CYLINDER BLOCKS

R-R Silver Dawn, Silver Wraith, Bentley MKVI, R type and S1 cars

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The problem with the RREC Hunt House HQ is that so much data is available that one would need to be well over 100 years old to have read it all and spent the entire time at that location. This, for the purist, is not mathematically worked out, but a guess. I would be completely open to any higher assessments.

The problem with the staff, from the top down, is that they are too helpful and will insist on directing you to mountains of information when you ask for that small, detailed request. Anyone not familiar with the location, or the staff, should be warned to take at least three meals and plenty of coffee whilst researching that small problem. If more than one search is envisaged an overnight bag can become a required accessory. An absolute mountain of data is available and one should be warned of the certain possibility of being side tracked and going off at a tangent from the target for the day's research. One advantage, if you are an automotive engineer, is the thought that this could be heaven in a different guise. One of my encounters at Hunt House involved research of automatic transmissions for the post war cars, this extended from about two full days to seven days and I have still not finished.

I did have an excuse for getting side tracked. During that time I was mindful of eventually needing to overhaul the engine, in my own R type chassis B87 UL, a 1953 automatic. Someone had just bored it, and fitted pistons, before I purchased the car. I decided that it was most likely to still have half liners fitted and eventually it would require attention again, probably rather quickly. I am very much aware that the cooling system is a weak link and I was determined that once the engine had been overhauled correctly, it should be given the best chance to outlast at least the owner's life cycle.

Some of the research into R Type automatics involved listing gearbox numbers and inevitably becoming engrossed and reading every chassis card in full. Most noticeable at the time were the chassis cards of cars destined for Australia and Switzerland. They told a story in which I was most interested, with my impending engine overhaul and interest in the cooling system. The details of these chassis cards are related later.

The object of this article is to highlight some of the potential problem areas arising from, and also causing, interference of heat transfer and the inevitable differential expansions that occur. Hopefully it may be of interest to the enthusiast contemplating an engine overhaul and also provide some food for thought as to how and why the cooling arrangements may be modified. All measurements throughout are in inches unless specified.

SOME COOLING RELATED PROBLEMS.

Prior to the early 1960's antifreeze inhibitors were not effective and the Ford Motor Company was one of the first to instigate standards to address the situation, whereupon the rest of the industry followed. The ineffectiveness of early inhibitors was the cause of heavy deposits of silt in the cooling systems. Some of the cars have had engine and cooling system overhauls since the introduction of better quality antifreezes. Silting on these overhauled units should be all but eliminated effectively by flushing and antifreeze changing at least every two years. There are still a number of cars that have drastic cooling system blockages and, worse still, owners who do not realise the potential damage that this causes, particularly in this side exhaust valve design. They would be well advised to attend one of the RREC seminars on the post war six-cylinder cars where their knowledge of the engines would be supplemented and their bank balance saved from going into potential dive mode.

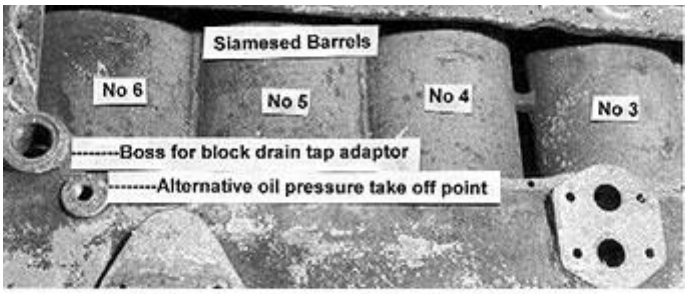

FIG1. Late type R type cylinder block RE 16217, with the side cut away.

The cooling system is based on a thermo syphon system assisted by the water pump flow. Pump output is directed along a removable cooling tube in the top of the cylinder block and discharged through rectangular ports at 90□c directly to the underside of the exhaust valve seats. After cooling the valve seats the coolant is directed against the top part of cylinder barrels, on the exhaust side of the engine. The joining together or siamesed cylinder barrels in the later engines effectively obstructs the lateral flow of coolant across the cylinder block. Even the very early blocks, with slight gaps between the barrels, inevitably silt up the passages causing the piston thrust side of the barrels to be in the shade regarding water flow.

Fig 1 shows very clearly the siamesed barrels in a late R type cylinder block, photographed from the carburettor side of the engine. Just out of view, on the left and above the outlet for the drain tap, is the coolant passage between No 6 cylinder barrel and the rear of the block. The small gap between No 4 and No 3 cylinders can clearly be seen, as can the link core adjoining the two barrels half way up the bores. Note how effectively No 5 cylinder barrel is sandwiched between the adjacent cylinders, this situation is also repeated with No2 cylinder. Clearly, the barrel cores restrict the water flow from the other side of the block. When heavy silt is allowed to accumulate, this side of the block can be full, up to 2 inches or even more above the top of the cylinder block tap drain outlet. The depth of the silt can give a false illusion that the cylinder block has been fully drained. In this view the holes around the block periphery are the result of a botched attempt to fit a side plate, in the mistaken belief that this would affect a cure for a cracked block.

Most of the water flow to the thrust or intake side of the block must take the torturous route through the centre between cylinders 3 and 4, or around the ends of cylinders 1 and 6. This situation is made worse if the cylinder block to cylinder head passageways is blocked or if a large amount of silt has been allowed to accumulate around cylinders 5 and 6. If that condition exists, any coolant flow around the rear end, generally the hottest part of the engine, becomes very restricted. If you are lucky and the cylinder block to cylinder head passages are actually free, some coolant may even find a way into the cylinder head. All this of course, even in a clean engine, depends, in the first instance, on whether or not adequate volumes of coolant actually reach the rear end of the cooling tube leading from the water pump. The pump, which normally runs at less than engine speed and with a large rotor to back plate clearance, must be struggling to attain adequate flow, particularly at idle speeds. Silt deposits naturally settle in the rear end of the block and at the bottom of the coolant jacket on the inlet side. My train of thought is that improved scouring and better heat dissipation can only be improved by more vigorous coolant circulation. Of course any minor silt deposits will not disappear and will need removing by regular flushing of the cooling system. Heavy silt deposits can only be removed by physically cleaning the cylinder block, with all the necessary access plates removed.

The radiator is also an effective filter of silt particles and needs clearing out and back flushing vigorously whenever cylinder block silting is found. Fine silt particles do eventually reach the bottom of the radiator. The water pump does not possess enough lifting capacity to bring all these heavier particles back into the cylinder block system, particularly when the pump outlet cooling tube is restricted. These heavier particles are drawn towards the bottom radiator hose outlet and collect in the area of the bottom hose radiator connection. A cooling flow restriction is then formed and if the blockage is severe, or the bottom hose is weak, the hose will collapse inwards under the influence of the water pump suction when the engine revolutions are raised. In this condition a very effective cooling system restriction is formed and the evidence of the hose collapsing is gone when the engine is inspected under idling conditions. Fitting a cooling system filter in the top hose will protect the radiator from excessive silt return, but it is important to frequently service the unit, particularly if the internal condition of the cylinder block is unknown.

Anyone studying the drawings of the pistons, liners and particularly the cylinder block cooling flow on these post war six-cylinder engines will quickly realise some potential problem areas. The engineering picture backs up practical experience in that these engines suffer from quite a degree of differential expansion, some quite uncommon, due to the side exhaust valve design. The peculiar heat transfer problems are confirmed by the wear pattern when these engines are examined internally. Although the company and the piston suppliers in particular, went to some length to circumnavigate the expected problems known to arise from differential expansion, they appeared to be only partially successful. The majority of original tests were conducted on engines that, prior to testing were in either new or very good condition. After many years of service, sometimes in conditions of neglect, like some of us, the cylinder blocks in particular, are showing the signs of stress.

These points however, should not be taken out of context; we all have some advantage over the original engineering and design staff, that of 50 or 60 years of hindsight. Furthermore, it should be remembered that these were, without doubt, the finest engines available at the time. It is a credit to the original engineering staff that the engines have a remarkable ability to keep running, long after other designs have long since died the death.

The object of test bed and road testing is to subject the engine to as much stress as necessary in order to highlight and eliminate any failings that may be expected to arise in service. Although testing cycles endeavour to parallel service conditions, it is very often the case that they fail to replicate faults that may occur after many temperature cyclic duties, or peculiar operating cycles, have taken place over years of service. In other cases, rather rare test bed failures occur, which are discounted as not likely to occur in service. Unfortunately these types of failure have a nasty habit of changing after years of service from rare to frequent. Test results have a remarkable ability to discount years of maintenance neglect and that specific animal, sometimes defined under the term driver, but more often better described as a motor mover.

At the risk of repeating myself, the importance of de-silting cylinder blocks and then regularly flushing cannot be emphasised enough. The hard silt in these blocks will not be removed just by inserting a pressure hose; the silt is so hard, almost like larva rock and requires very hard mechanical scraping. Owners are generally not inclined to remove the block side plates and apertures, but I am afraid there is no substitute. Running water through the block until it appears to egress clean has only removed the merest of silt traces and the dangerous hard silt, which is preventing good heat transfer to the coolant, will still be in place. It is unfortunate that owners are only too willing to convince themselves that they have flushed the offending silt away. In this respect a particular fact is clear. Once an owner has experienced the de-silting of a cylinder block and filled two or three large coffee jars with the offending matter, that owner does not require any further convincing.

HARDNESS

The components under discussion later invariably had a hardness specification and before describing individual parts in detail, some description of hardness is required.

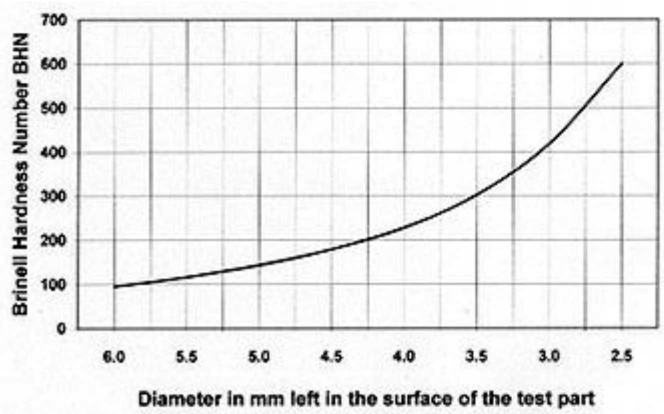

FIG2. Brinell Hardness number graph, showing BHN with a v3000 kgs load on a 10 mm ball.

Hardness is best described as the resistance to penetration by other objects. One must therefore have some means of testing the component part surface resistance to penetration, or deformation, and compare the results.

Brinell hardness numbers or BHN, were usually used for the larger castings. The BHN number is found by indenting the part to be tested by a spherical surface, which is forced into the test piece by a known force. The hardness number in BHN is calculated by taking the total pressure applied and dividing it by the curved surface area of the depression made in the test part by the applied load. In practice this curved surface area is related back to the diameter of the impression and is measured by a scale across the lens of a microscope, specially used for the purpose. A comparison is then made of the depression diameter, using known tables.

Fig 2 shows a graph, which gives the BHN numbers when a 3000 kg's load is applied to a 10 mm diameter ball and the resulting diameters of the depression. In very practical terms, if one took, say a fully loaded Phantom III and rested the entire weight on a 10mm ball resting on a piece of steel, if the resultant indentation on the steel was 4.00mm diameter, then the BHN number would be 228. This 228 BHN incidentally, is a very good hardness rating for a cylinder block but using a Phantom III is not a very practical way of measuring. Especially for smaller parts, quicker methods are used.

Usually either chromium or chromium and copper will be added to cast iron to improve its hardness, but other difficulties apply on large castings, such as keeping the temperature of the iron. The larger the casting the more difficult it can be to keep the temperature and very often chills or hard spots develop. A good example of hard spots can normally be found in the flywheel plates on the crankshaft dampers on these engines, where some of the surface is harder than the adjacent section.

The Vickers Pyramid Numeral system, VPN, and the Rockwell C method will occasionally be encountered. The VPN system uses a small pyramid diamond as the indenter and the test can be conducted on a Vickers machine without the use of outside power although modern machines use electrics and hydraulics for convenience. The different methods usually relate to the physical size of the part to be tested and the ease and speed of the test. An equivalent to Rockwell C49 – C55 would be a BHN of 472 – 547. Hardness at these high figures could be described as extremely hard as against, say, a hardness of BHN 300- 350, which would still be largely untouchable when using a file.

CYLINDER BLOCK VALVE SEAT CRACKS AND VALVE INSERTS

A major fault in a block casting is the dread of every enthusiast, but after some years of service, faults can arise within castings and especially take place in areas where differential expansion occurs. Some of the faults mentioned later had started to show up during testing and were well known, others have arisen after years of service. Certainly the availability of good and consistent quality foundry castings was a thorn in the side of most engine manufacturers during the 20 years after the war. All the foundries capable of producing large cylinder block and head castings in either aluminium or cast iron, experienced similar troubles. To a large degree, sustaining casting quality at the time, whilst being under the pressure of retaining production volumes, even stretched Rolls-Royce staff to the limit. These casting are the main foundation of any engine and it can be seen throughout the history of these engines that, in many cases, they were the source of many of the troubles that may be experienced. That is not to suggest in every case that the foundry was at fault, in the case of these six cylinders, no doubt the particular design also played a major part.

With the exception of the exhaust valve face and its seat, the hottest part of this engine is the bridge piece of the top cylinder block deck between the exhaust valve seat and the parent cylinder barrel. During the exhaust stroke the hot gases must follow an inverted ‘U’ shaped path from the cylinder to the exhaust port. One of the obstructions is the edge of the liner and also the resultant throat restriction between the top of the combustion chamber and the edge of the liner, on the exhaust valve side. Any obstruction will naturally cause power losses and localised hot spots; in this case they occur in an area that already suffers the highest heat retention. These particular issues normally limit the compression ratio to a maximum of 8.5:1 on this type of design, when natural aspiration is used. The exhaust valve will shed heat directly to the seat and, to a lesser degree, to the valve guide. The heat actually passed through the seat can only be transferred to the top deck bridge; this entire area relies heavily on reasonable quantities of water impinging on the bridge exhaust port and seat areas for final dissipation to the cooling system. These engines have little or no valve overlap whereby both valves are open together for a short time and some of the incoming cool charge can pass through the exhaust and assist in valve and seat cooling. The nature of the design of the side exhaust

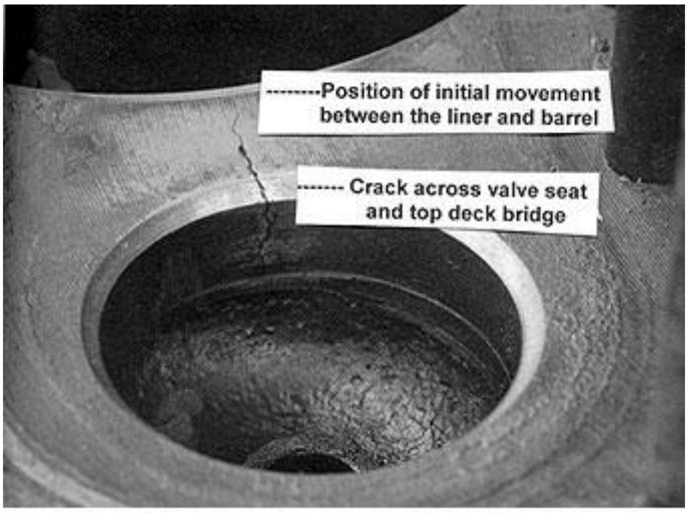

FIG3. R type cylinder block, showing cracked valve seat and top bridge.

valve and port area does cause extreme temperature gradients to occur across the top deck bridge and the valve seat and valve face have to contend with the 550 / 600 lbs. maximum pressure combustion loadings. The bridge temperature, I would assess, would be in the order of 400-500C and as high as 600C in severe cases of overheating.

Fig 3 shows the result of very hard driving on a 4.5 Litre; the crack rising in the valve port core has crossed the seat area and continued across the top deck bridge piece. Very often smaller cracks appear which are more over to the left or right hand side than that shown in Fig 3. All these cracks have one thing in common; they commence in the exhaust port under the seat and continue in the direction of the cylinder. Very rarely will a crack appear in the seat area on the opposite side to the cylinder bore. The company initially installed exhaust valve seats, on production engines, to combat this defect.

The later engine sizes are right on the edge of requiring valve seat inserts and it may be of interest to follow the history of these inserts from conception. Initial trials started in 1939 with the fitting of inserts to B80 engine no 5, this engine was subsequently fitted into chassis no 30 G V11, which covered 160,000 miles during war time. These particular inserts were made from Brichrome, the same 30% chrome cast iron used for the later top cuff cylinder liners.

As production gained pace after the war a few of the earlier 4.25 engines were fitted with inserts for production salvage reasons, usually due to incorrect initial machining. For historic interest, by August 1951, the following chassis had their engines originally fitted with exhaust valve seat inserts under this PSS scheme and many more were to follow.

BENTLEY MKVI

B198 AK. B258 BH B312 BH B346 BH B337 CD. B103 GT

B218 AK. B260 BH B326 BH B348 BH B405 CD.

B222 AK. B266 BH B328 BH B352 BH B413 CD.

B226 AK. B268 BH B330 BH B354 BH B439 CD

B5 AJ B274 BH B332 BH B362 BH B477 CD.

B7 AJ. B278 BH B334 BH B366 BH. B142 DA

B11 AJ. B280 BH B336 BH B368 BH B302 FV

B21 AJ B298 BH B340 BH B378 BH B354 FV

B35 AJ. B308 BH B344 BH B384 BH B476 FV

SILVER WRAITH

WYA 1. WYA 27. WYA 33. WAB 58.

WYA 7. WYA 28. WCB 5. LWGC 88

WYA 25. WYA 32. WGC 71. WME 38

The material used under the PSS scheme was Brichromium, a much lower chrome content iron than that first used in 1939. All these inserts were fitted by heating up the blocks and pressing in the inserts with a very high 0.0065 – 0.0085 interference fit. The company machined all these early PSS inserts from cast sticks. In 1951 service records for all the above chassis indicated that no problems had been experienced with the seat inserts. It is perhaps also significant to note that all these chassis were fitted with relatively low output 4.25Ltr engines, furthermore, with the exception of Silver Wraith WME 38; they would have been cooled by the high speed water pump and fan.

Although the production scheme had used Brichromium material for the inserts, as early as autumn 1948, Wellworthy Valmet, a 15% chromium iron, had been specified for the C60 single cylinder laboratory engine. This latter material was eventually to become the normal standard many years later. It was found to be very resistant to valve seat pocketing, particularly when rotators were used on the valves in B81 engines.

Later inserts were supplied ready made to the company in Brichromium, under part number EB 4027, to a BHN of 220/260. Normal 4.25Ltr engine cylinder blocks at the time were in the range of 202/240 BHN. Company interest was awakened to exhaust valve seat inserts once again, initially to salvage exhaust seats on higher power test engines.

Although a few valve seat cracks on 4.25 engines were experienced from very late in 1951, these had all occurred due to the presence of residue sand and flash in the water passages. However the situation changed and had become quite disturbing by the end of 1953. One early 4.5 Litre engine in Bentley B461 NY, suffered in particular, even though the block was unusually clean.

The 4.25 and 4.5 cylinder blocks are hard enough around the exhaust valve seats to cope with unleaded fuel, providing the engine speed is restricted. One shortcoming is that the seats are still subjected to distortion and cracking if the bridge piece heat transfer is interrupted. Distorted seats will quickly bring about failed exhaust valves and it is advisable to fit seat inserts when engines are removed for overhaul, for peace of mind if nothing else. This suggestion is not directly related to the use of unleaded fuel but to towards valve seat cracking, which if left unchecked, means that, the cylinder block will be scrap.

Fig 4 shows the underside of the top deck, or cylinder block ceiling, when the crankcase top has been sawn off. The smaller holes along the top are the valve guide and inlet push rod locations. The area where dangerous silt build up can take place, between the parent barrels and exhaust valve ports, is clearly shown. In practice it is difficult to clean this area, especially when the silt has compacted and hardened because access is very limited. Comparisons between fig 3 and this photograph will show that the cracked top deck bridge area runs directly above the cooling tube. One of the most important functions of the cooling tube is to sufficiently scour the areas around and between the exhaust ports, to prevent or displace any steam pockets forming. Later S1 crankcases had bulged exhaust ports to increase the area available to the coolant. The siamesed barrels and the difficulty of providing even cooling of the cylinder barrel areas can also be seen in this view.

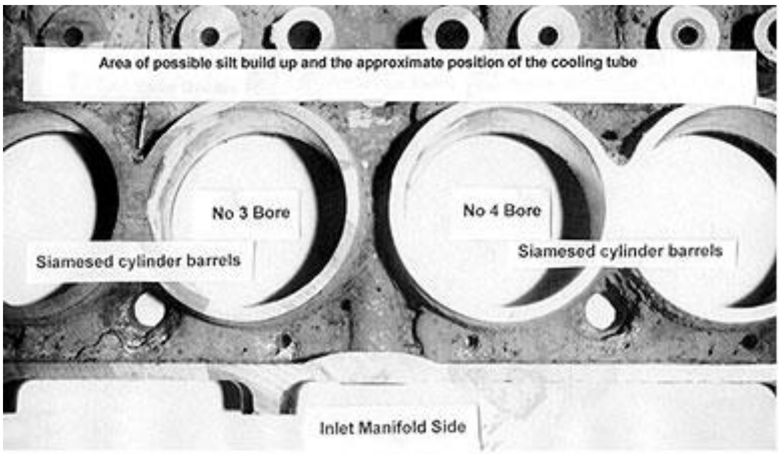

FIG4. Cylinder block top deck cut through on a 4.5 ltr R type.

The 4.5 litre engine in particular is known to be on the limit of cracking in the vital top deck bridge area. Only during the course of the last few days have I seen, once again, two different 4.5 Litre engines with exhaust seat cracking. On a few engines, problems of residual casting sand and flash caused some exhaust valve seat recession and cracking to occur, due to the interference of heat dissipation to the coolant. Cast iron, when it is subject to stress and high temperature cyclic gradients, will only experience a limited number of cycles before failure happens. A prime object of any owner wishing to retain the integrity of the cylinder block should be to look for ways to reduce any potential localised heat spots.

Differential expansion continued to be a problem. Even with exhaust valve seat inserts fitted, a number were reported loose in service. Most of these loose inserts on SI engines were traced to a batch of inserts made from the wrong material by the supplier. During 1956 the old exhaust valve seat inserts EB 4027 were finally superseded by the Valmet inserts RE 23855. Extensive engine overheating will, however, still distort the top bridge area together with the valve seat block counter bore. In cases where the inserts had been loosened and the block needed counter boring again, a 0.010 oversize Valmet insert RE 23983 was introduced. These final Wellworthy Valmet seat inserts were installed with a lower interference fit of 0.0025- 0.004. Valmet material is especially durable and, after hardening and tempering, has a hardness of 472- 547 BHN and a low expansion rate.

Considering that during the first year of production of the S series, the hardness of the cylinder blocks had reduced to 175- 185 BHN, the presence of these inserts was well timed. Present S1 owners should not have problems with block bridges cracking and the valve seat is very capable of running on unleaded fuels. That of course is providing and only providing, the correct exhaust valves are fitted and the coolant passages are kept clean. It is worth noting that a reduction of only 10 points BHN can make a drastic difference to valve and seat life. In the presence of unleaded fuels any exhaust valve seat area much below 200 BHN is not going to survive for long.

Even before 1954, repeated recommendations from tests and experimental records make the previously mentioned points very clear. I quote some extracts from one such report discussing exhaust seat cracks, which, I generally consider, contained very sound advice.

“If failures are to be avoided the following recommendations must be observed:-

Every effort should be made to ensure crankcase cleanliness in water passages around exhaust valve seats, ports and bridges”

“Exhaust valves with Bright ray coated heads and Stellate faces should be used on 3.75 inch bore engines and should be seriously considered for 3.625 inch, particularly Continental.”

“It would be desirable to have exhaust valve seat inserts on all engines but they are essential on 3.75 inch bore engines”

“The water rail does assist in cooling bridge pieces. Providing it directs coolant to them, as it does currently, the quantity is relatively unimportant above a certain level”

Bright ray was developed by the company and used to coat the exhaust valve seats of aircraft engines. The brightray coated exhaust valves were, of course, eventually instructed for production on the 3.625 inch bore Continental R type engine and the later 3.75 inch bore engines, under part number UE786. Valve heads, which have been coated in brightray, are extremely resistant to corrosion. In addition, they are well known for their ability to delay the onset of pre-ignition. If that condition is allowed to go unchecked it results in fantastically high and uncontrolled combustion chamber temperatures.

The last report statement is a little ambiguous, as it depends what that "certain level" may be, in terms of water volume. High volume flows along the water rail are important, not just for top deck bridge area cooling but also for those components downstream of the flow. These downstream areas receive the coolant after it has been directed at, and gains heat from, the bridge area. Most of the archive reports, quite naturally, were directed at the immediate problem then in hand and not particularly directed at other areas.

In our engines the main means of heat dissipation is by passing water through the radiator, this depends upon volume. As a general comment, the more volume of coolant that is passed through the radiator, the lower will be the coolant temperature. During the life of the Bentley Mk VI, as we shall see later, the fan speed was reduced. When bulletin BB141 dated 25/2/52 was issued, it stated that the reduced fan speed had "slightly reduced the cooling efficiency, but not sufficiently to cause overheating unless the safety margin is low for other reasons" These are not the words I would have used, the fan speed was reduced by 16.31%, but more significantly the water pump speed dropped by no less that 29%. It is also significant that the fan and water pump speed were reinstated in later years, for cases of overheating complaint. In addition, the first cases of reported valve seat cracking were reported in a November 1951 production car and less than two years after the water pump speed had been reduced. The matter of water flow is discussed later in more detail and owners can make up their own minds on the issue.

Unfortunately these original troubles highlight the weakness in this top deck area. The original foundry problem of leaving sand and flash in the cylinder block has now been superseded by corrosion and silt accumulation in later years. Whilst block cleanliness is essential for long engine life, this cooling system needs as much assistance as possible to disperse the heat. This becomes more evident when the engine is operated in a harsh environment, traffic jams or at altitudes when the coolant boiling temperatures are lower.

The engine cooling is not helped when the enormous exhaust backpressure is increased by exhaust silencer blockages. This is more common than may be first thought. Excessive exhaust backpressure is enough to overheat the exhaust valves and, just as important, the top cylinder block deck. If excessive backpressure is present, the engine will not fully exhaust the residue gases on the exhaust stroke and the incoming inlet charge is diluted. The result is poor performance in any case and intermittent misfiring down the exhaust system, at engine idle speeds.

Another drawback of the exhaust restriction is the resultant severe heat and pressure that prevails around the exhaust valve stem. The exhaust valve guides are well known to take up a bell mouth shape at their upper ends and exhaust pressure and carbon is directed in that case straight down the guides. The build-up of carbon on the top section of the valve stems does nothing for their ability to transfer heat to the guides. In the worst cases the valve to stem clearance increases alarmingly and sticking exhaust valves are a likely probability. The carbon works its way down the valve stem, creating havoc in the tappet chest and finally coming to rest in the crankshaft sludge traps. It is not surprising that when exhaust valve guides are renewed in these engines that noise reduction and cleanliness of the engine oil takes on other meanings. In extreme cases it is possible to detect the excess backpressure, from a rise in cooling temperature and repeated failure of exhaust manifold gaskets.

CRANKCASE AND LINERS

Engine design is somewhat of a compromise and differential expansion appears everywhere. However if expansion is allowed to get out of control, or the engine possesses features that allow high rates of differential cooling to occur, the results are inevitable. When these cars are operated during a cool English summer, with good engines and cooling systems, they will survive. Operating in a climate with a high ambient temperature is another matter. In my own opinion, very high traffic cycles with periods of slow idling in high ambient temperatures, interrupted by engine speeds in the upper revolution bands, causes the cylinder barrels to distort. The restricted coolant flow around the barrels has already been mentioned. Excessive heat transfer from the top cylinder block deck to the barrel, instead of to the cooling system, is the cause of early type top liners becoming loose in the cylinder block. In a short time, oil and fuel deposits then collect between the liner and parent block bore, only to provide their own insulating characteristics and preventing these vital areas from being cooled effectively. These problems can also occur with full length liners, especially if any engine overheating takes place. It is perhaps interesting to study the history of these liners and relate some of the problems that they incur.

The original small 0.062 wall thickness top cuff liners were introduced after the failure of chrome flashing technique at the top of the bores on the early 4.25 engines. The efforts, from 1948, to almost eliminate top end bore wear by the use of these short Brichrome 30 % chromium liners was catastrophic to piston rings. Wear steps also occurred at the joint of the liner and lower parent bore, causing piston ring breakages. This was due to the difference between the hardness of the liner and the natural cast iron lower bore. The step was actually present, albeit to only a small degree, when the engines were produced. Little wonder that service life was found wanting. The plagues of piston ring breakages and ring scuffing were well known from the inception of production.

Making matters no better was the introduction of a lower liner in the bottom end of the bore to form a dual liner arrangement, although this only became more common later, when a similar scheme was used to recover worn bottom bores. The original dual liner scheme originated as a salvage scheme to recover blocks with porous parent bores below the short cuff liner. This scheme was first introduced in the engine of B79 GT, which passed off production on 17th March 1950. Extreme use of chemicals or acids to clean out these blocks is not to be recommended as many of these crankcases suffered from porosity and cleaning out any residue from the internal casting pores is all but impossible.

Introduction of larger bore engines, fitted with hard cuff liners, increased the ring scuffing problems to an alarming extent. The problem is very apparent on the 4.5 Litre units. Heat transfer problems, coupled with hard surface liners that do not retain an oil film, are a recipe only for those people holding large shares in a piston manufacturing company.

A number of writers have suggested that the car engine full liner material specification was 30% chrome cast iron, the same as that used for the short top cuff liners. Nothing could be further from the truth. In practice a number of B80 engines were fitted with such liners, but only for use on very short journeys, in fire engines. Furthermore these engines were specifically produced with high clearance pistons and a known final honing surface. The life of the engine was expected to be short in mileage terms and the vehicle scrapped before piston problems arose. This material is not suitable for normal car engine use and indeed had been tested, and found wanting, when the experimental division had been at Clan Foundry. Many years later there is an archive record of a case when an engine seized no less than three complete sets of pistons before the company found the liners were in fact equivalent to 30% chrome cast iron. Having witnessed many years ago, a large pre-war O.H.C Wolseley engine seize its pistons within 50 yards after having these liners fitted, I am not surprised that this Rolls-Royce engine, overhauled by the company, did just the same before completing a road test.

A high phosphorous content cast iron material was finally chosen for the full liners, but only by default. At the time Sheepbridge Engineering at Chesterfield was having difficulties obtaining low phosphorous pig iron and tested a high phosphorous content iron, of which they could obtain sufficient quantity. In the event the high phosphorous iron was found to have much better wear capabilities and was standardised for Sheepbridge's Mark VIII type full length liners.

A number of companies were able to supply the liners in slightly different material content, but they all lay generally in the ranges of 2.9-3.5% total carbon, around 0.8-1.2% phosphorous and 0.2-0.6% chromium. The Sheepbridge liners had a tensile strength of 16 – 18 tons per sq. in and a BHN range of 240-290 and it is most probable that this is representative of the standards for the other manufacturers. This specification had become the accepted standard by late 1952 for the 4.9 engines and the full liner part number RE 17411 was the result. The requirement for the external diameter of the liner to be copper plated was withdrawn at the same time. Interestingly when these liners were tested it was found that they all distorted at the top of the bore adjacent to the exhaust valve by 0.001- 0.003 but above the position of the top ring travel. In other words they are inclined to distortion, at the top, in just the same way as the smaller top cuff liners. The full liners were, incidentally, used with chrome plated top piston rings. The latter still provide good wear characteristics providing a high chrome liner is not used.

The first full liners on 4.5 Litre production cars were fitted experimentally to a batch of cars in August and September 1952, in the middle of R type production. The following chassis were some of the recipients of these engines. Unfortunately, except for another 15 or so chassis, subsequent production continued with the very hard short top cuff liners until the demise of the 4.5Ltr engine. These particular full liners were Vacrit part no RE 19011, manufactured by Hepworth & Grandage; they had a phosphorous content of 0.6 – 1.0 %.

B36 SR, B52 SR, B98 SR,

B46 SR, B54 SR, B134 SR,

B48 SR, B56 SR, B142 SR.

In practice, the full liners were introduced into continuous production with the phasing in of the 4.9 Litre engine. The original intention had been to fit the engine into all models from March 1954. In the event, history records that the Silver Dawn and the Bentley R type had to endure the short cuff liners in the 4.5 engines for over another year.

Originally, the full liner was designed with a thickness of 0.093 but was finally fitted in 0.062 wall thickness. One reason for this about turn is very enlightening, and only proves a known weakness. The main reason for the reduction in liner gauge was that some of the 4.5Ltr blocks suffered from thin wall casting on the barrel sections. Unfortunately this same problem arose again with the introduction of the six inlet port 4.9 blocks. The other reason for the standardisation at 0.062 thickness full liners was that it followed the current practice of using this size in the short liners.

Fig 5 shows a particularly bad example of core slippage causing a thin barrel wall. Accurate positioning of the sand cores is obviously vital to produce a uniform cast iron wall thickness and the necessary dimensions; any collapse of the cores, or slippage from their original positions, will result in an imperfect casting. This example is one of the 4.9 Litre early UE450 crankcases produced by the Leyland Farington foundry. Later UE 2018 crankcases had bulged exhaust ports and were produced by Beans Industries as well as Leyland.

Anyone who has been unfortunate enough to bore through a barrel of one of these six cylinder blocks will no doubt have placed the cause down to corrosion of the barrel. In practice, however, they will have encountered an original foundry fault, which was not only confined to Rolls-Royce crankcases of the period, but a scourge of many manufacturers, only accelerated by corrosion.

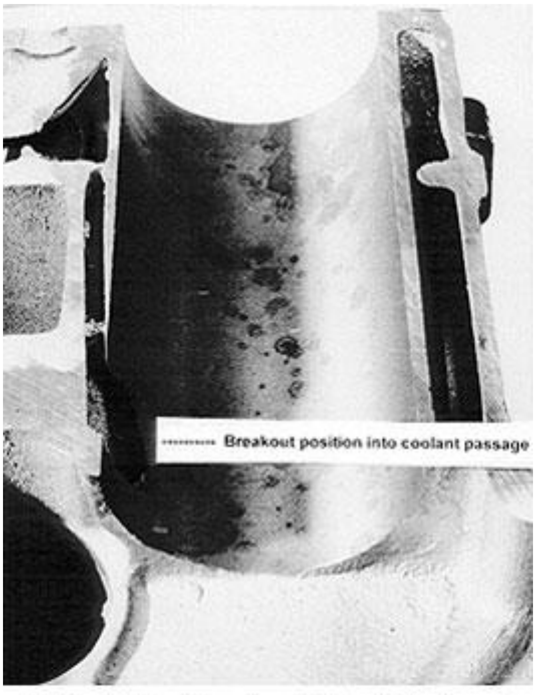

FIG5. Section through a 4.9 ltr cylinder barrel, showing break through of the bore.

When combining a cylinder barrel with varying thickness wall sections with a 0.062 liner, or worse, an oversize bored liner, it does not take rocket science to realise the potential for uneven expansion and wear of the bores.

The particular wear pattern on some of these liners, or bores, is not confined to earlier engines with either one cuff liner and parent cast iron or dual liners. In fact, it is possible to see the points of differential expansion on the liners of engines that have full liners. In particular the area of the liner adjacent to the exhaust valves generally shows the after effect of differential expansion, as do a series of expansion waves occurring down the bores. Accurate bore measurements show these irregular shaped wear patterns. This is the result of the barrels pulling and pushing against each other, when they are subject to differential expansion, accentuated by any variation in barrel wall thickness.

Dry liners do have the advantage of allowing a different material to be used for the cylinders than that provided by the parent bore of the cylinder block. In addition of course they also provide a method of reconditioning a block back to standard bore size after the maximum bore size has been reached.

They also have disadvantages, especially if the cooling system is anything but near perfect, or material has been taken out to provide the normal overbore sizes that service life demands. In the latter case, liners can become nothing better than slim wall, resembling no more than an oversize soft drink can and possessing only a little more mechanical strength. Possibly a slight exaggeration, but it serves to illustrate the point. A liner in this state, under cooled on one side in our case, is a recipe for disaster. Even in the standard bore size these liners will tend to move away from the parent cylinder block bore when subject to any overheating. As previously mentioned, this happens even more so at the point between the exhaust valve seat and the cylinder bore where the temperature gradients are very severe.

Movement between liners and barrels is self-generating and a dramatic cycle begins. First localised overheating occurs, and then contact is lost between liner and barrel causing more localised overheating and finally piston destruction. Full contact is vital to permit heat transfer. When localised pockets occur between dry liners and their parent bores, small as they are, they are the death sign for the liner and piston. The liner proceeds into piston destruct mode. In practice of course the alloy piston is blamed as it refuses to take up the irregularities of the bore as it passes at high speed through an orifice that it will not fit. After all the evidence is there, piston scuffing, seizure, closing of the split skirt and gripping of the gudgeon pin bosses. This cycle is all the more severe if cuff liners are involved because the 30% chrome iron material has poor oil wetting characteristics. In other words the material surface does not retain or hold an oil film. Evidence of lacquer on liners of these engines seems to show predominance for the liner bores to take up a more oval shape than normal when the engine is hot. Attempting to obtain extraordinary tight piston to liner clearances in these situations only brings the inevitable sooner. The piston rings have no chance of fully contacting the liner around their circumference if the liners are out of shape and high oil consumption is a forgone conclusion. When the engine design and cooling flow requirements are studied closely, it is not at all surprising that piston and liner problems are at the forefront on these 50 year old cylinder blocks.

All too often the liner is damaged and it is considered a waste of time to clean up the surface. Any attempt to check the bores with a Mercier gauge is discounted, after all a re-bore is required, is it not? In most cases a determined attempt to gauge the bores will show a series of waves down the liner where the liner has quarrelled with the barrel. Our oversized soft drink can has been subjected to overheating torture. It has been subject to overheating on one or more sides through poor cooling, initially, and is now subject to irregular heat transfer. The liner distortion in cylinders No 2 and No 5 is exacerbated by the siamesed construction of the barrel cores to the adjacent cylinders. These neighbouring liners having a mind of their own regarding the direction they would like to travel. The liners, especially if they have been over bored from standard, are sited indeed in a quarrelsome environment in a no win situation. There appears to be a clear case for always fitting new liners and not boring oversize when any piston seizure has occurred, in order to retain the mechanical strength of the liner. The original 0.093 wall thickness full liners may have been able to counteract this predicament to some degree, had they been retained. This additional wall thickness would have certainly provided some safety margin, when the liners were over bored in service. It should only assist the liner environment, if say 0.10 or 0.20 oversize O/D liners are used in current engine overhauls.

The cylinder barrel, of course, takes all this in its stride; after all it is stronger than our oversized soft drink can which has started to crumple under the strain. If by now the liner has not gripped the piston, then localised pockets are certainly present between liner and barrel, only awaiting the next high speed run or traffic jam. Of course, who can blame the unsuspecting owner, when he/she is presented with the costs of re-lining or a cheaper rebore, in choosing the latter? A rebore will remove the tops of the very minute bore waves. The localised pockets between barrels and liners still exist and, just as important the liners are now thinner adjacent to all the previous wave points. A new lease of life will be breathed into the reconditioned engine, at least for a short time, until the cycle repeats again.

The ravages of time have caused corrosion of the outside of the barrels, the ceiling area of the cylinder block and the vital area around the exhaust valve seats and ports. Heat transfer problems are then just around the corner. The use of completely different fuels and not least the actual conditions under which most of these cars are now used, invites the best possible cooling to be utilised. The average car is used on high days and Sundays. When finally exercised on that day out, the brief breakfast warm up period is ignored, as the car is immediately driven off and placed under load. Quite possibly it is expected to cover distances at motorway speeds, or worse, be operated in one of our legendary summer traffic jams. None of these conditions are conducive to long engine life and especially with an engine having a relatively long piston stroke and assembled to close clearances.

In new car engines the full liners were quite successful and they would perform well unless the cooling system passageways were allowed to silt up. The cooling system needed vigorously flushing out on a regular basis, for antifreeze with extremely good additives was still not available. Inevitably the day would arrive when the liners lost their battle in the heat transfer world. Boring out was usually the order of the day and the cycle related above commenced. Occasionally, a new set of liners would be fitted and the block thoroughly cleaned and then the engine would complete very long mileages once again. Today, with the availability of modern antifreezes, engines with very high carbon mileages are becoming more frequent, as more and more are being fitted with new full liners. High carbon mileage in this context meaning the mileage achieved without any dismantling, as compared with the total mileage completed by the engine.

BORING AND HONING CYLINDER BLOCKS

When the blocks were originally received from the foundry they underwent a form of stress relieving. Very high quality blocks, used by manufacturers that had low production volumes and no such stress relieving plant, were very often weathered to stress relieve the casting. In this case the weathering consisted of deliberately storing the castings outside, sometimes for months.

Considering that the R-R blocks have undergone quite an amount of torture before the enthusiast decides he or she can no longer delay the inevitable, weathering a cylinder block may be worth considering. Pressing in full liners under an interference fit produces stresses in the liner and block. It was certainly considered, by engineers in my time on the shop floor, which after inserting liners that blocks should be left for some time, at least before final honing. Liners do not just press into a block and make 100% contact and they creep for some time afterwards whilst trying to take up position. If time is not of the essence, nothing is lost, and probably much gained, by leaving the block at least seven days or even a month if possible, before boring and certainly before final honing. The compressive stresses produced in the liner and the block after pressing in the liners is considerable.

Perhaps further evidence may make this point and previously mentioned arguments, more forcibly. During early 1952, a further 0.015 was being bored out of the RE 10967 top cuff liners, before they were pressed into the blocks. This was being done to avoid ovality after boring. This fact is very interesting, considering that new liners were being fitted to new blocks, which incidentally, had never seen service or been subjected to overheating. Ignore for one moment, my previous warning that trouble should be expected with liners that have overheated and then bored oversize. Here was a production finished bore that tended to go oval in shape after being finish bored. Only forces on the outside diameter of the liner, the compressive forces, cause this type of ovality. The enthusiast may now understand better my previous points regarding the tendency for these liners to distort in service conditions and that it is unwise to bore out oversize, in particular, if the liner has sustained any damage. Boring out will only accelerate distortion and encourage parent barrel to liner gaps to develop, either result will have a predictable outcome. Consider that if the 0.062 liner is bored out to 0.040, the liner wall thickness of the liner is a mere 0.042. When liners are finally bored out to remove them from the barrels, at a 0.010 thickness they are extremely loose in the barrels. As I have previously related, liners, which are bored out, are not much thicker than the average soft drinks can and the compressive forces tending to cause distortion are severe.

Early in 1952 a full length liner for the 4.25 engine was detailed and the installation drawing SB 341 dated 16/5/52 states precisely that the block must have a smooth ground bore before the liner is installed. Grinding the bores originally predated the modern boring bars and it is doubtful if any bore grinding operations still exist, at least for motorcar engines. In current practice however it is questionable whether sufficient attention is given to this pre-installed finish to the barrels and ovality before the liners are pressed into place. The best that one can normally expect is that the parent barrels will be fine bored and oversize O/D liners fitted if the crankcase bores need attention. The outsides of the liners, however, are not particularly well finished anyway and both the parent barrel and outer liner finishes affect the final intimate fit. On a more personal basis, I favour copper plating the outside of the liners to assist not only in insertion of the liners but to fill in some of the fine gaps.

Originally this had been specified by the company but eventually deleted as a cost saving exercise, but it was retained by many manufacturers of high quality engines of the day and proved to be of good value in service.

It is doubtful if the average engine machining shop will carry out the boring and honing procedure correctly, on these particular engines, and this has an undesirable effect on engine life and oil consumption. Some Rolls-Royce specialists have their own machine shops and will be well aware of the procedure, but the owner would be well advised to ensure that these operations are conducted correctly and without fail. Unfortunately I have witnessed many boring operations where the methods have been ignored. The engineers of the company had clearly done their homework, in respect to the boring and honing specifications, and it is always a difficult task to get today's engineers to accept accurate measurements and not visual references. Even the small matter of storage is important as centrifugal cast liners will distort if they are stored in a laid down position.

The No1 cylinder bore is affected by the pull exerted from the studs holding the water pump adaptor plate to the front face of the block. This adaptor plate or a stress plate must be in position before liners are fitted and the block bored and honed. Failure to fit the adaptor or stress plate before boring, will result in the forward section of No1 cylinder distorting when the adaptor is finally fitted during engine rebuilding. The company specifically stated that the adaptor plate should be fitted before boring operations took place and this will not be known by the average machine shop.

For similar reasons, all the cylinder head studs must be fitted and tightened in the block, after boring, but before honing. It is not possible to fit studs before boring as the boring bar would have no surface upon which to rest. Final honing does, however, take out the top liner distortions caused by the stud pulling stresses. It would be an even further improvement if spacer tubes and washers were fitted to the studs and the head nuts tightened down. One of the reasons that this procedure is very often missed is because of the added difficulty of honing with the studs in situ.

Once again, further evidence may make the points. During experimental building of the later V8 engine it was found that even on this wet liner engine the head studs distorted the liners by up to 0.004. Considering this was a wet liner engine, the pull on the block was nevertheless distorting the liners. Admittedly the V8 block was of hollow construction to accommodate wet liners and is therefore weaker. In addition the V8 head stud torque was much more than the 375 lb. ins on the 4.5 Litre BSF stud block and 400-430 lb. ins of the UNF threaded later 4.9 blocks. Fitting of the studs before honing is, however, specified by the company for the six cylinder block, and not without good reason, liner distortion still takes place.

The actual company boring specification states that 0.003 diameter should be left in the bores for final honing. If you have ever experienced that feeling that your engine has not completed as much mileage as a new one before it starts to consume oil at a prodigious rate, look no further. The honing can be very time consuming and the cast iron stock is removed very slowly, in fact the task is laborious to say the least. If a very sharp boring cutter has been used, the boring marks will appear to have been removed well before 0.003 has been removed. I have used the word appear quite deliberately. The operator will usually accept what he sees. In practice, and with much experience, the actual boring will be completed leaving only about 0.002, or much less for honing. After all, the operator will know that after some 0.001- 0.002 has been honed out, the boring marks will have disappeared. How wrong he can be. In simple terms, what has actually happened is that the peaks caused by the boring cutter have been swaged over into the furrows and the surface burnished. This finish can be seen very clearly under a microscope and it is not a pretty sight. The piston rings will eventually fine polish the surface. Unfortunately the ring edges will get destroyed in the process and the original boring marks will reappear. Although it is quite natural for oil consumption to be a little high during the piston ring bedding process, this high oil consumption should quickly disappear. Very often this is not the case and the owner will be encouraged to continue running the car in order to run in the bores. Excessive running in this condition will eventually reduce oil consumption, but not to the extent that it should and heavy oil usage will reappear far too early, once again. At that stage a new set of piston rings will usually cure the oil consumption, because the old rings have now honed the cylinders mechanically. It is vital that honing takes place from a known measuring point, that being 0.003 undersize on the bore and then continued until the final cylinder size is correct, visual checking of the surface finish is not good enough. Anything less than a final hone of 0.003 will not take out either the bore cutter marks or the localised plucking in the tops of the liners, caused by the pull of the head studs.

Prior to considering any modification to the cooling system, which may help extend the life of a rebuilt engine, it may be prudent to convey some other points.

PISTONS

Although the main subject of this article is the crankcase, due to the intimacy of the pistons with the liners, a few points in connection with the pistons may be appropriate. The owner will usually be made aware of liner problems in service arising as a result of piston wear, ring pack breakages or piston seizure. Overheating causes and their results on liners and barrel distortions have been discussed. The piston is usually blamed. Although pistons may be tapered, elliptical and have expansion restriction blocks cast into the gudgeon pin areas, or all three, they will only take a certain amount of punishment from the liner movement.

References to the service manuals will illustrate the very close piston tolerances that were accomplished on the initial build. Early type piston skirt clearances at the top were listed between 0.003 – 0.0035 and on the skirt bottom at zero to 0.002 actual interference. Whilst these clearances are adequate for low engine speeds and engines with perfect cooling systems and no liner distortion, they do not seem to take lightly to the contrary situations. The actual result of this extra fine clearance should be put into perspective. Actual piston ring blow by on these six cylinders is about 18 cu ft. per hour at 3500 rpm on quarter throttle and around 45 cu ft. per hour on full throttle between 3500- 4250 rpm. These figures are 50 % less than what could normally be expected from an engine at that time, in the 4.5 – 5.0 Litre displacement range.

The piston and liner damage tends to show itself more on the rear two cylinders; probably this could be the result of overheating as already discussed. In some cases it could also be the result of crankshaft flexing around the rear two big end journals and imparting a twist to the connecting rod. This is the area where these crankshafts tend to show cracks developing and also the area that the company found to fail, when tests were carried out to deliberately damage the shafts. The area at the front of no 6 big end crankpin is the normal failure position.

A number of pistons are marketed for these engines, some of which do not perform at all well. Anyone fitting new pistons would be wise to consider the issue very carefully. On a personal basis, I favour pistons with larger clearances than those mentioned and having a 0.003 +0.001 at the bottom of the skirt and 0.005 +0.001 at the skirt top, measured on the thrust side of the piston. After examining a number of these engines fitted up at these clearances, none have shown any signs of cold engine piston slap. Pistons should perform well, providing they are given a chance. The cylinder bore is well catered for, in respect of oiling.

Only the very early engines lack a connecting rod hole to feed the bore thrust face and the piston skirt actually emerges out of the bore by 0.750 at the inner bottom dead centre position, therefore receiving adequate skirt lubrication.

PISTONS FROM B87 UL

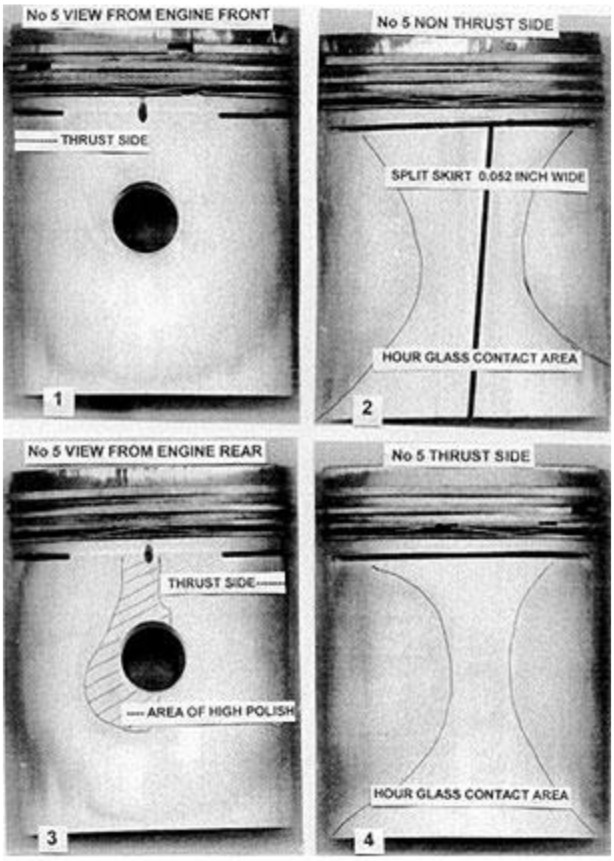

FIG6. View no 1 to 4 of No 5 Piston from chassis B87 UL.

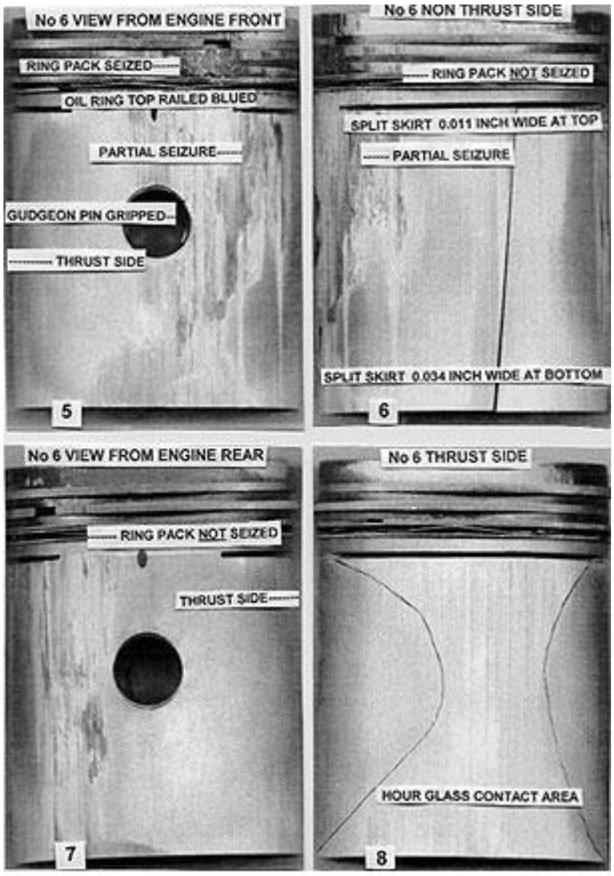

FIG7. View no 5 to 8 of No 6 Piston from chassis B87 UL.

Fig 6 and fig 7 show eight views of the pistons from No 5 and No 6 cylinders respectively, each view being numbered through 1 to 8.

Examination of the exhaust valve footprints in the cylinder block will show these are offset in relation to the cylinder bore centres, to accommodate the inlet valve push rod tubes. In particular it should be noted that No 6 cylinder exhaust valve is offset towards the rear of the cylinder block, whilst No 5 cylinder exhaust is offset to the front.

Unfortunately reflections in each view give the illusion that all the piston rings are carbonised. In fact this is not the case and all the rings are bright, so this point should be ignored. Views 1 to 4 show that number 5 piston is in good condition, except for the area marked on view 3, which is in fact a highly polished local area that corresponds with the siamesed block tying the barrels of 5 and 6 together in the cylinder block casting. The picture showing these siamesed links was formerly shown in fig 1. The split skirt slot is full width right up the length of the skirt and the normal hour glass shape contact area of the thrust and none thrust sides are normal. The highly polished area in view 3 does however show that this piston was starting to be pushed by the liner because of the siamesed jointing to cylinder 6.

Comparing view 3 on piston no 5 and view 5 on piston no 6, shows the piston damage that occurred in the liner area adjacent to the joint between the cylinders. These two piston positions were opposing each other.

Views 6 and 8 clearly show that the normal working surfaces of the thrust and none thrust sides of piston no 6 are normal, except for the reduction in the split skirt slot. This slot has closed up to 0.011 inch at the top and 0.034 inch at the bottom, from the original 0.052 inch wide position. The piston has been squeezed severely between the liner surfaces adjacent to the exhaust valve and the area adjoining no 5 bore.

Taking piston no 6, view 5 and view 7 are the most interesting. Reference to view 5 shows most damage has been caused by the front part of the bore, adjacent to the adjoining no 5 bore and the damage has extended right up to the ring pack. In fact the top ring is entirely free, but the oil ring rail is actually blue from heat, exactly on the line of the gudgeon pin. The lower two rings are entirely seized by distortion of the piston ring lands and the gudgeon pin itself was seized in the piston at this position. On the side opposing the exhaust valve and cylinder bridge, in view 7, piston damage is confined to the area below the ring pack and the gudgeon pin bore is entirely free and not distorted.

Before I purchased the car, the engine had suffered from an alloy gudgeon pin cap wearing away, most likely the result of an internal water leak, allowing the gudgeon pin to score the forward part of no 6 bore. The cylinders had been fitted with top cuff liners and these were over bored 0.035 inch to remove the scoring and genuine RE 15784 Wellworthy oversize pistons fitted. The original gudgeon pin end would have caused severe localised overheating down the forward end of no 6 bore at the junction with no 5 barrel. This would have distorted no 6 liner away from the parent barrel and even if the liner had returned to its position, a stress area and a localised pocket would have occurred, however small, between the barrel and liner. Over boring the liner only hastened the eventual failure. The push from the area between no 5 and no 6 bores would have forced the piston over to the other side of the bore where it also had the chance to pick up on the other bore high spot adjacent to the exhaust valve. Measurements of the bore showed the normal series of large waves running down the bore.

It is doubtful if the damage could have been avoided by modifying the cooling system. The stressing from the previous engine failure was already built in. Had a new set of liners been fitted, the story would have been different.

VALVE TRAIN, OIL FILTERS, OIL, TAPPETS, WATER LEAKS AND TAPPET CHAMBERS

Whether the engine is fitted with a bypass or full flow filter system, the oil supply to the valve train and geared timing drive is not filtered. Sludge and any metal particles will collect eventually inside the rocker shaft. The shaft and rocker bushes are the finest oil filters on these engines. For that reason these components require regular cleaning, in any case.

Any piston seizure, ring breakage or break up of the crankshaft bearings, will bring about a supply of metal particles along with the unfiltered oil flow, directly to the overhead valve gear and then to the cam followers. If piston seizure is the initial event, then metallic particles are likely to impregnate the main bearing shells. This will obviously be more likely to occur on the 4.25 Litre engines if they retain the original by pass oil filter. At any time, when it is known that metal particles have entered the oil stream, the overhead rocker gear and front gear train should be stripped and cleaned. Ignoring this procedure will quickly bring about extensive noises and very expensive repairs. The repairer or owner who lacks full knowledge of the oil feeds often ignores this vital procedure. The handbook text unfortunately gives the impression that the oil supply to the valve gear is filtered, although reference to the oil lubrication system diagram will clearly show the unfiltered low pressure supply. Strangely, the full filtered oil supply on the commercial B60 engines does provide filtering of the complete, low and high pressure, lubrication systems.

These same metal particles will also be fed directly between crankshaft and camshaft timing gears with longer-term disastrous results. The metal particles will be compressed onto the surface of the gear teeth and provide a very good cutting medium to self-destruct the gears. In this case the alloy camshaft gear will be severely affected unless the timing case is removed and the gear teeth cleaned of all debris. In these circumstances, what appears as an unusual gear tooth surface, can very often be recovered, providing the crankshaft gear teeth are cleaned with 2400 grade wet and dry paper dipped in light oil and the cam gear teeth are polished with a metal polish. Given this cleaning procedure the teeth will usually be recovered back to almost new condition and prevent future breakdown of the gear teeth.

It is good engineering practice on these engines, in fact any engine, to fully dissect the old oil filter element when it has been replaced. Alloy particles from scuffed pistons in the oil filter material should set alarm bells ringing.

The object of changing the oil is to remove as much of the old oil and debris as possible. I would suggest that the prudent owner completes this exercise when the engine has not been operated for some time. For those purists doubting this point the following is worth considering. The crankshaft is hollow and contains a large amount of oil, which takes days to drain down, as does the cylinder head and valve gear residue oil. When the engine cools, oil is drawn back out of the full flow filter bowl into the sump; this last point is mentioned in the S series handbooks but not in previous model publications. Should anyone have further doubts I suggest that they actually weigh the amounts of oil drained, using both hot and cold drain methods, when the case will be self-evident. A further advantage of this method is that any amounts of water residue will have separated out from the oil and will drain out initially, providing a warning of its presence. Many of these engines are suffering from minor internal leaks; owners' sometimes presuming that the very small water loss is caused by natural evaporation during normal driving. Some of these potential leakage areas are covered later.

It is vital to engine life that any internal water leaks are rectified. Any water accumulation will collect in the front crankshaft damper assembly. When the engine is stopped it will immediately separate from the oil and sink to the bottom of the damper. Over a period of time it will rust through, in turn, each and every one of the damper spring fingers, as the engine comes to a halt in different positions. This situation can be avoided by modification to the damper, but is beyond the scope of this particular article. Another peculiar wear pattern on these engines is the accelerated wear of the aluminium piston gudgeon pin plugs, when water is present in the oil.

The siamesed cylinder barrels have been mentioned in relation to differential bore distortions. This same feature can also cause other undesirable occurrences. If the engine cooling system is severely silted, even more of the particular components mentioned will be affected. The cam follower housings are apt to distort slightly, especially those controlling No 2 and No 5 cylinder valves. It is most probable, that as these centre barrels fight to retain position and shape, they are causing slight distortion of the cylinder block floor and the cam follower bores take some of the force. The inlet cam followers seem to be predominantly affected, and in particular, No 5 inlet. When afflicted, the follower shuttlecocks up and down the bore instead of following the camshaft profile religiously. Increasing the clearance very slightly between the follower and bore appears to provide a long term answer. The resultant noise is not usually constant, but comes and goes as the follower rotates. The bore sticking does occur without the presence of dirt, however it will obviously be more prominent if the followers or bores have corrosion pockets at the top bore entry point caused by the inevitable flush down of condensation from the top end breather, or water leaks

It should be borne in mind that the inlet tappets take all the return flow of the unfiltered oil from the rocker shaft oil feed, and hence initially all the dirt. The exhaust followers take spilled off oil from the inlet tappets. Earlier engines were fitted with rocker arm valve end restrictors to provide unequal oil flows, later types produced the restriction by the special design of the bush oil scrolls. These bushes are intended to divide the oil supply between the valve end and the push rod end of the rocker arm on an unequal basis, so that the greatest oil return is down the push rod. Many of these bushes have been replaced back to front and the valve gear and tappets suffer in consequence. This oil supply is the more vital because the overhead inlet valve gear and tappet are subject to over twice the loading of the exhaust valve tappet.

Very severe sticking of both inlet and exhaust followers will happen if they have been subject to tappet chamber water leaks. Coolant, in that case, will remain around the top of the follower and its mating bore, nicely rusting them into a partially seized position and taking its toll on the camshaft lobes and the alloy camshaft drive gear.

As in the case of the later S series, the cylinder blocks fitted to the MKVI and later R types were alternatively subject to bonding and then interjected into production as the necessary reworking or rejection was decided. The bonding or holding system is used by manufacturers to segregate components that do not pass the required quality standard. The components are bonded, whilst a decision is awaited as to the rework, rejection or concession that will eventually apply to the failed part. It can therefore be seen that to suggest definite chassis change over points of core and plugging operations is not possible in every case. These alterations affect the description of possible water ingress points as will be seen later. A general, but not exhaustive, picture of the cylinder block core history is given below.

In certain instances actual cylinder block part numbers are quoted throughout this article whilst in other cases the part number of the block assembly is quoted. The assembly comprises of additional parts in addition to the actual crankcase and is only quoted when there is doubt concerning the original crankcase part number.

Early 4.25 Litre blocks had only two coolant jacket circular cover plates part no EW 1431, on the carburettor side, with an aluminium KB 7303 core plug in the central position. A steel RE 9016 plug, later superseded this aluminium central core plug. These early two plate blocks were superseded in production and for service replacements with a three cover plate block RE 8812 assembly. Left hand drive versions had the rear cover plate EW 1431 replaced by a cover plate RE 8257 that had a cast lug to support the main fuel pipe by means of a bulkhead coupling. The special L.H.D cover plate was retained until the demise of the 4.5 Litre engine, all the cover plates were cast iron with a 202/241 BHN. Some owners find it difficult to remove these EW 1431 coolant jacket plates; therefore a section is devoted later to this problem. Cover plates for 4.9 Litre blocks had a much simpler fastening and were part number RE 20848.

The first 4.25Ltr blocks had fairly consistent foundry core positioning. The initial production of the 4.5Ltr blocks in 1949, however, reversed the situation. Many of these 4.5 Litre blocks had cores out of position on no 2 and no 5 cylinder barrels, causing cylinder wall thin sections and, in extreme cases, broke through the barrels during machining. This same problem, but affecting any of the six barrels, arose in later years during production of the S series cylinder blocks. The introduction of the 4.5Ltr type block started initially in the Bentley MKVI chassis GT series. They were cast to accept the initial full flow oil filter adapters but produced with by-pass oil filters, whilst still retaining the 3.5 inch bore, the cylinder block part number was RE 9421. By the beginning of the H series cars, cylinder blocks were generally 4.5 Litre, but a few 4.25 Litre blocks lingered on and were fitted to chassis B102 JO, B130 JO, B250 JO, B21 JN, B27 JN and B 151 JN.

It is not intended to provide an exhaustive list of potential water leakage points but some of the following areas are subject to leaks, particularly when the engine is hot. In general these blocks all have at least two core plugs in the rear of the tappet chambers. With the introduction of the 4.5 blocks, lugs were cast and two holes were drilled in the tappet chest, opposite no 2 and no 5 barrels, to check the barrel positions and the holes were plugged with hexagon set screws. Initially all 4.5 blocks were drilled in these locations, but later blocks were only drilled and tapped on a random checking basis.

The very first 3.625 inch bore crankcase, RE 16911, was fitted in the MKVI 4.5 Litre and R types to approximately chassis B255 SP. These blocks are easy to identify as they had two KB 7301 alloy core plugs fitted externally on the block face, immediately below the exhaust manifold, at the junction of the exhaust siamesed ports. The KB 7301 core plugs were subsequently changed on production and in service to cadmium plated steel plugs, part number RE 20740. Owners of MKVI cars will not find the RE 16911 crankcase listed separately in the parts manual, as it forms part of the listed RE 13404 and RE13835 crankcase assemblies.

The last 4.5 Litre engine block casting, part no RE 16217, was introduced around chassis B203 SP. This block was fitted with two inclined RE 9016 large core plugs in the upper tappet chest which, in situ, can only be felt by hand or viewed with the aid of a mirror. These inclined plugs correspond with the junction of the exhaust valve ports on cylinders no 2 and no 3 and the junction of no 4 and no 5. The core plug access holes provided a means of cleaning casting sand from these critical locations. The internal inclined core plugs are subject to leakage as they are directly affected by heat from the adjacent exhaust ports. In view of their important location, they need removing during engine overhaul in any case and any silt removing from the internal pockets at the exhaust port junctions. Like other core plugs they are also fitted with alloy washers, which are subject to corrosion.

It will be noted that the introduction into production of the RE 16911 and RE 16217 cylinder blocks, overlapped in the R type SP series chassis range.

In the case of S series blocks, with a possibility of a core collapse on any barrel, the tappet chambers were drilled and tapped adjacent to the barrels. These early blocks did not have any casting lugs to accommodate the tapped holes and therefore the wall thickness into which the plugs are fitted is very thin. Later blocks had casting lugs to overcome this thin wall situation.

Actual core plug removal from the tappet chest area can be difficult due to the limited access, which depends on the type of body fitted to the chassis, or due to corrosion of the threads. Each case will therefore be different, but usually the left hand wing will need to be removed. In practice the easiest way I have found to extract these plugs is to electric weld a hexagon nut to the face of the core plug. The heat from the welding provides enough expansion to loosen the thread and the hexagon aids removal with hexagon socket tools. New plugs can be treated in the same way by welding a hexagon nut onto the face, it is however necessary to reface the plugs before fitting to remove any distortion from the contact face of the plugs. The protruding hexagon nuts can be left in place, as, in none of the situations, do they foul any moving part.

The rear of the tappet chest is subject to a fair amount of differential expansion and this wall section can become porous or cracked, particularly in the area behind the exhaust valve springs. Porous or cracked sections are to be expected, and are, in fact, more noticeable at the rear of the engine where the block is inclined to be hotter. In addition this area is directly below the exhaust ports.

I would recommend in every case when the engine is to be overhauled, that the whole tappet chamber rear wall is first cleaned with a drill operated brass brush and then with thinners, to remove all traces of dirt and oil. If the engine is run to attain operating temperature, the area can be examined under a light to check for potential porous and cracked areas. Following an engine rebuild or any extensive de-silting of the cylinder block, it is essential that this area be checked again in the same way. Removing extensive silt very often causes leaks to appear in the tappet chest region. It should go without saying that de-silting also encourages leakages from the core plugs. These should be removed and, at the very least, their alloy washers replaced and the plugs either renewed or their contact faces machined, before applying Wellseal jointing to the threads and then refitting. Normally, cracking in the rear tappet wall area can be successfully repaired by slightly grinding out the crack, end drilling to stop the crack further developing, followed by interlock drilling and tapping, penning over and then applying a metal repair paste. Removal of the tappet chamber covers to view this area will involve removal of the front exhaust pipe and manifold to clear the front tappet door forward bolt on R types, but can be achieved with the pipe and manifold in place on most other models.

Although the main emphasis in this article is directed towards the cylinder block and crankcase assembly, and the differential expansion problems arising, it will provide a fuller picture if tappet setting is included. Long standing owners will no doubt be aware of the tappet noise that can become irritating on these six cylinders and the later advice of the company to set inlet tappets in the engine hot condition in annoying cases.

Predominately the noises originate from the combined overhead valve gear mechanism and to describe all the causes would take an article in its own right. Particular attention should be given to ensuring that the vast majority of the oil returning from the shaft assembly is directed straight down the push rods from the ball end adjuster of the rocker arm. It is advisable to remove the rocker shaft assembly and thoroughly clean out the shaft and oil ways before tappet setting if the engine has completed much carbon mileage. Advantage should be taken of the removed assembly to check that the valve ends of the rocker arms are not indented. If the ends of the valves have caused indentations, providing there is enough hardness depth remaining on the rocker ends, they will need radius grinding, otherwise the feeler gauge will not provide a true tappet clearance reading.

Enough has been stated previously to explain the effects of differential expansion. The overhead valve gear is no exception and is especially sensitive to clearance settings on No 2 and No 5 valves. It is advisable to check and adjust the clearances with the engine cold and then again with the engine hot and record the clearances. Providing the engine has no serious expansion problems, differences can be expected on No 2 and No 5 and they can be adjusted to conform as the object should be to have exactly the same valve clearances on all the valves in the running condition. If the owner is skilful enough, the clearances can be set whilst the engine is running at the normal operating temperature, this procedure provides exceptionally even idling, provided all other settings are correct. Tappet setting in the running mode takes into account wear variations in the valve gear operating assembly under operating conditions and a true 0.006 clearance can be attained. It is important to ensure that the push rods are revolving under the influence of the cam followers. Any sticking follower should be the subject of immediate investigation

The overhead valve inlet train is very much affected by expansion, although excessive wear may be a contributing factor, and the expansion of the alloy head is a major cause. Tappet noise can also be a warning of blocked cooling passages. The localised growth and expansion movement of the block should be watched. It is imperative that, should the clearances be checked in the hot condition and set at 0.006, they are again examined when the engine has thoroughly cooled down. On engines that have cooling circulation problems or are subject to localised overheating it is not uncommon to find that the cold clearances do not exist and, in extreme cases, the valves are lifted off their seats as the engine cools down. If the tappets are set at 0.006 hot it is not unusual to find cold clearances of around 0.002 when the engine is cold. Should the engine fall into that category where the valves are open when the engine is cold, deeper analysis of the cooling system passageways should take place and at the very least the tappets given at least 0.002 cold clearance.

COOLANT GALLERY ASSEMBLIES

At least three different coolant gallery tube assemblies are currently listed for the 4.25Ltr and 4.5 Litre families of engines, whilst a fourth was introduced for the 4.9 litre. The brass gallery assembly, which distributes the coolant pump output to the exhaust valve seat area, is made up of a number of components. Its cross section shape, position of outlets and dimension of the outlets varied according to the crankcase. In order to enable a better understanding of the differences, each type has been designated Type A, B, C and D and these should not be confused with the scheme types described for the water pump and fan drives described later, as they have no connection.

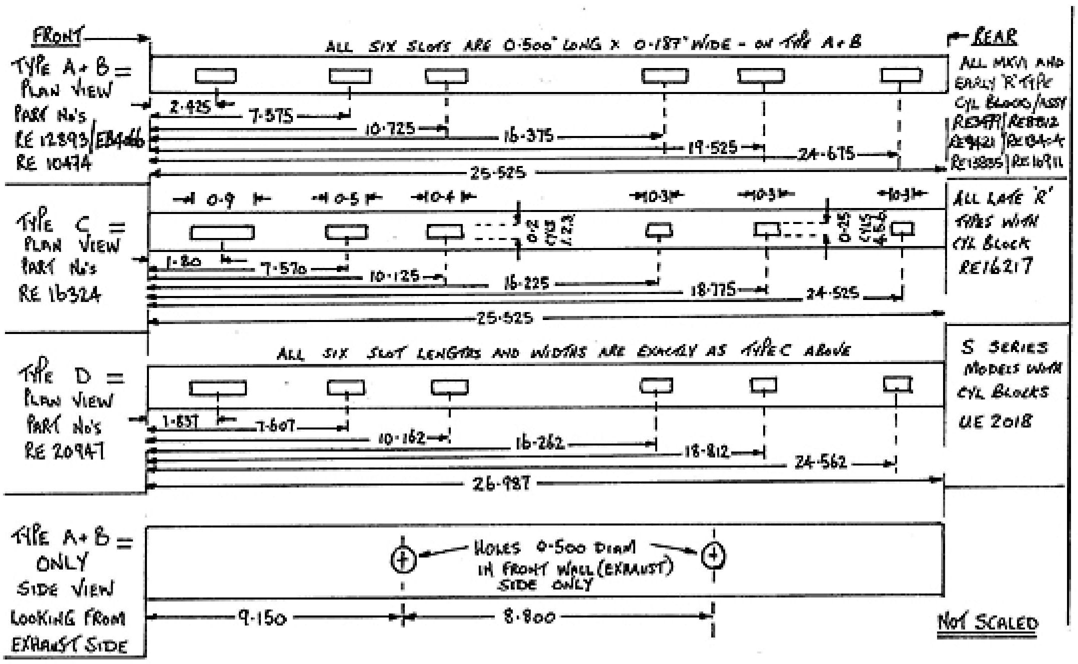

FIG 7A. Water Gallery Types used on Post War six cylinder engines in MkVI, R and S

Fig 7A sketch shows the position and dimensions of the slots in these galleries but has no relationship to the cross section shape.

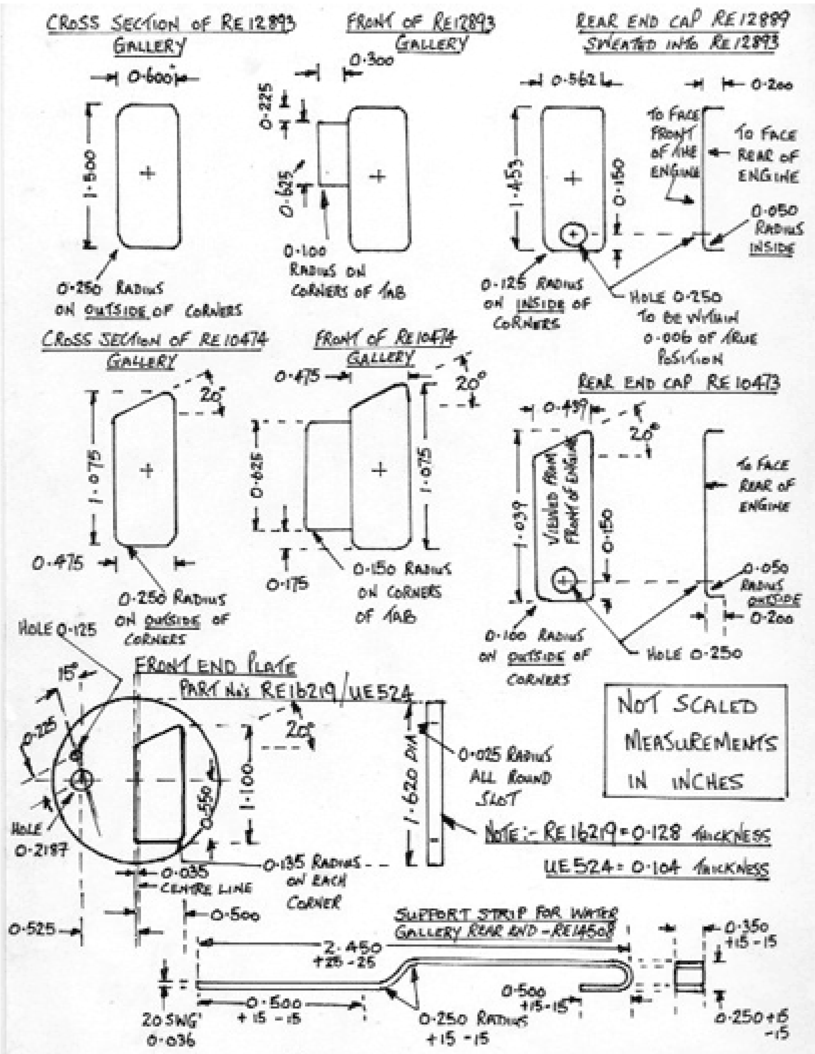

FIG 7B. Various Water Gallery Fittings.

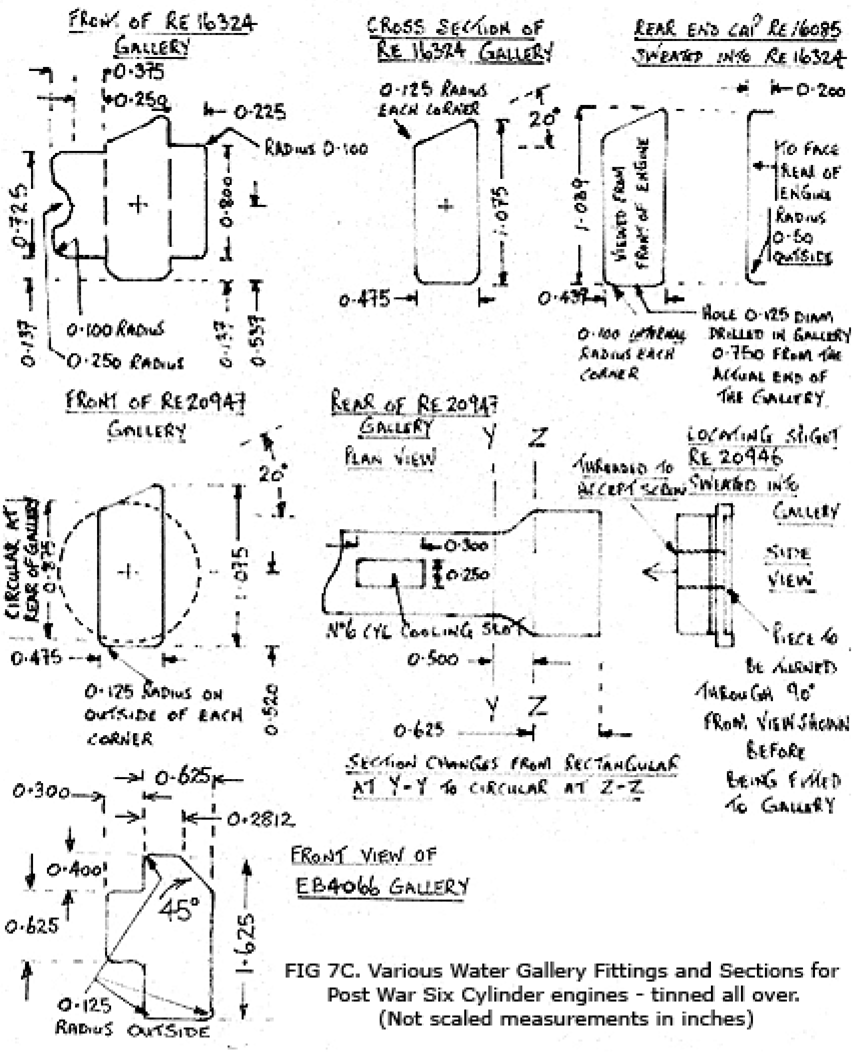

FIG 7C. Various Water Gallery Fittings and Sections for Post War Six Cylinder engines - tinned all over. (Not scaled measurements in inches)

Fig 7B and Fig 7C sketches show the various end caps, fittings and the cross section shapes that complete the actual water gallery assemblies. It should be noted that all the fittings shown in Fig 7B and 7C are made of 26 S.W.G (0.018) brass, except RE 16219 / UE 524 front end plates, which are made of 0.128 / 0.104 thickness BSL16 and EN 2 respectively, and the RE 14508 rear support strip which is 20 S.W.G (0.036) brass.

The actual cooling slot comparison dimensions, Fig 7A, and their individual positions in respect of the front face of the gallery is very interesting. Considering that the valve seat positions are very similar between different cylinder blocks, it is surprising that the centres of the gallery slots vary so much. Slot centre lines for cylinders no 2, no 4 and no 6 are very similar but the variances on the other three cylinders vary much more. Although the actual centre line positions could be due to variances in exhaust port casting dimensions, this would not explain why three cylinder positions are similar. In the absence of any other evidence, one can only presume that these centre line dimensions were altered in the light of development. Possibly this was after thermocouples had been used to find the localised hot spot positions more accurately on later engines. If this was the case, it seems very strange, not only to me, but also to other engineers who have been consulted on this subject, that the rear three cylinder slot positions on gallery types C and D are much shorter in length than the front three cylinders. Even stranger is the slot length on no 1 cylinder on these later units; it is three times longer than the slot on no 6 cylinder. One could be forgiven for believing that no 6 cylinder needs more cooling than no 1 cylinder. Having used engine test beds for temperature measurements on other engines, I can confirm that this is the normal case. In fact a temperature measurement of the front and rear exhaust manifolds on these six cylinder engines shows the rear manifold at least 25C hotter, even at idle. I can only defer to the company engineers on this score, as I have been unable to locate any evidence so far as to why these variations occurred on such a wide scale. In practice, however, it is normally the rear cylinders in which the overheating problems and distortions occur. We shall see later that indeed these slot lengths were eventually found to be wrong and they are discussed in more detail, under the section headed Later Developments.

No doubt the observant will have noted that the comparison slot centre lines between types C and D vary by only 0.037 on each cylinder. In the fitted situation this variance is largely eroded because of the thickness of the front end plates. Originally the late 4.5 Litre engines were fitted with a RE 16219 front end plate, which was 0.128 thick, front end plate UE 524 is currently supplied, which is 0.104 thick. These end plates both had a tolerance of 0.0075 plus or minus on thickness and the gallery slot lengths had a tolerance of 0.015. In the extreme, the type C and D gallery slot centre lines could be identical or even the type C gallery slots could be further rearwards than the type D gallery.

Each gallery type and the parts making up each assembly are detailed below.

TYPE C GALLERY ASSEMBLY

This unit was fitted with the introduction of the RE 16217 cylinder block part way through the SP series R type. The assembly part number RE 16345 consisted of a RE 16324 gallery, a RE 16085 rear end cap and a RE 14508 rear support strip. At the rear end the gallery stops 1.50 short of contacting the rear cylinder block coolant closure plate. The cross section shape is the same as the type B, but there the resemblance ends. All the slot dimensions and positions are completely different to the former types, as is the front end fixing arrangement. This gallery can, in theory, be withdrawn from the front of the engine.

On the cylinder blocks up to and including the RE 16217, the centre position of each exhaust valve can be referenced easily as the exhaust valve guide centre lines are exactly in line with the exhaust manifold attachment studs. From the front of the cylinder block the valve centres occur at 2.25, 7.875, 10.50, 16.682, 19.375 and 25.125. Whilst it should be remembered that the cylinder bore centres are offset by around 0.750 from the valve centre lines, it is interesting to compare the valve positions with the relevant water gallery slot centres in Fig 7A. The cylinder to valve offset varies fore and aft depending on the cylinder number. On cylinders 1, 3 and 5 the exhaust valve centre lines are nearer the front of the cylinder block, than the cylinder centres, whilst on cylinders 2, 4 and 6 the opposite is true. The flow of hot exhaust gases across the top bridge is therefore angled, if one presumes the centre lines as a datum point. Of course, in practice the gas has to follow a ‘U’ shaped path on its way from the cylinder to the exhaust port and take the offset exit from the cylinder. This torturous route no doubt causes a vortex in the exhaust port and bridge region and the gas impingement and the hottest point may well be offset from either the valve or cylinder centre lines.

Reference to the top of Fig 7C shows the sketches of the sections and the double tab location arrangement at the front of the RE 16324 gallery. The front end location plate RE 16219 originally fitted with this gallery, is shown at the bottom left of Fig 7B, but UE 524 is the current substitute. Strangely, this original front end plate was omitted from the R type parts manuals and the view of the engine block shows the end plate as part of the water gallery, a view that was still shown in the earlier versions of the S1 series parts manual. The RE16217 cylinder block was the first to be fitted with the front end location plate. This plate is very prone to corrosion and erosion from the blast of silt laden water egressing from the water pump, although it is preferable to having the front of the cylinder block corrode on the earlier blocks. In my own view the owner would be well

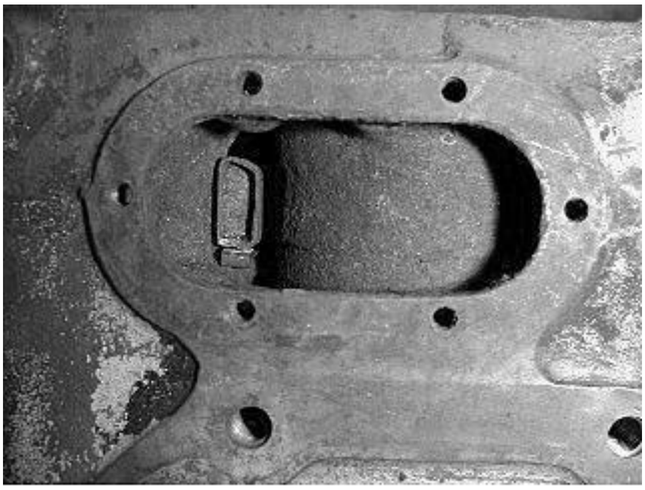

FIG 7D Photograph of Water Gallery in situ with Rear Support Strip.

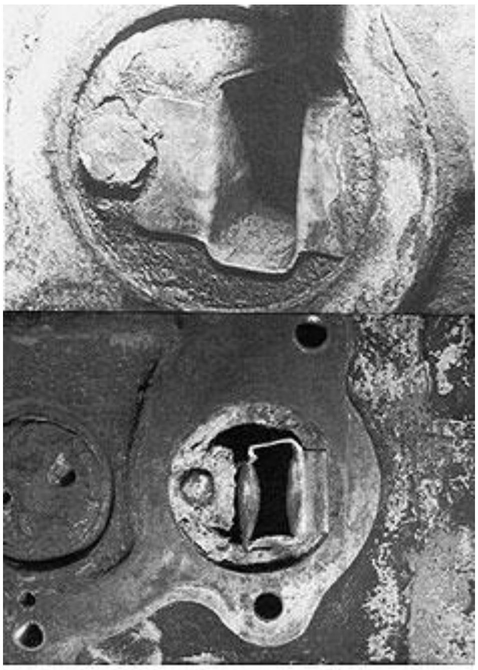

FIG 7E Photographs of corroded front end plates on the type C gallery

advised to make and fit a new front plate made from 0.125 thick brass plate and fit a brass retaining set screw, to eliminate any future corrosion.

The front end plate is fitted behind the tab locations of the gallery. In other words, to extract the front locating plate one would need to straighten the tabs at the gallery front end, not particularly a good idea with a brass component. All these galleries are located very tight in the cylinder blocks and the usual method of extraction is to remove the rear block plate and very carefully drift the gallery back and forth with soft wooden blocks. Of course this involves the removal of the engine and all that this entails.

Fig 7E shows, at the top, the initial corrosion of this plate and this is quickly followed by terminal corrosion, as shown in the bottom view. The owner does not need to be an engineer to realise that once the second stage is reached much of the water pump output is directed directly into the front end of the block. Once that position is allowed to persist, the rear engine cylinder liners, pistons and valves will overheat and quickly pass into a self-destruction mode.

Owners of R types, Silver Dawn and Silver Wraith cars which are fitted with a RE 16217 cylinder block should keep this weakness in mind, should the engine show unexplained signs of overheating. Fortunately these blocks are easy to identify as the RE 16217 part number is cast into the cylinder block face just forward of the oil dip stick hole. Prevention is better than cure in this case and it is well worth removing the water pump and the pump adapter plate to examine the gallery front plate on engines that have seen long service lives. Once the water pump adapter has been removed, the gallery front end plate can be viewed, with the aid of a mirror.

The owner who finds a plate severely corroded will immediately face a dilemma, as the front plate will almost have ceased to exist and the gallery will normally have moved rearwards in the block. The screw retaining the front plate can usually be extracted surprisingly easily, in spite of the corrosion. Even straightening the gallery tabs to fit a new front plate could be a test of the nerves, as the gallery will be fast in the block and attempts to draw it forward to its correct location may be frustrated. It is most probable that a corroded gallery front end plate is pointing to the fact that the cylinder block is in fact highly silted, as the engine is unlikely to have received recent attention. Understandably the need to remove the engine to enable the gallery to be withdrawn and a new front end plate fitted, is not to be undertaken lightly. The owner faced with this dilemma needs to console himself and consider that in the longer term many tears may be saved. Removing the engine and taking the opportunity to clean out the cylinder block through the side block access plates may well prevent a very expensive engine overhaul.

An engine operating with a front end plate in the condition shown at the bottom of Fig 7E is not going to operate for long without cooking all the major engine components. In point of fact, this particular cylinder block had already cracked some of the top deck bridges between the valve seats and cylinders. The engine must have been operated for some considerable time with very high temperatures. In all cases when it is necessary to remove the radiator on these cars, the opportunity should be taken to check the condition of the plate whilst access is good, failure to do so is false economy.

TYPE D GALLERY ASSEMBLY

This RE 20949 assembly was introduced for the 4.9 Litre cylinder blocks. It comprised of a RE 20946 rear end cap or spigot, a RE 20947 gallery and a UE 524 front end plate. These parts are shown in Fig 7B and 7C. A drain hole 0.125 diameter is drilled in the gallery base some 1.450 from the rear. The rear end of the gallery was located into a spigoted RE 20948 rear cylinder block closure plate and secured with a counter

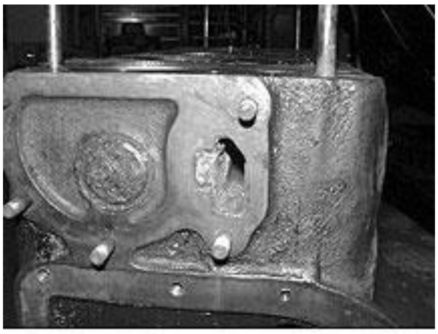

FIG 7F Photograph of RE 12893 Gallery fitted in RE9421 Block instead of Gallery EB4066.

sunk screw which passes through the block closure plate and screws into the gallery rear end cap. The gallery is firmly secured and located against the rear block plate and reference to Fig 7A shows that it is longer than the other gallery types. The advantage with this design is that the gallery is positively restrained from moving rearwards by the spigot location into the rear block plate and does not rely on any front end tabs.

The front of the brass gallery passes through the UE 524 front end cap and does not have any front end tabs. In this case, easy renewal of a corroded front plate is therefore achievable. Reference to the centre bottom of Fig 7C, shows that the gallery cross section changes from rectangular to circular at the rear end. This circular rear section will not pass forward through the cylinder block and extraction of the actual gallery assembly necessitates removing the engine.

My previous comment regarding fitting a brass front plate and retaining screw would also appear to be appropriate for this type of gallery assembly, as would be the requirement to be aware of the original front plate corroding.

LATER DEVELOPMENTS

In the introduction to this section on cooling galleries I commented on the fact that the lengths of the gallery rear slots, on types C and D galleries, are much shorter than the front slots. Additional to this I also knew that the rear of the cylinder head was much hotter than the temperature gauge, at the front of the head, indicates. Even on an engine with clean water passages the variance is in the order of 10 C; thus a temperature gauge showing around 75 C is indicating that the rear cylinder head temperature is around 85 C. One does not need to have a degree in maths to realise that some of these engines, which are running nearer 85 C indicated are in danger of boiling the coolant in the rear of the cylinder head.

Although I said “I can only defer to the company engineers on this score” I was unhappy with the situation even after fitting a high speed water pump pulley and fan and conducting some tests in August 2001. Details of the high speed water pump and fan together with the tests are related later in this article.

In Autumn 2001 I tried some simple tests by passing water down a pipe which was drilled with holes amounting to the same cross sectional areas and positions of galleries type C and D. These tests proved that the amount of water emitting from the rearmost holes was pitiful in the extreme and the flow from the front slot outlet was starving the rest.

The type C water gallery in my own engine was removed and the slots on cylinders 2 to 6 were lengthened to 0.800 inch, the slot on no 1 cylinder being left at 0.900 inch. The temperature gradients from front to rear now appeared even better and the engine ran even cooler than during the August 2001 tests. I should add that the engine is run with a high speed coolant pump and would not suggest to any owner who has retained the original pump speed that the gallery slots should be opened out to this degree. The car has now travelled some 3000 miles with this last modification and has stood the usual traffic jam tests, which England can provide, without the slightest indication of any major temperature rises.

Pursuing the matter further, more to satisfy and prove my own engineering instinct rather than finding new data, I searched the archives for information into the 60’s after the car engines had been replaced with the newer V8’s. This search was most rewarding.

The date of 1st April 1960 was rather apt for report EER 1688, titled:- B60 Engines—Modifications to engine cooling system to improve coolant distribution within the water jacket.

The forward to this report is worth repeating:-

1. In its standard form the distribution of coolant in the B60 cylinder head is uneven, there being hot spots at the rear and along the exhaust side of the head.
2. The use of a supplementary rear water outlet together with the standard front outlet and thermostat cures the above trouble. However, this arrangement may produce installation difficulties; and a second thermostat is required.
3. The use, instead of a revised water gallery in the cylinder block gives the advantages of (2) without the attendant drawbacks.
4. If (3) is adopted it will be necessary to carry out a valve life test to check that the bridge piece cooling has not suffered.

The system suggested in (2) above had already been used in Military B81 engines and found very successful. Valve life tests on B60's were indeed conducted, using this modified gallery, and the valve life was not only held, but as could be expected, valve life was very substantially extended.

This engine test was conducted using thermocouples around the head, coolant flow and outlet temperature recording and special bleeds placed in the cylinder head to detect water or steam. In addition, the rear head cover plate was replaced with a Perspex window to record the water level at the rear of the cylinder head. All the tests were performed at full throttle at, 1000, 1500, 2000, 2810 and 3750 rpm. The relevant details from this B60 engine test, which will interest the enthusiast whose car is powered by a very similar engine version, are as follows and at least gallery types C & D could be modified accordingly. My own view is that all the gallery types would show cooling benefits if they were modified to gallery D type slot positions with the slot lengths as detailed below.

The coolant flows, in imperial gallons per minute, using slots lengths as in galleries type C and D. Slots numbered from the water pump or inlet end.

SLOT #	**1**	**2**	**3**	**4**	**5**	**6**
FLOW (gpm)	3.87	1.82	1.88	1.85	1.82	1.93
LENGTH (in)	0.900	0.500	0.400	0.300	0.300	0.300
TOTAL FLOW						**13.96 GPM**

The coolant flows, in imperial gallons per minute, using modified type gallery C and D slots lengths as shown

SLOT #	**1**	**2**	**3**	**4**	**5**	**6**
FLOW (gpm)	3.31	3.08	2.91	3.11	3.05	3.70
LENGTH (in)	0.550	0.500	0.450	0.450	0.400	0.400
TOTAL FLOW						**19.16 GPM**

Further study of this report shows that with the standard water gallery C and D slot lengths the worst cylinder head temperature variance front to rear was 13 C. At an outlet temperature of 88 C, the exhaust and inlet side of the rear of the cylinder head were 101 C and 100 C respectively.

With the modified slots the worst variation was 4 C, whilst at 2000 rpm and above, the rear of the cylinder head was within 1 C of the outlet temperature, in the worst condition.

The engine revolutions appertaining when the test comparison was carried out are not recorded but from water pump output recordings the flows appear to have been compared at around 1500 engine rpm.

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This particular B60 test along with my own findings seems to vindicate my former modification of the water gallery slots.

BROKEN FRONT END TABS AND DAMAGE ON GALLERY ASSEMBLIES

There are still sources that are capable of supplying type A galleries, the type B and C are however in general short supply, therefore salvaging a damaged gallery may need to be considered. It is widely known that these cylinder blocks are prone to frost damage if the correct antifreeze procedure is ignored. Owners who may find cracked blocks and subsequently repair them successfully are apt to forget about the water gallery. These galleries are all drilled at some point at their rear end to enable correct draining down, however the drains, especially the smaller holes on the later galleries are liable to have become blocked. Any engine, which has suffered a cracked block, is therefore very likely to have a burst gallery, the consequences of which should not now need further explanation. If the block has suffered frost damage, it is sensible to remove and check the very vulnerable thin brass gallery.

It is probable that owners may be faced with the situation when the front brass tabs are broken on these galleries. Refitting the gallery in this condition will inevitable result in the gallery being driven rearwards under the influence of water pressure from the pump. Subsequently, the cooling slots will not be directing coolant to the correct areas and very severe localised overheating will occur, the effect on the engine will obviously be disastrous. These front end tab type galleries fall some 1.50 short of the rear block closure plate and therefore it is not beyond the realms of possibility to suitably extend the gallery at the rear and modify the rear block closure plate to prevent the gallery moving rearwards. In short redesigning the arrangement on the lines of the type D gallery arrangement.

REMOVAL OF THE EW 1431 COOLANT JACKET COVER PLATES

To remove the extremely hard silt deposits in these cylinder blocks it is necessary to remove the cover plates from the side of the crankcase. No amount of flushing will remove all these deposits that actually need removing with a blunt instrument.

Each cover plate is attached by six half inch long 2 BA steel screws part no K 2804/Z, which have slotted countersunk heads. The taper of the countersink is none standard. The paper type joint between each plate and the block is part no RE 4864.

The screw ends just penetrate into the block interior and the end section of the thread is subject to corrosion. If an attempt is made to unfasten the screws, this corrosion is immediately drawn back into the block threaded section, which is tapped 2 BA to fine limits. Although some screws may release in this fashion and some may actually withdraw, the fine limit on the block threads will be compromised and it is probable that some screws will jamb solid. In my own view, the best method of removal, which also preserves the thread limit better, is outlined below.

It is necessary to centre pop mark each screw head accurately. To achieve this, it will be found preferable not to even attempt to unscrew any screws, as inevitably the use of a screwdriver will foul the slotted heads and make accurate pop marking difficult. After the pop marking has been completed, drill each screw square down its centre with a sharp 3 mm drill, use a new 3 mm drill for each plate. Limit the depth of drilling to some 8 mm from the screw head face. This means the drill will not penetrate through to the block interior, but will drill through the head of the screw into the shank. Utilise a reversible electric drill for the process, a non-reversible drill is not ideal.

Using a larger drill size, and centring on the previous 3 mm holes, drill the countersunk screw heads, carefully, until the screw heads drop off without damaging the side cover if possible. Remove the side cover. This will leave the threaded shank of each screw in position complete with a centred 3 mm hole partially drilled into the shank. The initial pop marking will have shaken the threaded shank endwise and the initial drilling will have heated up the threaded area. With the screw heads now removed, reinserting the 3 mm drill and turning the drill chuck by hand will probably show a number of screws that are willing to screw into the block interior. Do not under any circumstances be tempted to withdraw the screws towards the outside of the block. If any screws are loose, but reluctant to screw down the hole, insert a tapered, not a plug type, 2 BA tap and attempt carefully to screw them into the block. Any screws, which are still tight, can be completely drilled through. Whilst attempting this, do so at a low drill speed as most screws will screw into the block interior as soon as the twist drill grips. When the screw finally passes through into the crankcase interior, reverse the electric drill and pull back, whereupon the drilled off part thread will drop into the block interior. Screws that are particularly stubborn and allow the drill to penetrate without moving, and they will be very few, will only have their threads engaged with the block as the centre section of each screw will have been drilled out. The threaded section will collapse easily and the residue either picked out with a sharp instrument or screwed out by applying a tapered 2 BA tap. This drilling procedure allows the threaded sections to be relieved of pressure and the drilled off screws to be screwed into the crankcase interior, preserving the fine limits of the block threads. Each screw end can be extracted easily from the block with the aid of a magnet. It is vital that the holes are not cleaned up with a parallel or plug tap, partial engagement only of a taper tap will suffice. If the cover plates have been damaged during the process of drilling the screw heads, they can either be cleaned up by accurate re-drilling or new ones made by a machine shop. Taking the time to carry out the process and preserving the fine limit threads will ensure that coolant does not leak down the screw threads after assembly. The removal technique can equally be applied to cylinder head or block rear end plates.

I have always used countersunk brass screws for re-assembly together with new stainless steel cover plates. In the case of the screw threads, I apply blue Hylomar jointing and, on the paper joint faces, Wellseal compound. Wellseal has the ability to seal but allow the surfaces to slide under conditions of differential expansion. The company originally developed both products. Wellseal is available in the U.K from Binneys in Birmingham **Tel:** 0121 4544545 and Hylomar from Halfords branches. In the U.S.A, I understand both products are available from Pegasus Auto Racing Supplies Inc., 2475 South 179th Street, New Berlin, WI. 53146-2150 **Tel:** 612 4762531.

HISTORY AND COOLING SYSTEM MODIFICATIONS

Without considerably modifying the cooling system and probably directing some coolant into the inlet side of the block, a few things can be done to increase both coolant and air flows. Drastically modifying the cooling system would starve other vital areas. Without a major coolant pump redesign, the best that can be achieved on a limited budget is to increase coolant volume flow and additionally open out the shutters to increase the air flow.

During the years of production, the radiator shutters were subjected to alteration on a number of occasions. At least four main variations were tried, all in the aid of increasing airflow through the radiator matrix.

The earlier MKVI's had ten shutters in each half of the grille, positioned at large angles to present a pleasing frontal effect, to the detriment of airflow. In very early 1948, from chassis B320 CF, the grille shutter angles were opened out to increase air flow through the radiator. Quickly following was the introduction of a deeper radiator in the middle of the DA series production. From the earlier part of the DZ series cars, the spacing of the shutters was altered by the deletion of one bar, resulting in nine shutters in each half. Problems were still encountered with inadequate air flow, particularly in the centre area of the matrix, the situation being made worse with the introduction of the 4.5 Litre engine. The shutter angles were opened up to around 37 degrees from the former 43 degrees and the company suggested opening up the shutter angles of existing cars. Initially, R types had a 36 degree shutter angle. During the early TN series production of the R type, the final shutter modification was made. This consisted of re-spacing and altering the angles of the nine a side shutters and inserting a centre bar to restore the frontal appearance. There is a worthwhile gain to be achieved by opening out shutter angles to improve airflow on all the cars.

KDA132

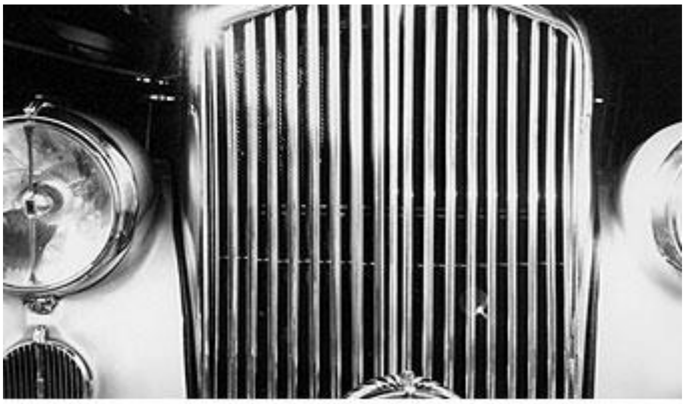

FIG8. Shutters opened out to improve air flow.

Fig 8 shows the appearance of B87 UL where the shutter angles have been increased beyond the production settings to the maximum possible angle that the shutter horizontal holding bars will allow. Although the radiator can be seen from the direct straight ahead position, moving only a few degrees right or left with the eye returns a pleasant facade.

At least two export sales markets were deemed to require extra cooling requirements. They were Switzerland and Australia. Research of the car chassis cards indicates that all cars destined for Australia and Switzerland were fitted with the high-speed fan and water pump pulley. The Australian destination is not specified in the parts manuals, I believe this was because the Australian importers, and not the company, specified the equipment. Part numbers for all the different schemes appear in both MKVI and R type parts manuals.

I had decided, a long time before overhauling my own engine, that afterwards I would retrofit the high speed water pump and fan pulley, together with the appropriate fan. The reasons for this decision have now been discussed. At this stage it is perhaps appropriate to study the specification of all these cooling systems, especially in connection with the fan and water pump drives. These specifications are followed by descriptions of the results and other tests that were carried out during the retrofit period.

In order to provide easier identification to the following descriptions, the 16.25, 17.75 and 17.25 fan assembly and drive arrangements have been designated schemes A, B and C respectively. A brief description and the introduction of these assemblies is provided below, followed by more detailed descriptions of individual parts.

SCHEME A GENERAL DESCRIPTION

Very early Mark VI engines are fitted with a 16.25 fan and a fan pulley providing a theoretical drive ratio of 1.11:1. In other words the water pump and fan are driven at a higher speed than the engine. Although the drive ratio of the fan is identical to scheme C, the pulleys are not physically interchangeable.

Scheme B general description

During the Bentley MKVI GT chassis series, the fan diameter was increased to 17.75 whilst the pulley diameter was also enlarged, resulting in a slower fan speed of 0.850:1. In their wisdom, Rolls-Royce stated that the fans were not interchangeable, although no specific reason, to my knowledge, was given at the time. The fan is actually physically interchangeable with the other schemes, but for the reasons given later this should not be attempted. There are very sound engineering reasons for not swapping fan types. The reason for the original change to this scheme was a quest for an even quieter driving environment by reducing fan noise.

Scheme C general description

In spite of the change to a scheme B assembly for most markets, this special configuration was retained and used on MKVI and R type cars for Switzerland and Australia, which, I believe, is a useful retrofit to cars in other markets today. The earlier drive ratio of 1.11:1 was used for these two export markets albeit with a special fan diameter of 17.25. This scheme was designated in later years for all markets in cases of complaints of overheating.

FIG9. High speed pulley, normal pulley and RE 6060 Fan assembly.

FAN ASSEMBLY DESIGN, RELATING TO ALL THE SCHEMES

Fig 9 shows the difference in diameters between the high and low speed pulleys and, quite plainly, the varying blade angles of the fan.

The complexity of this fan design would appear to have resulted from a very deep analysis of the frequencies that cause fan noise, or at least in depth experimentation to break up the rhythmic frequency. Close inspection reveals quite involved design issues for the time and great attention to detail, for which the company is well known. These fans are far removed from a simple and straightforward paddle blade construction.

All A, B and C schemes use a common fan flange, or central spider and rivets, part numbers RE 2017 and KB1962 respectively. These part numbers are not shown separately in the manuals as they are part of the fan assemblies listed for each scheme. Each of the five radial spiders of RE 2017 is twisted through two different radii of 3.00 and 2.84 to present a fan blade attachment point at 40 degrees to the centre flange. The spiders are disposed radially in what appears to be a group of two blades and another of three blades, simplification in the extreme. Much more detailed inspection shows that each arm is disposed radially at a different angle. These angles present the fan blades at 45, 90, 70, 60 and 95 degrees respectively to each other. The finale is that each blade is riveted to the spider in a canted position at an angle of 2 degrees from the true radial line along the length of the rivets.

When viewed directly from the front the fan blades on the fan assemblies of all schemes have the same tip radii. The larger radius at the attacking tip edge is 1.750 whilst the trailing tip edge has a 0.500 radius. Each blade is not uniform in shape but is wider at the tip end and profiled to produce a straighter trailing edge along the blade length. The curvature of each blade, when viewed from the end on position, commences at the inner end at a radius of 3.00 and enlarges uniformly along the blade length to produce the tip curve required for each fan scheme.

In each scheme the blades are riveted to the spider arms, commencing with two rivets in a line abreast position 1.250 apart and 0.375 from the inner edge of each blade. Three more rivets are used in line along the length of the blade at 1.00 pitches. Five rivets are used on each blade, the final rivet therefore being positioned some 3.375 from the inner blade edge.

Differences between the blades of each scheme are outlined below, together with the other component parts that make up each scheme.

PARTS OF SCHEME A

The 16.25 fan assembly, part number RE 4758 was drawn up in February 1946, having RE 3367 blades which themselves were drawn in August 1944. The blades are 5.125 at their longest point, 3.250 at the widest and their outer tip curve radius is 4.225.

The crankshaft pulley was part number EB 5014, the outside diameter (O/D) measuring 6.35. The top of the vee groove measured 0.930 across, with a combined angle of 36 degrees. This crankshaft pulley was replaced by RE4031, used in scheme B, when the bench type crankshaft damper was introduced around Bentley chassis B128 DA.

The fan pulley, part number RE4736, was fitted up to around Bentley chassis B213 GT. The O/D being 5.787, width across the vee groove 0.930 and the combined vee angle was 36 degrees.

It should be noted that the chassis number change over points in regard to the pulleys is approximate, as the changes occurred over a range of different chassis.

Originally the drive belt was part number EB4202, subsequently changed to RE 20941, this is detailed in scheme C, together with the relevant measurements.

PARTS OF SCHEME B

This scheme comprised of a 17.75 fan assembly RE 6060 having EW 2070 blades. These blades were 6.00 at their longest point and 3.375 at the widest with an outer tip radius of 4.225. As a digression, the blades originated from the EW 1514 blades first designed in 1937 for the proposed pre-war Wraith III. In fact all the dimensions are identical and the only difference is the change to Dural alloy material for the EW 2070 current blade. In June 1956 the fan assembly was increased in diameter to 18.00, but the part number of the assembly remained the same at RE 6060. The life of this fan assembly stretched from the Bentley MKVI GT series and through S1 series cars, except certain Continental models.

The crankshaft pulley is part number RE4031, having an O/D of 6.375, an included vee angle of 36 degrees and is 0.945 across the top of the vee groove.

A larger fan pulley, part number RE 11270, is fitted with this scheme. It has a 38 degree included vee, the O/D is 7.468 and width across the wider part of the vee groove is 0.930. This pulley can be recognised in service because the outside diameter of the fan pulley is in line with the circle formed by the water pump casing bolts.

The fan belt which matches this pulley is part number RE 20950, the included vee angle is 42 degrees, the wide part of the vee is 0.920 and the vee itself some 0.500 deep. The inside diameter of the belt is 15.160 and the makers code was V480/9R. However, an industrial belt, number C48, is an alternative fitment and can be positioned over the pulleys easily, unlike the original belt. The differences between the belt and pulley vee angles are quite normal to establish full belt contact under driving load.

PARTS OF SCHEME C

This scheme used a 17.25 fan assembly RE 19078, drawn in February 1947 and having RE 19076 blades. Each blade was 5.625 long and 3.30 at the widest point; their outer tip curve radius was 4.150. The drawing for this RE 19078 fan assembly, states that from May 1964 the assembly was no longer available. Additional advice was to produce the fan assembly by modifying a scheme B fan assembly RE 6060 to drawing R 4805. This fan assembly was reserved for cases when overheating could be a problem and finished life on the S1 series Continental, when that model had power steering fitted.

This R 4805 drawing actually shows 0.375 being clipped off each blade, however it is important to note that this reduction in tip diameter was taken off the later 18.00, RE 6060 assembly. Anyone therefore wishing to modify the earlier version 17.75 fan would be required to reduce each fan blade tip by only 0.250. The heading for the R 4805 drawing states that this modified assembly is to be used in cases of overheating complaints and for Switzerland.

The crankshaft pulley is part number RE4031, the same as used in Scheme B.

The fan pulley complimenting this scheme was part number RE 19043, having an O/D of 5.7875, the included vee angle being 36 degrees. This fan pulley and fan diameter therefore provide increased speeds of just over 29% and 25.4% for the water pump impellor and fan tip respectively above those provided by the B scheme arrangement. Coolant and airflows around the engine are therefore substantially increased with a drive ratio of 1.102:1. Physically, this higher speed pulley can be recognised, when compared with the B scheme pulley, as its outside diameter is smaller than the circle of water pump casing bolts adjacent to the edge of the pulley.

The belt, for this special pulley, is part RE 20941, having an inner diameter of 14.523, and an included vee angle of 42 degrees. Its top vee width is 0.920, the belt depth 0.50 and the original maker's code was V165/ 9R. This specification was drawn up September 1953 and it also replaced the EB4202 belt in the A scheme. An industrial belt, number C46, is an alternative fitment and can be positioned over the pulleys easily. The slide adjustment of the dynamo then takes up a position slightly further out than in scheme B, producing larger belt wrapping around the dynamo pulley.

TEST CAR DETAILS

In order to place all the results that follow into perspective, it is perhaps better to give some indication of my car and engine specification on which all the following tests took place. Individual specifications, for instance axle ratios, will have some bearing on road speeds at which certain problems are encountered.

The car used was B87 UL, a 1953 R type automatic. The engine had completed between 1000 and 2000 miles during the period of the tests since a very full and thorough rebuild, including new valves and guides, rockers, rocker shaft, push rods, standard bore full length liners, pistons and a crankshaft regrind with new bearings. The compression ratio of 7.2:1 was achieved by fitting a high compression Rolls-Royce cylinder head gasket and the static ignition timing was set at 3□ B.T.D.C, fuel was low grade unleaded. The previous owner had renewed the camshaft and the lobes were not worn. The complete cylinder block was clean, with no silt, and the radiator core was as good as new. A correct bellows type thermostat was fitted, albeit at a 73□c opening temperature, and the cooling system held a very high 70% ratio of antifreeze. Previously the radiator shutters had been modified, not just to the latest standard, but also beyond, to position the shutters at their absolute maximum open position and promote the maximum airflow through the radiator matrix.

The engine and cooling system could not be found wanting in any way, performance was exactly as one would desire, both under running and idling conditions. Even idling speeds could be attained down to very low engine speeds, so low as being able to discern each fan blade revolving if required or needed.

Carburetter needles of types SN, SP, SH and TV were used during testing, and as regards these particular tests, no variances were noted except for power and fuel consumption variation.

The exhaust system had been modified some time ago by eliminating all the internal baffles in the rear resonator silencers. A modification that was done on at least one R type from new by the company at the owner's request. The exhaust back pressure thus being at a minimum, with only a burble evident at idle and sounding much the same as the R Continental.

The rear axle ratio was the later ratio of 3.4:1, which is some 8% higher than the Bentley MKVI, tyres were Michelin 6.50x 16 XCA radial set at 32 lbs. front and 36 lbs. rear.

TEMPERATURE GAUGE CALIBRATION

To assist in clarifying any results I needed to know, on B87 UL, how the dashboard temperature gauge was performing compared with actual coolant temperature. The gauge only has two reference points, at 30C and 75C. An attempt to track down a drawing together with the correct calibration of the gauge was fruitless.

Drawing RD 3770-2 dated 29th October 1946, shows the gauge was originally fully marked up with the proposed 30, 50, 70, 90 and 100C positions. The important point to perhaps reference here is that the centre position was 70C, not 75C, as on the current gauges.

Drawing RD 3141-3, dated 8th January 1947, shows the dash panel assembly of the gauges and clearly the temperature gauge, marked as the current cars at 30C and 75C. The centre of the gauge position had therefore been raised by 5C, and it is probable that all the marks after the initial 30C starting mark may have been subject to an uplift of 5c, including the highest 100C marker.

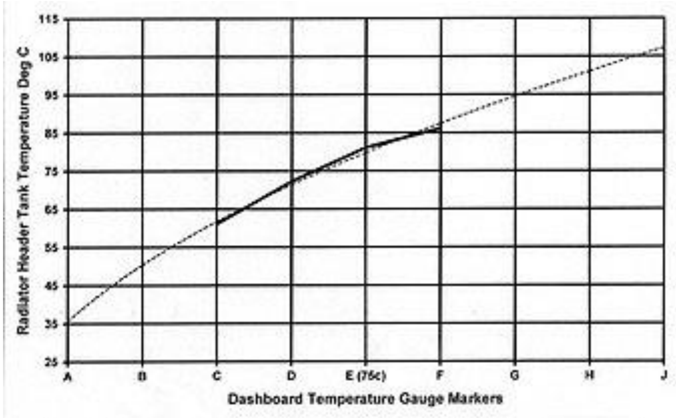

FIG10. Bentley R type temperature gauge, comparison graph with trend line.

Although immersing the bulb end of the capillary tube in water across a range of known temperatures should be carried out for true calibration of the gauge, I was not inclined to drain the cooling system off to carry out this full test. I did not want to introduce air unnecessarily into the cooling system, nor finish up with steam pockets in the cylinder head, which may have distorted the road test results. In particular, the water pumps are inclined to hold air in the upper casing for some considerable period after draining and refilling the cooling system. I therefore calibrated the gauge against header tank coolant temperature.

The Bentley temperature gauge has nine marked segments, the middle one of which is marked 75C. In order to have an understandable standard I have referred to these marks as A,B,C,D,E,F,G,H and J. The marker A, representing the lowest gauge reading of 30C. Rolls Royce car owners will note that the temperature gauges on the majority their cars are different to the Bentley versions, although the operating principle is the same.

The testing routine I followed involved using a digital cooking thermometer as the comparison. First of all, this thermometer was tested for accuracy at water boiling point and found to be within 0.5% accurate.

The figures recorded in fig 10 are the comparisons between the car dashboard temperature gauge and the temperature of the coolant in the radiator top tank. This exercise was carried out with the engine switched off and the coolant temperature checked by the digital gauge immersed in the header tank as the dashboard temperature gauge reading fell. The readings were completed after the car had been driven for 30 miles to attain true heat sink of the engine and surrounding components.

It should be remembered that the car temperature gauge is recording the coolant temperature just under the thermostat, whilst the check gauge was recording radiator header tank coolant. Inevitably there will be a differential in temperature between the two points, especially as the thermostat closes. In addition, because the gauge is mechanical, there is a slight delay in the recording temperature of the car gauge compared to the drop in temperature in the cooling system. Actual practice showed the temperature recorded at the header tank was 3C to 6C higher than that noted on the dash gauge. Of course the cooling system was not under pump circulation pressure and it is possible that this differential between the readings does not exist to such a degree in running conditions. Equally some of the differential may be caused by an out of calibration dashboard gauge or a combination of differences. The temperature difference, between the commencement of the mark indicating the normal 75 C running position, and the end of that mark, covered over 3 C (from 78.89C to 82.22C). This shows the scatter and difficulty in obtaining readings even within 3 C. Even so, the readings do provide the driver with enough data to have confidence that the car gauge is presenting the approximate signals.

I should add at this point that the rear of the engine cylinder head and block are by far the hottest points on these engines, the actual operating temperature of the engine is therefore much higher than the temperature shown on the gauge. Owners believing that their engine is running within normal temperature parameters can be therefore deluded and fall into the trap of believing all is well. S. H. Grylls, Chief Engineer, Motor Car Division, Rolls-Royce Ltd made this point well in his paper “The History of a Dimension” when he said, “ A little moral that can be drawn from this story is always to put the recording thermometer at the hottest part of a cylinder head, a practice which had been forgotten since 1937”

At the time of testing it was not possible, due to practical reasons, to test the full spectrum of the operating range of the temperature gauge. I was not prepared to boil the engine for the sake of calibration, for instance. The lower end of the scale is not of particular interest and in any case the thermostat closing masks the header tank temperatures. During calibration, readings were taken through dashboard temperature gauge marks C to F inclusive with extra intermediate readings taken between the gauge marks to provide better results. These extra calibration readings are not reproduced in the graph. The graph in fig 8 shows the C to F readings, together with a computer generated trend line, both above and below, my own readings. For illustration purposes and standardisation throughout, I have quoted the marker readings from the graph trend line. The main reason is that I believe the trend line is certainly not producing a lower reading than actual, therefore accepting this situation is erring on the safe side. Reading at a particular marked letter point on the trend line can be cross referenced to the marks attained when road testing with the high-speed pulley arrangement described later.

The top end marks are a little academic because if the coolant temperatures reach that stage, the engine is indeed in trouble. It is nevertheless interesting that the trend line indicates that the highest gauge mark J is higher than 100C. In practice this agrees approximately with test results kindly forwarded to me by Mr Dick Kress of Ohio, U.S.A who has calibrated the dashboard gauge on his car, an early R type. After testing the capillary bulb directly in boiling coolant of a 50 / 50 mixture and adjusting for the elevation at the test site, he believes the highest possible position on the gauge would be about 108 C. Strange as this may first appear, it is in fact very probable that the gauge was calibrated by the company above 100 C to counter for different cooling system mixtures and the potentially later introduction of a fully pressurised cooling system. Applying a trend line to his readings shows that the two trends are almost identical.

TESTING THE RE6060 17.75 INCH FAN WITH THE INCORRECT PULLEY

Earlier I made reference to the fact that the company had originally stated that the 16.00 and 17.75 fans were not interchangeable and this comment is physically underlined in the various manuals. Having regard to the company statement on fan interchange ability, I decided to carry out a series of tests by road testing the larger original fan with the high-speed pulley, to compare the different specifications only.

Very minute differences between one fan and another can have a remarkable effect on the fan performance, not least noise and vibration. These include the run out truth of the blades, blade angles and blade end shape at the tip, or differences in diameter and curvature. The results obtained from road testing with this fan and the incorrect pulley were therefore somewhat anticipated.

A fan pulley was produced to the dimensions of the RE 19043 drawing and fitted to the engine using a C46 belt and the original 17.75 fan. Commencing at 52 mph, the fan began to drone, this drone peaked at 55 mph and by 60 mph one had driven through the frequency. The speed was limited to 65 mph due to the newness of the engine at the time. The droning could be likened to sitting at the side of a propeller driven aircraft engine, most unpleasant and most worrying as to what was happening at the tips of those alloy fan blades.

Worse was to come. At 55 mph pressure waves could be felt travelling from the front of the engine to the bulkhead, receding as though the wave had returned to the fan and then the cycle repeated again. Like the fan roar, the pressure wave could be driven through. The car could be held at a road speed of around 56 mph, when a sympathetic driver, and one used to driving different cars of this model, knows that something is dreadfully wrong. Nevertheless it was noticeable that the higher water pump speed was having some effect on heat removal.

A most worrying aspect is the fact that some Bentley MKVI owners have in fact disregarded the company instruction and fitted the larger fans on a permanent basis. I have experienced this phenomenon before on the MKVI models but to a differing degree and this could be due to the radiator shutter angles being different. The air flow through the shutters of the MKVI is different to the later R Type in any case, and definitely different to B87 UL. The problem however, is still present with a wrongly matched MKVI fan drive. Having experienced the full-blown effect, I have no desire to be in the vicinity when the blades finally fatigue and part company with the spider. Having once had a set of blades come off the hub on an

Armstrong Siddley many years ago, I was amazed at how easily they dented the Bradbury four post lift vertical steel column. I don't believe a Bentley fan would be any different in its flight path or impact at the final destination. Those enthusiasts who have a MKVI and don't know the fan or pulley specification should take steps to investigate and quickly.

CONVERTING A RE6060 17.75 INCH FAN ASSEMBLY TO A RE 19078 17.25 INCH FAN ASSEMBLY

Rather than modify my original fan, I obtained a second hand RE 6060 fan assembly. The tips of the blades were trimmed by 0.250 to attain the 17.25 desired high speed fan diameter, according to drawing R 4805.

The tip trimming was carried out by marking out the shape of the original blade tip on a template and then cutting it out to the shape. This template was then position 0.250 down from the end of each blade and remarked in the new location. Finally I cut off the surplus material in small amounts with aviation shears and carefully finished the tips with a small file. The final act was to mount the centre of the fan on a wide ball bearing at eye level and carefully adjust the centre spider, not the blades, to ensure that each blade revolved true. This was achieved by measuring the distance from the engine side of each blade to the surface on which the bearing rested, with an aiming point of plus or minus 0.001 between the blades. By gripping the central spider between two hard wooden blocks in a vice, any adjustment was carried out. Then, by carefully bending the spider between the point where it attaches to the water pump flange and its attachment to the blade. It is imperative that the bending action is only fore and aft and does not alter the attack angle of the blade. Equally, the area where the fan is bolted to the flange must not be distorted or the blades will be forced out of true when the fan is finally bolted onto the water pump adaptor. Not surprisingly, some of the blades of this second hand fan were 0.250 out of true, a warning in itself to those who do not check any second hand fan for truth. This check is worthwhile completing on any car, in the interest of silent running.

The reworked fan was then fitted to the engine along with the new smaller pulley. Although it may look impossible, the fan can be removed from the exhaust side of the engine without any dismantling or removal of components except for any front mounted ignition coil. During removal it is necessary to turn the fan so that one cluster of the two or three blade positions are opposite to the direction of removal when the fan is manoeuvred through the gap.

TESTING WITH HIGH SPEED PULLEY AND 17.25 INCH FAN AS SCHEME C

The following temperature figures are those appertaining to the coolant temperature in the radiator header tank according to the trend line in fig 10, together with the letter allocated to the gauge mark for reference. The main reference to note is that at the centre gauge position marked 75 C, the header tank coolant was recorded at 80 C.

At an ambient temperature of 22C it was possible to leave the engine idling all day and the coolant temperature would stay on the E mark at 80 C. Formerly it would have crept up to the F mark 87.5 C. Road testing indicated no drone or other fan drive disturbance.

The ultimate test was carried out on 15th August 2001 when the temperature in East Anglia reached 33.5C (92F+), just the right day to check out the Bentley cooling system. The hot engine idle speed was set low at some 350 rpm and the carburettors were fitted with TV needles that are weak. The engine had now covered around 2000 miles but was still on the tight side. All conditions which would tend to raise the temperature of the engine.

The car was driven around the Peterborough ring road, about 22 miles distance, with some diversions off the ring road adding another 3 miles.

Once the engine had reached operating temperature, I drove some 8 miles at 60 mph, at this stage the temperature gauge was on the bottom end of the E mark 80 C. I then stopped for five minutes, leaving the engine idling with the car in the sun and the radiator shielded from any wind. By the time I drove away the temperature gauge was just above mark F at 87.5C. As the car reached 30 mph the gauge came down quite rapidly to just below mark F.

Joining the ring road again, the car was driven up to 70 mph and not allowed to fall below 60 mph at any time for another 8 miles. The temperature gauge stabilised exactly on Mark F or 87.5 C. The car was driven off the ring road about one mile and left idling for 10 minutes, in the sun and with the radiator again shielded from the wind. Ambient temperature was some 33C (92F). The gauge rose to just above mark G or 95C estimated. When driven away the temperature dropped very quickly down to mark F at 87.5 C and continued to maintain this at a consistent 70 mph.

It was noticeable that, as engine speed and car speed increased, the temperature dropped rapidly on each occasion, no doubt extra water circulation was having an effect. I believe that had I been in a traffic queue and raised the engine speed for 30 seconds or so at random times instead of just letting the engine idle, then the temperature would have been kept down. There seemed no indication that the temperature would have climbed higher than just above mark G at idle, even in this extraordinary ambient temperature.

Since the August test the car has been run in some traffic jams of 5 to 6 miles long and the extra water flow has produced outstanding lower temperatures compared with those normally expected in these conditions.

THERMOSTATS

These engines do rely on the correct type of thermostat, with the bypass control being fitted, to uniformly heat the inlet manifold and, in particular to prevent fuel condensation on the manifold walls feeding the No1 and No 6 cylinders. The gas speed to these end cylinders is somewhat slow and engines with incorrect thermostats fitted tend to develop a very rough idle and combustion roughness, with the occasional misfire on the end cylinders. This cylinder misfire situation is also compounded by the manifold design when fuel is centrifuged out of the air as the gas flow turns at 90 degrees to enter the end cylinders. The fuel droplets are deposited in the dead end section of the manifold, adjacent to each end cylinder. Although the inlet manifold design is outside this article’s scope, it is most probable that better idling could be obtained if these internal end pieces were blanked. Engines operated without a thermostat will quickly develop fuel bore washing problems and crankcase oil dilution. In particular, if the engine is fitted with the very hard chrome cuff liners, extensive damage to the pistons, ring packs and liners will soon occur. Any engine fitted with the high speed fan arrangement must also not be allowed to run too cool for similar reasons.

It is most probable that the high speed fan arrangement would allow a higher thermostat rating to be used as there is more cooling capacity available within the cooling system. There is, however, a danger when high rated thermostats are used in colder climates. On older crankcases and cylinder heads the influx of extremely cold water to an already hotter than normal engine can result in thermal shock so severe as to crack castings.

In these situations I would prefer to utilise a radiator blanking muff, positioned at the bottom of the shutters and covering about one third of the aperture. In fact using the same principle as the Rolls-Royce Silver Wraith radiator shutters. When used in conjunction with the normal thermostat this encourages a higher operating temperature, whilst still allowing an early coolant bleed to take place into the radiator.

The most efficient engine operation of a petrol engine is well above the 80C later specified for these engines, but the total heat extraction capacity of any cooling system should be enough to prevent the engine boiling under all foreseeable operating conditions. In practice this situation is almost impossible to achieve without a positive pressure cooling system and therefore these engines are running a very fine line between operational efficiency and overheating.

The operating temperatures were gradually raised over the engine design lifetime. I prefer to err on the safer side and run the engine at a lower temperature with a 73 C thermostat. Raising the temperature of an engine is normally easy, keeping it cool is more difficult. I favour having plenty of capacity in hand and allowing the thermostat to commence bleeding hot water into the radiator matrix at a lower temperature, to avoid thermal shock to the block and head castings. It should be understood that the thermostat opening setting is indeed just that and much higher temperatures need to be achieved in order to fully open the thermostat valve. In point of fact, the full valve opening temperature is only achieved when the by-pass port is finally cut off and the valve opening range takes place across a 20C temperature gradient.

Many different thermostats were fitted to the 4.25 and 4.5 Litre engines. All of them were produced to the same dimensions and only differed in opening settings and the closed by pass port temperature. A second phase of thermostat designs originating from 1955 incorporated an air bleed hole with a weighted restriction pin. For some markets, winter and summer units were produced and suitably marked, they correspond exactly with thermostats having the same initial opening temperatures except for their seasonal markings. Later, many of the individual units were consolidated and a smaller range was offered as replacements, some of the older unique units are still available from some sources. For reference, a limited list is reproduced below showing opening range specifications and the bypass port closure temperatures. The current suggested thermostat fitting is part number RH 9143 which commences opening at 78C.

The widest brass body dimension, where the unit fits into the recessed location in the thermostat housing, is 2.312 diameter. An adjacent smaller location flange is 2.124 diameter, this flange passing into the hole formed in the thermostat housing. The maximum valve lift is 0.275, this position always occurs simultaneously with the full cut off or closed position of the bypass port. Overall length of the thermostat is 3.343.

PART NUMBER	OPENING TEMP	BY PASS CLOSED
RE 15943	70-75□c	90□c
RE 11528	75-80□c	95□c
RE 22619 Winter	84-86□c	104□c
RE 22618 Summer	75-77□c	95□c
UE 3531	75-80□c	95□c
UE 4194 Summer	70-73□c	90□c
UE 4518 Summer	70-73□c	90□c
UE 4519 Winter	75-80□c	95□c

CONCLUSION AND NOTES

I would like to take the opportunity to extend my thanks to all the staff at Hunt House who have been more than helpful during periods of my research. In addition James Pate, Dick Kress, Robert Godbey, Bill Vatter and Ken Yokosawa from the U.S.A, Ron Venter from Canada and Bobbie King and Mike Kendrick from the U.K. They have assisted with experimenting with high speed pulleys, supplying photographs or generally reading parts of this article. Their assistance and helpfulness is appreciated.

Although this article has been written in relationship to the background of the Bentley models, the principles apply to their Rolls-Royce counterparts the Silver Dawn and Silver Wraith, with certain exceptions. The Silver Wraith L.W.B certainly has differences in the drive pulley arrangement. In all cases the models used the RE6060 fan assembly.

Throughout, the original parts numbers have been quoted as these correspond to part numbers in spare parts manuals that many owners possess. In some instances however the part numbers have been superseded by a modern equivalent.

Every effort has been made to check the historic information detailed in the article and to examine all the drawings and appropriate files. Unfortunately some data is missing from the archives and in other instances the minor details on drawings are not clear. Inevitably some mistake may have crept in, hopefully, if any, only of a very minor nature. The author would be pleased to receive any information from any member who identifies any error.

It is hoped that the case for not boring out cylinder liners and especially when they have been subject to partial seizures will be taken on board. Any owners who have engines that have suffered liner damage may hopefully contemplate that it is safer to fit new liners.

The down side of the high speed fan modification is that when operating in ambient temperatures of 16 C or less the engine will show an inclination to run cool, or at least, at no higher than the thermostat cracking temperature. In climates that are very hot in the summer and very cold in the winter, it may be more convenient to change the pulley types over for winter and summer, rather than change a thermostat.

Even on much later models the company documentation indicates that the automatic choke flaps tended to close at speed under the influence of air flow, even on warm engines. On these earlier cars, fitted with an automatic choke, one must also be aware that the choke coils are set so that there is no trend for the choke to come into operation. This situation was anticipated on B87 UL and the choke coils set to a weaker position, when the engine was rebuilt. This is not to suggest that the engine would run too cold but that the cooling capability is improved and, on a like for like basis, these engines will run cooler after modification.

The positive aspect is that the water flow around the engine and heat distribution is much better and I believe silt in the cooling system will have less chance of accumulating at the rear end of the engine cylinder block. The addition of this modification should keep a sound engine in the best of condition for some time, but it is not a cure for otherwise sick engines that have silted up cooling systems. In fact, it is possible that the extra water flow gained from a high speed pulley could circulate otherwise dormant material into the radiator from a highly silted cylinder block.

Whilst the engine design lends itself to cylinder liner distortion, I believe extra water flow cannot but help to keep the problem under control. This modification was obviously deemed appropriate originally for the markets where the cars could be subject to high temperatures and loading. As the frequency of higher speed running and high density traffic situations are now common practice, I believe this is one of the best specifications to apply to these chassis to help overcome any tendency to overheat. Operating in some of the highest temperatures we have experienced, in England at least, the high speed pulley and modified water gallery seems to have proved its worth.

Road testing with the wrong combination of fan and pulley confirms that when used in this configuration, stress in some form is taking place within the blade swept area. Whatever form those stresses are taking will more than likely eventually lead to blade failure. Owners of very early cars would be well advised to check that the correct fan has not been replaced by the normal R type fan in an effort to increase cooling capacity. Any early engine fitted with an oversize fan should respond to trimming the blades.

The high speed fan pulley RE 19043 is no longer available It can however be reproduced by any good machining shop without difficulty, using the normal pulley as a general pattern but extreme care need to be taken to avoid a foul occurring with the water pump by-pass casting section.

It does appear that one main factor affects the temperature gradients on all these blocks and that is the position and size of the water gallery cooling slots. Altering these slots to accord with the centre positions of a type D or RE 20947 gallery and adjusting the slot lengths to the measurements shown in the Later Development section of the discussion on water galleries, would seem to defer most overheating problems.

On 4.5 Litre engines that have RE16217 cylinder blocks and all 4.9 Litre version blocks the corrosion and erosion of the front water gallery plate should be watched. Corrosion of this plate will promote overheating.

During the tests, and to date, evidence of excess coolant pressure has not been found, although some higher internal pressures must be assumed. The radiator water level has remained constant and has not been topped up in over 2000 miles. It is possible that any extra water pressure might find a weak spot in old heater matrixes. B87 UL has the later hot water valve in the heater line, which provides some automatic restriction. Owners of cars, which are not fitted with this valve, would be wise to fit a restrictor disc in the heater pipe to restrict any high water pressure being applied to the heater matrix.