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Z MAGNETTE GROUP NEWSLETTER - SPRING 1999

95826-2235 11

A pre-production ZA with no quarter-lights, fog lamps, over-riders, and with plain hub-caps and dark-coloured wheels.

Dear Enthusiasts,

Well I've finally done it - I now have Email! You can reach me at: mgracer@ync.net. Please keep me informed of your Magnette projects adventures for the newsletters.

ZMG member Bob Williamson's Magnette website at: members.xoom.com/mgcars/index.html should not be missed. Bob and webmaster Sharon Putman have done an excellent job. They have photos, library ,message board, and music! Please support this website.

Our Magnette has been out to several events already this season and is running great. I reported last fall about the "clunking" noise in the front suspension, further inspection showed that the problem was the worn rubber bushings in the trailing rod of the lower suspension arm. The trailing rod ends at the bracket that bolts to the underside of the floor (just under your feet). The rubber had deteriorated, allowing the rods to be loose in the bracket causing the "clunk". Repairing that, installing new shocks and tires really did the trick, the Magnette feels as good as new. I made the rubber bushings myself from urethane, as well as modify the brackets to hold the bushings better in place. I will write a full description of the modification in the next newsletter.

I bought bias-ply U.S. Royal 5.60-15 tires from Coker Tire. Radial tires are better for daily use and all weather driving, but I really have to have those 2 1/4" whitewalls! The shocks are from NAPA - #94001 at $17.99 each. The tube had to be pressed out of the lower rubber bushing and the Magnette pin installed. This was done in a bench vise, the shocks then bolted right in.

The newsletter this month features an engineering dissertation on the ZA Magnette, submitted by ZMG member Eric Baker. This article was published in Automobile Engineer, a British trade magazine, in 1955. Everything that we need to know about the mechanical components of our cars is covered.

This was an expensive newsletter to copy and mail, and the last response for donations was rather low. If you have not sent in a $10.00 donation recently, please do so now so that I can keep these newsletters coming to you.

Safety Fast!

Jeff

NEW MEMBERS

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Seattle, WA 98144 (206) 860-6043

CARS FOR SALE

'56 ZA $3500.00 - OBO. New tires. Call for details.
John Allinson (317) 823-0781 (IN.)

CAR WANTED

ZB body shell with minimal rust or a decent driver.
Allen Bachelder (540) 544-7333 (VA.)

ZA or ZB in good running condition.
Oscar Dahms (630) 469-7332 (IL.)

PARTS FOR SALE

ZB cloth wire harness $150.00
Doug Jackson (415) 472-3067 (CA.)

Parting out ZA, call or Email your parts list.
Aaron Spaulding (310) 639-2582 (CA.)
aaronspaulding@compuserve.com

Windscreen and rear window seals for ZA & ZB (not varitone rear window).
Jeff Powell (708) 344-2268 (IL.)
mgracer@ync.net

M.G. MAGNETTE

$1\frac{1}{2}$ Litre, High Performance Saloon Car

SPECIFICATION

ENGINE: Four cylinders. Bore and stroke 73·025 mm (2·875 in) × 89·0 mm (3·5 in). Swept volume 1,489 cm³ (90·8 in³). Maximum b.h.p. 60 at 4,600 r.p.m. Maximum b.m.e.p. and torque respectively 125 lb/in² and 76·1 lb-ft at 3,000 r.p.m. Compression ratio 7·15:1. Three-bearing crankshaft, fully balanced. Overhead valves, push rod operated. Carburettors: twin S.U. downdraught, with $1\frac{1}{4}$ in diameter choke. Heart-shaped combustion chambers.

TRANSMISSION: Borg and Beck single dry plate clutch, 8 in diameter. Four forward speeds and one reverse. Ratios: top 1:1, third 1·374:1, second 2·214:1, first 3·64:1, and reverse 4·76:1. Hardy Spicer open propeller shaft.

REAR AXLE: Three-quarter floating unit, with hypoid bevel and banjo casing. Final drive ratio 8:39.

FRONT SUSPENSION: Double transverse link, with brake torque struts and coil springs. Girling, 1 in bore, telescopic, finned shock absorbers; length, $9\frac{3}{16}$ in—$8\frac{11}{16}$ in compressed and $14\frac{7}{16}$ in—$13\frac{15}{16}$ in extended.

REAR SUSPENSION: Semi-elliptic leaf springs with through axle. Girling, 1 in bore, telescopic finned shock absorbers, 12 in compressed and 19 in extended.

STEERING: M.G., rack and pinion type. $2\frac{3}{4}$ turns of the 17 in diameter steering wheel from lock to lock. Turning circle 37 ft 6 in.

BRAKES: Front, Lockheed hydraulic, two leading shoe. Rear, Lockheed hydraulic, one leading and one trailing shoe, with mechanical parking brake. Drum diameter 10 in. Shoe width $1\frac{3}{4}$ in. Total friction lining area 134·4 in².

TYRES: 5·50—5·00. Pressure: front 24 lb/in², rear 26 lb/in².

DIMENSIONS: Wheelbase 8 ft 6 in. Track: front and rear 4 ft 3 in. Ground clearance $5\frac{5}{8}$ in. Overall length 14 ft 1 in. Overall width 5 ft 3 in. Overall height 4 ft 10 in. Frontal area 20 ft². Curb weight 2,478 lb, with 5 gallons of fuel. Front/rear weight distribution 54·3/45·7. Engine dry weight 404 lb, power unit complete. Body weight 760 lb, without seats and trim. Gearbox and clutch weight 80·1 lb.

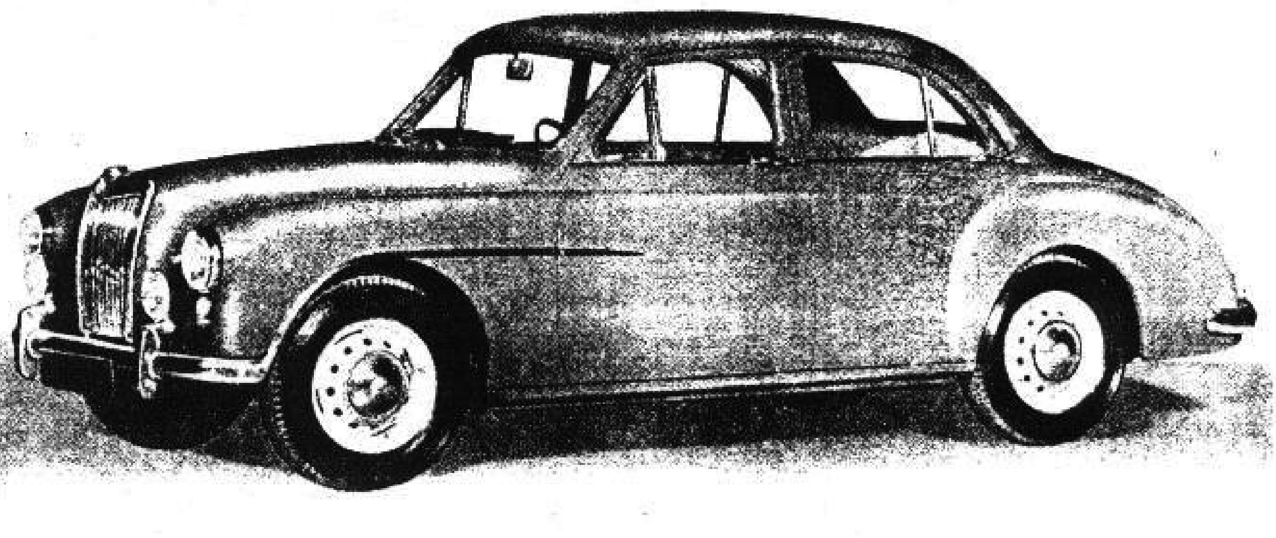
WHEN the M.G. Magnette was first conceived, it was intended to replace the $1\frac{1}{4}$ litre model. The aim was at a low-built, four-seater saloon of brisk performance, good road holding and steering, and with a braking performance to match its other performance characteristics. This vehicle has been designed for those who appreciate the finer points of driving and who are prepared to pay a little extra for good quality.

It is powered by the B.M.C., four-cylinder, 1,489 cm³ engine, which has a bore of 73·02 mm, a stroke of 89 mm, and a compression ratio of 7·15 : 1. The unit develops 60 b.h.p. at 4,600 r.p.m., and has a maximum torque of 78 lb-ft at 3,000 r.p.m. A three-bearing crankshaft is employed. The well-known heart-shaped type of combustion chamber, characteristic of the B.M.C. engines, is incorporated. A single camshaft in the side of the crankcase actuates the push rods and rockers of the overhead valve gear. This engine, complete with carburettors and air cleaner, weighs 404 lb dry; the gear box and clutch together weigh 80·1 lb.

The kerb weight of the vehicle, with five gallons of fuel, is 2,478 lb and the weight distribution is 54 per cent to the front and 46 per cent to the rear. The engine power per ton is 54·2 b.h.p. This should give lively acceleration. The brake lining area per ton is 104·5 in², a figure that is ample, even in mountainous terrain, for this class of car.

A particularly well-designed transmission system is incorporated. A noteworthy feature is the careful attention to the lubrication of the gearbox. This should help to give long, trouble-free life and, what is perhaps equally important, the noise level should remain low for an exceptionally long period. In less well-designed gearboxes, noise generally begins to develop as a result of wear of the bearings, as a result of which the gears mesh incorrectly. Although the gearbox is a standard unit designed for incorporation in other vehicles, and therefore can be fitted with a steering column gear shift control, in the Magnette a floor-mounted type of gear shift lever is employed, since this type is preferred by most drivers of high performance cars.

A conventional rear suspension system is employed, and a simple semi-elliptic spring layout has been found adequate. There is no anti-roll bar on this vehicle since it is low slung and shows no tendency to undue rolling. At the front, a double transverse link and coil spring layout has been adopted. Care has been taken to ensure that the linkage is sufficiently rigid to avoid undesirable steering characteristics, particularly under the influence of brake torque. This has been accomplished by employing additional links to assist both the lower single transverse link and upper wishbone in reacting brake torque and drag loads. Coil springs have been



employed because torsion bars are out of the question with such a low floor, but in any case, extremely satisfactory suspension characteristics are obtainable with the coil spring type of arrangement. To obtain positive steering characteristics, a rack and pinion type unit is employed. A flexible coupling is incorporated in the steering column to damp out shocks and vibration.

So far as the body is concerned, its low lines minimize air resistance and give the vehicle an appearance that is characteristic of sports saloon cars. Although the body is designed to form a rigid box structure, the strength and stiffness of the base is adequate to support the vehicle satisfactorily without the help of the roof. Generally, chassisless construction is inherently stronger and stiffer than a separate frame and body combination.

Clutch and gearbox

A Borg and Beck, single-dry-plate clutch with a sprung centre is employed. The outside diameter of the friction lining is 8 in and its inside diameter is $5\frac{1}{4}$ in. Six pressure springs are fitted, and the spring load, assembled, is 990-1,050 lb. The withdrawal thrust ring is of carbon-based material and there is no compensating device or engine tie, since hydraulic actuation is employed. A bell-housing that is integral with the die-cast aluminium gearbox encloses the whole unit.

The four-speed gearbox weighs 67 lb dry. Its oil capacity is four pints, and S.A.E. 30 grade is recommended. Synchro-mechanisms are employed with top, third and second speeds. The ratios obtainable are: top 1:1, third 1·374:1, second 2·214:1, first 3·64:1 and reverse 4·76 : 1. A conventional type of floor-mounted gear shift lever is employed and gives positive control.

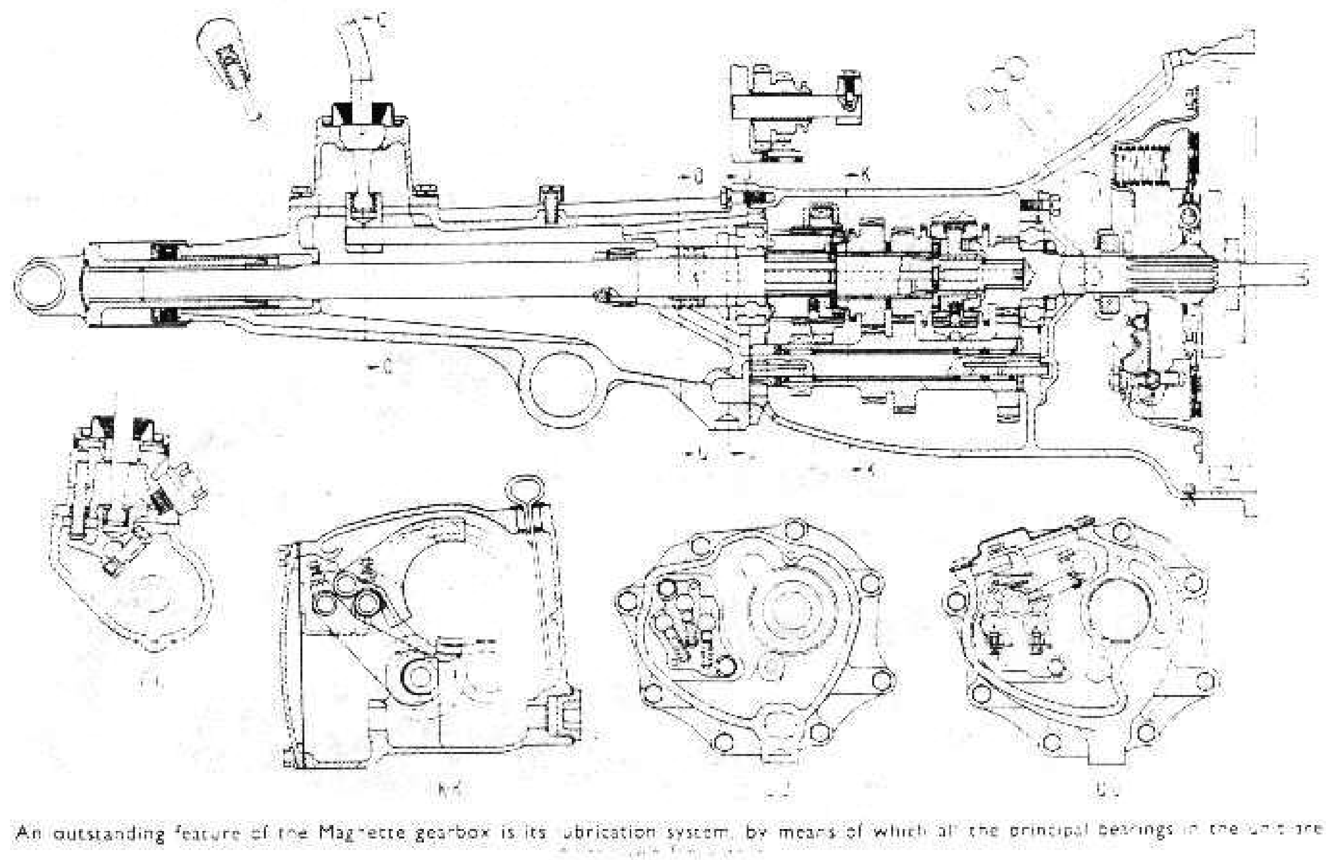
At the front of the box, a die-cast aluminium cover is bolted on to retain the primary shaft bearing. In it a scroll type oil return is machined in the bore through which the primary shaft passes. This cover positively locates the primary shaft assembly in the usual manner: that is, it retains a snap ring against a seating in a shallow counterbore round the bearing housing, and this snap ring is sprung into a groove in the outer periphery of the outer race of the ball bearing that carries the primary shaft assembly. The inner race of the ball bearing is pulled up against the primary gear by a ring nut, locked by a tab washer, on a $1\frac{1}{4}$ in diameter thread on the En 361 primary shaft. This shaft is $\frac{3}{4}$ in diameter at the spigot bearing at the front end, and 1 in diameter over the splines that carry the driven plate of the clutch. Integral with the front end of the gear and shaft are the cone and tooth ring for the first speed synchro unit. The tooth ring is $\frac{7}{16}$ in thick and the thickness of the primary gear is $\frac{3}{4}$ in at the pitch circle diameter.

An En 361 mainshaft is employed; its overall length is about $24\frac{3}{4}$ in. This dimension is unusually large because the rear extension of the gearbox is long. However, the distance between the centres of the ball bearings in the front and rear walls of the gearbox is only about $7\frac{5}{8}$ in. At the front, the diameter of the shaft is reduced to about $\frac{11}{16}$ in where it spigots into the $1\frac{1}{8}$ in long, needle roller bearing in the primary shaft. The bore in the primary shaft is about $1\frac{1}{2}$ in diameter.

A simple cup-and-cone type synchro unit for the top and third speeds is mounted on the $1\frac{3}{8}$ in diameter splined portion of the mainshaft, immediately behind the spigot bearing. Three spring - loaded - ball detents, housed radially in the centre component, register in grooves in the En 18C sleeve component. The included angle of the cones is 20 deg and the mean diameter of the conical surfaces is about $2\frac{7}{16}$ in for top and $2\frac{1}{8}$ in for third speed. Duravint bronze synchro-cups float in annular recesses, one in each end of the stamped En 18C centre component.

Three slots are machined radially in the material round each of the recesses. Projecting through them are lugs spaced 120 deg apart round the peripheries of the two synchro-cups. These lugs extend outwards as far as the roots of the splines in the sleeve member. To provide clearance for each lug and to form a V-shaped lead-in to centralize it in its slot after the synchronization, the ends of two of these splines are machined off obliquely at an included angle of 30 deg for a length of about $\frac{5}{16}$ in. The inner end of each lug, that is, the end facing towards the centre of the synchro unit, is chamfered to a V-shape to seat in the lead-in.

During the initial stages of gear selection, the sleeve, and with it the centre member and cup ring, are moved axially until the cup and cone engage. If the driving and driven assemblies are not synchronized, the cup is rotated



An outstanding feature of the Magnette gearbox is its lubrication system, by means of which all the principal bearings in the unit are through a small angle, which is limited by the clearance between the lugs and edges of their slots, and in this position the lugs baulk further movement of the sleeve. When synchronization has been effected, the chamfered edges of the lugs slide into the V-shaped seatings on the ends of the splines adjacent to them; thus they are centralized, so the sleeve can be slid into full engagement with the tooth ring of the gear.

The third speed gear, with its $\frac{7}{16}$ in thick integral tooth ring for the synchro unit, floats on a phosphor bronze bush. Its effective bearing length is $1\frac{1}{4}$ in, and the outside and inside diameters of the bush are respectively 1·3115-1·3120 in and 1·124-1·125 in. The bush is flanged inwards at its forward end and splines are cut in the flange to locate the component against rotation relative to the mainshaft. A $\frac{1}{4}$ in wide, oil groove is machined round its outer periphery, and a radial hole in the base of the groove allows oil, from passages drilled in the mainshaft, to pass through the bush.

Another phosphor bronze bush, behind the first, carries the second speed gear with its integral cone and $\frac{7}{16}$ in thick tooth ring. Separating the two bushes is a $\frac{3}{16}$ in thick phosphor bronze thrust washer with four slots cut in its inner periphery. Two diametrically opposite dogs are formed on the adjacent ends of each bush to register in these four slots and lock the whole assembly together: thus, neither the bushes nor the washer can rotate relative to the mainshaft.

Lubrication of the second speed bush is effected in the same way as that already described for the third speed bush, but the oil groove is $\frac{1}{2}$ in, instead of $\frac{1}{4}$ in, wide. Its inside and outside diameters are respectively 1·126-1·127 in and 1·3115-1·3120 in, and the effective bearing length is $1\frac{1}{2}$ in. This inside diameter is about 0·002 in larger than that of the third speed bush, which is the master component so far as location against rotation is concerned. To have made both bushes the same inside diameter, and therefore the same fit on the parallel shaft, would have introduced difficulties on assembly, since the second speed bush has to be pressed further along the shaft than the third speed one.

The second and third speed gear assembly is retained between two $\frac{1}{8}$ in thick, case hardened mild steel thrust washers. At the rear, the washer is divided diametrically and is fitted between the end of the second speed cone and a shoulder round the splines that carry the centre component of the synchro unit. The two parts of the ring are retained in position by the synchrocup, which surrounds them.

A different method is employed to retain the front washer. This component is internally splined and is assembled from the front on to the splined end of the shaft until it is stopped by a spring-loaded plunger projecting from a radial hole drilled in the base of one of the splines of the shaft. Then the plunger has to be depressed so that the washer can move over it. At this station, a groove is machined round the splines, and the washer is rotated in this groove until the plunger springs up between two of the internal splines and prevents further rotation. Thus, the splines in the washer are half a pitch out of normal phase relative to those round the shaft, so the washer cannot move axially. The head of the plunger is chamfered so that a small screwdriver can be used to push it down to allow the washer to be rotated back again to dismantle the assembly.

The mean diameter of the conical face of the second speed synchro cone is about $1\frac{13}{16}$ in. As in the third and top speed assemblies, the cup is carried in a counterbore in the centre component of the synchro unit. The first speed gear is formed on the sleeve of this unit. There are three spring-loaded-ball detents in the centre component, which is allowed a limited axial float on the $1\frac{1}{4}$ in diameter splined portion of the shaft.

Location of the complete mainshaft assembly is effected at the rear ball bearing. The inner race of this bearing is mounted on a $1\frac{1}{8}$ in diameter portion of the shaft and bears against a shoulder formed at the ends of the splines that carry the centre component of the second speed synchro unit. Behind this bearing is a $2\frac{1}{8}$ in long, distance sleeve and the keyed-on En 1A speedo gear. The whole assembly is retained by a ring nut tightened against the speedo gear and locked by a tab washer. The distance sleeve forms a muff to help to carry the oil to lubricate the mainshaft. Details of this lubrication system will be given later.

The outer race of the ball bearing is carried in a circular housing, which is spigoted into a circular aperture in the rear wall of the gearbox. This arrangement has been adopted so that the whole of the assembly on the mainshaft, including the bearing housing, can be assembled in through the aperture in the rear wall. The bearing and its housing are retained by the rear extension casting, which is bolted on. Behind the speedo gear and its retaining nut, the mainshaft is 1 in diameter and splines are cut on its rear end to carry the sleeve of the sliding joint.

A felt dust seal and a lip type oil seal are housed, with the oil seal innermost, in a cupped, steel pressing assembled on to the rear end of the extension. To hold the assembly on, the rim of the cup is rolled into a groove round the extension. A tubular shroud, mounted in a similar manner on the sleeve component of the sliding joint, surrounds the seal housing. The sliding joint operates in a $2\frac{1}{4}$ in long × $1\frac{1}{4}$ in inside diameter, phosphor bronze bush. A noteworthy feature of this gearbox is that a small breather is screwed into the top of the extension so there is no possibility of oil being forced out by fluctuations in pressure due to the pumping action of the sliding joint, or to build-up of pressure internally as the unit warms up.

All the layshaft gears are in one cluster on the En 361 sleeve. This component is carried on needle roller bearings on the $\frac{3}{4}$ in diameter, En 352 shaft. There are three rows of needle rollers, one at the rear and two at the front. The reason why two rows of rollers are employed at the front, despite the fact that the loading is heaviest on the rear, is that by far the greater proportion of running is done on the higher gears, so both fatigue and wear tend to be greatest at the front. All the rollers

TABLE I. GEARBOX DATA

	Number of teeth	Diametral pitch	Helix angle	Material	Thickness of gear blank at p.c. diameter
Primary gear:					
Constant mesh	21	10·0231	33 deg 21 min 26 sec	En 34R	$\frac{21}{32}$ in
Dog teeth	30	10·8 15·45	straight	En 34R	$\frac{7}{16}$ in
3rd speed gear	25	10·0231	33 deg 21 min 26 sec	En 34R	$\frac{5}{8}$ in
2nd speed gear	31	10	30 deg	En 34R	$\frac{11}{16}$ in
1st speed gear:					
External	28	8	straight	En 34R	$\frac{5}{16}$ in
Internal	27	10·8 15·45	straight	En 34R	—
Layshaft cluster:					
Layshaft gear	30	12	33 deg 21 min 26 sec	En 34R	$\frac{41}{64}$ in
3rd speed gear	26	12	33 deg 21 min 26 sec	En 34R	$\frac{5}{8}$ in
2nd speed gear	20	11·547	30 deg	En 34R	$\frac{11}{16}$ in
1st speed gear	11	8	straight	En 34R	$\frac{3}{8}$ in
Reverse speed gear	—	8 (topped)	straight	En 18C	$\frac{3}{8}$ in
Reverse idler gear	—	8 (topped)	straight	En 18C	$\frac{1}{2}$ in

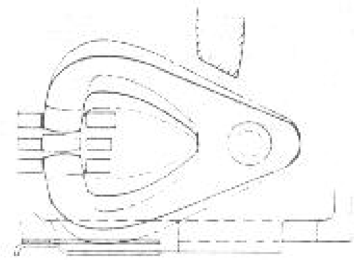
are $\frac{3}{8}$ in long, and a wrapped, steel distance tube separates the front and rear sets. At each end, a $\frac{3}{16}$ in thick. case hardened En 18C thrust washer is interposed between the cluster and the end walls of the gearbox. The shaft is carried in holes in the end walls of the box and is located between the front cover and rear extension castings. At its front end, it is shouldered diametrically, by machining away a semi-cylindrical portion, and the remaining portion projects into a semi-circular hole in the front cover and thus is prevented from rotating.

The front end of the $\frac{3}{4}$ in diameter En 32B reverse idler spindle is carried in a boss cast integrally in the base of the box and its rear end is in the rear wall. This spindle is locked at its forward end by a dowel-ended set bolt which is screwed radially through the boss and registers in a diametral hole in it. Pressed into the idler pinion is a phosphor bronze bush $1\frac{5}{8}$ in long. The running clearance between the bore of the bush and the spindle is 0·003-0·0015 in.

An outstanding feature of this gearbox is its lubrication system. When the unit is static almost all the layshaft cluster is submerged below the oil level. Lubricant splashed through the primary gear bearing falls down into a space between the front cover and the front end of the box. Thence it passes into an axial drilling in the layshaft and out through radial holes to lubricate the roller bearings at that end of the cluster.

At the rear of the box, the other layshaft bearing is lubricated in a similar manner, except that this end is always submerged below the oil level, so it is not necessary to supply the space between the rear cover and the end wall of the box from above. This space forms part of the balancer duct that maintains the oil levels equal in the main part of the box and the rear extension. The positive feeding of the roller bearings with lubricant is a good feature since, despite the fact that they are below the oil level, they might otherwise be starved by centrifugal action.

Lubricant splashed through the mainshaft bearing at the rear of the box is collected in a space in the front end of the rear extension casting. From there, it passes up a $\frac{1}{16}$ in diameter duct to an annular groove in the bore of the casting, which houses the muff around the shaft. A spiral groove round the muff passes the oil forward to another annular groove, this time round the muff, and thence through a radial hole into a $1\frac{3}{8}$ in long annular groove in the bore of the muff. From there, it passes through a radial hole into an axial one drilled from the front end of the mainshaft. As the oil passes forward along the length of the shaft, some goes out through radial holes to lubricate the second and third speed gear bushes and the remainder passes through a restrictor plug in the end of the shaft to lubricate the spigot bearing.



The ends of the arms of a C-shaped plate register in the striker forks so that only one gear can be engaged at a time

There are two subsidiary lubrication systems in the rear extension. One serves the speedo gear. A $\frac{1}{4}$ in diameter hole, drilled through the mainshaft bearing housing and the rear extension, breaks out of the casting immediately above the speedo gear. This hole slopes downwards towards the rear so that oil splashed into it runs down and drops on to the gear.

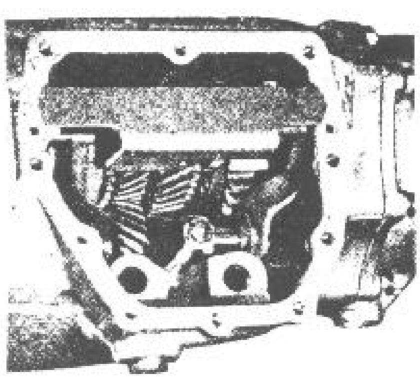
An overflow from the space behind the rear bearing runs into a trough that extends rearwards in the rear extension and feeds into a space immediately in front of the bush of the sliding joint. This ensures that, while the bush is adequately lubricated, the space in front of it is not flooded with oil, a condition which would lead to excessive quantities being pumped in and out of the sliding joint. A drainage duct is cored from the bottom of the space between the rear end of the bush and the oil seal, into the main chamber of the extension.

A turret for the En 3A striker lever is mounted near the rear end of the extension casting. On the lower end of the lever, a spherical head is formed, and round it is a phosphor bronze bush. This bush is in two semi-circular sections held together by a circlip in a groove round its outer periphery. It is housed in an En 3A fitting secured by means of a pinch bolt on the end of a remote-control rod. This fitting is positively prevented from rotating on the shaft by a Woodruff key.

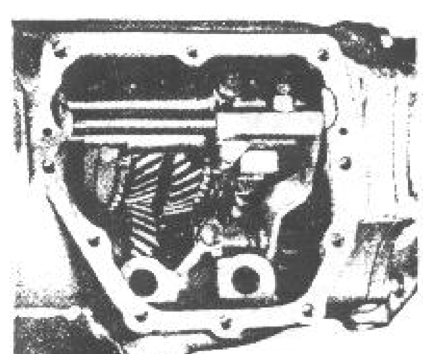
The rod is housed in the rear extension and its axis is parallel to the mainshaft. Its front end projects through a hole in the front wall of the extension of this front wall is drilled to carry the rear ends of the En 32B selector rods housed in the side of the gearbox and the spring-loaded ball type locks. Another striker lever, keyed on and secured by a pinch bolt, is carried near the front end of the remote-control rod. This lever projects in the usual manner into striker forks on the rear ends of the selector rods. These forks are secured to the rods by conical ended set screws inserted radially into their bosses to register in holes in the rods. This remote control layout is lighter, more compact and less expensive than the more usual separate control-housing arrangement bolted on top of the extension.

The interlock mechanism is somewhat unusual. In the rear extension casting, a C-shaped plate is arranged so that when the gear shift lever is in neutral, the ends of its arms project into the striker forks on each side of the striker lever. This plate is pivoted at a point to the rear of the forks, on a right angle bracket bolted to a lug on the casting under the pressed steel cover on top of the extension casting. Side-to-side movement of the striker lever turns the plate about its pivot and aligns the gap between the ends of its arms with one or other of the striker forks. When a gear is selected, the appropriate fork is moved into the gap between the arms of the plate, which is only just wide enough to accommodate it. Thus, it is impossible to select more than one gear at a time. A right angle bracket in the main body of the gearbox is slotted to form a gate, and tongues projecting down from the bosses of the striker forks register in the slots. The reason for the adoption of this interlock arrangement is that in other vehicles, this gearbox is used with a steering column gear shift control. With this type of control, the transverse movement of the striker lever is effected by moving the C-plate about its pivot. The reverse baulk mechanism is a spring-loaded plunger that bears against an arm on the fitting at the rear end of the remote-control rod. A cam on this fitting actuates the plunger of the reversing lamp switch, which is screwed into the side of the turret casting.

The front ends of the selector rods



Lugs project downwards from the selector fork collars into a gate assembled into the



The gearbox with its side cover and gate

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are supported in the front wall of the gearbox, and the selector forks are secured to the rods in a similar manner to the striker forks. The selector fork for top and third speed registers in a groove round the sliding member but that for second and bottom speeds is of channel section and fits round the first speed gear on the sleeve. All the forks are of aluminium bronze.

A die-cast aluminium cover bolted on the side of the box gives access to the selector mechanism, and a small pressed steel cover on top of the extension gives access to the striker forks. The oil level dip-stick is carried in a boss on top of the gearbox casing and the drain plug is below it. This gearbox arrangement is noteworthy for its compactness. The overall height of the gear assembly is about $6\frac{1}{4}$ in, and its overall width, complete with selector rods, is approximately 6 in.

Rear axle

A Hardy Spicer propeller shaft, with needle roller bearing universal joints, transmits the drive to the rear axle. The shaft is $2\frac{1}{4}$ in diameter and its length between centres is 50·38 in. The axle is of the three-quarter floating, hypoid bevel type in a banjo casing. The oil capacity of the unit is $2\frac{1}{4}$ pt, and Hypoid 90 grade is recommended.

The B.S.1490-LM4-M gear carrier is secured to the banjo casing by ten $\frac{5}{16}$ in diameter bolts. In the nose piece, the axis of the case hardened En 35, 3 per cent nickel chrome steel pinion is offset 1 in below that of the crown wheel and $\frac{1}{32}$ in from the axis of the differential pinion. Two taper roller bearings, separated by a $2\frac{3}{4}$ in long barrel-shaped distance piece, support the pinion shaft. The distance piece is a casting; it is less expensive than a drawn tube and its barrel shape gives it more resilience.

At the rear bearing, the shaft diameter is $1\frac{1}{4}$ in; at the front bearing it is 1 in. The whole assembly, comprising the two bearings, distance piece and companion flange for the universal joint, is pulled against a distance washer, which seats on the front face of the pinion, by a nut on the $\frac{3}{4}$ in diameter threaded end of the shaft. This distance washer is fitted to clear the fillet radius between the spindle and the pinion, and it also is used to adjust the axial position of the pinion. It is made in a range of thicknesses in increments of 0·002 in, from 0·112 in to 0·126 in. The pre-load of the bearings is adjusted by tightening the nut on the front end of the shaft until the torque required to rotate the assembly is 11-13 lb-in without the oil seal, or 14-16 lb-in with the oil seal in position.

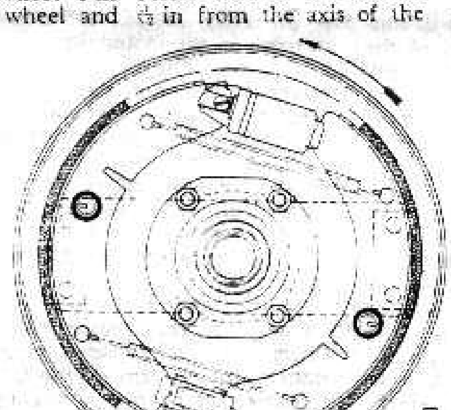
During assembly, the shaft, with the rear roller bearing on it, is inserted from the rear, the other components, including the front bearing, are then assembled from the front. The lip type oil seal is housed in the front end of the casting and its inner periphery bears on the boss of the companion flange for the universal joint. A cupped, pressed steel shroud is pressed on to the boss of the companion flange for the universal joint and surrounds the seal assembly to protect it from foreign matter. Lubrication of the bearings is effected in the usual manner through a large duct cored in the top of the nose piece and so placed as to catch oil splashed off the gears.

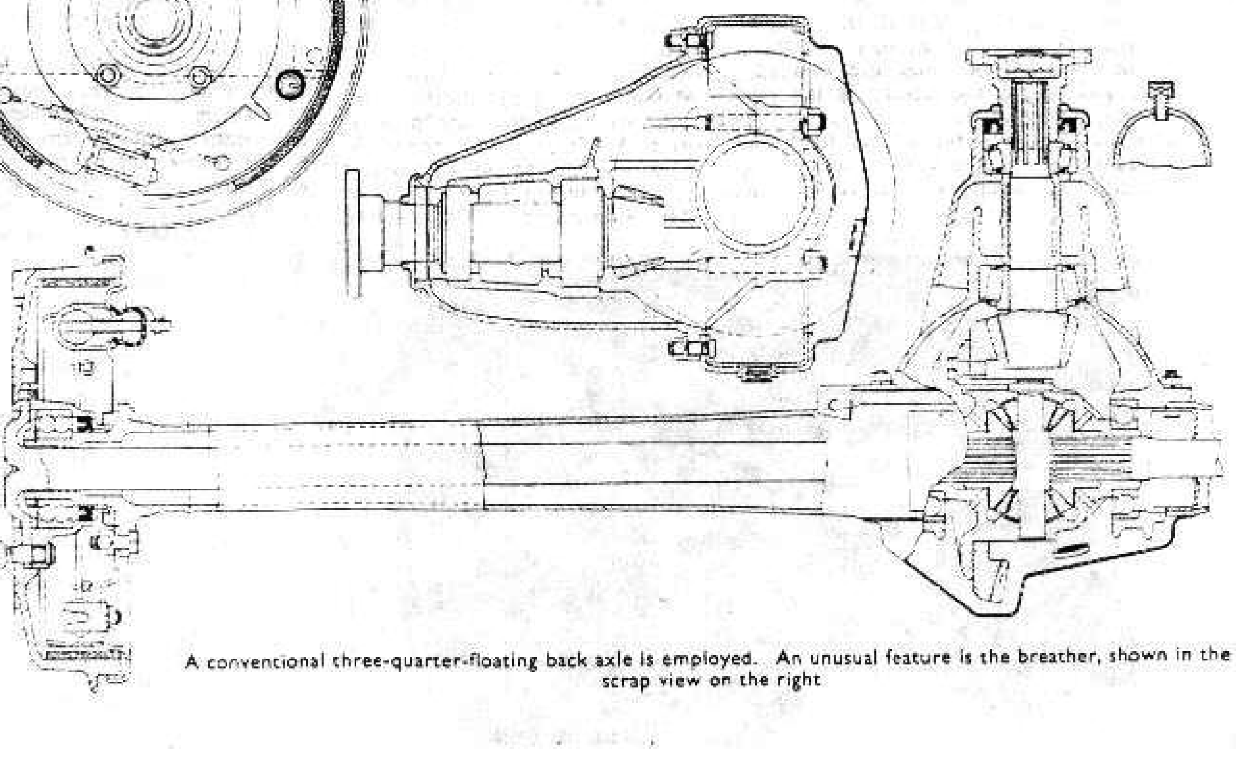
A case hardened En 35, stamped crown wheel is employed. It is $7\frac{1}{2}$ in diameter. Five $\frac{3}{8}$ in diameter set bolts and tab washers are passed through holes in the flange round the differential cage and screwed into the crown wheel to hold it in position. The B.S. 310/3 cast iron differential cage is in one piece. It is drilled to receive the $\frac{11}{16}$ in diameter, case hardened En 36 nickel chrome steel differential pinion spindle. This spindle is located in the cage by a $\frac{3}{16}$ in diameter peg driven diametrically through one end. Flats are machined on it to pass oil outwards into the pinion bearings.

Spherical thrust washers of B.S. 407/2 bronze, in the quarter-hard condition, are interposed between the differential pinions and the cage. They are $2\frac{1}{2}$ in diameter. The two pinions are of case hardened, En 36V steel, and the effective length of their engagement with the differential gears is approximately $\frac{1}{2}$ in. They are about $1\frac{11}{16}$ in overall diameter.

Flat thrust washers of the same material as those for the differential pinions are interposed between the differential gears and the cage. These gears are of En 36 and are $2\frac{1}{4}$ in overall diameter. Their bosses bear for a length of $\frac{5}{8}$ in in the differential cage and are $1\frac{3}{8}$ in diameter. The diametral clearance is 0·005-0·003 in. To assist lubrication, four $\frac{3}{16}$ in diameter holes are drilled through the cage and break through at points where the journal and thrust bearing surfaces meet. This practice is adopted by most manufacturers, but it is of interest to speculate as to whether or not the oil is centrifuged out of these holes instead of passed in through them to lubricate the bearings.

On both sides, the cage is machined down to $1\frac{5}{8}$ in diameter to receive the inner races of the ball thrust bearings that carry the assembly. These bearings are spaced with their centres approximately $5\frac{1}{4}$ in apart. The outer races are housed in the gear carrier and are retained by bolted-on bearing caps. A projection from the outer face of each bearing cap registers positively against





A conventional three-quarter-floating back axle is employed. An unusual feature is the breather, shown in the scrap view on the right the end of the adjacent axle tube to prevent spreading. The mesh of the crown wheel and the bearing pre-load are adjusted to the specified values by means of shims between the inner races of the bearings and the cage. Baffle plates are fitted in the inner ends of the axle tubes to prevent excessive quantities of oil from passing outwards to the wheels. On top of the right-hand tube is screwed a small breather, exactly the same as that on the gearbox. This is a good feature, since it prevents oil being forced past the seals by internal pressure every time the axle warms up.

The axle tubes are about 2¼ in diameter by [illegible] in thick at their outer ends, where the hub carrier, with its integral brake-back plate flange, is butt welded on. At a distance of 19⅛ in on each side of the differential pinion centre, the pressed steel, spring seating pads are welded on. The half shafts are of En 17V, manganese molybdenum steel. They are 1⅛ in diameter at their splined ends. The spline root diameter is 0·926–0·936 in and the depth is 0·080–0·090 in. Immediately outboard of the splines the shafts are reduced to 1 in and then are tapered out to 1⅛ in diameter near their extreme ends.

A nickel chrome cast iron brake drum is secured by countersunk set screws to the upset driving flange on the end of each half shaft. These screws also retain the En 3A housing for the wheel bearing, which is pulled up against the inner face of the driving flange. The five, shouldered bolts that carry the wheels are threaded $\frac{7}{16}$ in diameter. They are pressed into holes in the flange round the bearing housing and pass through clearance holes in the driving flange and brake drum. To prevent them from rotating when the nuts are being tightened, their shanks, where they pass through the flange round the bearing housing, are milled.

The wheel bearing is of the two-row, staggered ball type. Its outer race is retained in its housing by the driving flange, which is pulled against a distance ring interposed between the two components. The inner race is pulled on to the end of the hub carrier by a ring nut threaded 1½ in diameter. An oil return scroll is machined in the bore of the hub carrier and, as the half shaft rotates in it, oil is prevented from passing out to the bearing, which is grease lubricated. Leakage of grease or oil into the brake drums is prevented by a lip type oil seal carried in the inner

The gear carrier assembly of the back axle

end of the bearing housing. This end of the housing is lipped to form a thrower and is enshrouded by a dished steel trap from which any lubricant that might escape past the seal is drained away to the outside of the unit.

Rear suspension

Semi-elliptic springs are used in conjunction with the through-axle of the rear suspension. There is no anti-roll bar, either at the front or rear, on this vehicle. A rubber bump stop is fitted under each longitudinal side member of the body-frame structure. In the full bump position, the axle tube bears directly on the rubber. The rear end rate of 144 lb/in gives a periodicity of 95 cycles/min. To the fully laden position, the travel is 3·88 in and to full bump it is 6·55 in. The unsprung weight is 125 lb per wheel.

A conventional spring mounting arrangement has been adopted. U-bolts round the axle clamp the spring between the lower plate and a pressed steel seating bracket welded under the axle. The inner edge of the lower [illegible] is turned round the $\frac{9}{16}$ in diameter [illegible] that carries the rubber bushed eye at the bottom of the telescopic shock absorber. At the upper end of the shock absorber, another rubber bushed eye fitting is mounted on a similar pin welded in holes through a cross member of the body-frame structure. The shock absorbers are finned, for cooling, at their lower ends. Their centre-to-centre length is 12 in when in the compressed condition, and 19 in extended. Channel section rubbers, $\frac{3}{32}$ in thick, are interposed between the spring and its seating pad and retaining plate. Rubber is also fitted between the leaf retainer clips and the spring.

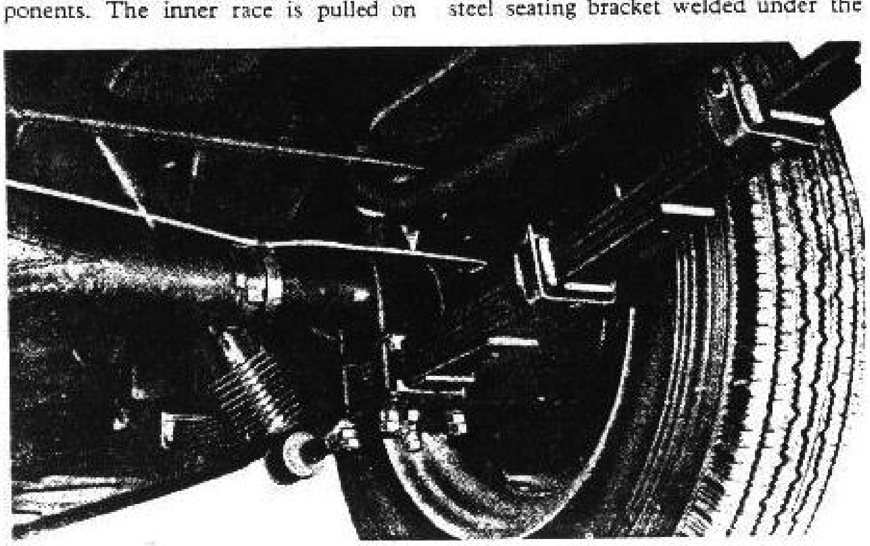
There are seven, 1¾ in wide, En 45A leaves in each spring. The first leaf is $\frac{7}{32}$ in thick and the remainder are ¼ in thick. Rubber pads are inserted between the ends of the leaves. The overall length between eye centres is 46 in, and the axle is mounted above the centre of the spring. Rubber bushes are employed in both spring eyes and also in the shackle pivot on the frame. The pins are of En 5C/R and are $\frac{7}{16}$ in diameter.

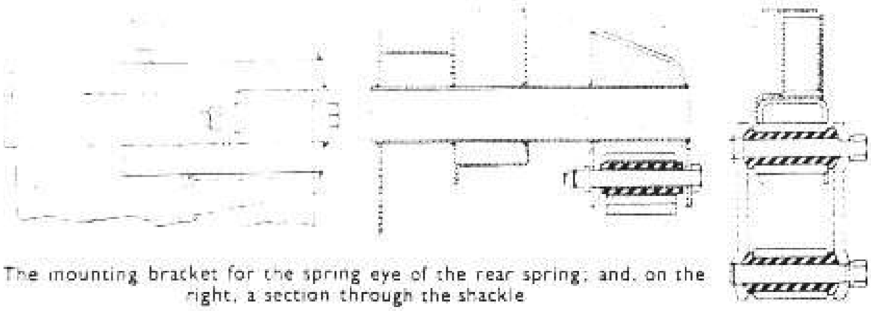
A Silentbloc bush is fitted in the front eye. It is carried between two ⅛ in thick lugs, which are reinforced locally round the bolt hole by $\frac{3}{16}$ in thick washers. This reinforcement, of cou[illegible] is necessary to increase the crush[illegible] strength of the material round the hole for the bolt. The lugs form part of, and extend down from, an overhung, two-piece box shape bracket welded to the side member of the body-frame structure. A ⅞ in outside diameter × 16 S.W.G. mild steel tube is passed through the frame side member and the bracket, and is welded to both. This tube forms the main support for the bracket.

At the rear, a simple, inverted U-section bracket carries the shackle. Round the holes for the pins, the shackle plates are dimpled to precompress the rubber bushes. Doubtless this pre-compression materially increases the life of the bushes, since it offsets any tendency for tensile stresses to be introduced in the rubber during operation. An unusual feature of the assembly is that the shanks of the pins are milled near the heads and they are pressed into holes in one of the shackle plates. Thus, only one spanner is needed to tighten the nuts. The other end of each pin is shouldered so that when the nut is tightened, the plate is clamped against the shoulder. This arrangement positively limits the amount of pre-compression applied to the bushes.

Front suspension

A double transverse link type front suspension, with a negative angle of trail, that is, an angle of lead, of 7 deg to clear the steering gear and engine sump, is employed; the cross member that supports the suspension is behind [illegible]





The mounting bracket for the spring eye of the rear spring; and, on the right, a section through the shackle

one is a single arm, of H-section. The differential loads due to brake torque are taken by two struts. One of these struts is attached to the outer end of the lower link and anchored to the body-frame structure to the rear of the suspension, and the other secured to a bracket near the outer end of the upper wishbone and anchored to the body-frame structure in front of the front suspension assembly.

At first sight, it is perhaps surprising that the lower strut is of $\frac{7}{8}$ in diameter × 10 S.W.G. tube, whereas the upper one is $\frac{3}{4}$ in diameter rod. However, there are two reasons for this. One is that the lower strut supports the single transverse link, whereas the upper one is attached to the wishbone, which is relatively stiff by itself and so requires less support. The other is that the drag loads are added to the compression loads in the lower link, but help to counterbalance those in the upper one. To have used a larger diameter solid rod instead of a tube would not only have increased the unsprung mass, but also, because of this increased mass, it would be liable to deflect under inertia loading as the suspension rises and falls. Such deflection is undesirable, since in most instances it would introduce an initial bow in the rod just when the compression loads due to drag are greatest.

One end of the tubular strut is inserted in a socket in the outer end of the lower link assembly; it is positively located by a pin, and is brazed in. At the other end of the strut, a $\frac{1}{2}$ in diameter threaded rod is brazed in to the tube to carry the rubber end-fittings and their retainers and the nut that secures the assembly. Two circular rubbers are employed. They are fitted one each side of the bracket to which the assembly is secured. A steel collar is fitted round the foremost rubber to stiffen it in compression. Both rubbers are spigoted into the holes in the bracket so that metal-to-metal contact does not take place between the bracket and the rod. The bracket is formed from two channel section pressings, placed back-to-back. To hold these two channels together, a strap plate is spot-welded to the flanges on each side. The upper edges of these plates are flanged and welded to the frame structure. To form a seating for the rubber, a circular rib is pressed in the metal round the hole through which the rod passes in each channel section.

A similar arrangement is employed at the front end of the upper tie rod, but the bracket on the frame is a single pressing and two separate seating plates for the rubbers are fitted one on each side of it. The rear end of this rod is swaged and drilled and is bolted between two lugs of a forked fitting mounted between the flanges of the front arm of the upper wishbone link. Apparently, the angular movement of the rods as the suspension rises and falls is insufficient to cause fatigue failure due to bending either at this point or on the front tubular tie where it fits into the socket in the lower link assembly.

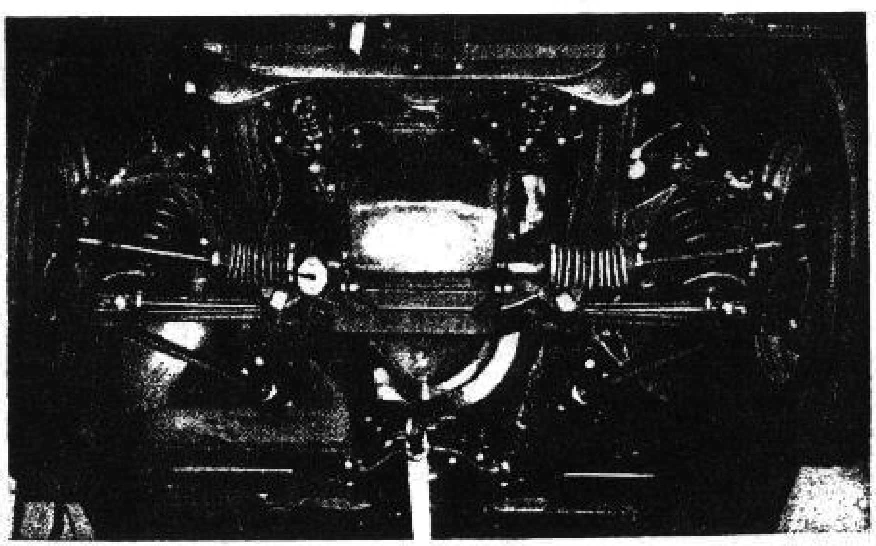
The centre of the rear attachment of the tubular strut is about 10$\frac{3}{8}$ in behind the axis of the lower transverse link, while the centre of the front attachment of the upper strut is about 16$\frac{1}{4}$ in. measured horizontally, in front of this axis. These positions, of course, were fixed mainly by practical considerations of the space available in the layout of the chassis. The angle the rear strut makes with the axis of the lower link is 40 deg. while that of the front strut from the vertical plane containing the axis of this link is 60 deg. Despite the relatively large rubbers employed at the ends of the ties, this arrangement gives a much more positive reaction to brake torque than is obtainable with more conventional double transverse wishbone systems. As a result, the accuracy of the steering geometry is maintained during braking, and the are better than would otherwise be possible.

Conventional coil springs are employed. Their rate is 184 lb/in. The rate at the wheel, including the effect of the rubber bushes, is 188 lb/in. This gives a periodicity of 94 cycles/min. The deflection of the springs to the laden position is 5$\frac{11}{16}$ in and to full bump 7$\frac{11}{16}$ in. Unladen, the static deflection is 4·14 in.

The spring is of En 15A, silicon manganese steel. Its free length is 14$\frac{5}{32}$ in and its length, as fitted, in the fully laden condition is 8$\frac{3}{4}$ in. The mean diameter of the coil is 4 in and the wire is 0·525 in diameter. There are 10$\frac{3}{4}$ coils, but only 9$\frac{1}{4}$ are effective. The upper end of the spring seats on a flanged ring, to which are projection welded the heads of the bolts that secure an inverted, flanged, cupped pressing bolted on top of the cross member to receive the upper end of the shock absorber. A doubling plate is fitted over the top of the cupped pressing. Two circular rubbers are employed in the end fitting on the shock absorber; they are spigoted into the hole through the pressing and doubling plate so that metal-to-metal contact cannot take place.

The shock absorber extends down through a hole in the spring pan. On its lower end, a rubber bushed eye is carried between two lugs, each of which is formed by a special head on an eye-bolt which, together with two more bolts, holds the lower spring pan down on a platform on the lower transverse link. A simple cupped pressing is fitted in the spring pan to provide the necessary radial location at the lower end of the spring. The 1 in diameter shock absorber is finned at its lower end to assist cooling. Its compressed length is given as 9$\frac{5}{16}$ in to 8$\frac{15}{16}$ in and the extended length is 14$\frac{5}{16}$ in to 13$\frac{15}{16}$ in.

A rubber bump stop is mounted under the frame cross member just inboard of the spring and bears directly on the lower transverse link. This link



The front suspension cross member and the steering unit are behind the engine sump is sturdy enough to take the fairly heavy bending moments imposed on it as it strikes the bump stop. The rebound stop is incorporated in the shock absorber.

Two channel section pressings, $\frac{1}{8}$ in thick with $\frac{7}{8}$ in flanges, form the arms of the upper wishbone. They are about $2\frac{5}{16}$ in deep at their inner ends and $1\frac{7}{16}$ in at their outer ends, where the flanges are reduced to $\frac{11}{16}$ in and $\frac{5}{8}$ in overall width respectively, to enable a clean pressing to be obtained. At the inner end of each, a tubular boss for the rubber bush is welded into a hole punched in the arm. These bushes are of the parallel, as distinct from the conical, type. They are carried on the ends of a common pivot pin, which is passed through the cross member. Each rubber bush is in two parts, and is held between $\frac{1}{8}$ in thick dished retainers. These retainers are so shaped as to compress the bush into the boss when the nuts on the $\frac{1}{2}$ in diameter threaded ends of the spindle are tightened.

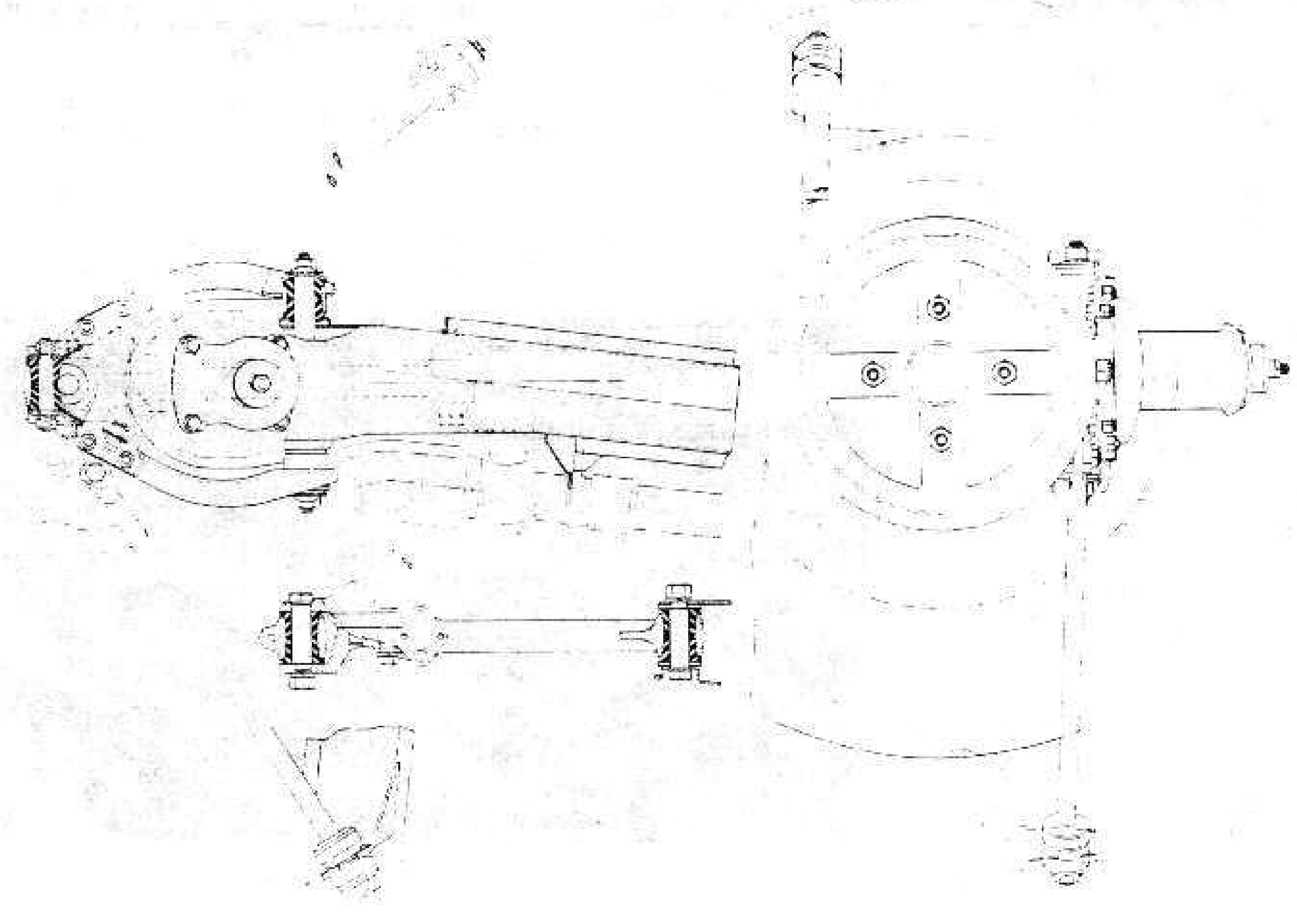
If flat retainers had been used, the excess rubber would have tended to be squeezed out of, instead of into, the bush. The pre-compression of the rubber offsets any tendency for tensile stresses to develop during operation, and therefore increases the life of the unit. A plain washer is interposed between the nut and dished retainer at each end of the spindle so that as the nut is tightened the retainer does not rotate and apply torsional pre-loading to the rubber. In the bearings, the pin is $\frac{11}{16}$ in diameter, but it is waisted to $\frac{9}{16}$ in diameter at the centre, where it passes through the frame cross member. It is shouldered at its ends to limit the amount of pre-compression.

At the outer end of the wishbone, a two-piece conical rubber bush is housed in an eye in a small En 3A forging assembled on to the swivel pin. The rubber bush is pre-compressed between the arms of the wishbone, and the $\frac{1}{2}$ in diameter pin is shouldered to $\frac{3}{8}$ in diameter to limit the degree of this compression. Both ends of the pin are threaded for the retaining nuts and spring washers. The two arms of the wishbone are held together near their outer ends by a $\frac{3}{32}$ in thick, pressed steel, channel section saddle bolted to their upper flanges.

The lower transverse link is a simple H-section forging. An eye at its inner end carries the two-piece conical rubber bush, in which is a $\frac{11}{16}$ in outside diameter distance tube. The pivot pin is $\frac{5}{8}$ in diameter. A hexagon head is formed at one end of the pin and the other is shouldered to $\frac{1}{2}$ in diameter and threaded to receive the retaining nut, by means of which pre-compression is applied to the rubber. The whole assembly is carried between two $\frac{1}{4}$ in thick lugs welded to the lower face of the frame cross member. Plain washers, also $\frac{1}{4}$ in thick, are interposed between the lugs and the rubbers. One of the two lugs is of channel section, and the head of the pin fits into it so that the nut can be tightened up without using a second spanner.

Near the outer end of the link is a platform on which the lower spring pan is seated. Immediately outboard of this, the link is forked, but is in two pieces. One piece is the main body of the link and forms one arm of the fork, while the second piece, which forms the other arm of the fork and the socket for the tubular strut, is bolted on and dowel located. A rubber bush assembly, similar to that at the other end, is carried between the arms of the fork. This bush is housed in an eye formed integrally with the lower end of the swivel pin. The head of the bolt that holds the assembly together is prevented from turning by a shoulder on the spot-face on which it seats.

As has already been stated, the eye for the outer pivot of the lower link is integral with the bottom end of the En 36 swivel pin. This pin is 0·8738 in diameter; the eye for the outer pivot



of the upper link is in a forging keyed on to a 2½ deg taper on its upper end. This forging is secured by a slotted nut on the ¾ in diameter threaded end of the pin. The swivel pin bearings are housed in the knuckle between these two ends. They are of steel backed, lead bronze and both are 1¼ in long. The bushes are assembled one from above and the other from below into their shouldered housings and their centres are spaced approximately 9·59 in apart.

They are lubricated through two grease nipples. One is screwed into an axial hole in the lower end of the pin. The grease passes through this hole and out through a radial one into the lower bush. The other nipple is screwed into the side of the knuckle and the grease passes through it, directly into the upper bush. The thrust bearings between the upper face of the knuckle and the lower face of the upper eye forging is of the three-washer type. Two of the washers are of case hardened En 2 and are pegged, one to the knuckle and the other to the eye forging. A Compo washer is fitted between them.

Integral with the knuckle, immediately behind the lower swivel pin bearing housing, is an eye into which the 5 13/16 in long, En 25W forged steel steering arm is inserted. The end of this arm is tapered and threaded for the nut which pulls it into the eye. Positive location against rotation is effected by a Woodruff key.

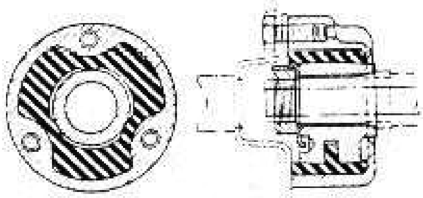
The stub axle is integral with the En 17T knuckle forging. At the inner wheel bearing it is 1·1812 in diameter and at the outer bearing it is 0·7875 in diameter. The two bearings, with a thimble-shaped distance piece between them, are assembled into the En 1 hub from each end, and the whole assembly is pulled on to the stub axle by a nut on the ¾ in U.N.F. thread on the outer end of the stub axle. A distance ring is interposed between the inner race of the inner bearing and the flange that carries the brake back plate. It is internally chamfered to clear the fillet. A lip type seal, housed in the inner end of the hub, bears on the periphery of this distance ring. At the outer end of the hub a pressed-in steel cap retains the grease and prevents the ingress of foreign matter.

A cast iron brake drum is spigoted and secured by countersunk set screws to a flange round the hub. The five, ½ in diameter wheel studs are secured to the flange in a similar manner to those on the rear wheels. Four lugs are cast near the outer periphery of each drum for balancing purposes. Unbalance can be the cause of serious vibration troubles and may adversely affect steering characteristics.

Steering

Positive control is given by the rack and pinion type steering unit, of M.G. design. A 17 in diameter steering wheel is employed. Its movement from lock to lock is 2¼ turns. On full lock, the inner wheel is at an angle of 30 deg 40 min and the outer wheel is 28 deg 50 min from the neutral position. The turning circle is 37 ft 6 in diameter.

A 1½ in diameter ×20 S.W.G. tube attached to the dash carries the steering column. At its lower end, where the column passes through the toe board, a rubber gaiter effects the seal to prevent noise, dust and draughts from entering the body. Impregnated felt, split bushes, ¾ in long, are pressed into each end of the tube to carry the column, which is a 11/16 in outside diameter × 9/16 in inside diameter tube, approximately 26½ in long. At the lower end of the column, a flanged, cupped pressing is welded on to form a companion flange for the flexible coupling.



Flexible coupling connecting the steering column to the pinion shaft

The coupling is of simple design and is noteworthy, since not only does it help to prevent the transmission of severe shocks to the steering column, but also allows for any misalignment due to flexure of the structure when traversing extremely rough terrain, such as is encountered in colonial countries. Moreover, assembly tolerances do not have to be maintained within such fine limits. The coupling is housed in a casting bolted to the flange at the lower end of the steering column. This cast casing is shaped internally to fit round a three-armed spider splined on the pinion shaft. Rubber is interposed between the spider and the casing, and the drive is transmitted from the spider, through the rubber, to the casing and thence to the flange. A nut on the end of the pinion shaft pulls the spider against split, tapered collet in a groove immediately below the splines. The split collet is retained in position by a circlip sprung into a groove round its outer periphery.

A three-piece, rack and pinion casing is employed It extends the full width between points about 2½ in inboard of the axes of the upper transverse links of the suspension system. The centre component is a 1½ in outside diameter tube. It is pressed into sockets in the inner ends of the two outer components, which are castings. One of these outer components houses the rack and pinion while the second serves only as a bearing for the other end of the rack.

Where the En 36T pinion shaft enters the casing of the rack and pinion unit, it is ¾ in outside diameter by 11/16 in inside diameter. A felt seal in a groove in the housing surrounds it at the upper end. The pinion is integral with the shaft and is near its lower end. Two En 32B thrust washers are fitted, one each side of the pinion; the upper one bears directly against the housing and the lower one bears against a bolted-on cast iron cover, which closes the lower end of the unit. The bore of this cover forms the bearing for the lower end of the shaft. A longitudinal groove in this bore distributes the lubricant over the bearing surfaces. A rubber sealing ring is fitted in an annular groove round the lower end of the bore.

The 1⅛ in diameter rack is of En 8B and the teeth that are subject to most wear, that is, those adjacent to the centre, are flame hardened. Immediately opposite the centre of the pinion a spring-loaded plunger, housed in a thimble screwed into the casing, bears on the rack to hold it firmly in mesh. This helps to compensate for wear as it takes place, but its main function is to provide a measure of damping to help to prevent the transmission of shocks.

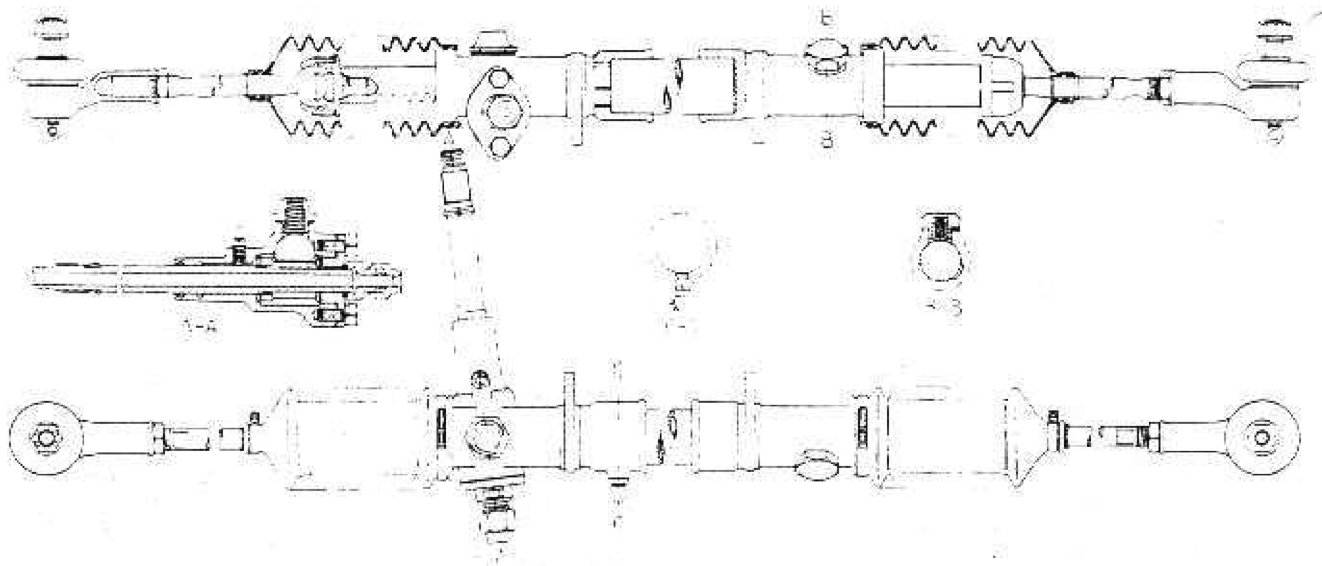
A second damper pad, of similar form, is screwed into the casing and bears on the other end of the rack. The ends of the casing are closed by rubber gaiters secured round them by clips and also clamped round the steering rods. Two lubrication nipples are fitted; one is on the pinion housing to lubricate the pinion and the other is on the adjacent rack housing to lubricate the rack, its bearings and ball joints.

The two-piece, cup component of a ball joint is screwed into a

TABLE II.

FRONT SUSPENSION DIMENSIONS

	Upper		Lower
Swivel pin angle		6 deg	
Castor angle		−3 deg	
Camber angle		−1 deg	
Toe in		Zero	
Length of links between centres	8⅜ in		12⅞ in
Inner bearings:			
Type		Rubber bushes	
Centre-to-centre spacing	5⅜ in		Single bearing
Horizontal distance from centre of bearing axis to chassis centre line	14½ in		9¾ in
Vertical spacing		8¾ in	
Pin material		En 3B	
Pin diameter	11/16 in		⅞ in
Outer bearings:			
Type		Rubber bushes	
Horizontal distance from centre of bearing axis to chassis centre line	22⅜ in		21⅛ in
Vertical spacing		9 7/16 in	
Pin material		En 3B	
Pin diameter	¾ in		⅞ in



Rack and pinion steering assembly, with scrap views showing the spring-loaded plunger arrangements: one of these plungers holds the rack firmly in mesh with the pinion and the other simply helps to damp out road shocks and vibrations

[illegible] in diameter tapped hole at each end of the rack. The balls are formed on the ends of the steering rods, which are more or less in line with the rack. With this arrangement it has, of course, been impossible to incorporate self-adjusting devices to take up wear. Therefore, shims are fitted between the two components of the cup. During assembly, the outer of these two components is passed over the steering rod and is screwed on to the inner one to pull the ball against the En 32B spherical seating housed in the inner component of the cup. The shims are fitted between the end of the outer component and a flange round the threaded portion of the inner one.

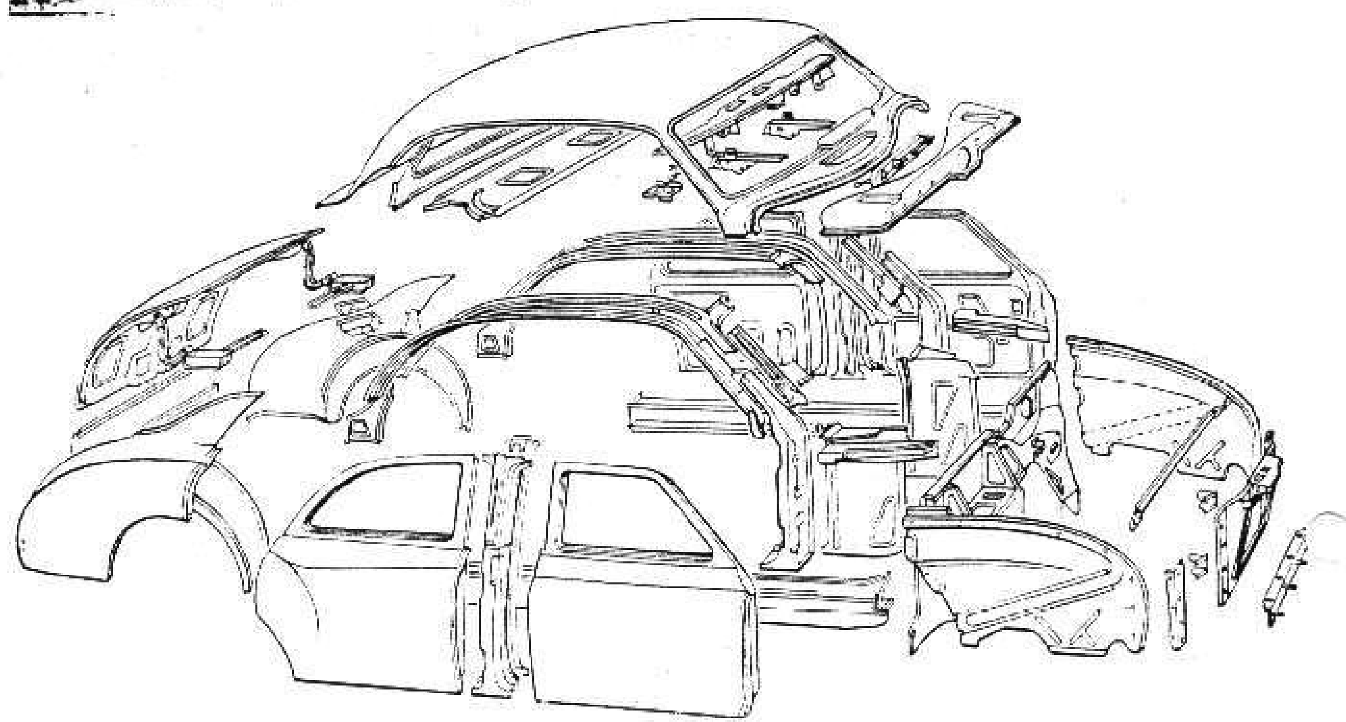
The steering rods are $\frac{7}{16}$ in diameter and their outer ends are threaded to receive screwed on ball joints of the self-adjusting type. A lock nut on the threaded portion of the rod is tightened against the ball joint fitting. Two flats for a spanner are machined on the rod adjacent to the threaded end. These ball joints are lubricated through a grease nipple. Lubrication of the ball joints on the ends of the rack is effected during assembly and further attention is not required until adjustment, by removing the shims, is necessary.

Brakes

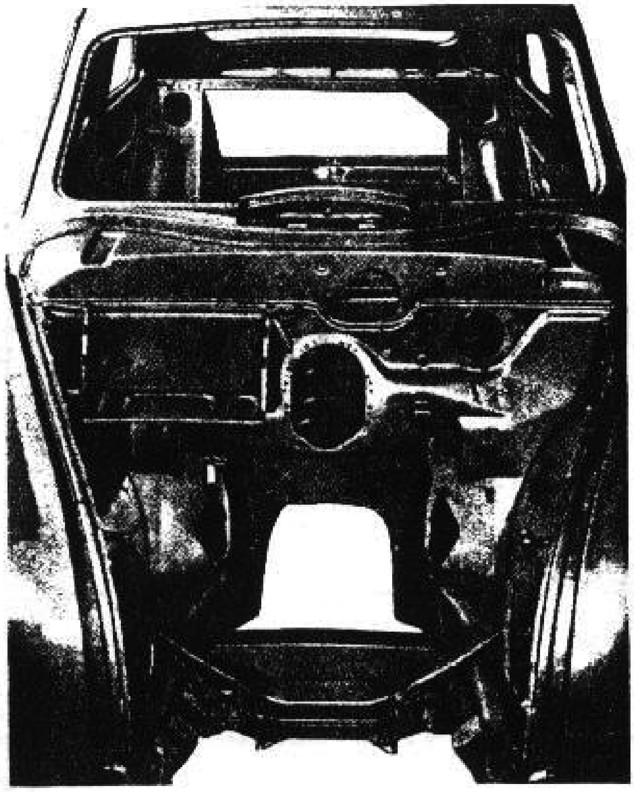
Lockheed hydraulic brakes are employed at the front and rear, with the usual mechanical hand brake arrangement to actuate the rear brakes for parking. A two leading shoe arrangement is employed at the front, while at the rear, a leading and trailing shoe layout is employed. The drums are of B.S.1452 grade 17, nickel chrome cast iron and are 10 in diameter. Slots are cut in the webs of the shoes to prevent squeal. The friction facing area is 134 in^2.

The pedal is of the pendant type with 6 in travel. Its lever ratio is 5 : 1. Compo, bronze bushes, 0·5665-0·5655 in inside diameter by 0·6705-0·6895 in outside diameter by 0·880 in long, form the pivot bearings for both the clutch and brake pedals.

The hand brake is of the pull-up lever type and is situated between the front seats. It is of the usual ratchet



type with a thumb button release at its end. The lever ratio of the handbrake is 7·3 : 1. A pressed steel yoke, the centre of which is pinned to the end of the cable from the brake control lever, forms the compensating device. The cables to each brake are, of course, attached to the ends of the yoke. All the cables are of the sheathed type.



A view looking into the front end of the body structure to show the dash assembly

Body-frame structure

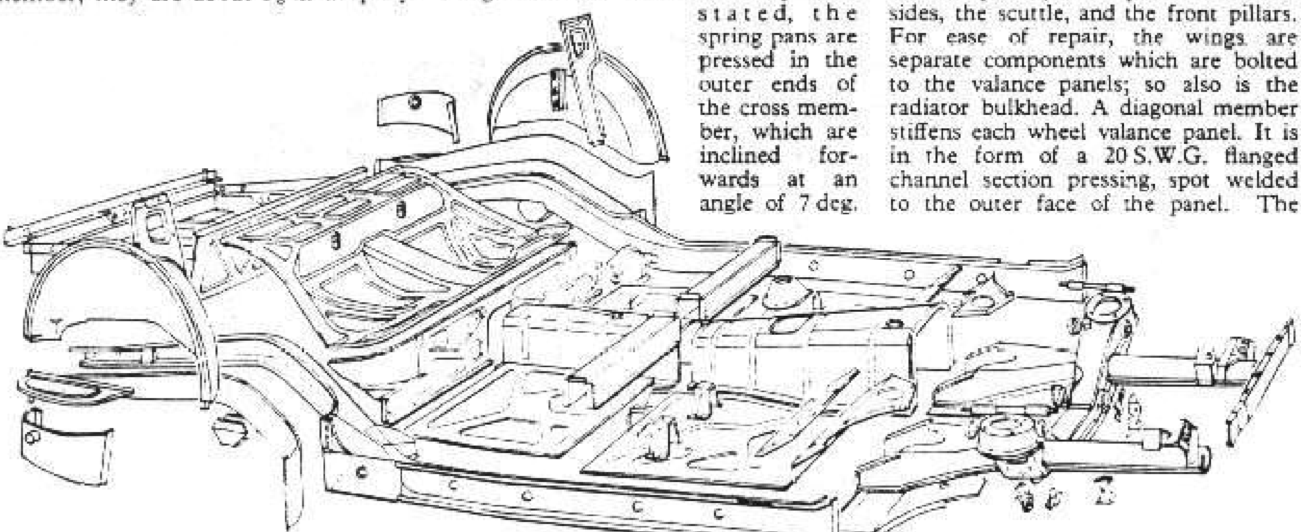
The weight of the body, which is a rigid box-like saloon structure, is 760 lb, untrimmed and without seats. The main longitudinal members of the floor structure comprise a pair of side members cantilevered forward from under the dash to carry the engine and suspension, and two more side members formed by the body sills, which, at the back, are swept inwards to clear the rear walls. A number of cross members are ingeniously incorporated by making good use of the dash, toe board, heel board and floor panels. Another longitudinal component of the structure is the propeller shaft tunnel, which stiffens the floor and helps to carry the loads between the dash and the heel board.

At the front, a 14 S.W.G. × 2½ in deep channel section cross member, with its 1 in wide flanges turned rearwards, is welded to the ends of the two cantilever side members. These cantilever members are of box section, formed by welding two 16 S.W.G. flanged, channel section pressings together at their flanges. Where they pass through the suspension cross member, they are about 3½ in deep by 3½ in wide over their flanges, but they are tapered to a smaller depth at their front ends. Welded on top of each are two brackets. One carries the engine mounting and the other the forward end of one of the brake torque reaction struts.

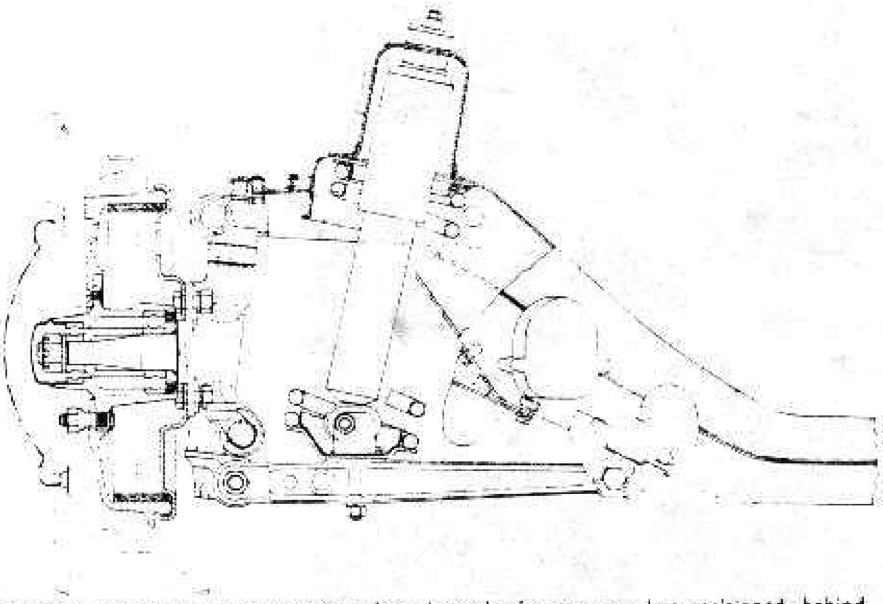
The 0·088 in thick suspension cross member is fabricated from two top hat section pressings welded together at their flanges to form a box section. At the centre, the overall depth of the member is 3 in and its width over the flanges is 4 in. As has already been stated, the spring pans are pressed in the outer ends of the cross member, which are inclined forwards at an angle of 7 deg. This sweep forwards has been adopted so that the centre of the member can be positioned further back to pass behind the sump, and also so that the rack and pinion steering gear can be accommodated immediately in front of it.

At the rear face of the suspension cross member, the lower channel section of the two that comprise each cantilever side member is terminated; the upper one is extended back under the toe board and splayed out to increase its width. Towards the rear, the top of this splayed out portion of the channel section is cut away and the upper edges of its side walls are flanged and spot welded to the 20 S.W.G. toe board. Its rear end is also flanged and is spot welded to the 2¼ in deep riser of a step in the toe board. This step forms part of a cross member. It is completed by a 20 S.W.G. flat panel welded on, to continue the ramp of the toe board down to the floor, and by another flanged plate welded horizontally under the floor, between the sill and propeller shaft tunnel, and extending forwards to close the bottom of the flanged channel section that forms the rear end of the cantilever side member. The upper edge of the toe board is extended forwards horizontally and is spot welded to the dash. This provides the necessary space above the toe board for accommodating the pivots of the pendant pedals.

The other components that form part of the front end structure are the 20 S.W.G. bulkhead on which the radiator is mounted, the 20 S.W.G. wheel valance panels, dash panel and dash sides, the scuttle, and the front pillars. For ease of repair, the wings are separate components which are bolted to the valance panels; so also is the radiator bulkhead. A diagonal member stiffens each wheel valance panel. It is in the form of a 20 S.W.G. flanged channel section pressing, spot welded to the outer face of the panel. The



The body sills and propeller shaft tunnel form the main longitudinal members of the floor structure. Extensions of the sills carry the rear suspension, and the front suspension units are mounted on a cross member welded to two box section cantilever members extending forwards from under the dash



The front suspension units are mounted on the ends of a cross member positioned behind the engine sump

function of this member is to take the loads that otherwise would be carried by the lower edge of the panel, which is not stiff enough to do so because it is cut away locally to clear the suspension components.

At the top, the dash panel is flanged and spot welded to the scuttle. Its lower edge is spot welded to the flanged front end of the tunnel, on each side of which it is pressed to form a vertical channel section, of large cross sectional dimensions, to act as a buttress to support the two cantilever side members. The sides of each channel section buttress extend down on each side of each cantilever member to which they are spot welded. The wing valances, of course, are spot welded to the dash; the dash sides are separate panels, also spot welded on. There is no scuttle side panel since the wings extend back to the front pillars, which are of top hat section, spot welded to the dash side.

Sills of large cross section are employed. They comprise an 18 S.W.G. inner member, of top hat section, with its flanges spot welded to the 20 S.W.G. outer member, which is pressed to conform with the exterior styling and to form the door shut. The overall dimensions of this section at the centre pillar are 5 in wide × 5½ in deep over the flanges. Forward of this, the sill section tapers in plan to about 8 in wide at the front pillar, but the depth remains constant. The cross section of the 20 S.W.G. front pillar is 3¼ in wide over the flanges × 3½ in deep, and that of the 20 S.W.G. centre pillar is 3½ in wide over the flanges by about 2¼ in deep at a section midway between its ends.

Mid-way between the front and rear pillars, a cross member is formed by an 18 S.W.G. top hat section with its flanges spot welded to the floor and with its ends flanged and spot welded to the tunnel and sills. It is continued under the propeller shaft by an 18 S.W.G., short box section member, the ends of which are flanged and spot welded to the inner faces of the tunnel side walls. This member is, of course, cranked downwards at the centre to clear the propeller shaft.

Another cross member is formed by the 20 S.W.G. heel board, the ends of which are spot welded to the sills. Above the level of the sills its ends sweep to the rear to conform with the radius at each end of the seat pan and are attached to the rocker panel. The floor panel and the propeller shaft tunnel are in two pieces; both are of 20 S.W.G. sheet. They are divided on a transverse line adjacent to the centre cross member. At the front, the floor is pressed to form the riser and toe board, as already mentioned. All the floor panels, including that in the boot, seat pan and the heel board, are extensively swaged to prevent drumming. Swages are pressed round the propeller shaft tunnel in transverse planes so that it maintains its shape when it is removed from the press, otherwise there would be a tendency for it to spring open.

Immediately behind the heel board, the inner component of the side member on each side is swept inwards to clear the rear wheels. It is closed by a 16 S.W.G. plate bent to conform with the sweep. At the rear, a 16 S.W.G. bracket is welded inside the section, immediately above the point where the lugs for the spring shackle are welded on. Additional reinforcement is furnished, immediately behind the heel board, by the 16 S.W.G. bracket that carries the spring eyes. This bracket is of box section, spot welded to the inner face of the side member. Immediately outboard of the bracket, the sill section, where it is concealed by the quarter panel, is completed by a separate closing plate. A steel tube is passed transversely through holes in the sill and bracket and is welded in. Welded to the rear end of the sill is the closing plate of the inwardly swept section.

A 20 S.W.G. dished pressing forms the boot floor. Its front end is bent up to form a riser to which is spot welded the rear edge of the seat pan. This riser also acts as a cross member and carries the upper ends of the shock absorbers. Another cross member is formed by the rear skirt panel. The wheel arches, with the supports for the ends of the seat squab welded on to them, are attached to the flanges of the inwardly swept sill section.

Electrical equipment

Lucas 12 volt electrical equipment is used throughout. A GTW/9A 2 battery of 51 amp-hr capacity at a rate of 10 hours is employed. It is charged by a C39/PV2 dynamo operating in conjunction with an RB106/1 voltage regulator and cut-out unit. The system is protected by two 35 amp fuses, one for the horn circuit and the other for the trafficators and other auxiliaries. The contact breaker and distributor unit is the DM2 type with a vacuum operated automatic advance and retard mechanism. It is supplied from an LA 12, oil filled coil. Other electrical equipment includes WT 614 horns and SF80 trafficators. The lamps are P700 Mk VI headlamps, with 42/36 watt bulbs, 489/1 side lamps, with 6 watt bulbs, 469 tail lamps with 4 watt bulbs, L538 stop lamps with 6/18 watt bulbs.

PLASTIC METAL FILLER

A COLD setting plastic metal amalgum called "Hermetal" has recently been tested by the National Physical Laboratory. The manufacturers of this material, the Kenilworth Manufacturing Co. Ltd., of West Drayton, Middlesex, state that these tests have shown that Hermetal has exceptional [illegible] scratches or holes in metals, but also it can be employed to build up continuous flats or curves and to conceal rivets, screws or any kind of joint or seam, whether welded, spot welded, riveted, lapped or rolled. It can be applied to iron, steel, wood or glass, and to all non-ferrous metals including [illegible] and to be non-shrinking, heat resistant and water-, oil- and petrol-proof.

To keep the material in good condition, it is packed in two tins. One contains the powder and the other the solvent. This avoids wastage, since it is only necessary to mix enough material for the job in hand. The makers point [illegible]