banks gives an inherently smooth unit because all noticeable torsional vices can be designed out and all primary forces, secondary forces and couples balanced out—(see the Appendix). Even firing intervals can be achieved but inlet manifolding is made difficult by uneven firing on each individual bank of cylinders. Alternatively a single plane (normal four-cylinder) crankshaft gives even firing on each bank and is comparatively easy to carburet but involves unacceptably large secondary vibrations.

The use of aluminium alloy is very much more open to criticism; it has the propensity, nay eagerness, to transmit noise freely which, coupled with a low modulus of elasticity and high notch sensitivity, makes it a difficult material to use for an automotive engine crankcase. A not unmixed blessing is its good heat-transfer coefficient.

In the search for silent and reliable operation a gear-driven camshaft is essential. This unfortunately meant push-rod-and-rocker valve operation with the attendant flexibility problems, even though the proposed maximum speed of the engine was only 4500 rev/min.

DESIGN

Perhaps it will be of interest to explain the designer's method of approach to the problem of starting with a blank sheet of paper and evolving the initial project design.

Before the advent of modern thin-shell bearings the areas of main and big-end bearings, coupled with torsional considerations, largely dictated the crankshaft design and hence the overall length of an engine. Nowadays allowable bearing loads have risen to an extent which makes cylinder centres the criterion of engine length—even when a two-bank arrangement with side-by-side big-end bearings is considered.

After the initial check of bearing areas and crankshaft torsional frequencies the cylinders are spaced as closely together as head studding will permit; Fig. 1 shows how closely the outer diameters of the liner flanges in the later large-bore versions of the engine are spaced.

This then is the heart of the matter and design progresses outwards from this centre.

THE 'WIDE' ENGINE

The crankcase

A fully heat-treated 4 per cent silicon–aluminium alloy (LM 8) is the material used for the cylinder block casting; in this condition a minimum Brinell hardness of 80 and an ultimate tensile strength of 15–17 ton/in^2 is attained and although reputedly it has only fair machining qualities, its excellent casting fluidity and good corrosion resistance make it one of the better light alloys for this purpose.

Upon the crankcase, together with the cylinder heads, depends the structural stiffness of the engine and, to a great extent, the ultimate smoothness of the power unit. The emphasis must be placed upon rigidity as a beam but beam stiffness is the product of the section modulus and the modulus of elasticity of the material. Since E is the unalterable factor, design effort must be confined to the attainment of the highest section modulus compatible with economic production.

To this end the skirt of the cylinder block extends $3\frac{3}{4}$ in. below the crankshaft centre line which, coupled with the extension of the front and rear walls to the full height of the cylinder banks and the tying together of the banks by intermediate ribs, largely contributes to the high beam stiffness achieved in the vertical plane.

In the horizontal plane also the skirt has been extended in width to provide additional rigidity.

This method of obtaining the requisite degree of crankcase rigidity has another advantage when the power unit, including the flywheel housing and gearbox, is considered.

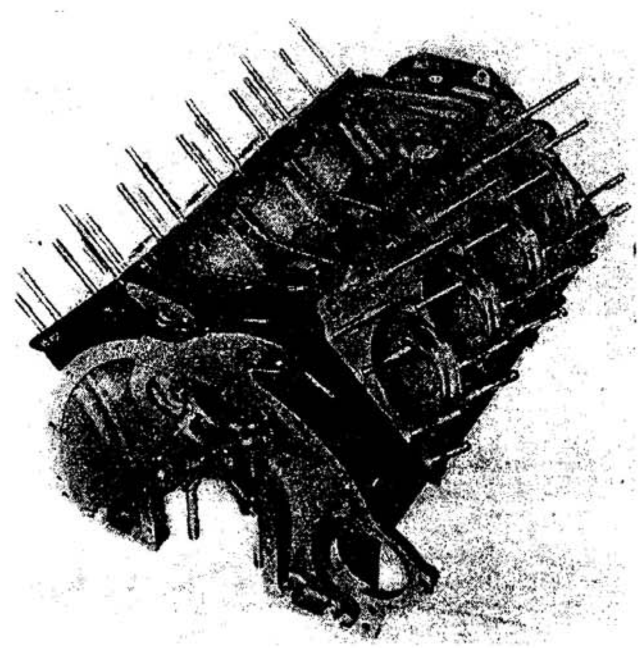


Fig. 1. Spacing of the cylinder liner bores

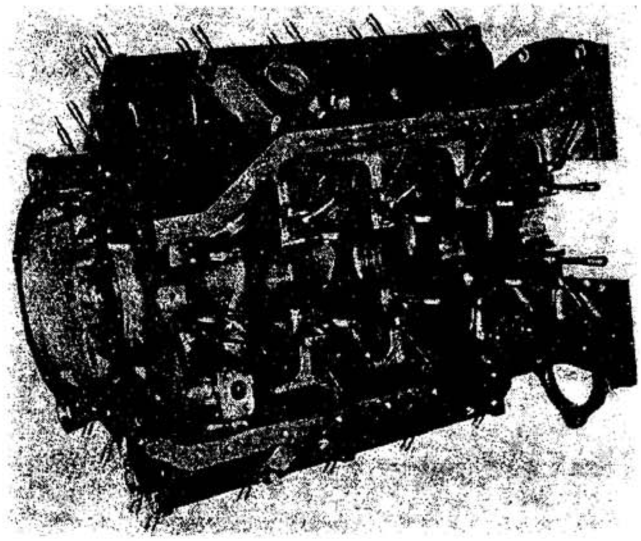


Fig. 2. Crankcase bottom face showing support buttresses

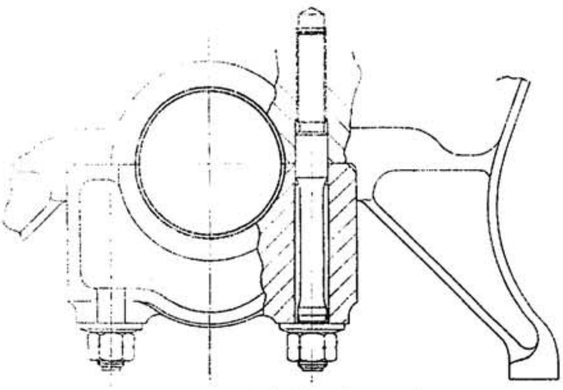


Fig. 3. Main bearing stud

The large abutment heel provided by the wide and deep rear face facilitates in turn the design of an almost conical shape of flywheel housing; thus a portion of the power unit assembly, where flexure is usually most evident, is provided with adequate stiffness at the cost of an insignificant weight penalty. Care must be taken to provide effective buttresses between the lower wings of the crankcase rear flange and the skirt if excessive deflection in this area is to be avoided and, as a corollary, the main load-carrying bolt bosses between the flywheel housing and the flange must be well supported in the carcases of the crankcase casting (Fig. 2).

All thread fixings, studs, setscrews and bolts are more difficult to engineer satisfactorily when securing light-alloy pieces than when ferrous materials are used. Interfaces where frettage is likely, such as main bearing caps and where a maximum freedom from distortion must be achieved, deserve special attention.

Consider a simple stud-fixing of a bearing cap as shown in Fig. 3. When the nut-tightening load is applied, it is well known that heavy stress concentration occurs in the threads, both male and female, nearest the interface. If the female thread is in light alloy excessive local distortion will occur; it is therefore prudent to counterbore all tapped holes to a considerable depth when dynamic loads are to be carried, thus avoiding large load variations across the mating faces.

Again at the nut end of the fixing a normal thin washer will distort excessively and indent the light alloy adjacent to the periphery of the hole to a stress beyond the limit of proportionality; the washer should therefore be made thicker and larger in diameter than usual to distribute the clamping load as evenly as possible over an area sufficient to keep the pressure to a reasonable figure; in this case a figure of about 6 ton/in^2 was chosen.

Even a conventional cast-iron cylinder block suffers bore distortion if the bolting stresses in the cylinder-head stud bosses are allowed to penetrate to the bore surface; how much more consideration must be given to head-studding when using light alloys! Only four load-bearing studs per cylinder were employed but these were designed to penetrate deeply into the structure and, in consequence, bore distortion was extremely small.

The long studs of both cylinder head and main bearing caps together combined to reduce the volume of crankcase material called upon to carry tensile loads and where material was in tension very generous sections were provided without increasing unduly the overall structure weight.

The cylinder liners, produced from centrifugally cast, high-phosphorus iron pots, were sealed by direct contact between the underside of the top flange and the cylinder-block counter bore, the 'nip' being controlled to approximately 0·002 in. Separate coolant and oil seals at the skirt end of the liner were provided by rubber rings located in grooves machined into the cylinder block. 'Tell-tale' drillings between the grooves were made to indicate any oil or coolant leakage past the rings but to the best of the author's knowledge not one case of leakage has ever occurred.

The bores were honed to a finish of approximately 30 micro-inches to provide an oil-retaining surface and the outer diameters of the liners were given a protective coating of lacquer on the surfaces exposed to coolant.

The cylinder head and valve gear

The cylinder heads, produced in LM 8 and to the same heat-treatment as the cylinder block, had a basically hemispherical combustion chamber modified to provide a squish ratio of 20 per cent and had a centrally disposed sparking plug.

A feature of the cylinder head was the very short exhaust elbows and a good deal of time was spent reducing the area of coolant over these ports in order to avoid the necessity for a larger radiator than that previously employed to cool the smaller 4·9 litre in-line engine. In fact the heat-to-coolant was reduced from 76 per cent of the power output in the case of the in-line engine to only 54 per cent in the V8.

The initial design employed only two rows of studs which were sufficient to carry explosion loads, coolant and oil transfer between block and heads being provided by

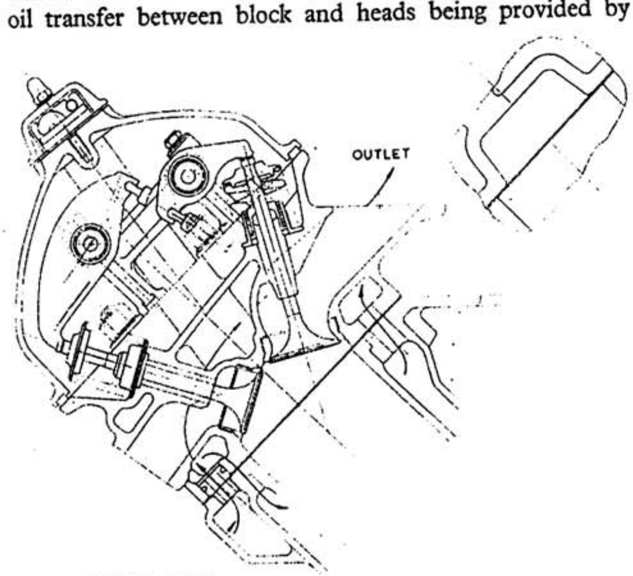


Fig. 4a. 'Wide' cylinder head and coolant circulation

bobbins with rubber O-rings but because the O-rings tended to adhere to the cylinder-head material, head removal was difficult and a further two rows of smaller diameter studs were later included with a nip-line outside the transfer area to overcome this trouble.

Later still in the 'narrow' cylinder-head design all studs were made the same diameter to avoid the necessity for different torque loadings during servicing.

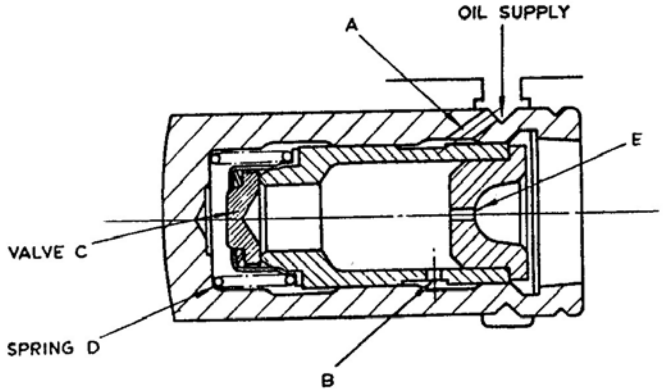
Contrary to contemporary American practice, the gear-driven camshaft was carried in four bearings and rotated in a trough formed as an integral part of the crankcase.

The objects of this departure from the more usual arrangement of five bearings and a camshaft exposed to crankcase oil-splash were twofold. Firstly, the reduction in total bearing length allowed more room for cams and tappets. Secondly, an oil-bath ensured better low-speed lubrication of the tappets when the peak Hertz stress was a maximum.

Tappet diameters were 0·875 in. having flat bottoms offset to provide positive rotation and a peak Hertz stress of 126 000 lb/in^2 based on a cam width of $\frac{1}{2}$ in. The first designs employed 'solid' tappets running directly in the crankcase material but they would have been excessively noisy and a design change was made immediately to hydraulic tappets running in cast-iron tappet blocks. The separate tappet blocks would facilitate production and maintain a more constant diametral clearance under all running conditions.

The reasons for assuming that solid tappets would be noisy were largely due to the fact that in this aluminium engine, the differential expansion condition, with hot coolant and cold oil, could account for an increase of 0·020 in. valve train clearance. It was essential, therefore, that a 'lash adjuster' or hydraulic tappet should be employed, if we were to achieve our standard of mechanical silence, as an extended ramp on the cam form would have meant sacrificing the area under the lift curve and hence performance.

The hydraulic tappet (Fig. 4*b*), as with most tappets of this nature, receives its oil from the engine lubrication system, entering the inner chamber of the tappet through holes A and B.



A, B Oilholes.
C Valve.
D Spring.
E Hole for oil supply.

Fig. 4b. Hydraulic tappet

With the tappet running on the base of the cam, spring D takes up the clearance in the valve train and oil is transferred through valve C into the lower chamber of the tappet. As the tappet reaches the flank of the cam, valve C closes and the valve train is operated through the medium of the column of oil in the base of the tappet.

An intermittent supply of oil to the rocker-arm socket is effected through the tubular push-rod via hole E, and this takes place during the period when the tappet is in contact with the base of the cam.

The use of case-hardened steel cams and chill-cast tappets, whilst entirely satisfactory in the previous in-line engines, gave rise to catastrophic failures in the V8 but an alteration to chill-cast iron cams tapered at an angle of five minutes and running against hardenable iron tappets with a 50-inch base spherical radius largely cured the trouble.

The inlet valves had a head diameter of 1·750 in. and the Stellite faced and tipped exhaust valves 1·5 in. The same cam form was used for each, giving a valve lift of 0·345 in. and a maximum acceleration at change-over of 0·0006 in./degree^2. The mean gas velocity at 4500 rev/min was approximately 200 ft/sec. Inlet valve guides were of cast iron but to avoid valve-stem scuffing troubles the exhaust guides were made of phosphor-bronze.

Valve timing was I O 23° B.T.D.C. (before top dead centre), I C 100° A.B.D.C. (after bottom dead centre), E O 55° B.B.D.C. (before bottom dead centre), E C 26° A.T.D.C. (after top dead centre) at 10 and 15 thou. clearance respectively.

Off the ramps at 0·030 in. clearance the timing was I O 8° A.T.D.C., I C 39° A.B.D.C., E O 36° B.B.D.C. and E C $5\frac{1}{2}$° B.T.D.C.

The comparatively small valve overlap chosen was necessary to reduce steady tick-over speed to a minimum, a feature essential to reduce the tendency of the car to 'creep' when the engine is to be used in conjunction with an hydraulic coupling.

The long exhaust rocker necessitated by this valve arrangement presented a major problem in stiffness per unit of polar moment of inertia. Studies were made to produce the best design of rocker, and the resulting form is shown in the drawing of the head, which also gives, in greater detail, other features, chiefly the coolant circulation within the engine (Fig. 4*a*).

The coolant circuit

Coolant issues from a double-volute pump into two cast galleries, on the upper and outer extremities of the cylinder banks. It does not at once enter the cylinder blocks but is directed immediately upward into the heads; coolant is then deflected to scour the valve seats and spark-plug bosses and leaves the heads to collect in water rails cast integrally with the induction pipe situated in the centre of the vee.

The coolant pump is designed to provide a flow of 40 gal/min at a pressure of 20 lb/in^2 and the cylinder-block flow has a characteristic of low flow rate at high pressure to reduce, somewhat, the likelihood of cavitation erosion

(To be continued)

A NEW EVENT!

Organised by the Morris Minor Club and largely motivated by the poorly organised 'Terribly British Day, I have spoken to the Citröen people and hopefully we can put on a joint display of our mutual hydraulic interests. As usual our aim is to educate the public (and probably ourselves) about Rolls-Royces always in the hope of entrapping new enthusiasts who can share the load of keeping these cars on the road.

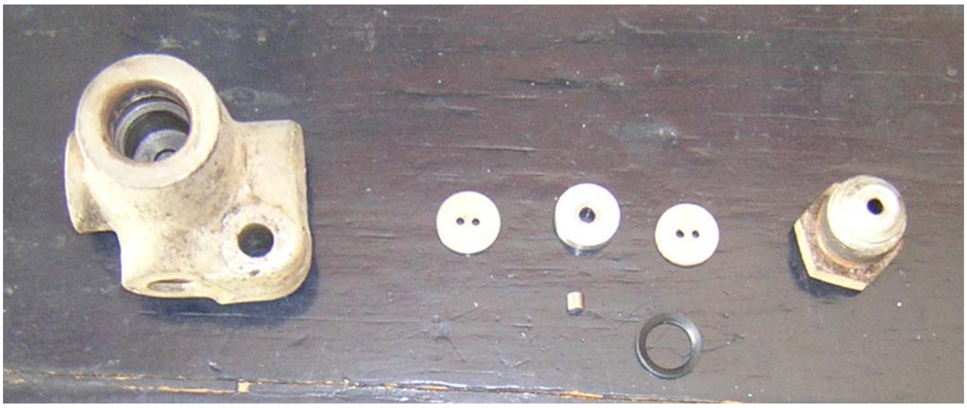
If you have any interesting bits bring them along and we can have a mutual 'you show me yours etc' and hopefully provide some material again for the public to inspect.

I also hope to produce information posters which can be stuck on the windows. It would be a help if you could be on site by about 9.00 AM to help get set up. See you there.

☸

RESTRICTED

This strange little object with the pipes protruding from it can be seen if you care to lie under your Shadow at the rear, is a hydraulic restrictor. It is one of two and among other functions

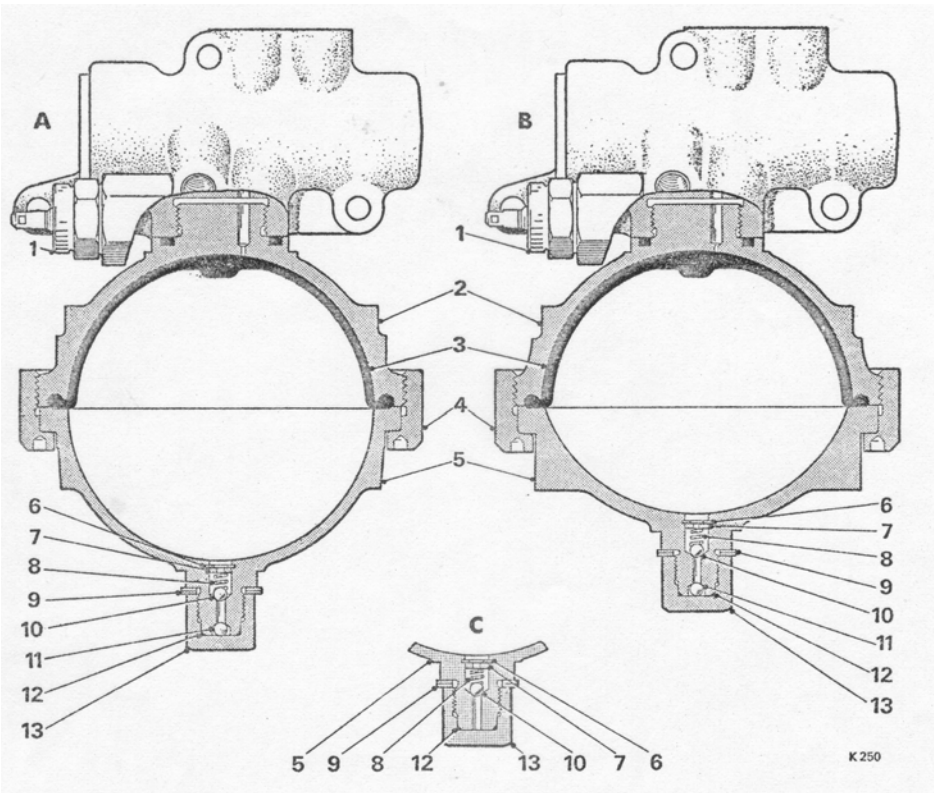


protects the levelling valves from the full onslaught of pressure from the accumulators. The picture below shows the restricted opened. It is very simple but if the slightest bit if dirt gets in there it can produce noises that no contemporary composer would contemplate. The system has to be exhausted before the pipes are undone and the system (number 2) bled afterwards. Note the tiny jiggle pin – something not to be lost!

☸

MORE ON INFLATING SHADOW ACCUMULATORS

A number of owners are intent on having the accumulators on their Shadows re-pressurised before they damage their diaphragms. Most would not bother or would want to afford the equipment to do this since it not only requires a cylinder of nitrogen but also a high pressure regulator and gauges which today cost over $700! Nitrogen originally at 1000 psi is going to leech through the 'rubber' diaphragm no matter how well made it is. So it is no reflection of the car when the number of pumps you can get out of your brakes starts dropping. The diagrams below illustrate a fully charged sphere with the diaphragm jammed hard up against brake fluid entry point. The charging valve at the bottom consists of a steel ball and light spring retained by a washer and tiny circlip. When recharging these spheres the nitrogen must be piped in slowly lest the whole ball and spring assembly be blown out of their socket. If there is any resistance it is wise to pass a small drill but up the charging hole to ease the ball off its seat. Note also the plastic crush ball on the valve which has to be replaced each time the valve is opened. The sphere on the left was originally fitted to Shadow I's (sic) and the other to II's which explains why the latter has less capacity than the early cars. It is also common to find either or even both on some cars.



WEB SITES YOU SHOULD HAVE ON YOUR COMPUTER

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A Dutch private web site with an excellent forum

All the above sites have free forums where you are welcome to share your knowledge and ask your questions. Or write to me - Bill Coburn Post Office Box 827 FYSHWICK ACT 2609 Australia or tuppercharles@bigpond.com.

If undeliverable please return to Post Office Box 827 FYSHWICK 2609 ACT AUSTRALIA