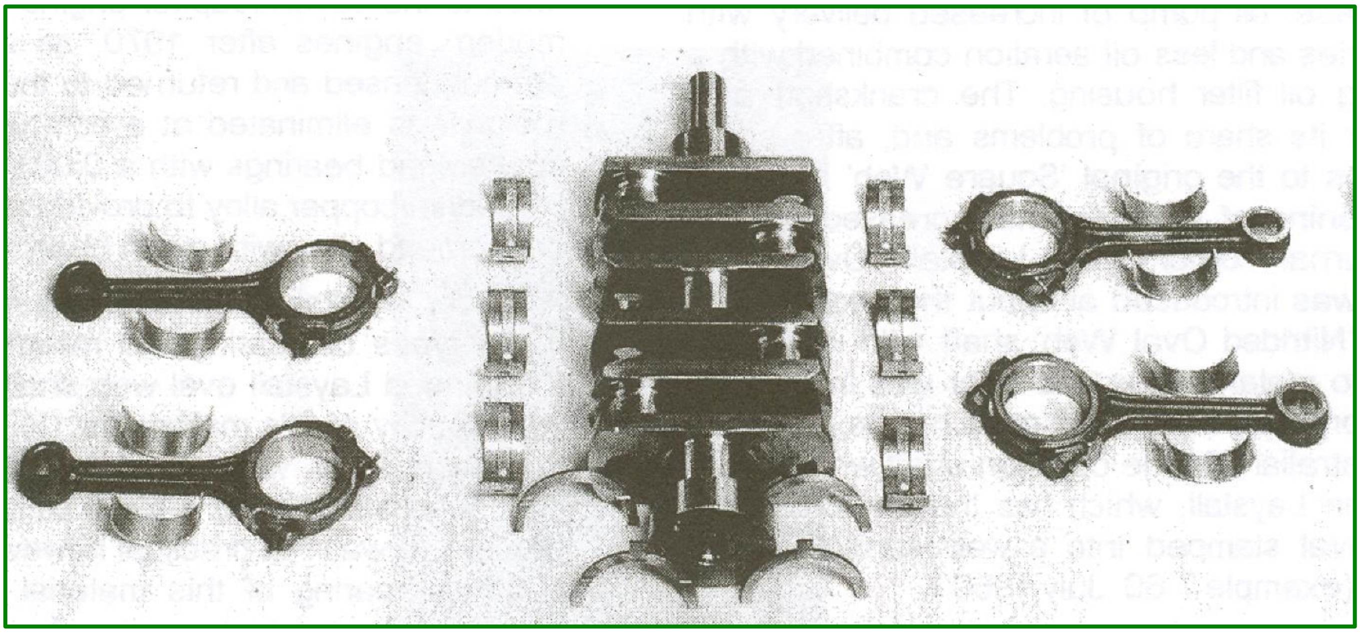
TECHNICAL NOTES SERIES

JOWETT JAVELIN PA, PB, PC, PD & PE

JOWETT JUPITER SA & SC



Above: With thanks to North American Jowett Register. Featured items are the internals of the R1 Jowett Jupiter.

PART IX – CRANKSHAFT BEARINGS

INCLUDING:

GLACIER 20% TIN-ALUMINIUM BEARINGS

Compiled by, Mike Allfrey – May 2004.
Revised – March, 2024.

CONTENTS

WARNING!
ASBESTOS COULD BE PRESENT IN GASKETS AND FIBRE WASHERS

INTRODUCTORY COMMENT FOR TECHNICAL NOTES

These introductory notes should be read prior to reading Part IX of the Technical Notes Series.

The Jowett Technical Notes Series have been an ongoing activity for several years. That means that some techniques and specifications may have been superseded in later notes on the same, or associated topics in the series. Also be aware that some topics and recommendations may be specific to certain Engine Serial Number ranges. It appears that, in Australia, the various State Main Agents did not carry out Service Bulletin information during Jowett active times. A set of known Service Bulletins is in Part III.

Some of the notes are restorations of what was written by members of the Jowett Car Club (UK), the Jowett Car Club (NZ) and by members of the JCCA.

Over the years of involvement with matters Jowett, and with the dawning of the personal computer age, a personal decision was made to help members of the Jowett Car Club of Australia Inc. with technical information. Included with the Technical Notes are ‘restored’ versions of the Javelin and Jupiter Maintenance Manuals and the associated Spare Parts Catalogues. Future generations will prefer to flick through images on their personal device screens, rather than leafing through pages in a tattered and oil stained book to access information.

The term 'restored' has been used because it soon became apparent that, as with our efforts in restoring Jowett vehicles, we desire excellent quality of workmanship in the reproduction of Jowett related documentation. Not for us the crude, and crooked, photocopies that have been issued over the years. These have, even though accurate at their time, become partly out of date.

Hence the decision to ‘restore’ the publications and documents that have come to hand.

It should be noted that the Javelin and Jupiter Spare Parts Catalogue is a combination of all the catalogues that were to hand (from 1948 to 1953).

The Maintenance Manuals were originally written on the assumption that they would be used by skilled motor mechanics who had attended service training courses conducted by Jowett Cars Limited and after works closure, were made available for owners who had reasonable mechanical knowledge of motor car maintenance and overhaul.

Included with the Technical Notes Series is a Lucas Overseas Correspondence Course, which can be of great assistance when trouble-shooting electrical problems related to your Jowett, or any other British vehicle of the same period.

Please be aware that this is an ongoing project . . . .

Mike Allfrey. – February, 2024

JOWETT JAVELIN AND JUPITER CRANKSHAFT BEARINGS

NOTE: This information originally appeared in ‘*Flat Four*’ the newsletter of the Jowett Car Club of New Zealand, in November, 1978. The article has been updated by its author, Neil Moore, in 1994. Thanks are due to Neil for the work he has put into this, and for permitting this full reprint. There is some Australian content.

The subject of engine bearing life has interested me ever since I became the owner of a Javelin in 1950. As my knowledge of the engine grew, it seemed the Javelin was plagued by bearing problems more or less from its beginning. The list of engineering changes show this with a number of modifications to improved bearing materials, stiffer connecting rods and crankcase, oil pump of increased delivery with larger galleries and less oil aeration combined with a non-draining oil filter housing. The crankshaft also came in for its share of problems and, after some modifications to the original ‘Square Web’ including flame hardening of journals and increased radii in bearing journal corners, the Laystall ‘Oval Web’ crankshaft was introduced at about the beginning of 1954. The ‘Nitrided Oval Web’ shaft with increased resistance to metal fatigue and wear was introduced from 1958 on. This style shaft must not be confused with the Australian Meade cast iron shaft which looks similar to the Laystall, which has the Laystall name within an oval stamped into a web surface, along with a date (example 7 60 = ‘July, 1960’). This gave a fairly satisfactory life of 50,000 to 80,000 miles (80,465 to 128,744 kilometres) between major overhauls if copper/lead bearings were used with the Laystall oval web shaft, and Jowetts gained a fair reputation among the motoring public for speed rather than longevity.

However, in New Zealand it seems we had another problem with inferior bearings. These gave rise to shorter life engines and some unwarranted reputations. This was due to Model CC Bradford and Javelin big end dimensions being the same, but the Bradford engine's specification calling for white metal bearings, and the Javelin/Jupiter engine's specification calling for copper/lead bearings. However, for approximately ten years, New Zealand produced bearings were of one type – white metal for both and the problem was further compounded by the manufacturer producing white metal shells for the Javelin front and centre main bearing sets. (The rear main bearing is white metal with thrusts anyway as this is the only way it can be manufactured.)

It must be borne in mind that this is ancient history – 1950s to 1960s – and while the local manufacturer was mistaken in producing them anyway, it was a situation created by very harsh import controls imposed by the New Zealand governments of the period. Allied to this was the fact that Jowetts were no longer being produced any more and the only people who were aware that white metal bearings were inferior in Javelins and Jupiters were the Jowett agents around New Zealand, who were kept informed of developments by Jowett Cars Limited (later Jowett Engineering), England, up to 1966 when the firm finally closed down. Boxes of these white metal bearings would be placed at the back of the shelves and have unfortunately in recent years come to light again with a lot of ex-Jowett agents quitting their little piles of parts left after all this time. All the copper/lead bearings have been used and, the white metal bearings are all that is left and now (1978) unsuspecting Jowett owners of this decade snap them up as 'genuine' parts – "Caveat Emptor" – let the buyer beware!

The inherent problem with these alloys were that they were prone to corrosion from the organic acids produced during combustion of the fuel in the engine which find their way into the engine's lubricating oil. Such corrosion is increased by a breather valve as fitted to the Javelin/Jupiter engine and to most other modern engines after 1970, as crankcase gasses are condensed and returned to the engine. The corrosion was eliminated at a cost by over-plating the copper/lead bearings with a 0·001-in. (0·0254 mm) layer of lead/tin/copper alloy to provide a high load bearing copper/lead alloy with a soft bearing surface.

A harder crankshaft surface was recommended with these types of bearings to minimise wear. (Hence the nitrided Laystall oval web shaft.) The load carrying capacity of this material is 1½ to 2 times greater than babbitt or white metal bearings, although the cost is greater due to a more difficult manufacturing process. It was not practical however to manufacture a thrust bearing of this material, so Jowetts (and several other vehicle makers) ended up with a white metal rear main c/w thrust and copper/lead overlay bearing for centre and front main bearings, though it will be noted that the rear main bearing is a little wider than the two other main bearings which partly compensates for its lower load carrying capability. This then was the state of things about the end of 1954 when Jowett Cars Limited ceased production. Since then another improved bearing material has been developed. This is an aluminium-tin alloy of approximately 80% aluminium and 20% tin, and has the same or slightly superior load carrying capacity to copper/lead and, as a bonus, is corrosion resistant and can be used with a soft shaft. It is also cheaper to produce than the copper/lead bearing surface material. This is now the standard bearing material used by ACL in making nearly all replacement bearings for the New Zealand market. These bearings are designated by a number followed by 'AL'.

Bearing materials are only part of the story because although they play a very significant part in engine longevity, the crankshaft and its oil way design is another significant factor. Jowett were obviously worried by their numerous crankshaft breakages and the eventual oval web design overcame this defect; however, it did not solve the bearing failures. Bearing life was increased by reducing the clearances in the Series III to something like ¾ of a thousandth of an inch (0.00075") but this, while it kept bearing leakage to a low level until wear increased the clearance, it did not get at the cause of the problem. Research on crankshaft design since 1954 has shown that the engineers at Jowett Cars Limited were "backing the wrong horse" by feeding the oil to the big ends at the trailing side of the bearing. Centrifugal force was the culprit. It is reasonable to assume that the faster a crankshaft turns (spins), the more force is exerted on the oil at the extremities of the crank and any excess clearance there could give quite a leakage. If this exceeds the feed of oil to the main bearings, they will suffer oil starvation and wear.

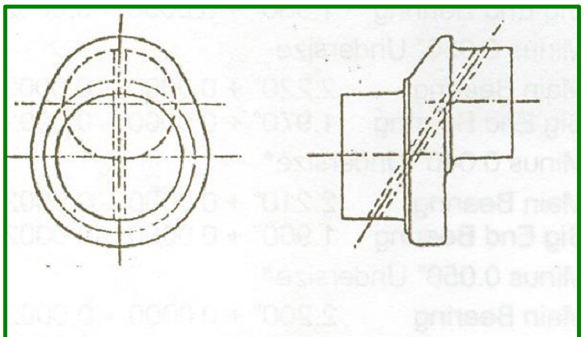


Figure 1a.

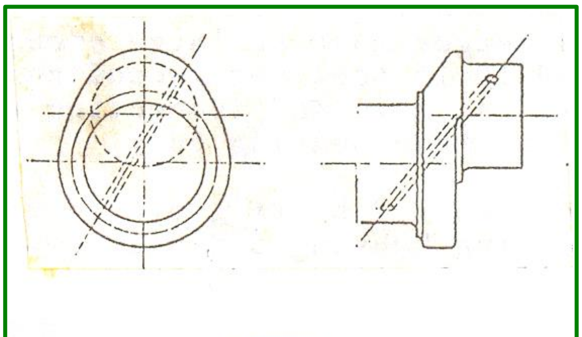


Figure 1b.

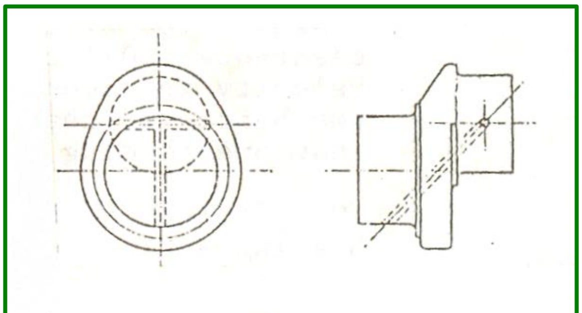


Figure 1c.

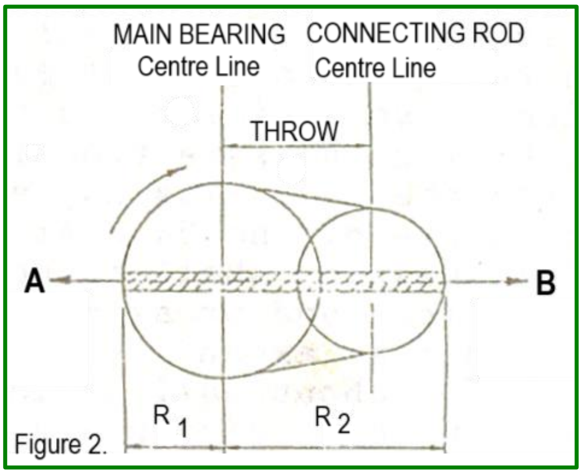


Figure 2.

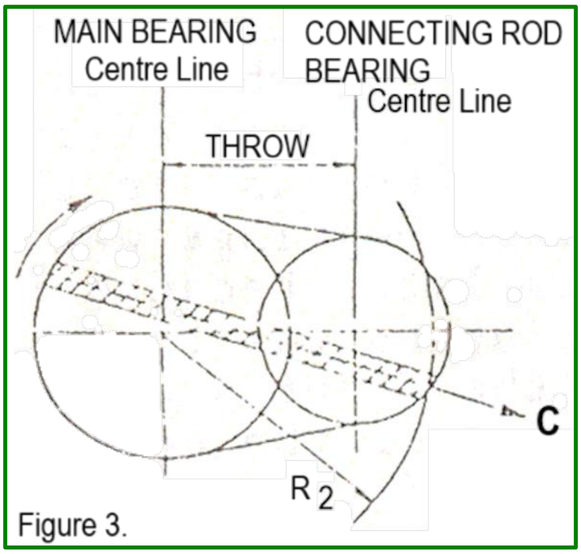


Figure 3.

At *Figure 2* the following applies:

A = Formula – Pressure = $PW^2 R_1^2 \div 2$

B = Formula – Pressure = $PW^2 R_2^2 \div 2$

At *Figure 3* the following applies:

C = Formula – Pressure = $PW^2 R^2_2 \div 2$

On the Javelin/Jupiter engine the centre main bearing feeds two big end bearings, so there is twice the draw-off, while the front and rear mains only feed one big end each. In the Javelin/Jupiter Series III engine a groove was machined behind the bearing shell (slipper) in the crankcase around the housing to give continuous oil feed and, as well, a huge groove was machined around the shell on the bearing face – which was in fact one third of the surface area of the bearing. Thus we had a situation of oil being flung to the big end cranks by centrifugal force and a huge drain was supplied to help it get away!!

Some relevant theory on this problem and the ways of combating it follow, per courtesy of '*Engine Design*' by JG Giles:

"Quantity of oil required (for each bearing) will be determined by the leakage area of the bearing which is related to the diameter, length and clearance. Oil passages must be large enough to restrict the oil velocity to around 10–15 feet per second, and where one passage feeds two bearings, that is, main and connecting rod, it should be larger than one feeding the main bearing only.

"Oil is fed to the connecting rod bearing by a duct or drilling in the main bearing and/or bearing housing and oil pressure must be sufficient to overcome centrifugal effects due to crankshaft rotation on the oil in the cross-drilling in a main bearing. This centrifugal effect opposes entry of oil into the cross-drilling on one side of the main bearing and assists outlet into the connecting rod on the other, tending to create low pressure. Pressure therefore must be sufficient to overcome the centrifugal force and also maintain the rate of flow at which the oil is being thrown out at the connecting rod bearing.

"Centrifugal oil pressure can be minimised by consideration to the position of cross drilling. *Figure 1*, illustrates three methods of cross drilling. Drawings at, *Figure 1a* and *Figure 2*, shows drilling on a line joining the main to connecting rod journals and maximum centrifugal forces will be evident on main and connecting rod journal diameters.

“The centrifugal pressures can have an important influence on the flow of lubricant to the various crankshaft bearings and may lead to actual starvation at high speeds when there is some wear in the bearings. Consequently the small reduction achieved by the angled drilling, Figure 1b, can produce a useful gain in this respect.

“It will be noted from *Figure 2*, that the effective radius on the connecting rod side and hence centrifugal pressure, has been reduced. A further reduction of course is achieved by the cross-drilling shown in *Figure 1c*, although this operation adds a further drilling operation to the manufacturing process and thus incurs a small cost penalty.

“This has extra advantages of:

1. Oil feed to journal at two positions;
2. Oil holes positioned where minimum load occurs;
3. Inertia force on oil is not increased by an increase in connecting rod journal diameter; and
4. Oil flow is not biased towards one side of bearing as is evident in methods ‘a’ and ‘b’.”

The reader will digest all this and say, well this is very nice – but what can I do to incorporate all this theory into my rebuilt, in 1994, Javelin/Jupiter?

The answer with respect to the crankshaft theory is, very little, but maybe we can do something to the bearings. To get an idea of what has been achieved we need to look at racing engines and the other similar engine to the Jowett, the Volkswagen. Firstly, it is standard practice in Lotus Cortinas and other similar engines to replace the main bearing shells with a full oil groove all round, with a pair, one of which is plain and the other with a groove. This reduces the supply of oil to the big ends to half the crank’s revolution and retains it in the mains which seemingly is adequate! Racing ‘Minis’, sometimes reaching 9,000 rpm, according to Mr Ted Thompson (balance expert from Kumea), sometimes have bronze plugs with smaller cross holes in them fitted to the cross drillings of the crankshafts which feed the big ends, again the purpose of which is to restrict the oil flow to the big ends which is adequate at high revs due to centrifugal force anyway. And, of course, the Volkswagen engine which is very similar to the Javelin/Jupiter engine was, in the early 1960s, having the same troubles with centre main bearings wearing out for exactly the same reasons, albeit ten years later because of the unstressed nature of the Volkswagen engine. Their solution was to fit a completely plain bearing shell in both halves (the front and rear were made in one piece and pressed into the crankcase) for the centre and plain front and rear except at holes and the shaft journal had a small groove ground into it for about half an inch around the circumference area in line with the hole, giving a squirt of oil each time the groove passed the holes.

Main Bearings

So, here we have some solutions to the centrifugal problem, but which is the easiest (and cheapest) way of applying this to the Javelin/Jupiter? Well, as the crankshaft is nitrided (case-hardened and hard, it is probably best to leave well alone; the plug idea did not appeal due to balancing problems. This left the main bearings, which as the originals had a much-reduced surface area due to the one-third groove, to be improved. A search of the bearing manual shows that the Perkins three-cylinder diesel engine as fitted to the MF-135 Massey-Ferguson tractor (and numerous other vehicles) had a big end shell of the same dimensions as the Jowett main. How convenient! A three cylinder too, providing a boxed set of three pairs of shells – in aluminium too! So, we machine the shells. Not as with the Lotus engine, because it is an in-line engine, so we put a small groove from the oil hole to the join, which in the Javelin/Jupiter is vertical. This is ideal because in a horizontally opposed engine the load is at the centre of the shell. With an in-line engine, the load is on the bottom so a plain shell can be fitted to the bottom half and a grooved one to the top half of the main bearing. We can go one better by installing three sets of shell bearings throughout the mains and fitting some thrust washers to the rear faces of the rear bearing. Again, consulting the bearing catalogue, we find that almost all standard thrust washers are 0·093-in. thick. This means that the crankcase will have to be machined to allow the thicker washers to be installed. So much for theory, let’s put it into practice.

Firstly, the crankshaft, find one that will stand grinding to the next undersize and take it to an engine reconditioner who understands Volkswagen crankshaft grinding requirements. Ask around, as the Volkswagen shaft is nitrided, of similar shape to the Jowett shaft and **requires the same 0·100-in. radius at the journal corners**. Most engine reconditioners who do these shafts regularly will have a special wheel which they install on their grinding machines, for which they may charge a little extra. If in any doubt, contact someone who knows, or contact the Volkswagen Club.

Those connecting rods with sharp corners at the big end bearing bore, must have a suitable chamfer machined at this edge. Otherwise the connecting rod will pinch at the crankshaft big end bearing journal radii, when the big end bolts are tightened. The chamfer should have a 0·125-in. face at 45° cut angle.

Have the shaft ground to one of the following sizes, depending on the condition of the shaft you have.

Crankshaft Grind Data

Standard Size	
Main Bearing	2·250 + 0·0000, – 0·0002-in.
Big End Bearing	2·000 + 0·0000, – 0.0002-in.
Minus 0·010-in. Undersize	
Main Bearing	2·240 + 0·0000, – 0·0002-in.
Big End Bearing	1·990 + 0·0000, – 0·0002-in.
Minus 0·020-in. Undersize	
Main Bearing	2·230 + 0·0000, – 0·0002-in.
Big End Bearing	1·980 + 0·0000, – 0.0002-in.
Minus 0·030-in. Undersize	
Main Bearing	2·220 + 0·0000, – 0·0002-in.
Big End Bearing	1·970 + 0·0000, – 0·0002-in.
Minus 0·040-in. Undersize*	
Main Bearing	2·210 + 0·0000, – 0·0002-in.
Big End Bearing	1·960 + 0·0000, – 0·0002-in.
Minus 0·050-in. Undersize*	
Main Bearing	2·200 + 0·0000, – 0·0002-in.
Big End Bearing	1·950 + 0·0000, – 0·0002-in.
Minus 0·060-in. Undersize*	
Main Bearing	2·190 + 0·0000, – 0·0002-in.
Big End Bearing	1·940 + 0·0000, – 0·0002-in.

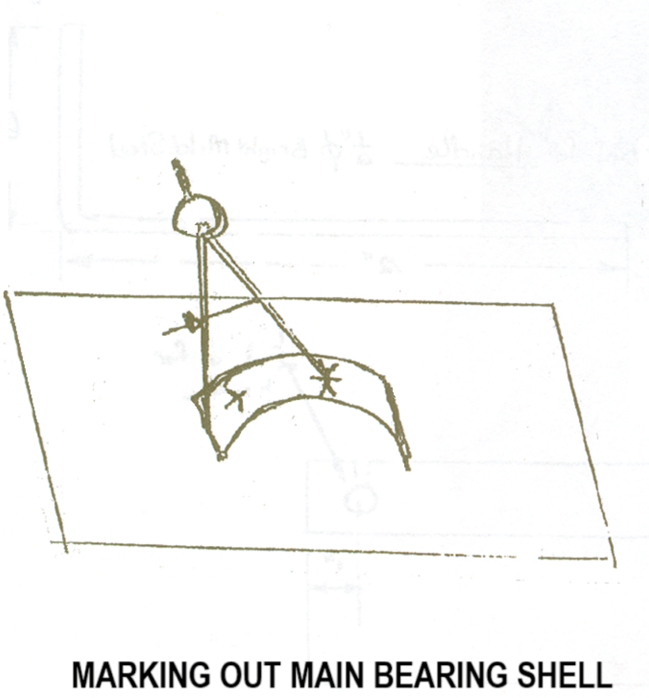
* Ensure that bearing sets of undersizes 0·040, 0·050 and 0·060-in. can be obtained before commencing the crankshaft grinding operation. These bearing sets are usually available from Jowett Club Spares, ex-stock.

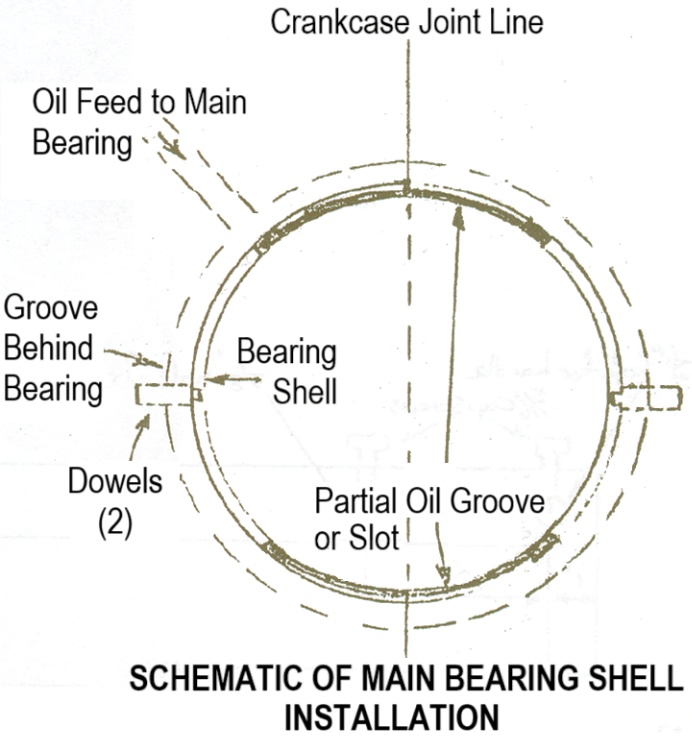
Should you purchase your main bearing set from your Perkins Engines dealer, or from a Massey-Ferguson dealer, ask for big end shells for either P3-144 (2·36 litres) or P3-152 (2·5 litres), or, Massey-Ferguson 3A-152 engine for MF-35, 35X and 135 tractors (agricultural), their Part Number 85035. Incidentally, these bearings may carry the ACL marking ‘3B1012AL’, (Perkins Power Part Number 85036-B, Glacier GS8955 in England) on them as they are manufactured in Australia by ACL and packed for their respective suppliers. Also, it is worth mentioning here that these bearings are a plain shell for the big ends of a three cylinder diesel engine. Perkins also happen to make four and six cylinder engines with the same physical dimensions. The three cylinder set is best because it ideally suits our purpose by providing the right number for the Javelin/Jupiter mains.

A bonus for us is that a large number of Massey-Ferguson tractors are still being overhauled. There are a number of specialist tractor parts suppliers that stock these bearings. In addition, the JCCA is currently holding stocks of copper/lead with white metal rear main bearings from England.

These bearings must be modified to fit into the Javelin/Jupiter crankcase, shown below, using the following procedure:

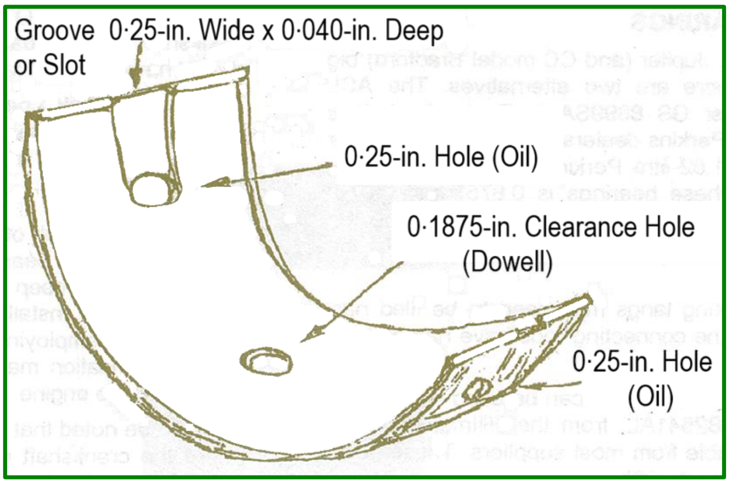
1. Carefully file the tangs off the bearing slippers, so that the outer edge of the shell is flush and will not distort as the crankcase is assembled. Some may ask, why not use the tangs? But, if the way the Javelin/Jupiter crankcase was machined is considered, an explanation will result. The two halves were bolted together and the crankcase was bored for main bearing and camshaft bores as one piece. Thus the centreline of the crankcase may not coincide with the centre of the bearing bores. Therefore if one half circle is smaller than the other, the shell would not fit. So by placing a dowel at right angles to the join the bearings will fit and still stop being spun in the block.
2. Mark out the three holes per shell as follows – take a pair of dividers and, placing the shell on a flat surface, scribe a half line from each join side to find the centre of the outside. Then, taking an old Jowett supplied shell, mark from it the two oil supply holes in roughly the same position. Then equidistant from each edge mark the three holes on the centreline.





Above: Figure 4. Marking out Perkins bearing shell and fit-up schematic sketch.

3. Centre pop and drill them – the dowel hole having a clearance of 3/16" (a Number 11 drill can be used) and the oil supply holes being ¼-in. De-burr the holes after the drilling operation.
4. This part is the crux of the bearing – machining the groove – and will depend on the model of the block you are to use for your engine. Bearing in mind the preamble about oil flow and centrifugal force and so on, I have concluded that the easiest method of obtaining the desired effect from the various alternatives, is to modify the bearings. Now if the crankcase is an early PA, PB, PC or SA, it will have no oil groove machined behind where the bearing shell fits.



Right: Figure 5. Dimensions for grooving Perkins bearing.

If it is a later PC or SA model, it will have the centre main housing grooved only. If it is a PD, PE or SC crankcase it will have all three main bearing bores machined with a groove. All early engines reconditioned by Jowett Engineering were modified in this manner. All PA, PB and early PC and SA – a thin groove will need to be machined around all bearing bores approximately 0·040-in. deep and no more than 0·125-in. wide. The later PC and SA

– the front and rear main bearing bores will require machining as above, but the centre main will be machined differently as follows (remember it feeds two big ends).

Taking the later PC, SA, PD and PE – again about 0·040-in. deep and 0·25-in. wide machine a groove from the oil holes to the join line of the bearing. This can be done with a rotary file bit and an electric drill, or machined on a lathe or milling machine. When the two bearing halves are held together the grooves will join oil hole to oil hole across the top and bottom (low load area) of the bearing.

5. Bevel the edges of the shells at 45° as they are a fraction wider than the original shell and may foul the large radius on the corners of the shaft journals – front and centre only.
6. Check the shells for any burrs caused by drilling or machining and remove same, thoroughly clean the shell set and install.

   Note: The dowel holes could want slotting with a round file a little on one side to ensure no binding. See *Figure 5*.

It will be obvious, of course, that the PD, late SA, PE and SC crankcase sets with grooves behind each main bearing will give the best results, and the earlier crankcase sets will only be a compromise. However, if the engine re-builder desires a PE result from a PA, PB or SA crankcase, the answer is to machine the block with a groove at all three bearing housings. This can be done without removing the dowels by machining a groove slightly to one side of the dowel and making a shallow dent with a die grinder or burr in a drill to mate up with the oil holes or, the dowels can be removed by forcing grease, through the hole provided, equipped with a needle point nozzle. The crankcase halves can then be set up and machined as per the PE and SC crankcase, using a 5/8” diameter spot facing drill (end mill).

Separate Thrust Bearings

The next stage is the fitting of the rear thrust washers. First purchase thrusts for, ACL part number 2T2167 (Ford Cortina Mk 3), or Repco part number 2T3188 (Ford Zephyr Mk 1 – Glacier W2058). Four individual half washers per engine are required. These washers are 0.093” thick, the original Javelin/Jupiter thrust flanges on the rear main bearing are 0·073-in. thick. This means that 0·020-in. will have to be machined off the crankcase set at front and rear of the rear main bearing faces. Reducing the thickness of the separate thrust washers is an option, but it should not be used because, for future overhauls, non standard parts will be required – thus making the job somewhat difficult and very time consuming. It is much better to machine the crankcase set and be done with it – in future, thrust replacement will be that much simpler.

A special tool can be made-up for machining the rear main bearing front and rear faces by hand. For details see sketch on Page 12. The items that need purchasing are:

- 2 off Sealed bearings, SKF RLS8 or equivalent Hoffman LS10.
- 2 off Pairs of old standard size front and centre main bearing shells.
- 3 off Cap screws $^{5}/_{16}$ x ¾-in. (Whitworth or SAE).
- 1 off Thread tap set to suit cap screws.
- 1 off Length of 1-in. diameter bright mild steel bar 19-in. long. Ends de-burred.
- 1 off Length ½-in. diameter bright mild steel bar 18-in. long.
- 1 off Piece of ⅜-in. diameter tool steel, HSS.
- 1 off Mild steel collar with 1-in. bore and with cap screw.
- 1 off Front hub distance tube, Jowett part number 50383 to use as a spacer.
- 1 off Set of feeler gauges.

The following procedure should be used to make-up the machining tool:

1. The piece of tool steel is sharpened for about 1-in. of its length by being ground flat to the half-way line, taking care not to overheat while grinding. A light hone with an oil stone will give a good cutting edge on either side.
2. The 1-in. diameter bar is drilled as shown in the drawing on Page 12.
3. The ½-in. diameter bar is bent to form a handle with a 6-in. grip area.

4. The standard size bearing shells should be installed in the crankcase set, at front and rear main bearing supports, with a with a piece of Cellotape, wrapping the RLS8 bearings, with one layer to take up clearance. The front bearing has to be held firmly in place as this will control depth of cut.
5. The ball bearings will sit in the shells and the two crankcase halves are then bolted together with four tie bolts – two at front and two at rear – to hold the ball bearings in a nipped condition. Make sure that the two joint faces are thoroughly cleaned before joining.
6. The boring bar will then slide through the ball bearings.
7. The cutter blade is fixed by a cap screw in the boring bar at right angles to the face to be cut.
8. The front hub distance tube and collar are slid on to the boring bar against the rear bearing's centre race, at the outside end, and with the handle inserted at the opposite end to the cutter blade.
9. The collar is set with the cutter blade pressed against the crankcase, and a suggested 0·020-in. feeler gauge is placed between the collar and the spacer. The collar is set and locked on to the boring bar with a cap screw. The cutting depth is now set. Cutting material from the rear face can commence by turning the handle, while maintaining a moderate pressure on the boring bar until the collar sits on the bearing spacer and the 0·020-in. thickness of material has been cut off. The aluminium cuts quite easily and smoothly with only moderate pressure applied.
10. The process is then reversed by placing the cutter blade on the front face of the rear main bearing web, changing the handle to the other end and setting the collar gap again. It should be noted that on the inner face of most crankcases there is a partly machined step where the original flanged bearing was located. The higher portion requires to be carefully cut down to the original bearing flange level and then the 0·020-in. removed from the face.

    Note: The rear main bearing supports require machining at front and rear faces to provide 0·002–0·004-in. crankshaft end float. Equal amounts must be cut from both faces of the support.

Next is the job of drilling and tapping the crankcase set for the thrust washers:

11. Mark the steel face of one of the thrust washers $^{15}/_{16}$-in. (24 mm) from the crankcase centre line edge, and centre the screw hole in the washer half. The $^{15}/_{16}$-in. dimension ensures that the oil gallery is avoided.
12. Using a pedestal drilling machine (it is essential that the holes are square and straight), with a 2·5 mm drill, the correct tapping size for 3 mm thread, centre-drill the two holes and then drill **right through** the thrust washer.
13. Use the drilled washer as a template to mark the crankcase, each half, and again using a pedestal drilling machine, drill right through the rear main bearing support from the rear. Use a suitable cutting fluid or the drill will grab, as will the 3 mm tap later, and clear the drill frequently.
14. Tap the holes, with a 3 mm x 0·5 mm pitch tap, to halfway point, from each face, to allow enough thread for the screws, clear the swarf frequently. It is essential that a threading lubricant, suitable for use in aluminium, is used while cutting the threads. A broken tap could certainly result if a lubricant is not used. The best lubricant is Tapmatic Dual Action 'Plus 2' or, copious amounts of clean kerosene can be used.
15. Lightly clamp the thrust washers in their correct position and, using the 2.5 mm drill, mark the backs of the three remaining un-drilled washers. Accurately mark with centre punch and drill the all washers with 3 mm drill. Countersink on the bearing surface side so that, when the screws are tightened, their heads are just below the white metal (or bronze) on the washer face.

    It may be necessary to slightly countersink the threaded holes in the crankcase, so that the screw heads clamp the thrust washers firmly against the rear main bearing support faces.

    Note: Mark individual thrust washers with a dot punch to identify their positions on the crankcase, the hole centres may vary slightly, due to the drilling of the bearing support.
16. Install the thrust washers, using 3 mm x 12 mm countersunk screws, and check that the crankshaft has the correct end-float using a feeler gauge. A small drop of Loctite 'Nutloc' under the head of each screw will keep them secure. Do not apply Nutloc to the thread, otherwise it will not be possible to remove the screws when replacing the thrust washers.

IMPORTANT NOTE: It should be noted that there may be some instances where the crankshaft may have been ground oversize between the thrust faces at the rear main journal. If this is the case, then less material than the stated 0·020-in. will have to be removed. Consideration will have to be given to aligning the crankpins with the centreline of the cylinder bores. A rough indication of correct crankshaft axial position is when the front journal is flush with the front face of the front main bearing support.

Big End Bearings

For the Javelin, Jupiter (and CC model Bradford) big end shells, there are two alternatives. The ACL 2411AL (Glacier GS 8899SA in England) shell is available from Perkins dealers, as a big end shell for the 4-cylinder 1·62 litre Perkins 4-99 diesel engine. The width of these bearings is 0·875-in. and 0·075-in. should be machined from the tang side to stop the edges fouling on the big end journal radius of the crankshaft.

Note: The locating tangs may need to be filed narrower if the connecting rods have narrow tang grooves.

The second alternative that can be used is the ACL, part number 4B2641AL, from the Hillman Avenger 1970 on, available from most suppliers. These bearings are the correct width.

The bore diameter in the Javelin/Jupiter (and Bradford CC) connecting rod for the big end bearing shells is 2·1445 – 2·1450-in. The bore diameter in the Hillman connecting rod for the big end bearing shells is 2·1460 – 2·1465-in. Since no original Jowett big end bearing shells are currently available, it is advisable to have the connecting rod big end bores checked for roundness and correct size. They can be honed to 2·1460-in. (*0·0015-in. larger – a well used connecting rod bore could have stretched a little*) prior to installing new bearing shells.

Those connecting rods with sharp corners at the big end bearing bore, must have a suitable chamfer machined at this edge. Otherwise the rod will nip at the crankshaft big end bearing journal radii. The chamfer should have a 0·125-in face at 45° cut angle. *There was a Service Bulletin detailing this modification, but has not yet been found in Australia.*

The tangs may also need to be filed to suit the connecting rod tang grooves, so that the bearing can be located in the centre of the connecting rod bore.

JOWETT JAVELIN/JUPITER MAIN BEARINGS – UPDATED INFORMATION

Introduction To 2020 Update

In the recent past, we have been using as shell bearings, the connecting rod big end bearing shells that suit the Perkins 3-cylinder A3-152 (3A-152 in Massey Ferguson terminology) and the Perkins P3-144 diesel engines.

NOTE: The Perkins BIG END bearing has been/is used as the Jowett main bearing (after modification).

Usefully, these bearings come as sets of three main bearings. A Perkins P4 and P6 will also provide bearing sets.

The modification for these bearing shells is described in Technical Notes, Part 09 – Crankshaft Bearings.

Bearing Information

Data from an ACL catalogue for the Perkins bearing shells is as follows:

Relates to ACL Bearings for Perkins A3-152 (2,490 cc) Diesel and P3-1244 (2,360 cc) Diesel

Year	Part No.	Material Code	OEM No.	Standard Shaft	Std. Tunnel Bore
01/57-12/64	1012AL	F820	31131171	2·2485/2·2490”	2·3950/2·3955”
01/51-12/64	1012AL	F280	957E-6211A	2·2485/2·2490”	2·3950/2·3955”

O.E.M. means original equipment manufacturer. It should be noted that Part Number ‘957E-6211A’ could refer to the same engine that was used in the Fordson Dexta tractor, London taxis, fork lift trucks and so on. Frank Perkins Limited, diesel engine manufacturers, was absorbed into the Massey Ferguson company in 1959 and was subsequently acquired by Caterpillar in 1998.

Situation As Of October, 2020

Perkins bearings that suit Javelin/Jupiter main bearing tunnel are also available from a company called BEPCO, who specialise in spare parts for older farm tractors. It is understood that the company was formerly called Powerpart, which is understood to have commenced trading as Vapourmatic (1950s). BEPCO information is listed in the table above.

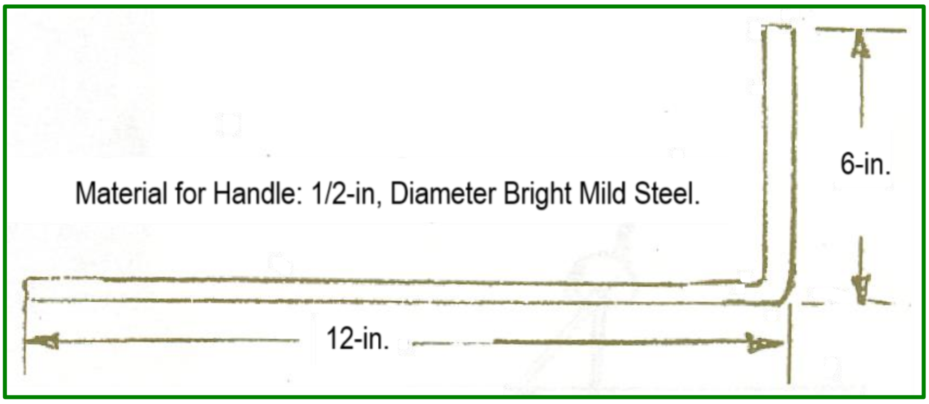
There is also a source of 'genuine' new-manufacture Jowett specification main bearings from Bill Lock and Associates (UK). It is understood that these feature white metal rear main bearings with thrust flanges. That means no machining for separate thrust bearings and that, should a crankcase have been previously fitted with separate thrust bearings, then possibly the appropriate solution would be to remain with separate thrust bearings and order two sets of main bearings to ensure same bearing material at all three main bearing positions. Also, the front and centre main bearings feature copper/lead surface material

NOTE: A white metal rear main bearing will require accurate balance of the engine flywheel and clutch assembly, with the crankshaft if at all possible.

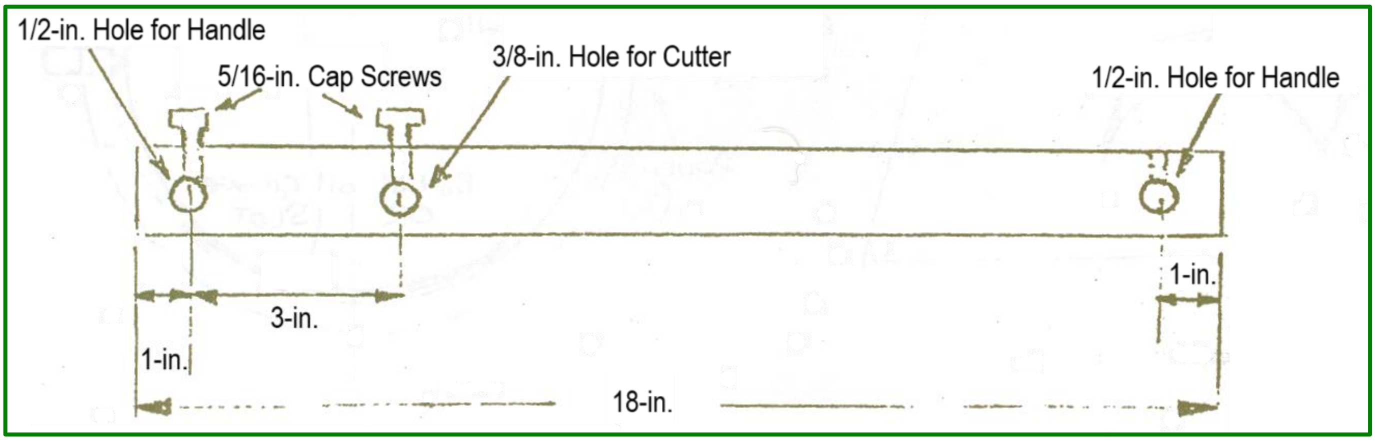
Conclusion

The Jowett Car Club of Australia Inc. can source main and big end bearings from New Zealand. They also keep in stock a quantity of thrust washer sets for installation in crankcase sets. The advantage of employing separate thrusts is that the same specification main bearing sets can be used throughout the engine.

Cutting Tool Sketches – For Separate Thrust Bearings



Dimensions for Tool Rotating Handle



Material for Bar 1-in. Diameter Bright Mild Steel

GLACIER 20% TIN-ALUMINIUM BEARINGS FOR AUTOMOTIVE ENGINES

Introduction

Glacier 20% reticular tin-aluminium engine bearing material has been developed in recent years by the Glacier Metal Company as a less expensive and in some cases superior alternative to tri-metal copper-lead material for heavy duty main and connecting rod applications in high performance petrol engines and high speed truck diesels. Glacier 20% reticular tin-aluminium is a high strength homogeneous material with surface properties proved to be acceptable in engine types varying from small high speed petrol (gasoline) passenger car units (1 litre/60 cu. ins. capacity) to diesel truck units 11 litres/670 cu. ins. capacity). It is used for both connecting rod and main bearings and against crankshafts varying in hardness from 200 Brinell upwards. It is successfully used against both steel or nodular iron crankshafts.

In Europe, as engine speeds and outputs rise, increasingly large quantities of Glacier reticular tin-aluminium bearings are used. In the United Kingdom alone, the four major automotive companies, B.M.C., Ford, Vauxhall and Rootes, together with commercial vehicle and other engine manufacturers, have fitted Glacier reticular tin-aluminium bearings to nearly 6,000,000 engines up to mid 1967.

Engine development in Europe is proceeding along a path which, in a broad sense, imposes two quite different operating conditions upon the connecting rod and main bearings.

The first condition is to be found in diesel engines which are highly rated either by means of turbocharging, or by a sophisticated method of supercharging. Connecting rod bearings are subjected to a very high gas load often at low engine speeds and in such conditions the fatigue strength of the bearing material is critical. The peak unit load on the bearing, although of comparatively short duration, can be as high as 8,000 p.s.i. and 10,000 p.s.i. is under consideration. Glacier reticular tin-aluminium bearings are used in these engines.

The second condition is found in high speed petrol engines used in passenger cars. An engine of this type when used at high speeds, say up to 6,000 r.p.m. for long periods at sump oil temperatures in excess of 120 °C (250 °F) imposes a particularly severe bearing condition, The load is inertia inspired and although of a reduced magnitude compared with the first condition, say 5,000 p.s.i., the load duration relative to the combustion cycle is extended. Extremely thin hydro-dynamic oil films are generated and in such conditions, in addition to high strength, very good bearing surface proper-ties are needed if acceptable bearing performance is to be achieved. Glacier reticular tin-aluminium bearings are fitted to these engines and perform with equal success.

The specifications of a selection of current production engines to which Glacier reticular tin-aluminium connecting rod bearings are fitted are shown in Tables 2 and 2 on Page 20.

General Requirements Of A Bearing Material

The increase in the specific power output of internal combustion engines of all sizes has been spectacular in the last thirty years. Developments in materials, metallurgy, fuels and in the thermodynamic design of engines have been applied to increase the power obtained from a given size and weight of engine. This increase in engine ratings has been in part contributed to, and has in part required, changes in the materials used for engine bearings. Glacier 20% reticular tin-aluminium bearings have been developed to satisfy, as far as any material can, the requirements of a bearing material for the engines of today.

The properties required in an ideal bearing material may be summarised as follows:

1. High mechanical strength to resist the high, fluctuating pressures in the lubricant film.
2. High melting point to resist damage by high temperature lubricant films.
3. High resistance to corrosion to resist attack from degraded and acidic lubricants.
4. Good embeddability to absorb dirt passing through the bearing and prevent scoring at high loads.
5. Good conformability to yield easily when the mating shaft is misaligned or mis-shapen.
6. Sufficient hardness to resist abrasive wear and to resist cavitation erosion.
7. Excellent boundary properties to resist seizure when the bearing is loaded but when speeds are not high enough to provide thick, hydrodynamic films.

It is doubtful whether it would be possible to provide a material which had all these desirable qualities but any plans for developing a new bearing material must be made with these factors in mind. For a particular application, it is possible to ascribe physical values to all of these properties. Mechanical strength, for example, can be defined in terms of the bearing not sustaining fatigue damage when a specific load above a certain value is applied for 100,000,000 loading cycles. Corrosion resistance can be defined in terms of the bearing suffering no damage when immersed in an oil of a certain acidity. Generally, however, the value of each of these factors will vary for different applications and the performance of a bearing material in a particular application will depend more on how the factors are combined together rather than to their separate effects. For instance, the apparent fatigue strength of a material in an application would be much affected by its hardness and its conformability if it were being assessed in an engine whose crankshaft was subject to misalignment and deflection. Nevertheless, the effect of the factors individually is worth study and the reasons for the choice of the 20% reticular tin-aluminium material will be seen more clearly after an examination of the properties of its rival.

PLEASE NOTE that The Glacier Metal Company makes no claim that the quantitative data on bearing strength published here are in any way comparable to the data published by other bearing manufacturers who in general do not provide sufficient detail of the test rig conditions for their data to be useful.

Despite all the development work of recent years, there is no bearing material which has all the desirable properties present to the extent that they were in the original babbitts for the conditions then pertaining. In corrosion resistance, embeddability, conformability and in boundary properties, they have no rival. Unfortunately, their mechanical strength and therefore their fatigue resistance is low. Furthermore, their strength is reduced by increase of operating temperature so that in the high speed, high load operating conditions of the modern engine they find little place, although there are still a few applications in the more lightly loaded main bearings of some passenger cars. Their use here has persisted mainly due to their ability to run at extremely small bearing clearances and thus provide a bearing which is quiet in operation. The low melting point of the babbitts and the absence of any supporting metallic structure of higher melting point render them unsuitable from this aspect in high speed engines.

When it became obvious to designers that babbitt was becoming inadequate, attention was turned to copper-lead materials. These contained between 20% and 30% lead with 2% to 4% tin being added to the lower lead alloys. These materials were much stronger than babbitt and thus could sustain higher engine loads, but they were also much harder.

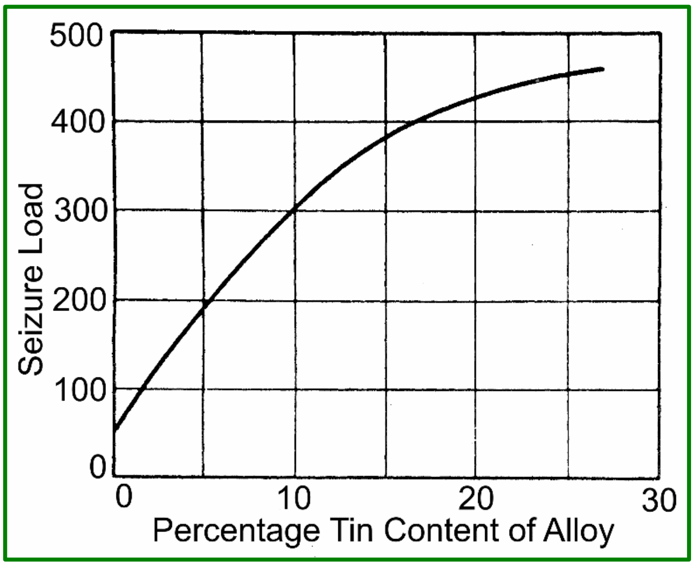
A typical figure for the hardness of babbitt, at normal operating temperatures, would be 20 Brinell, whereas the stronger copper-leads would be as hard as 60 Brinell. This greater hardness reduces seriously the embeddability and conformability of the material, with the consequential result that due to the inevitable shaft deflection in an engine and the necessity for the presence of the two above properties, the full load capacity of the material cannot be realised.

Indeed some of the load capacities quoted for copper-lead can be obtained only in special test machines where shaft deflection and misalignment are absent.

Another grave drawback of a plain copper-lead material is its poor corrosion resistance. In a highly rated engine it is difficult to maintain the oil in a neutral condition, but the lead phase of a copper-lead bearing is rapidly dissolved by acidic oil, removing from the bearing its boundary and wear-resisting properties.

The disadvantages so far described can be eliminated at least temporarily, by the use of electro-deposited overlays. These are either of lead-tin or lead-indium and have a thickness usually in the range 0·0005 to 0·0015-in. These overlays are very soft and have excellent embeddability and conformability. Unfortunately, because of their low strength and their thinness, they can be rapidly removed by abrasive particles and their presence cannot be guaranteed without reference to the operating conditions and particularly the cleanliness of the engine.

The overlays, although thin, are also liable to fatigue damage and, in some cases, the full load capacity of the copper-lead cannot be employed because of the possibility of fatiguing and thereafter removing the overlay and exposing the underlying lead phase to corrosive attack.



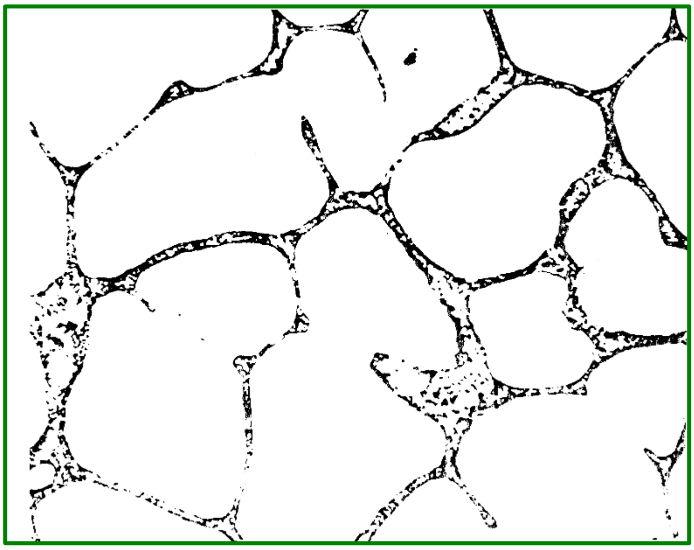
Right: Figure 1. The effect of the of tin to aluminium on the seizure resistance.

It was in the light of these limitations that the Glacier Metal Company Limited decided to examine the possibilities of developing an aluminium-based bearing material. Aluminium bearing alloys were not new. In Germany, before World War II, there had been developed a range of alloys, consisting of hard metallic compounds in an aluminium-base matrix with a structure similar to that of the tin or lead base babbitts. These alloys were exemplified by Quartzal which contained 2% to 15% copper. These alloys were widely used but they were very hard and embeddability and conformability were low, requiring hardened shafts, clean oil and good alignment. About the same time, in Britain and in the United States, alloys were developed of aluminium containing up to about 7% tin, which are still widely used. The presence of tin reduced the solidus temperature of the alloy and Hunsicker showed in 1949 that the presence of increasing quantities of tin up to about 25% improved its seizure resistance. This effect is shown in *Figure 1*.

Unfortunately, with the addition of tin at these levels, the tensile strength, the yield strength, and the ductility fell off sharply due to the tendency of the tin to envelop completely the aluminium grains. This was unfortunate since the properties of the high tin alloys were extremely attractive. However, work sponsored by the Tin Research Institute at the Fulmer Research Institute in England showed that the envelopes of tin which surrounded the aluminium grains could be broken up by working and annealing.

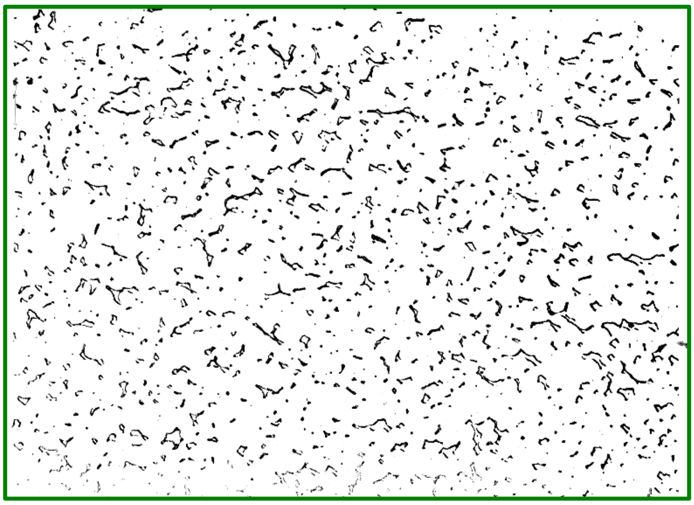
After this treatment the tin remained continuous along the grain edges, but not across the grain faces, so that a strong continuous aluminium-base matrix was established. The tin phase thus formed a network, and from this the term ‘reticular tin’ derived. The development of the bonding of this high tin alloy to a steel backing was carried out by the Glacier Metal Company Limited and the Tin Research Institute in collaboration and a roll-bonding process was established. The establishment of the alloy and the process has been described by Forrester to the Institute of Metals in 1960 and to the Institution of Mechanical Engineers in 1961. The decision on what tin-aluminium alloy to be used was based on consideration of the scuffing load curve, *Figure 1*, the cost of tin, and the required strength in the finished lining. It will be seen that the scuffing resistance is nearly at its peak at a tin content of 20%. It is higher at 30% tin, but at this level the tin content, involving a costly metal, has gone up by 50%. Further, due to the greater volume of tin in the alloy at the 30% level, the mechanical strength of the material, particularly at high temperatures, would be less than at the 20% level. All factors, then, pointed to 20% being the optimum tin content and it was fixed at this level.

Figure 2 is a microsection of a 20% tin-aluminium billet before annealing and roll bonding to the steel billet backing. *Figure 3* is a microsection through the finished material. The steel backing, aluminium foil layer 0·001-in. thick and the now reticular tin-aluminium lining can easily be seen.



Right: Figure 2. Structure of 20% tin-aluminium as cast and unrolled.

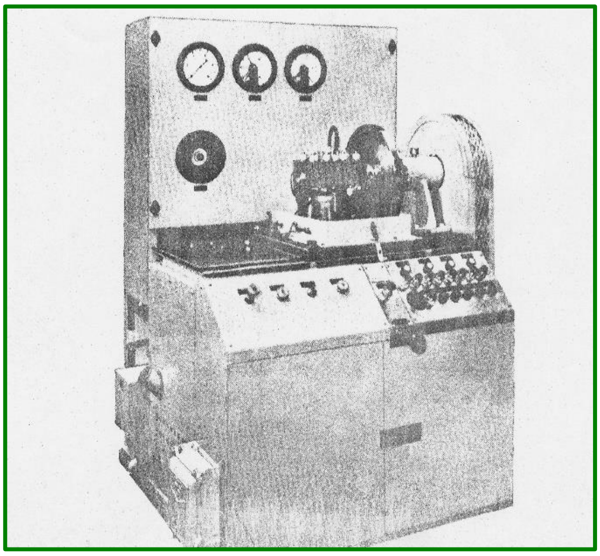
The first assessment to be made of the effective load-carrying capacity of the 20% tin-aluminium material was obtained in a test machine which loaded the bearing dynamically under conditions of operation which were as near ideal as could be obtained. The machine is shown in *Figure 4* and the system of loading the bearing can be seen in *Figure 5*.



Right: Figure 3. Structure of 20% tin-aluminium, roll bonded to steel.

The machine is hydraulically loaded with the test bearing carried on an eccentric journal on a very stiffly supported shaft. There was negligible edge loading on the bearing and the dynamic load was unidirectional. The limit of loading for fatigue at 5 x 10^6 cycles for the 20% tin-aluminium was 6,400 p.s.i. Comparative results obtained in this machine for other bearing materials are shown in Table 3.

These tests indicated that an increased usable load capacity was being obtained and that the pattern and appearance of the fatigue damage did not suggest that the mode of failure would be different with this material from those in common use.



Right: Figure 4. Single-head, dynamically loaded bearing fatigue test rig.

Following this, another series of tests were carried out on another fatigue test machine, shown in *Figure 6* and in detail in *Figure 7*. In this machine the load was applied to the bearings by a fairly flexible shaft which rotated at 4,000 r.p.m. and carried two eccentric masses. The masses could be adjusted in weight and were capable of applying specific loads up to 10,000 p.s.i. to each of the four test bearings which supported the shaft. This machine, with its rotating loads and flexible shafts, applied a load to the test bearings which was more typical of that existing in an automotive engine and therefore the fatigue strengths of typical bearing materials was lower than in the other machine. In this case, the effective fatigue strength of the 20% tin-aluminium material was 4,500 p.s.i. The comparative fatigues strengths are shown in Table 4.

It should be noted that the effective fatigue result for 70-30 copper-lead on steel in Table 4 is not consistent with that in Table 3. This is because of the shaft deflection and the rotating load in the second machine. The fatigue of the unplated copper-lead was aggravated by the occurrence of incipient seizure of the material.

These results indicated that in terms of dynamic load capacity, the 20% tin-aluminium was equal to any of the other materials which had the overall combination of properties which made them suitable for use in internal combustion engines. It should be noted, that in Tables 3 and 4, no strength comparisons are made with other materials which, although stronger, would have unacceptable wear or corrosion properties. The next property of the material to be examined was corrosion resistance and a lengthy series of tests was carried out with the 20% tin-aluminium bearings immersed in acidic and alkaline oils and additives. This work was done by Glacier and by several oil companies, particular attention being paid to the effects of alkaline attack. This was because of the known effect of alkalis on aluminium itself, but no damage was sustained in any test.

One of the most important properties of a bearing material is its wear performance exemplified by the wear of the material itself and also by the wear of the associated shaft material. To a large extent, the important wear figure in a bearing is the total change in clearance since this determines whether the engine must be dismantled for bearing overhaul and, to some extent, whether the crankshaft must be reground.

Glacier 20% reticular tin-aluminium had been designed to run against unhardened shafts (down to 200 Brinell) and long-term tests were set afoot in the Company's and customers' motor vehicles to assess wear performance with soft shafts. In recent years, the distance run by vehicles between over-hauls has increased considerably, and this means that very large distances must be run to assess the material. The copper-lead bearing materials which run with hardened shafts have excel-lent wear properties as long as the lead-tin or lead-indium overlay is not worn off. Once it has been worn through, after say 75,000 miles, the shaft wear rate against the hard copper-lead or lead bronze is extremely rapid. Typical comparative wear rates are shown in *Figure 8*. Initially the wear rate with both materials is small, although slightly more rapid with the aluminium-tin material, until at, say 60,000 miles, the overlay is worn off and the copper-lead wear rate increases. If the engine is dirty or filtration neglected, the increase of the wear rate of plated copper-lead will occur earlier in the life of the bearing. *Figure 9* shows a typical set of main bearings where the wear rate of the plated copper-lead bearings has become accelerated. The overlay has been removed in certain areas and the hard copper-lead ex-posed. It was also found that if engines containing tin-aluminium bearings were inspected after short periods of running, early in their life, they appeared to be wearing more quickly than plated copper-lead It was quickly found that this was a visual effect due to the absence of an overlay and could be ignored.

The final formal wear test of the material was a 60,000 mile run at 60 m.p.h. with 1,000 cold starts of six 1,000 c.c. British cars, three fitted with Glacier 20% reticular tin-aluminium bearings and three with a competitor's overlay plated lead bronze bearings. In each case, the total change in bearing clear-ance in the vehicles fitted with Glacier bearings was less than in the others. The results are summa-rised in Table 5.

A set of the 20% tin-aluminium bearings after completing the test is shown in *Figure 10*. In order to demonstrate that the material could survive not only under, somewhat sheltered test conditions, a 10,000 mile test under racing conditions, including participation in actual races, was carried out in a B.M.C. Cooper car. The condition of the bearings, which is excellent, is shown in *Figure 11*. Since this development stage, considerable service experience has been gained, where the reticular tin-alumin-ium bearings operated under severe conditions for long periods. An interesting published example of this is an article in the British '*Engine Design and Application*' for September 1965, where Perkins Engines Ltd. describe the successful performance of one of their 6-354 engines in 320,000 miles uninterrupted service. In this case, in common with other engine components, the reticular tin-alumin-ium bearings were in excellent condition and fit for further service after engine overhaul.

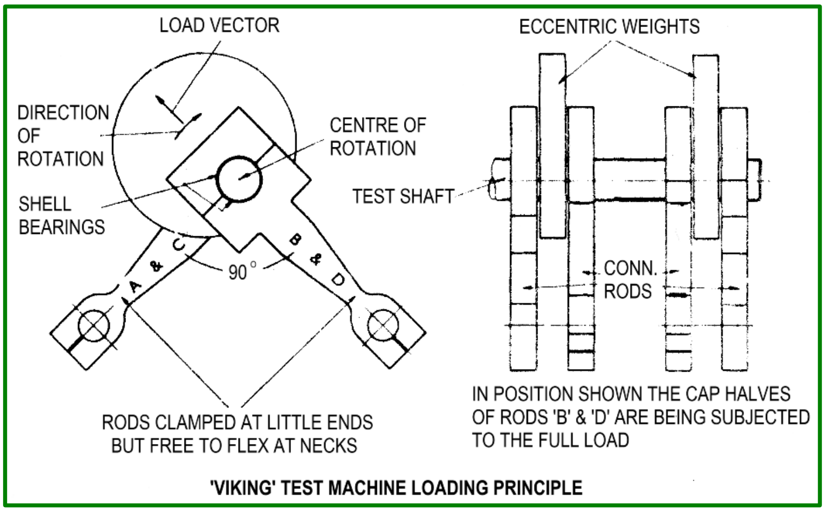
A very useful additional benefit from the reticular tin-aluminium structure – and one of particular use to the engine reconditioning industry is that the bearings are resizable.

Prefinished white metal and plated copper-lead cannot be resized for obvious reasons but the homogeneous structure of reticular tin-aluminium enables engine reconditioner to resize these bear-ings to suit a particular shaft size should this be necessary.

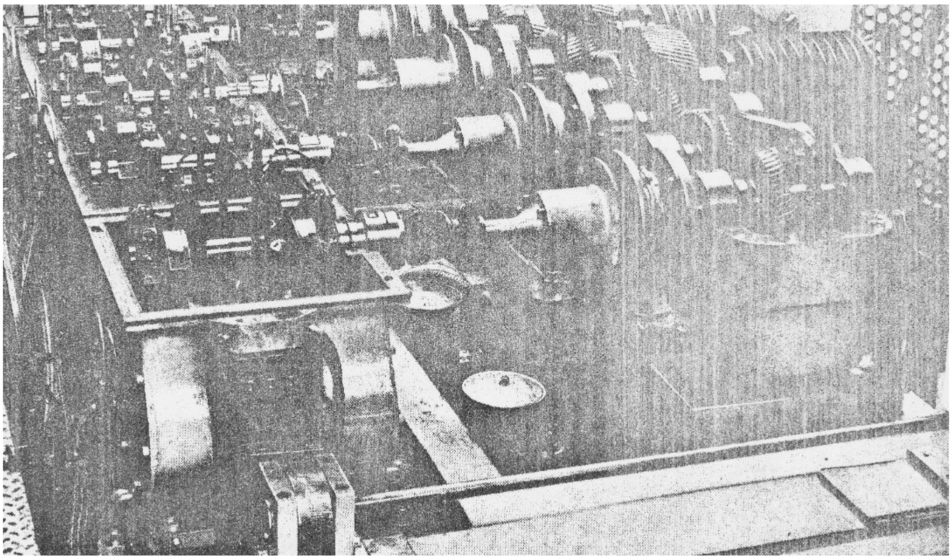
The Current Applications Of Glacier 20% Tin-Aluminium Bearings

Since reticular 20% tin-aluminium was introduced by Glacier and its Licensees some ten years ago, it has taken a prominent place in bearing materials used in Britain and in Europe. It is used as original equipment by B.M.C., Vauxhall, Ford (England and Germany) and Rootes, in their passenger cars and trucks and is also specified by the larger British commercial vehicle manufacturers, A.E.C. and Leyland. A list of users of the material is given on Page 22. The material is also widely used in larger stationary and marine-propulsion diesel engines, in diameters up to 10-in. It is also being used almost universally as the material around which new engines are designed and developed. This success arises from the fact that for the first time, a bearing material has been specifically formulated and developed for the work it has to do and the market it has to serve. Into Glacier 20% reticular tin-aluminium bearings are built the qualities of strength, melting point, corrosion resistance, embedda-bility, conformability, hardness and surface condition in the proportions which are required to fit it exactly for the bearing requirements of the engines of today and of the future.

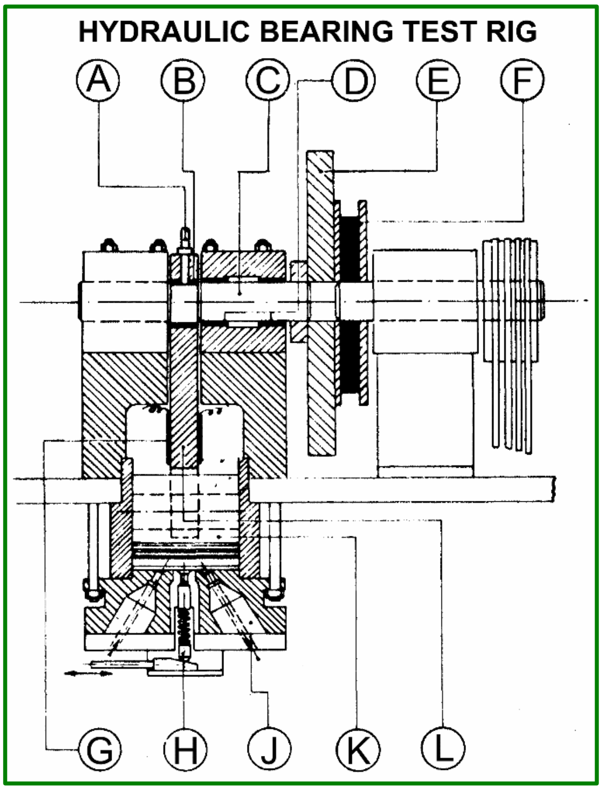
Right: Figure 5. Layout of loading system for machine in Figure 4.



Right: Figure 6. Multi-head, dynamically loaded bearing fatigue test rig.



Right: Figure 7. Arrangement of test head on machine in Figure 4.



Legend for *Figure 7*:

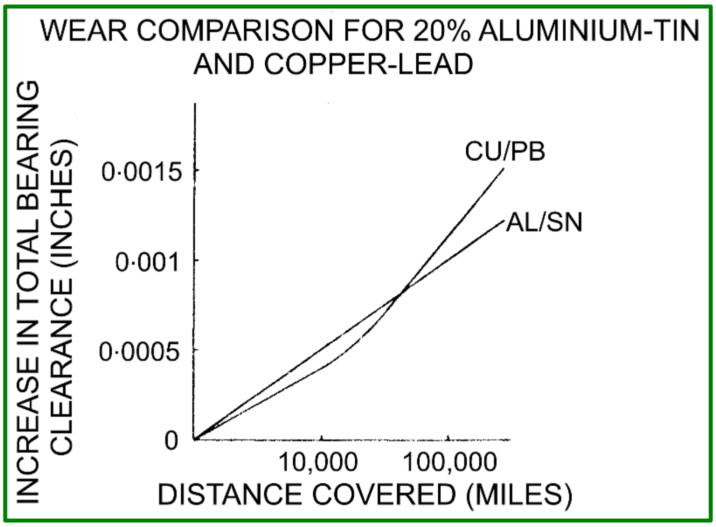
A – Lubricating oil inlet. B – Test bearing.

C – Test shaft. D – Slave bearings.

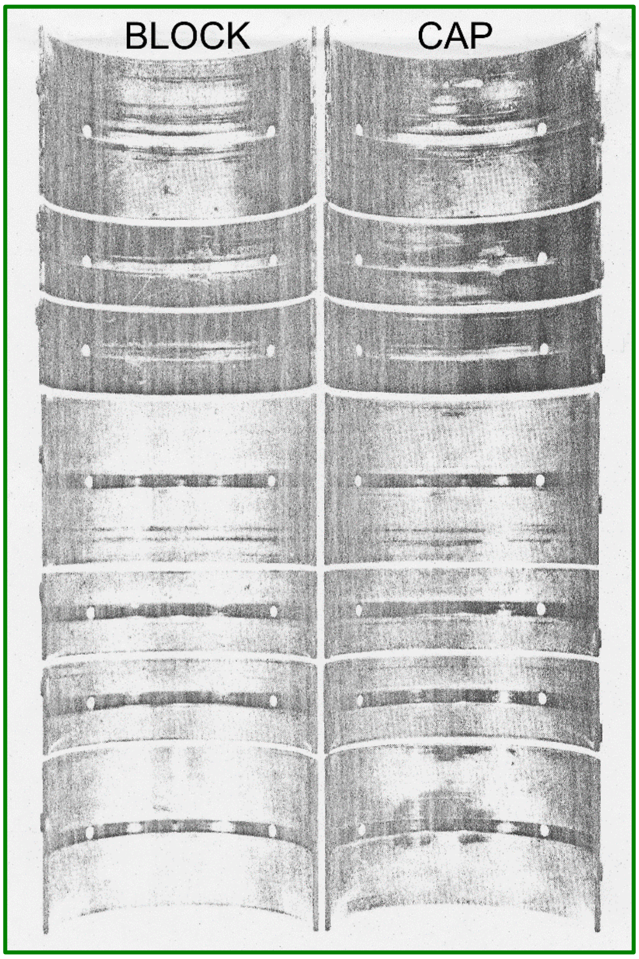
E – Flywheel. F – Flexible coupling.

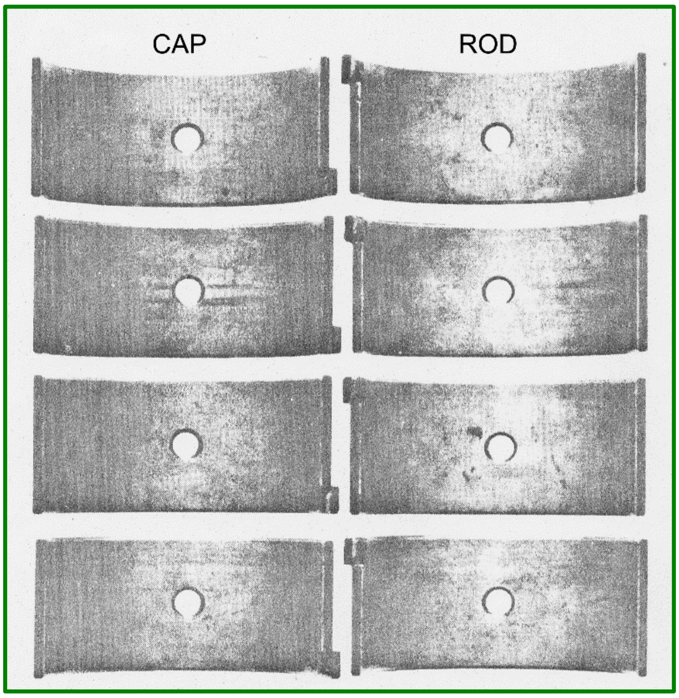
G – Strain gauges, H – Adjustable peak pressure valve. J – Hydraulic oil inlet valve.

K – Piston. L – Connecting rod.



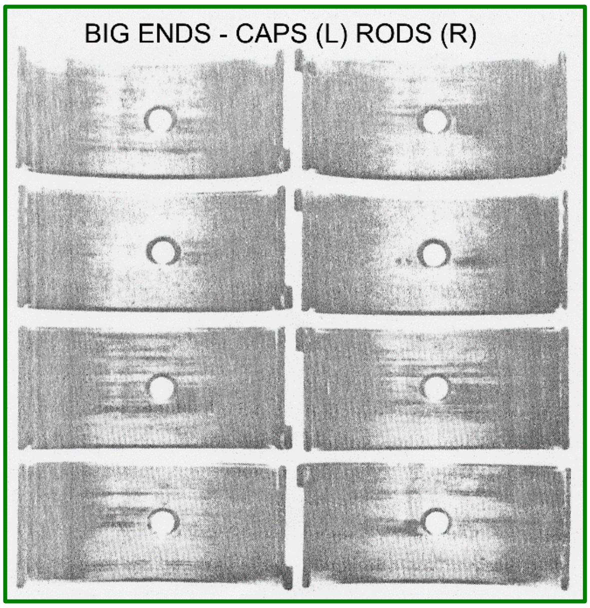
Above: Figure 8. Comparative wear of bearing materials.

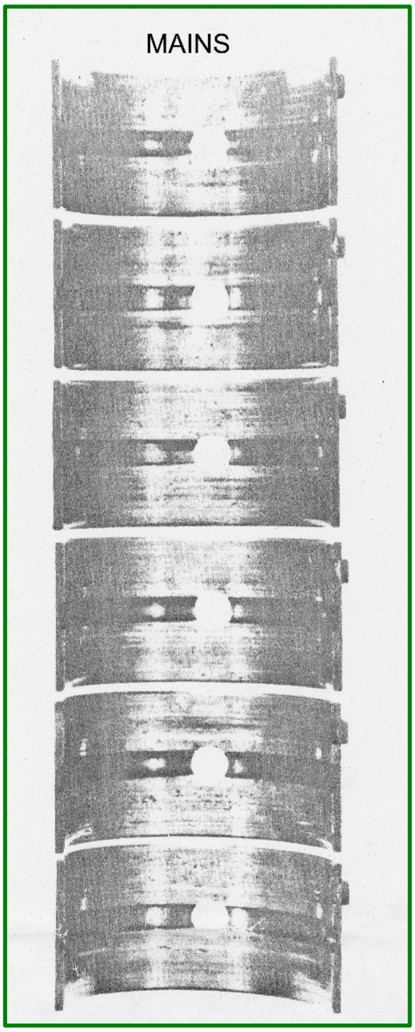




Above: Figure 10. Glacier 20% tin-aluminium bearings after 60,000 miles.

Above: Figure 9. Comparative wear performance for Glacier 20% reticular tin-aluminium material and an overlay plated copper-lead material.





Above left and right: Figure 11. Bearings from B.M.C. Cooper engine after 10,000 racing miles.

Table 1.

TYPICAL APPLICATIONS FOR GLACIER AS15 IN PETROL ENGINES

Capacity (Litres)	Engine Type	Bore (inches)	Stroke (inches)	Conn. Rod Bearing Load at Max. Torque (P.S.I. at R.P.M.)	Conn. Rod Bearing Load at Max. Speed (P.S.I. at R.P.M.)
1·0	4 cyl.*	2·458	3·200	4,100 at 3,600	3,150 at 6,000
1·5	4 cyl.*	3·188	2·867	4,350 at 3,600	3,020 at 6,000
0·9	4 cyl.*	2·677	2·377	4,725 at 2,800	2,650 at 6,000
0·9**	4 cyl.*	2·677	2·377	–	3,750 at 7,500
1·7	4 cyl.*	3·205	3·260	3,970 at 2,600	4,080 at 5,500
1·7	V4	3·920	2·275	5,096 at 3,000	3,069 at 6,000
2·0	V4	3·920	2·845	5,743 at 3,000	3,473 at 6,000
1·8	4 cyl.*	3·160	3·500	–	5,210 at 6,000

* 4 cyl. In Line. ** Uprated Version of Above.

Table 2.

TYPICAL APPLICATIONS FOR GLACIER AS15 IN DIESEL ENGINES

Capacity (Litres)	Engine Type	Bore (inches)	Stroke (inches)	Conn. Rod Bearing Load at Max. Torque (P.S.I. at R.P.M.)	Conn. Rod Bearing Load at Max. Speed (P.S.I. at R.P.M.)
8·5	V8	4·250	4·500	4,450 at 1,000	3,030 at 3,290 4,980 at 4,500*
5·8*	6 cyl.^	3·875	5·000	6,314 at 2,000	6,000 at 2,800
5·4	6 cyl.^	4·0625	4·250	4,980 at 1,500	3,140 at 3,000
5·8^^	4 cyl.^	3·875	5·000	8,080 at 1,350	5,300 at 2,800
5·8	6 cyl.^	3·875	5·000	4,115 at 2,000	3,616 at 5,500

^ 6 cyl. In Line. ^^ Super Charged. * Turbocharged.

Table 3.

'EFFECTIVE FATIGUE STRENGTH' OF BEARING MATERIALS:
UNDER NEAR IDEAL CONDITIONS

Babbit (Tin Based) 0·020-in. Thick on Steel	2,250 p.s.i.
Babbit (Tin Based) 0·006-in. Thick on Steel	3,400 p.s.i.
Overlay Plated Copper-Lead on Steel	5,000 p.s.i.
70-30 Copper-Lead on Steel – No Overlay	5,200 p.s.i.
Reticular 20% Tin-Aluminium on Steel	4,500 p.s.i.

Table 4.

'EFFECTIVE FATIGUE STRENGTH' OF BEARING MATERIALS:
UNDER SIMULATED OPERATING CONDITIONS

Babbit (Tin Based) 0·020-in. Thick on Steel	1,650 p.s.i.
Babbit (Tin Based) 0·006-in. Thick on Steel	2,250 p.s.i.
Overlay Plated Copper-Lead on Steel	4,300 p.s.i.
70-30 Copper-Lead on Steel – No Overlay	3,800 p.s.i.
Reticular 20% Tin-Aluminium on Steel	4,500 p.s.i.

Table 5.

TOTAL CHANGE IN CLEARANCE OF BEARINGS AFTER 60,000 MILES
AT 60 M.P.H. AND 1,000 COLD STARTS

		Mean Change In Clearance
Vehicle 1.	Glacier 20% Tin-Aluminium	0·0003-in.
Vehicle 2.	Glacier 20% Tin-Aluminium	0·0004-in.
Vehicle 3.	Glacier 20% Tin-Aluminium	0·00035-in.
Vehicle 4.	Overlay Plated Copper-Lead	0·0005-in.
Vehicle 5.	Overlay Plated Copper-Lead	0·0006-in.
Vehicle 6.	Overlay Plated Copper-Lead	0·00055-in.

USAGE OF GLACIER TIN-ALUMINIUM AS15 ENGINE BEARINGS BY BRITAIN'S MOTOR INDUSTRY

Passenger Car Usage

AS15 hearings are fitted as original equipment in the following current production models:

B.M.C.	Austin A40	Austin Healey Sprite
	Austin 1100	MG 1100
	Morris 1100	Vanden Plas 1100 Princess
	Morris Minor 1000	Austin A60 (Petrol)
	Austin Mini Minor	Austin A60 (Diesel)
	Morris Mini Minor	Morris Oxford (Petrol)
	Austin Cooper	Morris Oxford (Diesel)
	Morris Cooper	MG Magnette
	Riley Elf	Riley 4/72
	Wolseley Hornet	Wolseley 16/68
	MG Midget	
ROUTES GROUP	Hillman Imp	Sunbeam Rapier
	Singer Chamois	Sunbeam Alpine
	Hillman Minx	Singer Gazelle
	Hillman Super Minx	Singer Vogue
	Humber Sceptre	
VAUXHALL	Viva	Velox
	Victor 101	Cresta
	VX 4/90	Viscount
FORD	Anglia	Zephyr 4
	Cortina	Zephyr 6
	Corsair	Zodiac

Truck And Tractor Usage:

B.M.C.	Austin and Morris Commercial:	1·5 litre 4 cyl.
		2·2 litre 4 cyl.
		2·55 litre 4 cyl.
		3·4 litre 4 cyl.
		3·8 litre 4 cyl.
		5·1 litre 6 cyl.
		5·7 litre 6 cyl.
VAUXHALL – Bedford	214 cu. in. Petrol	200 cu. in. Diesel
	220 cu. in. Petrol	330 cu. in. Diesel
		60/70 Engine Diesel
A.E.C.	A410/A470 Diesel	A590/A690 Diesel
FORD	E1DDN Industrial Diesel	New Dorset Diesel
	Dearborn Diesel (Tractors):	Dexta 2,000
		Super Dexta 3,000
		Major 4,000
		Super Major 5,000
PERKINS		
	'P' Series	3-144
		3-152
		4-192

		4-203
		4-288
		6-305
	Remainder of Perkins Range:	4-270
		4-300
		4-107
		4-99
		4-236
		6-354
LEYLAND	370/400	600/680

Restored by Mike Allfrey. – February, 2024.
Taken from a faint photocopy handed down by the late John Taylor.
From a Glacier publication – with thanks,
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