This engine started life in DB2 JUK 700 as LB6B/50/544, a 6.5 : 1 compression version with white metalled bearings but, in 1957, was converted on the Vantage specification of 8.2 : 1 compression ratio, shell bearings, twin exhaust system, etc. Cost then was £344. I parted with my 1½ litre 1935 Mark II Long Chassis Tourer, after twelve years, and took over the DB2 from Ray Eve.

by DL 'Dicko' Dixon BSC, C Eng, MI Mech E, MIEE

DB2 ENGINE OVERHAUL

At approaching 93,000 miles after the rebuild, signs of water in the oil were noticed – the usual 'mayonnaise' at the oil filler and the breather pipe. Although not uncommon in some engines where it arises from condensation, it was something quite new in this case and therefore to be investigated. The head gasket was first suspect but quickly exonerated. This suggested the worse problem of a liner seating seal which, on sump removal, was confirmed. As mileage was approaching the 100.000 figure recommended between overhauls, I decided the time had come.

The comments which follow are not intended for the experts with ample workshop facilities but for those less well equipped but willing to have a go. They are intended to supplement the official Workshop Manual '55 (WSM) and Dudley Coram's 1956 Manual (M), it being assumed that anyone embarking on such a major overhaul will at least have bought, begged or stolen one of the two!

Since the work was completed some three years ago, Brian has been bullying me, in the nicest possible way, to give my pen some exercise but with the impending move to Wales, and the thousand and one jobs waiting here, it has not been possible earlier. Fortunately I did keep notes, so here goes.

Engine Removal Where plenty of height and space are available, the engine and gearbox may be lifted out as a unit but not having either, a different approach was necessary. A simple cradle, to take the weight of the engine while the gearbox was removed through the floor of the car, was made up. This consisted of a length of 2" x 1" steel channel supported by two 2" x ⅛" steel straps shaped at their upper ends to hook over the main chassis tubes. These side pieces were loosely assembled to the channel with long bolts which on tightening could lift the assembly to free rear support. Next, having removed the head all main accessories, the clutch and flywheel were removed. Before removing the bell housing, the possible lift of the flywheel flange in the housing was measured with feelers and found to be an excessive 0.006". This was later found to be a combination of rear main bearing wear and freedom of the corresponding cheese in the crankcase. The check was made in cold January. Crankshaft end float was also checked and seen to be 0.008", within limits. Then, having removed the bell housing, it was possible to lift the block by tackle suspended from an RSJ thoughtfully put in when the house was built. Height available was just enough to allow it to be raised well above the bonnet line and turned through 90°. Having protected the windscreen, the car was pushed forward to allow the bonnet to be lowered. The car could then be pushed clear and the block lowered on to a stand.

Pistons and Connecting Rods. I was surprised to see that several big end split pins had been incorrectly fitted – they did not lie in line with the crankshaft. There is always the risk of movement and wear if they are fitted to lie across the shaft as it were. When dismantling the big ends and extracting the pistons and connecting rods, particular care was taken to identify each individual part and to note which way round each rod and piston was fitted – were the stamped identity numbers to the near side or off side? Nuts were replaced on the bolts from which they were removed.

The good condition of the pistons, rings, little ends and rods exceeded all expections. In fact the only essential replacement was one segment of an oil scraper ring on piston No. 5. This was broken into two equal parts suggesting that it had been overstrained on fitting. All other rings had excellent faces but the top ring of each piston was replaced as the gap, measured at the cleaned top of the liner, was about 0.027", i.e. approaching the maximum recommended. Also the groove clearance of these rings was about the maximum, 0.006", whereas that of all others did not exceed 0.003"*, well clear of the 0.010" lower limit.

Cylinder Liners Cylinder bore wear is an important factor, revealing the need or otherwise for liner replacement. Internal micrometers are not the easiest of instruments to use but a simpler method is to hand if you have a spare piston ring available. Or better still, a scraper ring segment. Place this squarely in the cleaned top of the liner and measure the gap with feelers. Measure again with the ring pushed down to the position of maximum wear. Push the feelers firmly into gap so as to expand the ring into the space available. (Not normal practice with feelers!). Note the difference in readings, then divide by 3.1416, say 3, and you will have the diametrical bore wear. In my case the difference was only 0.003", implying a bore wear of 0.001". Most gratifying!

Crankshaft. Before the crankshaft can be extracted from the block, it is necessary, as the WSM puts it, to 'Unscrew the starter dog nut – –'. This may prove to be a far tougher nut to crack than the instruction implies! A good fitting tubular spanner is needed, long enough to be supported at its outer end on a stout wooden block. The 'tommy-bar', ½" – ⅝" dia. needs be about 12" long. A sharp blow on the end of bar, operating against the inertia of the crankshaft, may free the nut. But not so in my case. I did not favour applying excessive blows to be resisted by twist in the crankshaft, so packed the front web of the shaft and No. 1 big end to the crankcase sides with hardwood blocks and wedges. Then, with someone else applying an initial strain to a steel tube slipped over the tommy-bar, a few heavy blows did the trick. Having removed the dowels and oil transfer bolt, removal was straight forward. As suspected, the rear cheese was free in the housing but the other two required progressive tapping via a hardwood rod to ease them out.

Having removed the cheeses, it was apparent that the crankshaft journals had all been reground 0.010" undersize during the 1957 rebuild. All the shells were in reasonably good condition with the exception of the rear main which showed wear markings. The main journals were of excellent appearance and measurement with a precision vernier gauge showed each to be 2.489"d. A more accurate determination could not be made at the time as a suitable micrometer was not available. The WSM gives the standard dimension as 2.49875" (minus. 000"). Take away 0.010" for the regrind, leaving a minimum size of 2.48875". The actual regrind size is of course unknown. Suffice it to say there had been damn all wear!

The big end journals all had similar excellent surfaces, measuring by vernier 1.989"d. Fortunately, as ovality and taper were to be checked, a suitable micrometer was to hand. The standard crankpin diameter is given as 1.99875" minus 0.0005" i.e. a minimum of 1,99825", which on allowing the 0.010 regrind, becomes 1.98825". Each journal was measured in four positions, two to the front to give minimum and maximum thence ovality, and similarly two to the rear. The difference between these front and rear readings gave the taper. The minimum of all the measurements taken was 1.9885" but here again the actual regrind size is unknown. Three crankpins were without taper, two showed 0.0001" and the fourth 0.0002". Ovality ranged from 0.0001" to 0.0003" with four at 0.0002". Clearly nothing to worry about here. The Crankshaft was thoroughly cleaned, crack tested and the journals lightly burnished before oiling and putting away until required.

Removal of Cylinder Liners. Before commencing I decided to find out exactly where the leak had occured. The head was refitted and the water outlet blanked off by a steel plate to which a tyre connector had been fitted. The block inlet was also blanked off. It was then possible to pressurise the water ways to 15lb/sq in with a tyre pump. Soap solution quickly showed the leak to be between Nos. 5 and 6 cylinders.

Great emphasis is given in the WSM to the importance of fitting the liner securing plate when the head is removed and the crankshaft need be turned. However, this does not apply after 90.000 miles! Accumulated rust and deposit necessitated considerable force to unseat them all, care being taken to hold a block of wood squarely across the lower mouth of each before striking with a 2lb hammer. 'Lift out' indeed!

Liner Positioning & Sealing. These liners are seated on a flange at their lower ends,

* The new rings reduced the groove clearance to 0.004" & the gaps to 0.016", well clear of the 0.010" lower limit.

height adjustment and sealing being provided by copper gaskets of various thicknesses, the whole being clamped down by the head and gasket. So far, so good. But then W.O. must have suffered a severe mental aberation in deciding that the liners must seat in pairs on figure eight gaskets! Even the draughtsman, preparing the Spare Parts List could not believe it! He shows separate rings. Look at what this implies. The liners must be in matched pairs for length from top to seating face and the seatings in the block must be equidistant from the top face. The lapping-in called for must not upset any of these relationships. (WSM pg. 83). To complicate matters, the adjacent seatings in the block overlap so that any removal of metal by lapping from one seating, removes metal from the other **but only in the area of overlap.** Thus, if the block seatings are perfectly matched initially but one liner is a thou or two longer than the other, lapping it down on its own seating will cause a step in the other seating. The lapping itself is a tricky procedure. There is nothing to get hold of and the liner must be kept upright in the block. A chunk of rubber pushed into the liner can form a handle of sorts and pieces of shim material inserted in the clearance between liner and block at the top will assist in keeping the former square. Comparative lengths, and squareness, of liners are easily checked by rotating each, top downwards, on a surface plate with the probe of a dial gauge resting on the sealing flange. It is all a question of trial and error, requiring much patience.

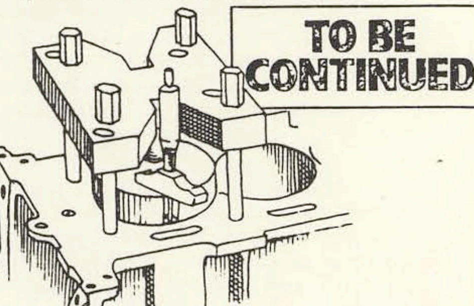
In my own case the problem was aggravated by the fact that No. 5 liner was almost 0.004" longer than No. 6. Furthermore, careless machining had left a gouged depression running part way round the seating. These faults were doubtless the cause of the leakage. To avoid some of the secondary effect mentioned above, No. 5 was not lapped directly on its seating but on a spare gasket placed on the seating. When equality with No. 6 was attained, both were lapped on their respective seatings. Similarly, to avoid upsetting liner lengths where a block seating requires matching, a discarded liner, or as in my case a dummy, may be turned up, and used as a lap initially.

We now come to the most contentious of all aspects concerning this class of engine – the projection of cylinder liners above the top of the block. This subject has occupied the minds of several members over the years as will be seen by a perusal of the AM Mags and New Sheets – all arising from the uncertainty of securing a good head seal if the 'official' instructions are followed. Starting from scratch as it were, with reseated liners, I had to decide on the best compromise in the light of this experience bearing in mind that I had not had a gas blow out or water in the cylinders since I had had the car. More recently our member Alan Wheatley has put his thoughts to me and I find we are in close agreement in our assessments of the problem.

First, look at the WSM instructions which in effect say – using a dial gauge, measure the discrepancy between liner top and block surface. Add 0.004" and select gaskets to bring the liner by this amount proud of the block. These are stated to be available in 4, 6, 9, 12 and 14 thou thicknesses. (Note, these sizes are listed in by my Spare Parts List as applicable to the VB6J 3 litre engine. Those listed for the 2.6 litre are 9, 12, 14, 15, 17, 24, 28 and 32 thous and not mentioned in the WSM).

With gaskets fitted dry, pull down the liner with a 'special tool', listed on pg. 29 with applied torque of 25lb ft. The liner top should then be flush with the block top. It not, reselect gaskets to comply. (This is easier said than done with the thicknesses available. Further lapping may be necessary.) This special tool is not described but by all accounts is similar to that detailed for the VB6J engine by Alan Puckett in his article appearing in AM Vol. 10, No. 26 of 1966, pg. 75 for the VB6J engine. It consists of a 5½" square steel plate, 1½" thick, drilled to fit on the four studs adjacent to the liner being checked and to apply load, through a ¾"d. steel ball, to a circular steel plug ⅞" thick shouldered to register in the opening of the liner. The plug has a centre dimple to locate the ball. The upper plate is cut away to allow access for a depth micrometer or dial gauge probe. Cylinder head nuts are used to apply the specified loading. Check that the dimensions will suit your own engine.

It is of interest to speculate on how the 0.004" is to vanish. The four studs, tensioned to 25 lb ft each, will produce a total load of about 6 tons. This will compress the liner itself by about 2½ thous. the other 1½ thou. is lost in block stretch and deformation, and possibly some compression of the seating gaskets. On checking used gaskets I have not been able to find any significant change of thickness from the specified sizes. The block is of such irregular shape that any direct calculation cannot be made.



TO BE CONTINUED

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Liner positioning + sealing Cont'd:

by DL 'Dicko' Dixon BSc, C Eng, MI Mech E, MIEE

DB2 ENGINE OVERHAUL

In the final assembly additional factors become involved. The load applied through the head gasket, 50 - 60 tons when the 14 head nuts are 'torqued' to 65 lb ft, is distributed but not evenly, over the whole contact area. The initial projection of the liners is provided to ensure, hopefully, that the top seal will withstand the fired gas pressure which is likely to be between 500 and 1000 lb/sq in. The oil and water pressures are negligible in comparison. (Assuming no immediate water leak on assembly, any water leak into a cylinder is almost certain to have been preceeded by a gas blow-through which may usually be confirmed by bubbles appearing at the radiator filler. Water will seep through on standing and be drawn in during the suction stroke when running.)

Assuming the ideal case of perfectly flat block and head surfaces and liner projections equal at 0.004", if the head is then fitted, for the purpose of this argument, with an unyielding gasket, the initial 36 tons applied should just bring the liners flush with the block leaving the balance for general oil and water sealing. However, this additional force of say 20 tons, **will not add to the sealing force on the liner tops.** They are already flush with the block top and the sealing load can only be increased if they are depressed **below** the flush position. Thus if a higher sealing force is agreed to be necessary, a projection greater than the recommended 0.004" is essential. The question is – by how much?

In practice of course, ideal conditions never exist, particularly in engines 20 to 30 years old, although the old Aston racing department aimed for near perfection by hand lapping each head to its block before fitting the liners. This is where the head gasket comes in as the means of absorbing small irregularities without loss of sealing ability. The standard for these engines is the laminated copper type with each orifice edge sealed by forming the outer lamina of one face through the orifice and folding it over the opposite face. A copper-asbestos-copper types does exist but should not be used. (One was included wth a lot of spares I bought from a member in 1976, origin unknown).

Various liner projection figures have been suggested – even up to 0.010"/0.015"! There are practical limits. The more force diverted to seal the liners, the less left available to make effective oil and water seals. Slight weeping to the outside of the block, although undesirable, may not be serious but there may be risk of distortion of the liners. The most vunerable spot for oil leakage is at the camshaft feed joint just forward of No. 1 liner – but more of this later.

Assuming the above are figures acceptable in giving some idea of the forces involved, each extra thou of projection should require an extra 1½ ton loading per liner – 9 tons in all – to bring the liners flush, which represents a 25% increase in sealing force. However, this is unlikely to be attained in full as a result of gasket yield. Weighing it all up, I do not think it adviseable to go beyond an extra couple of thous for normal use. This would mean that those with the preloading tool available should be looking for a projection of 0.002" with the 25 lb ft torque applied. It would be inadvisable to increase the torque to bring the liner flush as this would stress the liner more than when head and gasket are fitted.

My own liners were refitted before making the study outlined above but having in mind the general feeling that 0.004" was inadequate and, on the other hand, that I had had no trouble with top sealing at any time, I aimed for 0.005". Not having a preloading tool available, I made up a simplified version consisting of a turned plug, with centre hole, to fit the liner top, through which was passed a threaded ⅜" steel rod down to a length of channel steel across the bottom of the crankcase. A spherical faced washer was inserted between the plug and the top nut. The object of this was to make a check of projections with all waviness of the gaskets flattened out.

All were then assembled using RR Hylomar Unipart jointing fluid. (recommended by Aston Service Dorset.) This was very messy to apply having a jelly like quality and no tendency to spread but the job was made easier by fitting a small bore tube to the container cap. An empty ball pen refill with ball end ground off did very well.

This leaves the question of what one should do when refitting the head after a straightforward decoke, the liners being undisturbed. Doubtless the liner retaining plate is a very wise precaution if the head is lifted for any reason shortly after they have been fitted. In my case, and others I know of, the liners were firmly held by rust etc. and did not appear to expand upwards. For this to happen they must have been held at their upper ends but it is also likely that both block and head take on a degree of permanent set when subjected to high stress for considerable periods. This would reduce the effort to break out. The development of these sets is confirmed by the need to resurface. Paul Jackman, a development engineer in the racing dept. at AM during the fifties, regards a gap of 0.002" shown by straight-edge and feelers, as the sign that a head requires resurfacing. The tendency to deform after resurfacing will be reduced.

I have usually followed the WSM and fitted a new gasket after decoking, with one exception when I re-used the old. As this gave no trouble in the ensuing 40.000 miles I see little point in disturbing liners without other good reason. In all cases I have used Red Hermetite thinly applied to all faces. Should doubt exist, or a blow-through occur, I pass on a tip from Paul Jackman – place one of the thinnest sealing gaskets, 0.004", at the **top** of each liner.

Crankshaft refitting It had been my intention to fit new bearing shells from the start and a new complete set had been obtained from Aston Service Dorset as soon as the required size was known. Having found the crankshaft in such excellent condition, straight replacements, 0.010" undersize, were ordered. There is no doubt that most of the removed shells could have run on for thousands of miles – but wisdom prevailed. The new shells were fitted using the existing housing – 'cheeses' – except the rear where a new one was fitted. The front bearing shell was drawn into position using the same threaded rod and washer assembly used to remove the old, great care being taken to see that the location tab lines up correctly. The shell must lie central in the housing so that the projection at each end will centralise the crankshaft end float washers without protruding beyond them.

The easiest way of refitting the crankshaft assembly to the crankcase is to stand the latter on end, front downwards and to lower the shaft into position by suitable lifting tackle. The cheeses are slightly stepped in diameter, the rear being the largest, so all three attempt to enter their housing simultaneously. This can be more than a handful if attempted in the horizontal position. **Check that the inner thrust washer is in position** before lowering. Take particular car that each cheese will line up with the oil transfer plug which passes through the crankcase side and also locates the cheese. It may not be possible to rotate the cheese into line once in its housing. In this position the cheeses may be tapped home progressively and squarely using a hardwood or brass rod in a very controlled manner.

Pistons and Connecting Rods I have already commented on the very good first impressions obtained on these items when removed. Closer examination before cleaning showed very symmetrical marking as between the thrust areas and those adjacent to the gudgeon pin ends. There was no perceptable rock in the little ends and the pins were firm in the pistons, so I decided to leave well alone. As a matter of interest, the piston bottom skirt clearances were measured with feelers when inserted into the cleaned liner tops and found to be 0.004" at right angles to the gudgeon pin and 0.008" in line with the pin. The latter figure seems to be a little small according to the WSM pg. 21. but this obviously has been of no consequence.

Reinsertion of the rods and pistons into the bores is no problem provided a good ring compressor is used. To make the attempt without is not worthwhile. As the big end of each rod approaches its crankpin, make sure that it is the correct way round. No problem was encountered with the new shells which had been previously assembled and tried for freedom on the crankshaft. The nuts, after fitting in the individual previous positions, were taken up to the specified 30lb.ft and the position of the split pin slots noted. In several cases it was necessary to take the slightest skim off the nut face to allow the split pin to be correctly fitted. This was done in the lathe as it is very difficult to hold the nut square if rubbing down by hand is attempted. The face was given a trace of convexity to match very slight depressions observed on the mating faces of the big end caps. New nuts would have required similar treatment otherwise they would have seated on the corners of the flats. The alternative to this procedure would have been facing off the big end caps but this was not justified.

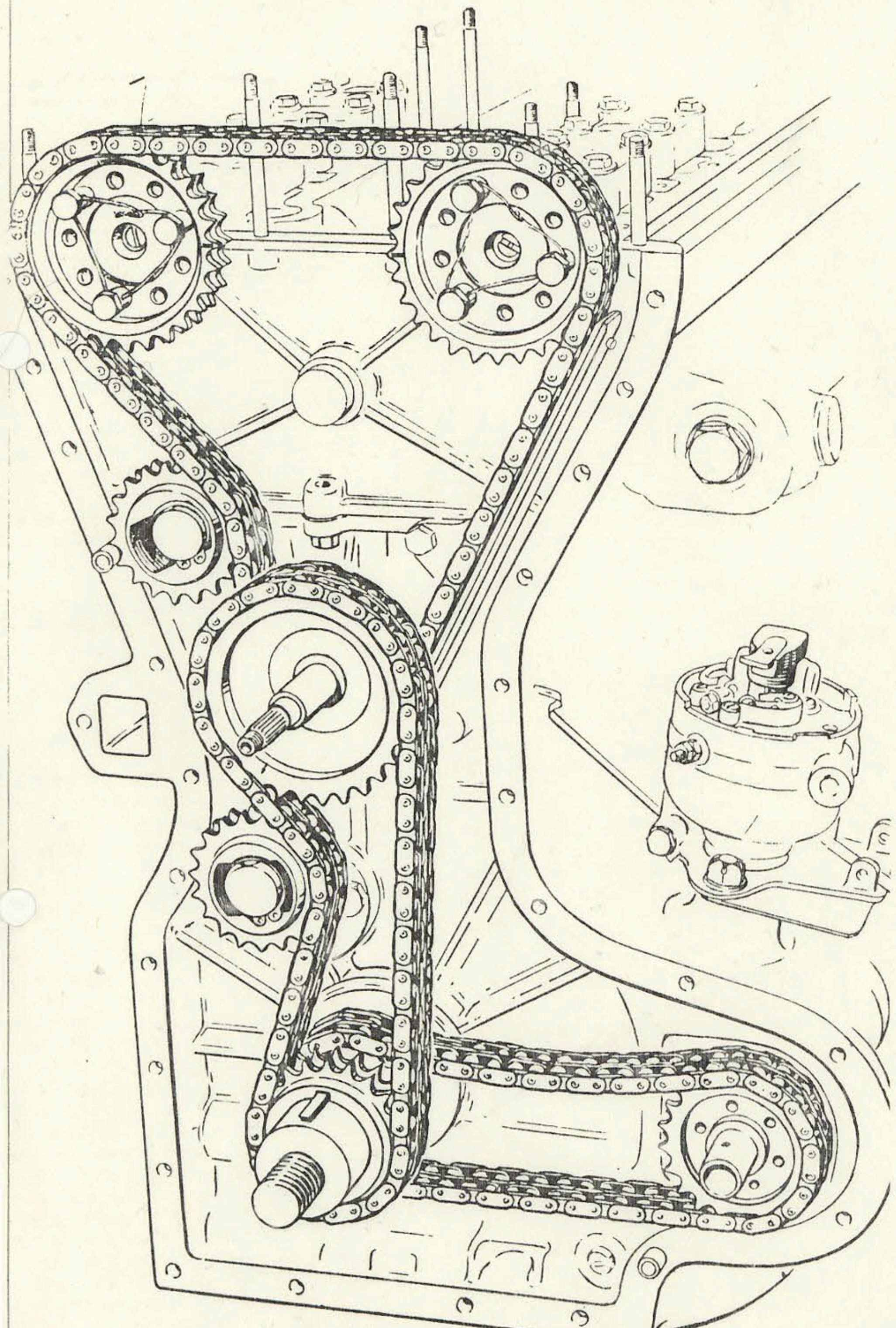
Camshaft, Oil Pump and Distributor Drives. In the original layout, as shown in Plate IV. of my DB2 Instruction Book, the final drive chain of 104 links passed over an idler sprocket just below and midway between the two camshaft sprockets. An oil hydraulic tensioner was positioned to engage the inner side of the chain below and to the left of the idler. This layout was simplified from engine No. VB6E 1269 onwards in that the idler was eliminated and the tensioner applied to the outer side of the chain which was reduced to 84 links. The timing case cover casting was unchanged and retained the provision for the discarded idler sprocket shaft. This may have puzzled some owners. The 1956 Manual Plates XV and XVI clearly show this. Incidently these two Plates refer to the DB2 and 2/4 2.6 litre engine, not the 2.9 litre as stated. Note also that the forward end of the camshaft cover is open to atmosphere – the draughtsman's aberation this time! My engine was modified to this revised drive during the 1957 overhaul.

As I had new sprockets and chains available I decided to fit them although only the chains had signs of wear. Normally one does not fit new chains without new sprockets. However, comparative diametrical measurements, taken across silver steel rods held between the teeth of the removed and new camshaft sprockets showed no sign of wear. Remarkable! So the old sprockets are held as spares for the next time.

The drive to the oil pump and distributor is via a 56 link chain to a 'gear shaft assembly'. A six tooth skew gear is integral with the shaft, also a flange to which the chain sprocket is riveted. The components of the assembly are not listed separately in the AM Spare Parts List but are as follows:–

A54749	Gearshaft
A50071	Sprocket (21 teeth)
111218	Rivet ⅛"d. (6 required)

The skew gear engages with a twelve tooth skew gear which is integral with the oil pump shaft and gives the 2:1 reduction required by the distributor. These gears are known to require replacement after considerable mileage and such was the case with mine.



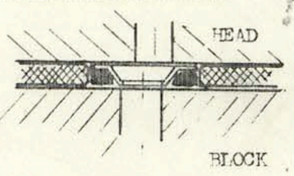


Fig 3

Camshaft Oil Feed Seal.

Flatten upper edge of conical washer and rub down flat before fitting.

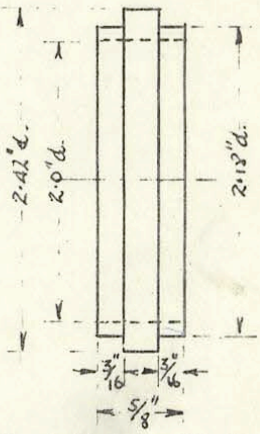


Fig 5.

Camshaft Cover Oil Seal.

Turn from steel tube or aluminium alloy.

Fit 'O' ring dry to each shoulder, section 0.210", 2 1/16" I.D.

Ref. PS 330 Oil proof.
Weston Ref. W21021
James Walker Ref. JW 50/330

Smear grease in cavities to ease fitting.

The Oil Pump As emphasised in the Manuals, this component requires considerable care in overhaul. The gear end running clearances are most important as after standing, perhaps for considerable periods, the pump must provide sufficient suction to draw the oil from the sump and to generate adequate pressure at the required output. In my case the axial end play of the steel gear keyed to the shaft was 0.002″ and that of the aluminium driven gear 0.003″ to 0.004″. These checks were made with feelers using a straight edge across the top face of the pump body. Lengths respectively were 1.9966″ and 1.9950″. The replacements were lapped to 1.997″ to give a nominal clearance of 0.001″ after wear grooving in the cover casting had been lapped out. Although the old gears showed definite attrition wear – they are not protected by the oil filter – the pump body was in good order and not replaced.

In spite of the Manual exhortations regarding meshing, there is little one can do to 'ensure correct clearances' even if these were specified. The gear centres are fixed and at most one can only carefully remove any tooth edge burr and assemble in the positions giving best rotational freedom. The gear ends may then be lightly marked to identify this relationship which, con[illegible] to normal practice with gears, is main[illegible]ned thoughout their operational life. The Manuals also recommend testing before refitting – but how? And to what specification? I would suggest care – and crossed fingers!

Flywheel and Clutch The flywheel was refitted in the same angular position as when removed although there was the temptation to move it round by 90° to bring unwarn starter ring teeth to the positions of maximum wear. To have done so would have necessitated rebalancing and drilling of another timing pin hole in the new correct postition. (T.D.C. determinations by piston top movements are not very accurate for translation to angular movements on the periphery of the flywheel.) Tooth wear was small in any case. Remarkably, the flywheel face showed only the slightest trace of wear, only a degree of polish where the clutch lining had engaged. True, some oil had got through the rear main bearing into the clutch housing during the later stages of the ninety odd thousand miles but this could not be the whole explanation. Racing starts, sloppy gear changing and hanging on the clutch, would have produced a different story!

A note is necessary here on flywheel sec[illegible]g bolts. The WSM gives a torque loading of 40 lbft as being applicable to the ⅜″ shouldered bolts of the DB/A engine. No mention is made of the 5/16″ nut secured bolts of the 2.6 litre engine. In the 1956 Manual pg. 205, correct reference is made to the 5/16″ nuts of the LB6 and VB6 engines but the shouldered bolts used on the DB Series are given as 5/16″. They should of course by ⅜″. Further, a torque figure of 40 lb ft is specified indiscriminately. This is excessive for the 5/16″ nut. A figure of 25 – 27 lb ft would be more appropriate.

In contrast, the clutch itself was in less satisfactory condition. One spring was broken, and the various operational parts showed signs of wear. Some years earlier, clutch pedal 'flutter' became noticeable which was traced to a badly worn pivot bush in the operating fork. This had thrown the thrust assembly well off centre resulting in very uneven wear of the carbon thrust ring. A new bush and thrust ring cured the trouble. The clutch itself did not appear to have suffered in any way. However, in view of the poor condition, and the fact that a new Borg & Beck cover assembly was readily available from a local firm at a very reasonable price, I decided on replacement. This new clutch carried the number 45692/15 whereas the replacement number given in th 1956 Manual pg. 300, is 45692/14. I therefore phoned B & B and was told /15 had superceded /14 as the direct replacement. On the question of balance I was assured that the cover assembly had been pre-balanced to suit any rpm my engine was likely to be run at. This of course may not hold good for the complete assembly of crankshaft, flywheel and new clutch and strictly speaking I should have had the whole assembly rebalanced. At the time I could not face the prospect of dismantling again so proceeded to fit the new cover assembly having had the existing centre plate relined.

It is essential that the centre plate be gripped in a concentric position between the flywheel and the clutch pressure plate during assembly, otherwise it will be impossible to insert the gearbox primary shaft through the centre plate and into the recessed ball race in the flywheel. The trouble of turning up a dummy shaft as recommended is well worth while.

The Head and Valve gear When dismantling the engine great care was taken to identify all parts, either by marking if not already numbered, and keeping in separate groups. All were thoroughly cleaned, the plugs being removed from the camshaft bores to enable all sludge to be removed. The camshafts provided a surprise or two. The inlet was found to be marked Part No. 50421 as listed, but the exhaust was No. 50422, listed as the inlet for the VB6J 3 litre engine! However it seemed to work very well so my only initial reaction was a raised eyebrow. We all know how individual these Astons can be! The cams were in excellent condition and the tappet thimbles had mirror like surfaces. Not a trace of scruffing.

Each camshaft was assembled dry in its bearings and the clearance lift in various positions measured with a dial gauge. The inlet lift was about 0.001″ throughout and the exhaust varied from 0.00075″ to 0.002″ at the drive end. All journals were in good smooth condition. Wear was barely half a thou as measured across worn and untouched ground diameters. As the reactive load from the valve operation is taken by the bearing caps, they are likely to be subject to wear shown by the lift checks. Local arcs of clean surface showed where the load had been taken. On this basis it was decided to follow the expedient, not acceptable with big end caps, of lapping the seating faces of the caps where necessary to reduce the lift to not more than 0.001″ the minimum specified. Each shaft was checked to turn by hand with all caps tightened down.

All valves were in reasonably good condition as removed. Stem and guide wear was not considered sufficient to justify replacement although the inlet clearances will merit closer attention at the next decoke. Interesting to note that the inlet guides were fitted with oil seal rings as specified for the later 3 litre engines. These were replaced. If one has a lathe available, the valves may be freed of deposits using a lightly held blunt hand tool and burnished with a fine grade of glass paper which will not harm the valve surface. Exhaust valve tops will usually require a tougher treatment. After cleaning I had the exhaust valves refaced before the normal grinding in. The refacing angle is 45° which does not appear to be given in either manual.

On valve clearances, both the WSM (pg.66) and the Manual (pg. 197) are a little misleading. I find the initial check is most conveniently made without the springs being fitted provided the valve in question is firmly held to its seating. Pop the tappets into their correct positions, fit the camshafts and secure the caps, then, with the head on its side, insert the inlet and exhaust valves of one cylinder into their guides. A domed block of wood, shaped to hold the valves on their seatings, is held in position while the head is turned to rest on the block. Keep the head level with packing as may be necessary. The clearances are then measured and recorded progressively.

Length adjustment may be quickly made by lightly offering the valve stem end squarely to the side face of a fine grinding wheel whilst held and rotated in a V block. Aim to provide the top limit of clearance as this will allow a thou or so for settlement. If the internal contact face of the tappet shows a slight depression, slightly chamfer the end of the valve stem before grinding to length and making your check clearances.

Valve Timing Here we come to the pièce de résistance of the whole exercise. No trouble of course if you happen to have the correct timing jig and your camshafts are not mixed!

Before going further it is necessary to dispose of a confusion which appears in the WSM (pg. 67 et seq) and has been perpetuated in the Manual (pg. 199 et seq). This concerns the reference to 'firing stroke' when setting the valve timing.

Both inlet and exhaust valves must be firmly closed at the moment of firing and the cams nowhere near the tappets. (See Plates XVI and XVX of the Manual and do not be misled by the illustration on pg. 199 – if the exhaust valve has been timed first as indicated by the instructions, the exhaust cam will be nowhere near the position shown. Also the diagram on pg. 198 is too incomplete to have significant meaning.) It is not until after the firing stroke that the valves are required to operate. The exhaust will open at the end of the stroke and normally close slightly after TDC is reached and the inlet start to open slightly before. Thus both valves are a little open at the same time; usually refered to as 'valve overlap'. (This is necessary to assist gas flow into the cylinder at normal running speeds.) Thus, to speak of valve timing on the firing TDC is quite misleading to say the least. There are two convenient alternatives. First, set No. 1 piston at TDC at the end of the exhaust stroke, and with distribution rotor facing rearwards, and proceed as instructed, or secondly, set No. 1 piston at TDC on its firing stroke with rotor facing forwards, and carry out the timing procedure on No. 6 cylinder where the valves will then be in the overlap positions.

To resume, but first some basic data. Unlike the DB engine, no flywheel degree markings are provided on the earlier engine flywheels. This omission may be rectified by scribing a line on the rim, where it is accessible through the starter bulge in the bell housing, to line up with a temporary pointer, secured to the bell housing. The timing pin for TDC should be in position when this is done. The flywheel diameter, in my case, is 11 3/16″ i.e. circumference = 35.15″. This is equivalent to 10.24° per inch from which –

8° = 0.78″ or 19.8 mm
14° = 1.37″ or 34.7 mm

The specified positions, 8° before TDC for the exhaust and 14° after TDC for the inlet, can then be scribed. Also mark at 10°, an inch near enough, before TDC for ignition timing later.

The camshaft sprockets have 28 teeth i.e. 12.86° per tooth, and 9 fixing holes, this combination giving possible adjustment steps of 1.43° at the camshaft or 2.86° at the flywheel.

The 9 holes in the new sprockets are ¼″d, one of which must be increased to 5/16″d to take the 5/16″. screw which ensures correct replacement. Not intending to make any change in the timing, the new sprockets were clamped to the old in the same tooth/hole relationship and the appropriate ¼″ hole opened out. With chains in position, the sprockets were offered up to their respective camshafts in what was adjudged to be the correct positions. (Note – the new chains had joining links as distinct from the original which were continuous. These are best fitted with the spring clip to the front and of course, the rounded end towards the direction of travel.)

A dial gauge was mounted in the manner shown in the Manual. Taking the inlet first, at 14° after TDC the valve had opened 0.107". The maximum specified lift of 0.087" was reached at 10° after TDC. Suspecting that this may be a feature of the cam profile, the actual maximum lift was measured and found to be 0.358" to which must be added the tappet clearance of say 0.01" i.e. a total of 0.371", for comparison with the specified 0.345" which includes the clearance. (WSM pg. 19.) This higher lift is virtually the same as the specified 0.375" for the later DB/A 3 litre engines from No. 1075 onwards. This is remarkable as the camshaft is marked F50421, shown in the Parts List as standard for the VB6B/E engines! This would account for the higher lift at 14°. As a further check, the original sprocket was temporarilly refitted with the same result.

Turning to the exhaust side, and knowing that camshaft No. 50422 was listed as the **inlet** for the 3 litre VB6J engine, one could hardly be surprised if the specification for the VB6E exhaust did not fit. And so it was. Instead of 0.082" to 0.087" at 8°, a measured opening of 0.050" was obtained, indicating that the valve was closing early relative to the VB6E specification. A closing gap of 0.085" required 17° before TDC. In this case also, the actual lift plus clearance was as specified for the DB/A 3 litre No. 1075! It would have been of interest to experiment with different settings but as results had been so satisfactory with the present timing, any variations were to be left to a future date. In the meantime, if anyone with a good memory and who was familiar with the AM camshafts, would care to throw a little light, I am sure many owners of the earlier cars would be interested. (Better still if velocity and acceleration diagrams are available.).

One further point before leaving this subject. When making the exhaust valve check, turn the crankshaft in the opposite direction of rotation beyond the 8° mark and come back to the marking. This will place the driving side of the chains in tension.

Ignition These engines were originally fitted with platinum pointed 10mm plugs, P TEN L 70 being the recommendation for general use. (WSM pg.96). Cost 18/6 ea. against the usual 5/-. Ray Eve had tried plugs of many different ratings and dozens were received with the car. To 'soft' a plug in the pre 70 mph days was to be avoided but with the 'harder' types I was always plagued with spitting and banging when getting away from a temporary slow down. This lead to a discussion with K. Gough of KLG in 1962 who produced a set of TENL 70 tungsten point plugs for me to try. These had been produced to replace the expensive platinum point type. Since then the engine has not missed a beat. Note – the P plugs are referred to on pg. 339 of the Manual but omitted from the recommendations on pg. 384. Best to forget them!

Timing should present no difficulty using the 10° marking already scribed on the flywheel in conjunction with a test lamp in the contact breaker circuit.

Lubrication The exceptionally fine condition of practically all wearing parts as found must be mainly attributable to the oil used exclusively since I've had the car – Castrol XXL. This is – or was, as it is no longer available – a fairly heavy non detergent grade recommended for high ambient temperature conditions. It caused heavy sludging throughout the engine and reluctant starting in winter but rather than change grades in autumn and spring, I fitted a Bray mains electric heater in the water circuit under the dynamo. The Exide 'Double Life' battery fitted in 1962 was not replaced until 1972, so the duty could not have been too onerous. As the Double Life was no longer available, (Exide told me it would put them out of business:) an Exide 'Supreme' was fitted. Back to oil, Castrol GTX is now being used.

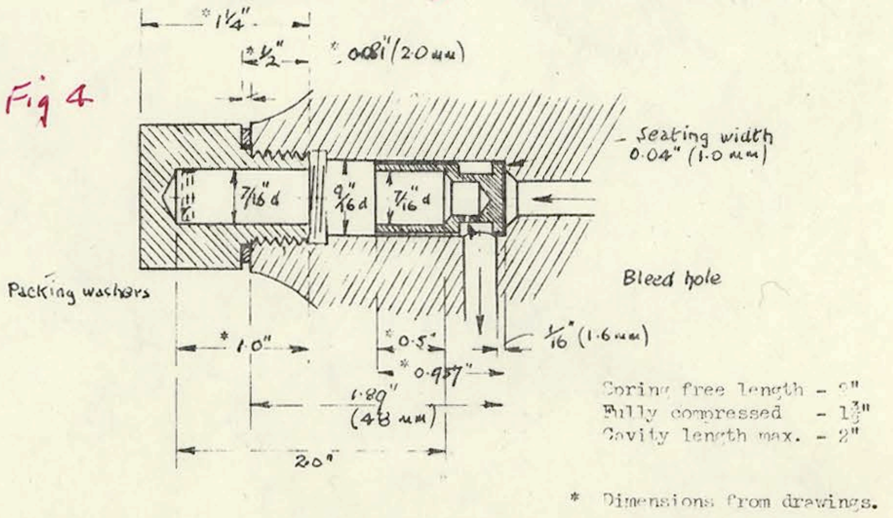
Oil feed to the camshafts. On lifting the head I had found the camshaft oil feed sealing washer well on its way up the oil way in the head, fortunately without having stopped the flow. Measurements showed the holes to be considerably out of line. This problem has been noticed before and cures suggested. However, the solution I am now trying is shown in Figure 3. The brass countersunk washer is of the type used in joinery where countersunk screws have to be used. The outside diameter must be large enough to bridge the head hole eccentricity and yet fit within the gasket hole. This arrangement provides a little extra compression to the rubber washer while preventing it making an upwards excursion.

Oil pressure relief valve A greater degree of oil pressure adjustment is available here than generally recognised. Tickover pressure with these engines when hot can be disturbingly low, even when relatively new or reconditioned. The pump has not the capacity at low speeds to deliver the hot oil flow necessary to maintain pressure at the many outlets. A pump, even in fair condition, may not be able to maintain the desired 55 – 65 lb/sq in pressure at higher speeds if the bearings are slightly but still acceptably worn. Page 16 of the Manual gives an idea of the greatly increased flow. As indicated on pg. 92 of the WSM, the cavity of the hexagon plug may contain small flat washers to back up the spring and so obtain the relief pressure required. Additional washers may be added but there may be doubt as to how far it is safe to go. Fig 4. will help resolve this. Most of the dimensions shown are taken from my own engine but they agree, within reasonable tolerances, with the few AM drawing dimensions which Capt. Forshaw has been able to supply. Unfortunately, the crankcase machining details are no longer available and I have assumed a discharge orifice diameter of ¼". My spring had a free length of 52mm, a little over the 2" (50.4mm) specified, but more important in this context, a fully compressed length of 35mm (1⅜"). Without any packing washers the cavity is just long enough to accommodate the spring and a valve movement of some 16mm, say ⅝", would be necessary to almost compress the spring solid. Assuming the extreme condition, unlikely to be met in practice, of the discharge hole being fully uncovered, the valve will have moved about 8mm (5/16") This leaves a further 8mm before the spring becomes solid and indicates the maximum thickness of packing it would be advisable to fit. The necessity for a packing approaching this thickness also indicates worn bearings and the need for overhaul.

As an interim measure this adjustment can be very useful and more effective than an overhaul of the oil pump. However, two words of warning; do not aim for too high a pressure on tickover hot otherwise it will not be possible to move off cold without wrapping the gauge needle round its stop! And if cylinder bore wear is to be minimised the engine should be put under load – with moderation – as soon as possible. Better to put up with a low tickover pressure hot and give the throttle an occasional blip when standing. The second point concerns the amount of oil thrown about the crankcase if one aims for too high a pressure with worn bearings. At normal temperatures the amount thrown up the bores may be more than the scraper rings can cope with and will be shown by a smokey exhaust and oiled plugs. Thus the whole thing boils down to finding the best compromise.

When ressembling, make sure the small bleed hole from the interior of the valve is clear. This allows oil, which may leak through to the spring cavity, to escape before build up would prevent the valve from operating. Also check the valve seating. It is not unknown for the cast iron to have been eroded away by the continued passage of the unfiltered oil. Redressing the seating is not likely to be a simple matter and will probably require expert attention. One problem is the complete removal of all debris after the operation.

Camshaft Cover Oil Seals The standard cork strips held by brass channel rings do not remain effective for very long and various alternatives have been suggested. By far the best in my experience is the one devised by Australian member Mike Jones and brought to our attention by Alan Puckett some years ago. It was illustrated in the New Sheet at the time but for those unfamiliar with the details the sketch is reproduced here Fig. 5. Each seal consists of a double shouldered steel ring carrying an 'O' ring on each shoulder. The original sizes are shown but some variations in the castings may necessitate some adjustment. In my case the shoulder diameter was increased to 2.20" If it can be arranged, a trial assembly before the camshafts are fitted can save much time. A simple method of checking for leakage from these seals, which may be confused with leakage from the camshaft oil feed joint, is to pull a pipe cleaner round the lower part of each seal and leave it in position for checking at intervals.



Oil Pressure Relief Valve.

Carburetter Tuning The standard S.U. Carburetter is very fully described in the Manual, pg 277 et seq, including instructions for setting up twin carburetters. However, after cleaning but before assembly, a simple test will show whether the two are likely to work 'in sympathy' with each other. Insert each piston fully into its suction chamber and holding one piston between finger and thumb so that the communicating hole into the suction chamber is covered, invert the pair over the bench – well padded – and allow the suction chamber to fall clear. Count the seconds and repeat with the other. With some dexterity a direct comparison can be made with one in each hand. If the times are equal, you will be lucky. If there is a significant difference, say 2 – 3 seconds, some attempt to equalise is advisable before proceeding. In my own case, one gave 4 – 5 secs, the other 12 sec. Interchanging the pistons produced 5 – 6 sec. So, breaking all the rules, I eased the piston giving 9 sec until the timings were equal. It is doubtful whether there is any 'official' timing for this drop test, which I believe was suggested by one of our members years ago, but I consider it advisable to get this aspect of balance right before following the instructions.

And so eventually, with engine and gearbox back in the chassis, all accessories [illegible], fuel and electrical connections remade, water in, – the moment of truth arrives. Are all the bits and pieces back in place? No vital bit left in a tin on the bench? Valve and ignition timings correct? – in spite of the instructions! First, with plugs unfitted, a cautious turn over by hand, then on the starter until the oil pressure gauge showed signs of life. Plugs in – and now the starter button. She goes first push! Never expected otherwise! But no matter how confident one may feel, there is always a feeling of relief when all goes to plan, especially when neighbours who have viewed apparent chaos for months, are determined to be in at the kill!

Some of the foregoing may prove a bit of a bore to some of our more experienced members but if it is of some help to those with interest and who cannot afford to rush off to the experts every time something requires attention – then I am well satisfied. I would add that my comments regarding mistaken information in the 1956 Manual, which in practically all cases originates in the AM Workshop Manual, are not to be construed as implied criticism of 'Sir' who did the Club an immense service in compiling so much information in one volume. My object has been to set the record right. Perhaps I may conclude with a few items of more general interest.

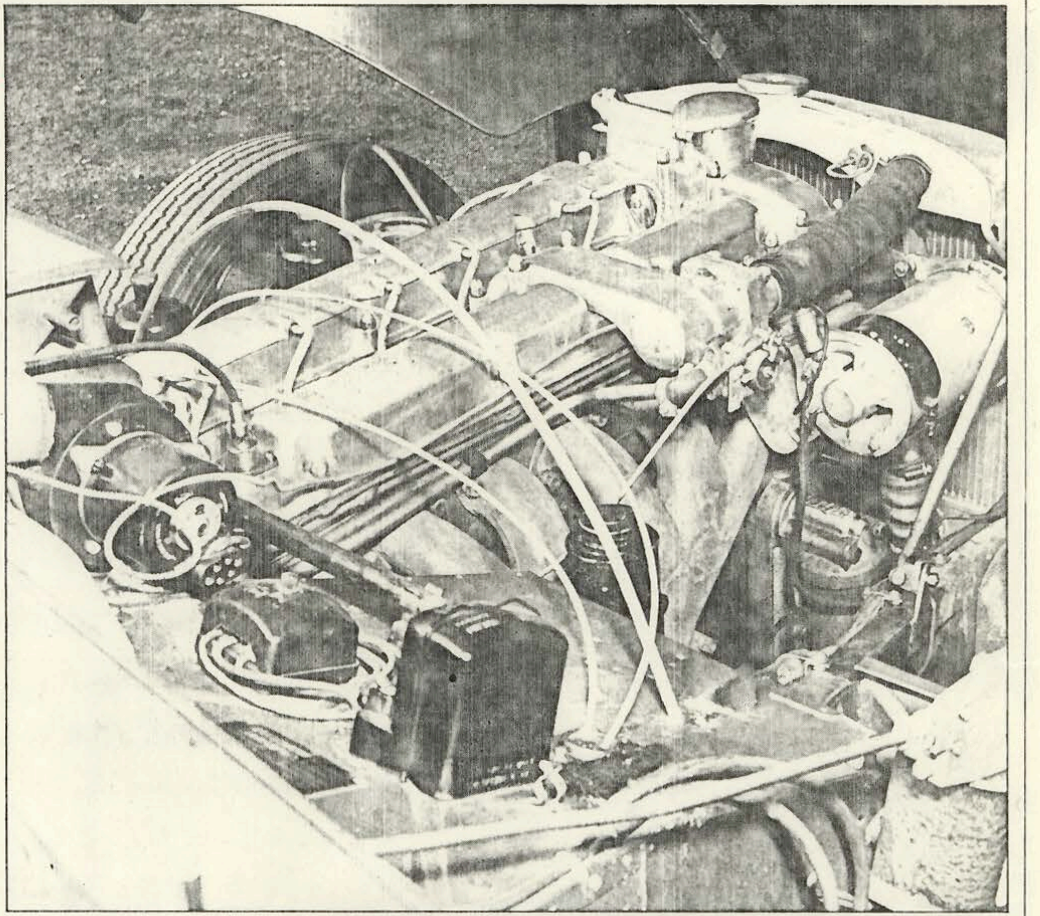
Although it may be very desirable to keep the car to the exact original specification, safety factors should, in my opinion, over rule that desire if the car is to be used to any extent. To this end the following modifications have been made to JUK 700 over the years.

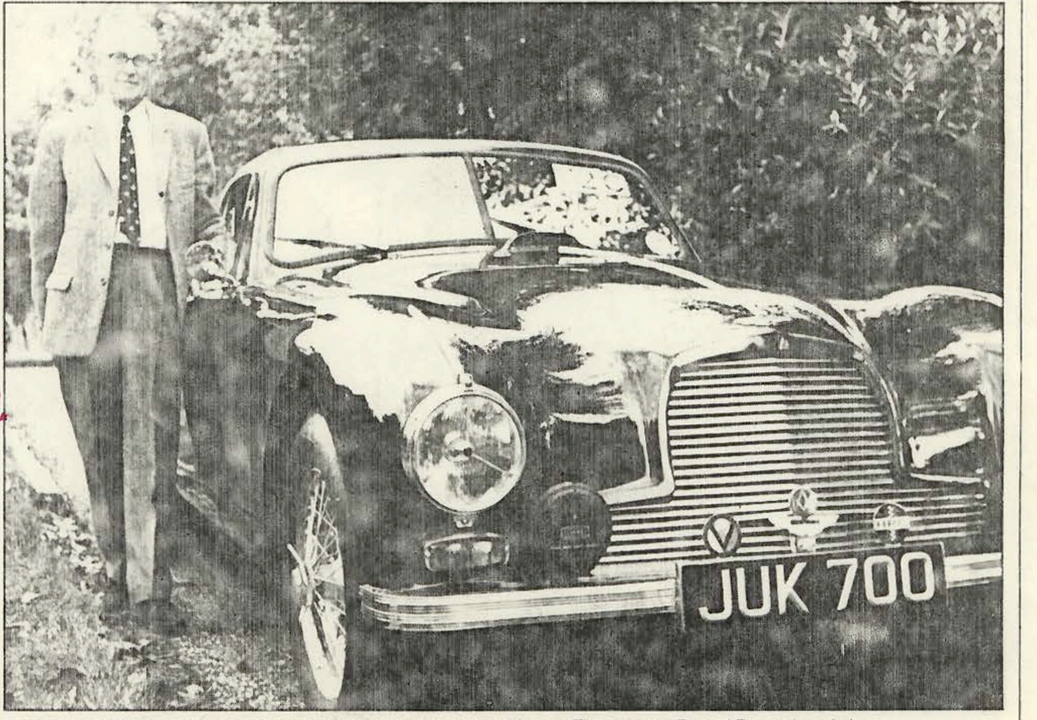
Very early on the useless semaphore trafficators were disconnected and the single lens front side lights replaced by Lucas combined side and indicator lamps, as used on the Triumph Herald. These lamps, which I think look better on the flat area below the headlamps, needed to be packed out about ½", (I used stiff rubber), to give clearance behind. The two front single lamps were fitted with red lenses and mounted on the spare wheel cover lid as rear indicators. A finger tip headlamp flasher and indicator stalk was mounted on the steering column and a Lucas Hazard Warning unit added later.

Volvo seat belts, single hand operation bought in Stockholm, were fitted. This necessitated the replacement of the forward half of the aluminium transmission tunnel with a similar piece in 18SWG galvanized steel. The wings were made wide enough to bolt to adjacent chassis parts. The rear belt fixings were to the respective wheelarches, the bolts also picking up 1" x ⅛" steel straps secured to the rear damper anchorages. Chassis side members were drilled for the side fixings.

A Clayton Dewandre MOT-A-VAC servo brake unit was neatly accommodated behind the screen washer bottle once the circular container (for a quart can of oil?) had been removed. The suction pipe was readily banjo jointed into the end of the rear manifold. Other pipework was straight forward and I would recommend the use of the Yorkshire Copper Co. 'Kunifer' piping which is corrosion proof and claimed to outlast the car! In 1968 the belt driven fan blades were removed and a homemade unit consisting of a Smiths heater motor with Kenlow fan was mounted in front of the radiator. This was controlled by a Kenlow thermostat inserted into the top water hose. An override switch and indicator were provided on the dash. It is remarkable how infrequently the fan operates even on a hot day. Originally the system was not pressurised but a replacement radiator from AM in 1963 was fitted with a 4 lb sq in cap.

The standard petrol pump position on the DB2 is very inaccessible and a change to a higher place was made by AM with the DB 2/4. Similarly my pumps are now mounted alongside the radio power unit and any doubts I had on their ability to raise petrol from the tank have proved quite groundless.





Top: An early DB2 engine of the type discussed in this article **Above:** The author, 'Dicko' Dixon beside his superbly maintained DB2.