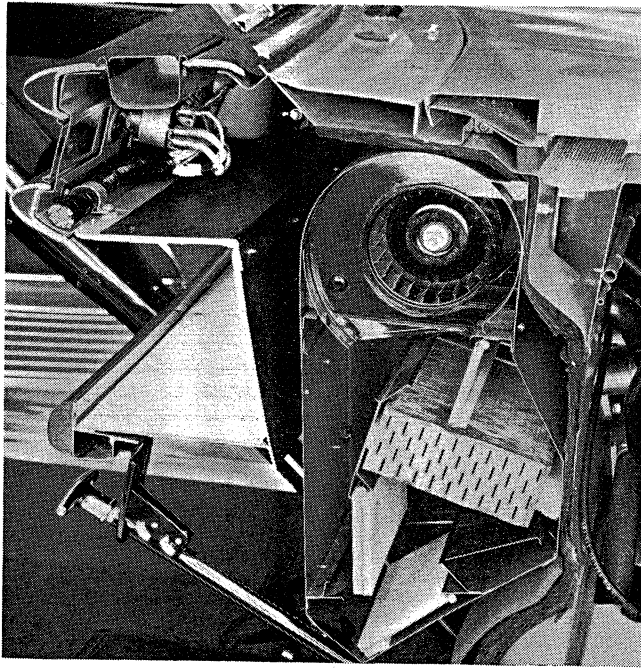
One of the two transversely disposed fans, the matrix and the flap-valves, can be seen in this sectioned exhibit of the interior heater

directing the air on to the windscreen are short and promote rapid demisting or defrosting.

A large volume of unheated air can be admitted to the interior of the car through a ventilation system entirely independent of that of the heater. From each end of the radiator grille, a 4 in diameter flexible hose connects an inlet duct to an outlet beneath the end of the facia. The two outlets each comprise two sets of louvres in rectangular mouldings of Shell Polypropylene; these mouldings* are made by Wall and Leigh (Thermoplastics) Ltd. The assembly is illustrated on page 51. Two rectangular shape frames are assembled one in front of the other. The front frame contains six polypropylene louvres disposed horizontally; each has an integral hinge and, to direct the flow of air in an upward or downward direction, the angle of the louvres can be varied by a lever in a quadrant at one side of the frame. Four similar vertical louvres in the rear frame also have integral hinges, and their angle can be varied by a lever in a quadrant above the frame, to direct air towards, or away from, the occupants of the front seats.

The body is phosphated and primed in a seven-stage roto-dip plant. This ensures that all surfaces, including those in the interiors of box-section members, are treated. Synthetic enamels are used in the painting process, which follows normal practice.

Electrical equipment

Joseph Lucas Ltd. supplies the principal components of the 12 V electrical system. These include a type C40 dynamo, which has an output of 22 A and is driven at 1·7 engine speed, a type RB340 voltage control unit, and a D11/13, 11-plate battery, the capacity of which is 50 Ah at a 20 hour rate. To special order, a D13, 13-plate battery, of 57 Ah capacity, is available; a positive earth is employed. For the type M418 starter motor, the lock-torque is 16·5 lb-ft at 450 A and, at 1 000 r.p.m, the torque is 7·7 lb-ft at 260 A.

Two features of particular interest in the electrical equipment are the two-level system for changing the intensity of illumination of the brake lamps and the rear direction

★ *Circle reply card* **1**

indicator lamps, and the grouped connectors for a three-piece wiring harness. The Lucas two-level system is not new, of course. When the parking lamp circuits are energized, resistors are brought into use in the circuits for the stop lamps, and the rear direction indicator lamps. Hence, the intensity of illumination of these lamps in daytime is high, whereas, at night it is reduced to prevent dazzle to other drivers. A further safety precaution is the installation of a repeater direction indicator lamp on each front wing, above the wheel arch.

For the wiring system of the A.D.O. 17 model, the British Motor Corporation stipulated the following requirements: the harness to be in two or more sections, and no printed circuits to be employed; provision for fused circuits and additional circuits; to be readily adaptable for special models and special purpose vehicles; to accommodate the likely increase in the installation of electrical components during the next 10 years.

Working in close co-operation with the British Motor Corporation and with Ripaults Ltd, who supply the wiring harnesses,* Aircraft-Marine Products (G.B.) Ltd., the manufacturers of the grouped connectors°, evolved the system. The harness is divided into three parts: the first is in the engine compartment, the second contains wiring for the switches and instruments on the facia, and the third serves the components in the rear of the car. These three parts are joined by plug-in, multiple connectors. An obvious advantage of this arrangement is that all the electrical connections to the components on the engine can be made on the pre-mounting assembly line for engines. Subsequently, when the engine has been installed in the car, the harness is plugged in, at a multiple connector in the dash, to the remainder of the system. Also, in service, it is clearly an advantage to be able to isolate one part of the system from another, simply by uncoupling the connectors.

The main multiple connector—that between the engine-harness and the facia-harness—is shown in an accompanying illustration. It is bolted to the edges of a rectangular

★ *Circle reply card* **2**   O *Circle reply card* **3**

These two multiple connectors—normally hidden behind the facia—join one of the three wiring harnesses to switches in the steering column assembly
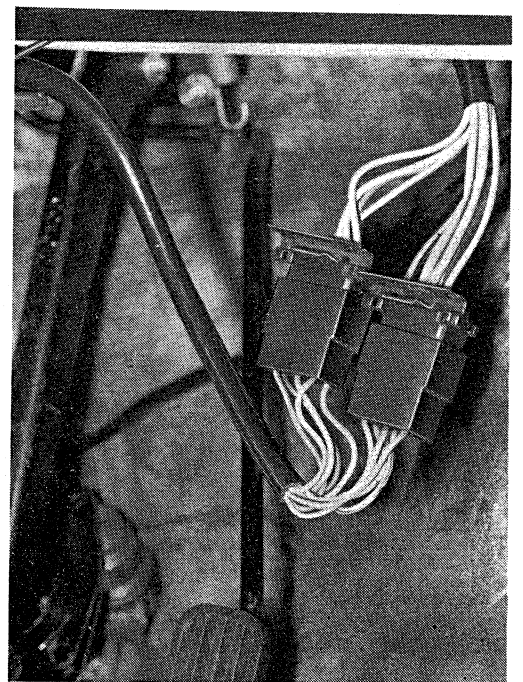
opening in the dash, and can be reached both from the engine and the passenger compartments. The base is a Cycolac plastics moulding and, in the lower part of the front face, five sockets are formed. A group of blades protrudes from the base of each socket, and a thin neoprene pad seals the gaps between the blades and the moulding. All the connectors in the plugs that engage with the blades in the socket are crimped to the cables; they are free to float in the plugs—also Cycolac plastics mouldings—to align with the blades in the sockets. A tab on each connector retains it in the plug. When this tab is depressed the connector can be removed.

Three of the plugs are used, and two are spares; the mouldings for the plugs have been designed so that they can be inserted neither in the wrong socket, nor in an inverted position. In the upper part of the unit three 35 A fuses, and two spare fuses, are mounted. The three fused circuits are for components supplied directly from the battery, those that are controlled by the ignition switch, and those controlled by the switch for the parking lamps. A smaller, but similar, eight-way connector is provided between the facia harness and the rear harness. The same type of connector is employed also in the circuits for the switch for the direction indicators on the steering column. It is claimed that, compared with a conventional harness, this three-piece harness has effected a reduction of 10 per cent in the amount of wire used. Altogether, there are 22 circuits in the system. Additionally, there is provision for 10 spare 22 A and six spare 45 A circuits.   *Concluded*

The wiring harness in the engine compartment is plugged in, at this multiple connector in the dash, to the remainder of the electrical system of the Austin 1800
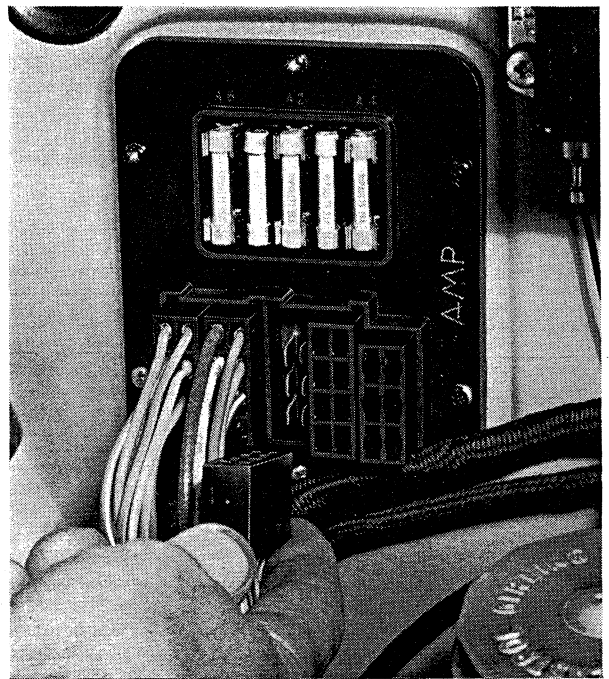
Engineering Courses, Conferences and Seminars

MANUFACTURING COSTS

SEMINARS under the title of "Estimating Manufacturing Costs" are being organized by Industrial Education International Ltd. The aim is at helping participants in the task of estimating, and improving the accuracy of estimating, manufacturing costs. These are two-day seminars, and they will be held in Birmingham on 17 and 18th February and London on 25th and 26th February. *Circle reply card* **7**

PRODUCTION ENGINEERING TRAINING

THIRTY-SEVEN separate courses have been planned by PERA for 1965. The programme includes five-day general appreciation courses, five-day courses on specific techniques, and two-day refresher courses. The subjects have been chosen to attract managers, production engineers, project engineers, foremen, chargehands, toolroom personnel and senior apprentices from Member firms of PERA. The programme is available free of charge.   *Circle reply card* **8**

FLUID LOGIC AND AMPLIFICATION

THE British Hydromechanics Research Association and the Production and Industrial Administration Department of the College of Aeronautics are proposing to hold a conference dealing with the new and rapidly expanding area of fluid elements for computing and power amplification. It is felt that this conference is needed, since a great deal of interest is evident as increasing numbers of small research groups in industry and colleges are actively investigating or

planning work on various aspects. The aim is at bringing these isolated groups together to generate discussion and co-operation. A programme is in preparation and the organizers will be grateful to hear from anyone who may wish to attend, in order that an estimate can be made of the probable number of delegates. Offers of papers are invited and further information can be obtained.
Circle reply card **9**

HIGH-SPEED ENGINES

A POST graduate course on the design and development of high-speed engines is being offered by the Brighton College of Technology in conjunction with Ricardo and Co. Engineers (1927) Ltd. It is to be held on 5th and 9th April 1965, and the closing date for applications is 23rd February.

The subjects covered include revision of basic thermodynamics and the calculation of performance by digital computer. Fundamental principles of gross and local heat flows will be discussed. Experimental techniques will be dealt with, and this will include dynamic measurements on valve gears. Also included are normal and abnormal combustion phenomena and combustion chamber design. Other subjects include combustion in diesel engines, engine dynamics—revision of basic theory—and performance and selection of petrol and diesel engines. Qualifications for admission are: a first degree, Diploma in Technology, Higher National Diploma or Certificate in Mechanical Engineering or an equivalent qualification.   *Circle reply card* **10**