This article was written by Steve S.Virginia, USA,who certainly knows his onions.It is over 33.000 words so you may want to print it off

Tuning The 1800cc MGB

There are few mysteries about the engine employed in the MGB.

During the era in which the B Series engine was designed, hydraulic lifters for automotive applications were still in their technological infancy, therefore the engine was designed to use solid lifters. This offered the designers the opportunity to wisely leave the camshaft exposed to the crankcase so that its lobes could be lubricated by a spray of oil emitting from the lower ends of the connecting rods. This desire to lubricate the lobes of the camshaft and the lower sections of the tappets dictated the thickness of the connecting rod big end. Adequate bearing support was achieved by using a large diameter big end design.

Its Heron-type head used Weslake-patented combustion chambers which were a marked advance beyond previous technology, allowing for superior flow characteristics while permitting excellent flame propagation. The incoming fuel/air charge was directed toward the spark plug and away from the hot exhaust valve, minimizing preignition and allowing less ignition advance to be used. The siamesed intake ports, like some other features of the engine, were largely the result of production economics. By using siamesed intake ports the intake manifold could be of simple design and thus be inexpensive to produce. The tappets and pushrods could also be neatly situated between the ports, thus keeping the head as narrow and light as possible. The placement of the intake and exhaust manifolds on the same side of the head meant that only one mating surface need be machined, and fewer manifold mounting studs and their attendant threaded bores were required. It also allowed the distributor and generator to be placed on the opposite side of the engine, thus greatly simplifying maintenance.

There are also some distinct engineering advantages to this approach. By placing the intake ports with their cool fuel/air charge next to the hotter exhaust ports, this area of the head is better cooled than it would be in a crossflow design, precluding warpage and possibly extending the life of the exhaust valves, although this configuration allows more heat to accumulate in the walls of the intake ports and thus is detrimental to intake charge density and hence lowers power output potential. The small-bore long-stroke configuration gives better thermal efficiency and thus better fuel economy. The bore centers are the same as those on the earlier, smaller displacement versions of the engine, so the new engine could be produced on much the same tooling, thus keeping costs within reason.

Although the B Series engine design is truly a compromise, it's a brilliant one that modern mechanics recognize as one that was far ahead of its time when introduced. It was further improved with the introduction of its 5 main bearing version. A higher capacity Holbourne-Eaton oil pump was provided to supply the bearings which were 1 1/8" wide for the front, center, and rear bearings, and 7/8" wide for the intermediate bearings. They all had diameters of 2.125", a full 1/8" greater than that of the previous 1622cc three main bearing version. This produced an almost unbreakable crankshaft with lots of overlap between its journals and counterweights. Certainly there were other new engine designs that were even more advanced in the mid-to-late 1940s, but this one was intended to be available in cars that ordinary people could afford to own and operate. In those days, that made it special, and its designers had every reason to be proud.

During an era when full race engines struggled to reliably produce 1HP per cubic inch, when the 18G Series arrived in 1962 it boasted 95HP from a mere 110 cu. in., giving it a specific output of .864hp per cu. in., and this was an engine that could reliably be used as a daily driver! In its heyday, it was impressive indeed.

Pretty fantastic for a relic whose design is well over a half century old! A true classic engine for a true classic car!

Everybody who is about to rebuild their tired engine entertains the thought of improving upon the power output of this classic engine design. However, nobody wants to end up with a temperamental beast. Since you're rebuilding the engine, this is a good opportunity to do it the Peter Burgess way. As a former professional mechanic who has built custom engines, I can assure you that I have thoroughly read both of Mr. Burgess' books "How to Power Tune MGB 4-Cylinder Engines" and "How To Build, Modify, And Power Tune Cylinder Heads," and that his theories are both sound and logical. His reputation as the MGB engine tuner is well deserved. His books should be in every MGB owner's library. His website can be found at http://www.mgcars.org.uk/peterburgess/ . If you have not studied his books, they are available from Veloce Publishing at http://www.veloce.co.uk/newtitle.htm . I wholeheartedly agree with his statement "The entire engine system needs to be considered as a whole, otherwise the gains from component changes may not be fully realized."

Before you begin, you will need to have a proper Service Manual. I would recommend that you purchase

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If your engine is a post-1967 North American Market model, then it is equipped with an antipollution system. To get better performance out of the engine, it will be necessary to remove some of the components of this system. Prior to doing this, check with your State Officials to find out if this is illegal. Be advised that in some states where it is illegal to tamper with a vehicle's antipollution system it is not required to be maintained once a car has reached a certain age, so specifically inquire about this issue as well. Be aware that it is desirable to retain certain items of this system, so don't start by simply stripping everything off. Instead, proceed with the same methodical approach that you would use toward any other part of the car.

If yours is a 1964 through 1969 GA through GF Series engine equipped with a PCV Valve, it should be retained to reduce atmospheric pressure inside the engine. However, if the compression rings start to fail, oil mist from the engine will saturate the oil separator tube of the early version of the front tappet chest cover and be transferred into the combustion chambers through the induction system, the consequent carbon buildup eventually resulting in problems such as preignition, sometimes called "pinging." The front tappet chest cover from the later 18V engines (18V-797-AE, 18V-798-AE, 18V-801-AE, 18V-802-AE, 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, 18V-891-AE-L) is preferable due to its better breathing characteristics and for having incorporated into its cover design an oil reservoir/return chamber which minimizes the transfer of oil mist into the induction system. When replacing the gaskets on the tappet chest covers, use sealant to glue the gaskets to the covers and allow it to harden overnight so that the gasket will not move during installation.

If you choose to not remove the hose that leads from the Gulp Valve to the fitting on the center of the intake manifold, it can be simply blocked with a plug, or, after removing the intake manifold, threads can be tapped into the intake manifold with a 1/4" NPT tap and a nipple installed.

At this point you may remove both the hoses and the Check Valve that connect the Air Pump to the Air Injectors atop the head. Next, remove the Air Pump, its air cleaner, and the attendant mounting brackets. When the engine is equipped with the Air Pump, the Gulp Valve is necessary to prevent backfiring when closing the throttle at high engine speeds, so remove the Gulp Valve along with its hoses and its attendant hardware as well. At idle the intake manifold vacuum is in the order of 18 to 20 Hg, while on the overrun it rises to 23 to 25 in Hg without the Gulp Valve. This is not enough to make a significant difference in terms of the amount of fuel pulled out of the jet, thus the Gulp Valve is unnecessary once the Air Pump is removed. Next, remove the Air Injectors and replace them with 7/16"-20 fine threaded bolts 3/4" in length. Do not be tempted to use Allen head plugs because they will have to be bottomed out into the head, projecting into and thus creating an obstruction to air flow in the exhaust ports. Finally, if your engine is from a post-1974 model, remove the EGR Valve and its hose and control pipe, the fuel shutoff valve, and the vacuum advance valve.

You should retain the Anti-Run-On Valve fitted on the 1973 and later models as its purpose is to apply such a strong vacuum to the chamber above the fuel in the float bowls that the fuel cannot exit the fuel jets when the ignition is switched off, thus preventing the car from running on. When the ignition is turned off the ignition switch energizes this solenoid-actuated valve to close it, then the oil pressure switch releases it after the engine has stopped and oil pressure has fallen. When the engine is running the Anti-Run-On valve is open, allowing fresh air to be pulled through the adsorption canister, clearing it of the vapors that have expanded into it from the fuel tank and the carburetor float bowl chambers, then through the rocker arm cover and tappet chest into the induction system to be consumed in the combustion chambers. The rocker arm cover is equipped with a restrictor tube to prevent the fresh air being drawn in from overly diluting the fuel/air mixture and causing lean running. This Anti-Run-On system can be readily retrofitted onto 1970 through 1971 18GJ and 18GK engines as well as the 1972 18V-584-Z-L and 18V-585-Z-L engines, all of which have the necessarily modified fuel tank, adsorption canister, non-vented oil and fuel filler caps, and restrictor tube equipped rocker arm cover as standard equipment. Do not remove or disconnect the Vapor Separator that connects the fuel tank to the Adsorption Canister.

It is important to retain the crankcase ventilation system. Properly maintained, crankcase gases are drawn into the combustion chambers of the engine by the vacuum created by the fuel induction system, either through the intake manifold as in the 18GB through 18GF engines, or through the carburetors as in the later engines. This permits the crankcase to function in a partial vacuum which causes oil mist inside the crankcase to be drawn upwards towards the camshaft and tappets. Without the partial vacuum provided by this system, the pressurized gases inside the crankcase of the B Series engine would cause oil to be blown past the pistons into the combustion chambers leading to carbon buildup and consequent preignition problems. In addition, an excess of these pressurized gases and oil mist would also be vented partially through its rocker arm cover, pressurizing the adsorption canister and interfering with its function, rather than traveling down through the pushrod passages as they should to aid in the lubrication of the lower ball ends of the pushrods and the upper sections of the tappets. For the excess pressurized gases in the crankcase to arrive at the rocker arm cover they would have to travel up the past the pushrods. This means that the gases would be forced upward around the tappets, depriving their upper sections of the additional lubrication supplied by the oil mist and the oil running down the pushrods from the rocker arm assembly. It thus must be understood that all of this is prevented by drawing all of the pressurized gases inside the engine out through the tappet chest cover and into the induction system under vacuum, and as such the system contributes to a prolonged engine lifespan. These procedures having been performed, you can now set out on a quest for more power.

You must accept the fact that more power will increase both wear and stress on your engine's components. Hence it is important that the basic components of the engine provide a sound foundation.

Remember: if anything is worth doing, it's worth doing right. Have all of your components, including the crankshaft, block, heads, connecting rods, and rocker arms hot tanked to remove the years of accumulated crud that is to be found in all old engines. Prior to this being done insist that all of the gallery/core/frieze plugs be removed. Remove the aluminum Engine Number Tag from the block prior to hot tanking as the caustic chemicals will dissolve it. After hot tanking, all of the internal passages should be chased out thoroughly with brushes and flushed. Be sure to tell your machinist that the area inside of your block around the rear cylinder is commonly a trap for sediment and to be sure that all of it is removed. All threads in the block should be chased with a tap and all holes should be reamed. Also insist that new oversize bronze plugs be shrink-fitted slightly beneath the surface of the block so that they won't interfere with proper gasket sealing of the sump and end plates. Bronze has a higher coefficient of expansion and contraction than iron. To "shrink-fit" them into the block, put them into a ziplock bag, turn the thermostat all the way down on your deep freeze, and leave them in there overnight. That "shrinks" them to a smaller diameter. When you're ready to install them, take them out, spray them with WD-40 to displace any moisture on them, then seat them into the block with a flat-nosed punch. When they warm to room temperature, they'll be in there good and tight because they've expanded! The only way to get them out is to drill and tap threads into them and use a puller! Because bronze expands more than iron when it gets hot, there's no way that they'll ever come out while driving down the road. Stainless steel Frieze plugs should be used for the same reason. Their high chromium content also means lots of expansion when hot, so they won't pop out, either. Make sure that they have a good concentric seating surface by specifying that an end mill bit be used to clean up their seating surfaces in the block. Not the cheap way to do it, but it always works.

Never reuse old gaskets, seals, oil gallery plugs, frieze plugs, core plugs, bushings, bearings, valve springs, shims, thrust washers, piston rings, circlips, wrist pins, rocker bracket studs, rocker shafts, head mounting studs, manifold studs, connecting rod bushings, connecting rod bolts, or the main bearing cap studs and/or nuts. None of these items are expensive, and recycling them into your engine is not only false economy, but an open invitation to future mechanical failure.

Be sure that all bearing support surfaces are line-reamed and their oiling holes carefully deburred. If possible, it would be wise to have the rocker arms, heads, block, crankshaft, and connecting rods magnafluxed or, better yet, x-rayed to be certain that there are no cracks. All of the rocker arm faces should be resurfaced on a contour grinder and rehardened if they are not to be replaced by new ones.

Warped mating surfaces are the major contributing factor in leakage and in the development of cracks in the head casting. While today’s sealants are excellent and today’s gaskets possess greater compressibility than those of the past, they can compensate for warped mating surfaces only to a very limited degree. Use a Payen or Fel-Pro head gasket or one that is marked FRONT/TOP as these should be quality gaskets. These gaskets are resin-impregnated, have copper sealing rings, and require no additional sealing coatings. The resin softens when it gets hot and makes a better seal. They are particularly appropriate for use on engines that have been converted to aluminum heads as they handle the differing coefficients of expansion between a cast iron block and an aluminum head quite well. Do not allow the gasket to overhang into the bore of the cylinder as this will lead to a blown gasket and/or internal damage to the engine. You will need to retorque the cylinder head immediately after the initial running of the engine.

During the course of an engine rebuild it’s common to find that the block is warped along its longitudinal axis, so we’re always prepared to line-bore the main bearing and camshaft journals. However, we rarely stop to consider that this warpage should also extend to the mating surfaces elsewhere on the engine. The necessity of skimming them flat just as one would the deck of the block and the mating surface of the head should always be explored. To check for warpage in your garage, simply clean the mating surfaces and smear a very thin stain of machinist's bluing or petroleum jelly on them. In a smooth, perpendicular motion, place a clean plate glass or a mirror on the surface and then gently pull it away. Hold it up to a light and look for any gaps in the bluing/petroleum jelly outline. If you find any, you’ve got warpage. This technique will work with any mating surface. Get the mating surfaces flat and you’ll have gone a long way towards having an oil-tight engine.

Paint the engine before reassembly with a thermoconductive enamel engine paint only. Hirsch has an excellent engine enamel which, being unique in that it was originally formulated for use on jet engines, will withstand temperatures up to F 600 and is an exact duplicate of the shade of red ("MG Maroon") used on the 18G through 18GK Series engines. It remains glossy almost indefinitely and can be applied directly to cast iron without primer. Hirsch has a website at http://www.hirschauto.com/ . Do not allow paint to get onto any gasket mounting surfaces or into any threaded holes. Do not paint the front face of the engine rear engine plate where it mates up with the gasket to the back of the engine block. Also, do not paint the area of the rear engine plate where the starter motor mounts, because the starter needs a solid electrical ground in order to work properly. Instead, these gasket areas should be masked off prior to painting. Once the masking is applied to the surface, place the component onto the plate and scribe around it with an Exacto knife, then simply peel away the masking from the area to be painted.

The most desirable engine blocks for a high output engine are the early 18V blocks of the 1972 through 1974 Chrome Bumper models. These later blocks have bolts instead of studs for securing their stronger main bearing caps. These can be readily identified by their engine numbers: 18V-584-Z-L and 18V-585-Z-L from the 1972 model year, and 18V672-Z-L and 18V-673-Z-L from the 1973 through 1974 model years. These later main bearing caps have shallow recesses for the heads of their mounting bolts while the earlier caps have deeper recesses for their washers and nuts. The later main bearing caps can be used in the earlier engines only if their appropriate mounting bolts are also used and only if they are line-bored.

Be aware that the later 18V blocks from the 1975 through 1980 Rubber Bumper models have a repositioned motor mount boss on the camshaft side of the block and so will not fit into earlier cars. These can be readily identified by their engine numbers: 18V-836-Z-L, 18V-837-AE-L, 18V797-AE-L, 18V-798-AE-L, 18V-801-AE-L, 18V-802-AE-L, 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, 18V-891-AE-L, 18V-892-AE-L, and 18V-893-AE-L.

The crankshaft with the best balance and wear characteristics is the flat-sided five-main-bearing cast iron version found in the early 18V engines (18V-584-Z-L, 18V-585-Z-L, 18V-672-Z-L, and 18V-673-Z-L). Although slightly weaker than the alternate steel crankshafts used in five-main-bearing engines and seven pounds heavier than the earlier three-main-bearing steel crankshafts (32 lbs Vs 25 lbs), it is strong enough for the streetable enhanced-performance engine that is the goal of this article. Advise your machinist that the crankshaft main bearing caps and the connecting rod end caps are individually matched paired sets and hence are not interchangeable. Following this, the crankshaft should be indexed and the lengths of its throws matched. Be sure to tell the machinist that you want the journals radiused at the web to reduce the chances of breakage under heavy loadings. Check both ends of the crankshaft for any grooves worn into it by the old seals. If they cannot be polished out by your machinist, then a Speedi-Sleeve will be necessary (Moss Motors Part# 520-515). Moss Motors has a website at http://www.mossmotors.com/ .

This having been done, the effective length of the connecting rods (eye center-to-eye center distance) should be matched. If possible, have the connecting rods balanced end-for-end. Have both the piston/ring/wristpin assemblies and the connecting rod assemblies matched respectively to within .10 of a gram. Pistons that use only three rings are lighter than the older-design four-ring and obsolete five-ring designs. The reciprocating masses having thus been matched, the crankshaft and the flywheel should then be dynamically balanced separately. Advise the machinist that you would prefer that the balance of the crankshaft be achieved by wedging rather than by drilling. These procedures are fundamental to producing the smoothest running engine possible and will provide a bit more power that would otherwise be lost to the production of vibration, in some engines perhaps as much as 3 HP. Lightening the flywheel to a minimum weight of 16 lbs will cause the engine to pick up and lose RPM faster with the clutch disengaged and thus enable faster shifting, although at the price of increased vibration and a tendency for the engine to stall due to decreased flywheel inertia. Should you choose to have this done, advise the machinist that the material to be removed should be taken from the front and back faces and not from the clutch friction surface.

Electropolishing and shot-peening of the connecting rods is necessary only if you're going racing. Note that exotic lightweight connecting rods such as those marketed by Carillo (590 grams) are primarily intended for racing use and are unnecessary for use in all but the most radical of street engines, although their lower reciprocating mass will reduce both horsepower loss and vibration. The obliquely split connecting rods first used in the three main bearing 18G and 18GA engines used a smaller-diameter (.750") wristpin. Both it and the obliquely split connecting rod of the five main bearing engine (18GB through early 18GH Series) weighed in at a ponderous 980 grams. Not only are they heavy, they are notoriously weak in highly stressed engines. The horizontally split connecting rods with balance pads used in the late 18GH through early 18V engines were a lighter 845 grams. The final version of the connecting rod used in the late 18V engines had no balance pads and were the lightest, weighing 760 grams. These can commonly be found on engines whose identification numbers start with 18V-883-AE-L, 18V-884-AE-L, 18V-890-AE-L, 18V-891-AE-L, 18V-892-AE-L, or 18V-893-AE-L. Be aware that the connecting rods used on the 18GB through 18GF engines use connecting rods that used the larger 13/16" wristpins that floated in a press fitted bushing in the small end of the connecting rod. This bushing was later eliminated in those of the 18GH through 18V engines. These later engines also used the larger 13/16" (.8125") diameter wristpins which were press fitted into the connecting rods, so your pistons must be chosen accordingly. However, the small end of the later connecting rods can be machined to accept the earlier bushing if floating pistons are desired. If you desire lighter connecting rods to further reduce vibration and its attendant power loss, the late Original Equipment ones without the balance pads found on the late 18V engines will fit this requirement at minimal cost.

When installed, the oil squirt holes of the connecting rods must face the side of the engine opposite the camshaft to cool the piston and lubricate the load bearing surfaces during the power stroke. Failure to do this will eventually result in extreme piston pin wear within the piston itself, plus create the very real likelihood of piston failure, not to mention increased bore wear as well. Positioning the connecting rods so that the oil squirt holes face the camshaft is not necessary as the camshaft receives excellent lubrication from both the pressure galleries in which its journals spin plus residual oil flowing down the pushrod bores from the rocker arm assembles, as well as oil sprayed from the crankshaft's main bearings and connecting rod big end bearings at the crankshaft. Be aware that on some connecting rod bolts, only one side of the bolt head is chamfered to provide sufficient clearance for the camshaft, so note this fact when you reassemble them.

Aside from matching the weights of the reciprocating components and dynamic balancing of the crankshaft and the flywheel, perhaps one of the best ways to create a smooth engine is to equalize the compression and thus the power impulses occurring in each cylinder. Once the crankshaft and the connecting rods have been indexed, this can be accomplished by making sure that the combustion chambers are of equal volume so that the compression ratio in each cylinder will be the same. The volume of each combustion chamber can be measured after the head has been skimmed flat by using a clear piece of sheet plastic with a small hole drilled in it. Simply put a bead of chilled grease around the edge of a combustion chamber and press the plastic down onto it so that the grease forms a seal. Using a syringe or an eyedropper with a scale of measurement on it, carefully fill each combustion chamber with light oil, keeping a record of how much is necessary to fill each one. Next, use a Dremel tool to gently remove small amounts of metal from the smallest combustion chamber. Work slowly. The walls of the combustion chamber should be kept perpendicular to its roof to ensure the best flow characteristics. The roof should be flat and finished with a sanding disc, care being taken to not undercut or groove the base of the wall where it adjoins the roof. Take care that in an attempt to unshroud the intake valve you do not attempt to remove too much material from the combustion chamber wall near it as this can lead to preignition and breaking into a coolant passage. Unshrouding is best left to experts. Instead, remove material evenly from the roof of the combustion chamber. Measure the volume of the combustion chamber repeatedly until it matches that of the largest one. Repeat this process on all of the combustion chambers until their volumes all match. You should now have equal compression on all four cylinders, making for a smoother engine.

Be sure that both the mating surface of the head and the deck of the block have been skimmed flat and that the stud mounting holes are chamfered, or at best you'll ultimately experience a blown head gasket or at worst a cracked head. Flycutting lacks precision and should be used only as a cost-cutting measure for removing metal prior to the final precision cut. An end mill produces a superior finish for street machines, the grooves left behind by the end mill providing a surface that the head gaskets can bite into and thus produce a better seal. After milling, the ridges at the edges of the grooves left on surface of the deck of the block by the machining process should be removed. Surface grinding is good for racing engines that use only copper head gaskets and face frequent disassembly. Note that the deck of the block must be parallel to the axis of the crankshaft. The Original Equipment specification piston crown to deck depth is .040".

It may be necessary to shim the rocker stands after the head has been skimmed so that standard-length pushrods can be used. If so, be careful to maintain the original symmetry of the thrusts of the opposing ends of the rocker. The edges of the combustion chamber and the valve seat recesses must be carefully deburred and smoothed to preclude the possibility of "hot spots" developing and thus prevent preignition from consequently developing.

While all of the connecting rods used in the B Series engine use the same center-to-center length (6.500”) to produce a connecting rod to stroke ratio of 1.86:1, they differ greatly in design details. Connecting rods from the 18GB through the 18GF Series all use floating wrist (gudgeon) pins that ride in small end bushings and are retained by circlips, while those of the 18GK and 18V Series use press-fitted wrist (gudgeon) pins, although the wrist (gudgeon) pins are of the same diameter (.8125"). These press-fitted pins typically require a 3 to 5 ton press to install. Prior to installation the ends of the pins should be checked to be sure that they have been lightly chamfered to prevent them from damaging their bores in the piston. The safest installation technique is to chill the pins in a deepfreeze overnight to cause the metal to contract to a smaller diameter and to boil the pistons in water to cause the bore into which the pins fit to expand just prior to pressing in the chilled pins. Although the pistons themselves are interchangeable between the five-main-bearing engines due to their identical small end and big end bearing sizes, they must be installed complete with their appropriate connecting rods and wrist (gudgeon) pins. Pistons of the three-main-bearing 18G and 18GA Series engines used wrist pins of .7500” diameter and as such cannot be used in the connecting rods of the later five-main-bearing engines

Because of the B Series engine's relatively short connecting rod to stroke ratio of 1.86:1, the engineers at MG insisted on forgoing the use of the usual split skirt Lo-ex piston normally installed in other versions of the B Series engine intended for use in the more sedate family sedans and instead chose to specify solid skirted pistons to minimize the effects of the greater side thrust loadings resulting from the higher engine speeds attainable with dual carburetors, thus gauranteeing reliability. The Original Equipment Hepolite pistons are of excellent quality and need not be superseded by specialty racing pistons. They also have the distinct advantage of having their oversize number impressed upon the forward part of their crowns to ease reassembly. These are available from Advanced Performance Technologies. They have a website at http://www.aptfast.com/ . Prior to boring the block each piston should be measured with a micrometer to establish its optimum bore size. They should be fitted to a clearance tolerance of .003” to .0035”. Because of the close proximity to the bore of the studs, the bores of the cylinders tend to distort slightly when the head is torqued down. Some machinists will try to compensate for this by sizing the bores to their maximum factory-specified clearance, but this will result in a shorter piston and bore life. The proper approach is to mount a blanking plate to the deck of the block and torque it to the same specifications as would be used when mounting the head in order to simulate the stress of a torqued head, then bore the cylinders. The top of the bore should be then be chamfered to reduce the development of "hot spots" that are precursors of preignition and to allow easier installation of the piston/ring assemblies. When using a hone to crosshatch the cylinder bore, bear in mind that it is the fine grooves created by honing that hold oil to lubricate the pistons and their rings. A groove angle of 120 degrees is optimum. After honing, use a plateau hone to clean off the ridges of the crosshatch grooves and thus facilitate easier seating of the rings. Afterwards, both the pistons and bores should be again precisely measured and the pistons paired to their optimum bores.

If you choose to install the Original Equipment 8.8:1 compression pistons with their 6.2 mm dished crowns of the earlier 18GB through 18GK engines with their connecting rods into the later 18V engine, then the smaller 39cc combustion chamber volume of the 18V heads will boost the compression ratio to about 9.4:1, presuming, of course, that the machinist hasn't removed too much material from the block or the head, in which case it will be higher. On the other hand, if the later low compression ratio pistons of the 18V engine (8:1) with their 16.2mm dishes are installed into an engine equipped with the head from an 18G through 18GK Series engine with their 43cc combustion chambers, then the compression ratio will be a very low 7.7:1. Fortunately, the UK/European market pistons for the 18V engines were available in a 9:1 compression ratio. Don't go over 9:1 on the compression ratio with unmodified combustion chambers or you'll most likely regret it when it preignites on America's newest federally mandated development: Oxygenated Gasoline. Without professionally modified combustion chambers, any increase in compression beyond 9:1 will give only a moderate increase in power at the expense of streetability.

Replacement of your ancient-and-probably-stretched-by-now head studs with new stock ones from Brit Tek (Brit Tek Part # HSK001) or stronger ones made of 8740 steel from ARP (Brit Tek Part # HSK002) is also recommended. Stretched head studs will not hold their torque settings and will lead to a leaking or blown head gasket and possibly a warped and/or cracked head. Repeated retorquing with stretched head studs will likely result in a cracked head. Retap the threads in the block prior to installing them with antisieze compound on their threads and do not attempt to torque the studs down as this may lead to cracking of the block. Torquing of the head compression nuts will accomplish this task just fine. Don’t make the all-too-common mistake of running steel head studs all the way down into the block until they bottom out. Steel studs have a different coefficient of expansion than that of a cast iron block and preloading them will aggravate this factor. If they’re bottomed out in the block they can cause the deck to distort as they expand more than the block, and that could lead to a blown gasket, or even a cracked deck of the block. When the block cools, being a casting it will tend to return to its original flat shape if it hasn't cracked. Never use a thread locking compound as it will result in damage to the threads whenever the studs are removed, rendering them useless. Should the stud spin in its threads when torquing, check to be sure that the studs aren't undersize.

Use either the original hardened thick head stud washers or replacement items of the best quality (thick and with machined faces) on the head, never thin mild steel ones. Make sure that the washer seating surfaces are machined flat with an end mill after the head has been skimmed so that they will be a parallel plane to the mating surface so that torque readings will be accurate. Put an anti-seize compound on the threads prior to installing the head compression nuts and torquing them to the head. While oiling of the threads is commonly done to protect from rust, but the antisieze compound will do an adequate job of protecting the threads from corrosion. If you're really paranoid about the threads corroding, then use acorn nuts!

A word about valve materials: For many years the standard exhaust valve steel was EN52. This steel was first introduced over 70 to 75 years ago and has a hardness of 25 to 31 HRc. Improved engine design has lead to increased compression ratios and higher operating temperatures, and improved fuels with an increased octane rating and the addition of tetra-ethyl-lead have lead to an increasing tendency to prematurely burn out the valve. This steel is classed as "semi" corrosion resistant as they are attacked by Chlorine and Sulfur compounds. As a result, this material is no longer considered suitable for exhaust valves, although it is still perfectly satisfactory for inlet valves when used with unleaded gasoline.

About 1960 a new steel, Austenitic 214N (Stainless), was developed. This steel has a hardness of 30 HRc and retains its hardness even up to temperatures of 800 Degrees C and possesses excellent rupture strength under high temperature conditions combined with good creep and impact values. Its high Chromium content gives good scaling resistance, and has greater corrosion resistance against Chlorine although is still not immune to sulfurous attack. This is the preferred material for use with the higher combustion temperatures attendant with unleaded gasoline.

A more recently developed material, Nemonic 80A, has a hardness of 32 HRc and has an increased operating temperature over Austenitic 214N as well as higher corrosion resistance. Due to its high cost, it is commonly seen only in very high compression ratio engines built expressly for racing.

Tufftriding (AB1 or TF1, the process used depends upon the specification of the valve) gives a hard layer of between 72 to 74 Rockwell ‘C’ over the complete valve of approximately 10 to 20 microns in depth, and gives excellent wear properties in a cast iron or bronze guide with the added benefit of stress relieving the valve. This type of treatment produces a black mottled finish all over the valve. Hard Chrome Plating gives the stem added durability by depositing chrome on the stem to guide area of the valve of approx. 32 to 72 microns in thickness. This gives good compatibility if the valve is made in Austenitic 214N (Stainless) and is used in a cast iron guide. This type of treatment is only applied to the valve stem. A Stellite 6 deposit can be applied to the exhaust valve seat face which will enhance the seat hardness (Rockwell ’C’ of 38 to 42 HRc) which will enable it to be used with unleaded fuel or in highly stressed engines. A Stellite 12 deposit can be applied to the tip of the valve stem which will further enhance the tip hardness (Rockwell ’C’ of 48 to 52 HRc).

When installed, all valves and valve guides should be of equal respective heights. Because valve guides will frequently distort when being pressed into their bores in the head, they should always be reamed to their manufacturer's recommended clearances after installation to assure a consistent internal diameter. Do not waste money on exotic tuliped valves. Due to the side draught configuration of the B Series Engine's ports, they will actually flow less than an Original Equipment flat-topped valve and will increase reciprocating mass in the valvetrain unnecessarily. Like a waisted throttle shaft, waisted valves are nice, but they really won't have much effect in a street engine. They're primarily for very-high-rpm racing use with a camshaft like a Piper 300 and full race heads. The risk with waisted valve stems is that they can vibrate like a tuning fork at maximum lift during high engine speeds. The vibration can cause metal fatigue to set in prematurely and then the valve stem will fracture, the valve head being sucked into the combustion chamber, there to do all sorts of evil things. That's why they're never reground and reinstalled by racers. Short Fatigue Life. Don't ever try to recycle them. Once the seating faces are worn, toss them in the trash. I would make one suggestion that Mr. Burgess does not mention in his book: for use in a street engine, once you've had the three-angle face made on the valve, it should be either stellite-plated or (preferably) tuftrided after lapping it in. Neither of these improvements is overly expensive and will help to ensure a long, long service life in street use.

Don't be tempted into trying to repair a cracked head by taking it to a welder. Welding cast iron is a very tricky thing, requiring the right tools. Contrary to what some welders might tell you, as a former Tool & Diemaker I can explain why it can't be done on a bench in the garage. The problem lies in the fact that a casting is essentially just a bunch of bubbles held together by metal. There is always the risk, even though the alloy of the block and the alloy of the welding rod may be the same, that the density of the weld will be different from that of the density of the casting. This results in different rates of expansion and contraction when the casting heats and cools. If the density of the weld is not the same as that of the head, the casting will crack where it adjoins the weld and you'll find yourself back where you started.

However, because creating a weld is nothing more than a matter of heating the metal alloy of the rod to the point that it flows into and heats the metal of the casting to the point that it liquefies and blends with the molten alloy of the welding rod, it is possible to achieve the same density if certain conditions are met. First, the temperature of the molten metal of the welding rod should be no higher than that necessary to attain a molten state. Second, the casting should be heated in a heat treating furnace until it almost melts. The white-hot casting then is removed and the weld applied, then the casting is quickly placed back in the furnace and very slowly brought down to room temperature in controlled stages. Although this controlled cooling process will help to allow stresses to even themselves out, the casting may be warped and require machining.

Why is it so necessary to heat the casting in a furnace instead of just heating it with a torch on a welding bench? So that the temperature of the weld will be as close as possible as that of the casting. Why is that so important? First, because of the density issue already described above. That requires a degree of precision control that a welder can't attain with a blowtorch, even though he may sincerely believe that he can. Face it, the man is a welder, not a trained Tool &Diemaker or a trained Mechanical Engineer. He simply does not know any better. Secondly, because the thermal stresses created by the extreme heat of welding will be minimized and not be isolated to the area immediately around the weld due to the fact that the heat differences are not as localized. Cast iron conducts heat very slowly, so the closer the temperature of the iron of the casting to that of the weld when the welding process begins, the less thermal stress is generated in the areas adjacent to the weld. This elaborate procedure is necessary to eliminate the possibility of cracking due to induced thermal stress, which is a separate issue from that of weld density. The whole idea behind the process is often called "stress relieving," a process that I'm sure that you've heard of. Now you understand just what it is.

Needless to say, this process is expensive. If the problem is with a crack in the head, I would just scrap it. There are many used heads available in good condition for far less than what the above-described process costs. You'd have to pay for the machining costs on the head either way that you choose to go, so why bother?

The better shops will do most or all of the aforementioned machining and engineering procedures as a matter of course. If the shop you're considering can't provide these services, they are merely tradesmen rather than professionals: go elsewhere.

I would also suggest that you use the later type of crankshaft oil thrower that is common to all five-main-bearing engines and its matching timing cover which uses a neoprene seal rather than the leak-prone felt seal of the earlier timing chain cover. A duplex-type camshaft drive chain tensioner, the 3/8” pitch duplex camshaft drive chain and sprockets of the 18G through 18-V-584-Z-L and 18-V-585-Z-L Series engines, plus a nitrided rocker shaft (Advanced Performance Technologies Part # RSB-T) will aid in achieving long-term durability. In addition, an adjustable sprocket (Brit Tek Part # PGS001), although expensive, will enable you to easily keep the camshaft operating in phase with the crankshaft as all camshaft drive chains wear and thus "stretch." However, the same objective can be attained in a less expensive manner by using offset keys to adjust the timing of the standard camshaft sprocket, although adjustments made in this manner are far more troublesome and tedious. The reuse of old camshaft drive sprockets is false economy. A set of worn sprockets will result in uneven and accelerated wear of a new camshaft drive chain, thus causing its length to oscillate. This will accelerate wear of the camshaft drive chain tensioner. The oscillation of the chain will cause both the valve and ignition timing to "wobble" inconsistently, playing havoc with performance. Install a new slipper pad on the camshaft drive chain tensioner and check that the mechanism is functioning properly. Be sure to inspect the bore of its adjuster body for ovality (+. 003" max.). Should it prove to be worn out, a new one can be obtained from Advanced Performance Technologies (APT Part # BCT-1).

Replacement or refurbishment of your tired old harmonic balancer is highly advisable as it reduces torsional stress on both the crankshaft and the camshaft, as well as reducing wear of the camshaft drive chain, coolant pump, and alternator due to reduced oscillating stress loadings. Advanced Performance Technologies' stainless steel version (APT Part # 18CSP-2) has provision for easy removal. However, your Original Equipment harmonic balancer can be rebuilt by a specialist (Damper Dudes, 6180 Parallel Drive, Anderson, CA 96007 (800) 413-2673).

Although the 18V-672-Z-L and later versions of the 18V engine sacrificed dual valve springs for single valve springs in an effort to reduce production costs, it should be remembered that these later engines reached their maximum power output at the notably lower engine speed of 4,800 RPM than the earlier engine's 5,400 RPM and thus spring surge was not a problem. However, at the higher peak operating speeds and greater valve lifts that a power-enhanced engine attains, a single valve spring is inadequate to avoid valve bounce and spring surge. Spring surge can result in a valve failing to close rapidly enough to avoid clashing with the piston on the upstroke, while valve bounce can lead to a broken valve. Dual valve springs are thus a necessity for an enhanced-performance engine in order to control spring surge at the high engine speeds which can be achieved, especially if a hotter camshaft that relocates the power output peak to a higher RPM is utilized.

Be aware that the early type spring caps with square groove cotters used on the 18G through 18GF/2159 non-overdrive and 18GF/530 overdrive engines will not work with the later type round groove valve spring stem cups. Larger valve sizes with the square groove machined for the earlier size keepers are not available. This is just as well, as the round groove type is better. You will therefore need to use the later type dual spring caps used on the 18GF/2160 non-overdrive and 18GF/531overdrive through 18V-585-Z-L engines to go with the round groove keepers. You will also need the valve spring collars of the 18G through 18GK Series engines to go under the inner valve spring in order to locate it properly.

Old pushrods can be trouble. Because of the fact that the central axis of each of the tappets is offset from that of the camshaft and the the tappets have a .002” dome on their faces which bear against lobes’ surfaces which are obliquely slanted away from the rotational axis of the camshaft, the tappets spin in their bores when being lifted by the lobe of the camshaft, thus reducing both friction at the tappet/lobe interface and consequent wear. Should a pushrod become bent, it will prevent the tappet from rotating in its bore, ruining the tappet/camshaft interface and rapidly wearing out the lobe. (You weren't really going to reinstall those ancient pushrods in a blueprinted engine, were you? Know what metal fatigue is?) If you should choose to reuse your old Original Equipment pushrods, they should be inspected for signs of bending and excessive end wear. Remember that the ball ends of the pushrods have mated to their individual tappets and rocker arm ball adjusters (11/32") have mated to the cup ends of the pushrods over the years, so when you take them out, keep them in ordered sets and make sure that they are oriented as they came out of the engine (cup end up). Because the rotating faces of the rocker arms have also mated to their adjacent rocker stands over the years, even if you intend to replace the old pushrods with new ones, be sure to keep them in the same order as that in which they were previously installed or you may have problems aligning the center of the thrust faces of the rocker arms over the valve stems.

Clean the pushrods thoroughly, then put a very thin coat of machinist's bluing or petroleum jelly on their shafts. Roll each pushrod on a clean piece of plate glass and then examine the stain on the glass for gaps. That'll tell you if the pushrod is bent. Be aware that a bent pushrod can cause its tappet to stop rotating, resulting in uneven wear of the tappet, which in turn will make accurate setting of the valve clearances impossible and eventual ruination of the lobe of the camshaft. When reinstalling them, make sure that you put some fresh motor oil onto the upper end of the tappet and also down the pushrod passages to lubricate the cups on the pushrods and the tappets.

Unlike Original Equipment pushrods, tubular chrome-moly pushrods do not deflect at the higher engine speeds that an enhanced-performance street engine can produce, plus they have less reciprocating mass and thus will give more accurate valve timing at high engine speeds. This is a problem for both the early short pushrods (72 grams) used in the 18G through 18GK Series engines and the later long pushrods (88 grams) used in the 18V Series engines as they tend to deflect as much as 5/64" at high engine speeds. Crane makes an excellent set of 5/16" diameter 18V pushrods (64 grams) for this purpose (Crane Part # 905-0004) and can supply them in custom lengths if necessary. They have a website at http://www.cranecams.com/ . Due to their larger diameter (.3125" Vs .280"), it will be necessary to relieve the passages in the head for the pushrods in order to eliminate interference. Be aware that simply boring these passageways to .660” to accomplish this may leave insufficient material to permit portwork to be done.

The shorter (1 1/2" length), lighter bucket tappets (45 grams) introduced on the 18V-584-Z-L engines will also assist in the goal of reducing reciprocating mass. Due to their having identical diameters of 13/16” (.8125”), the early long barrel tappets (81 grams) and the later short bucket tappets are interchangeable when paired with their length-appropriate pushrods. The later OE tappet/pushrod assembly is 13% lighter than the earlier OE long barrel tappet (2.298")/short pushrod (8 3/4") combination used in the earlier 18G through 18GK Series engines. The reduced deflection angle of the longer pushrods (10 1/2") decreases side thrust loads on the tappets and thus enhances their lifespan. Crane's lighter chrome-moly pushrods will also reduce inertia in the reciprocating mass of the valve train by about 20% when compared to that of the later Original Equipment 18V short bucket tappet/long push rod combination and by 30% when compared with the earlier Original Equipment 18G through 18GK Series long tappet/short pushrod combination.

Be aware that the heads used on the 18G through 18GK Series engines and those used on the 18V

Series engines are of different thicknesses due to the different depths of their combustion chambers and redesigned coolant passages of the 18V Series engines. As a result, the heads used on the 18G through 18GK Series engines are taller (3.172") than those of the 18V Series engines. As a consequence of this, their pushrod/tappet combinations have different included lengths (277mm 18G through 18GK, 274mm 18V). As a result, if you should choose to install the later 18V bucket tappets and longer pushrods into an engine equipped with one of these earlier heads, it will be necessary to screw their rocker arm ball adjusters 3mm further towards the bottom of their travel. This will result in an increase in the effective length of the fulcrum arm of the rocker, with a consequential slight decrease of valve lift.

If coupled with new Original Equipment-specification dual valve springs and their valve spring cups as used in the pre-18V engines, this reduction in reciprocating mass should be sufficient to easily protect the engine from valvetrain float and valve/piston clash up to at least 6,700 RPM when used in concert with camshafts and rocker arms that have the standard amount of lift, plus reduce both camshaft and tappet wear as a result of their lower inertia loads. These valve springs should have a free length of 1 31/32" (inner spring) and 2 9/64" (outer spring), and for proper preload they should have an installed length of 1 7/16" (inner spring) and 1 9/16" (outer spring). Taken collectively, all this should ensure more accurate valve timing resulting in a smoother, more powerful output at high engine speeds.

Although simply fitting a stiffer set of valve springs as a less expensive alternative to reducing reciprocating mass in the valvetrain is possible, in reality it's a poor practice. The additional pressure on the cam lobe/tappet interface and the increased stress on the camshaft drive chain and sprockets will result in accelerated wear of these components. In extreme cases the increased torsional stress can also cause the camshaft to distort along its axis at high rpm, playing havoc with valve timing and risking the breakage of the camshaft itself. Should you elect to use a camshaft with an amount of lift greater than .450" you should consider further reduction of the reciprocating mass further by substituting a set of lightweight alloy spring caps for the heavier steel Original Equipment items. Should you choose to employ them, light alloy spring caps should be checked for deformation at the time of every valve adjustment in order to prevent the valve from pulling through the cap, resulting in a dropped valve. Always use valve springs with rates and lengths that are recommended by the manufacturer of the camshaft. If the installed lengths of the new springs are to be greater than that of the Original Equipment items (Inner: 1 7/16", Outer: 1 9/16"), it will be necessary to counterbore the spring seat surfaces in the head to the proper depth to attain the manufacturer's recommended preload setting length for the springs. Many amateur engine builders will attempt to prevent the springs from binding by being sure that when they are installed they have a certain minimum of .XXX" clearance between the coils. Unfortunately, there is no such "magic clearance figure" that will universally insure against this. Always follow the spring manufacturer's recommendation on this issue, just as you would on the issue of installed height. Peter Burgess recommends a .050" difference between the compressed height when the valve is at full lift and the fully compressed height to avoid valvetrain compression damage.

Just as gasoline is the food of an engine and its cylinders are its lungs, so oil is the life blood of an engine and the oil pump is its heart. I cannot overemphasize the importance of this fact. If your engine is to live a healthy life, its oil pump must be immaculately rebuilt. Unless you're building a highly stressed high output engine, your Original Equipment-specification oil pump will be adequate to the task. This is due to the fact that its design is of the eccentric rotor type. Its rate of flow increases in direct proportion to the engine speed. Any increase in pressure beyond that of the oil release valve spring rating results in the opening of the oil pressure regulating valve and the excess oil falling into the oil sump. Properly rebuilt, it should deliver 60 to 70 PSI at idle when oil temperature is 200 degrees Fahrenheit. The early version of this pump used on the three-main-bearing version of the engine had a problem of having its pressure fall off above 5,500 RPM, an issue which was addressed on the five-main-bearing engines by machining the pump cover and providing a second inlet port to the sump.

A high-volume/high-pressure oil pump will only require more power to function as well as increase stress and consequent wear on both its spindle gear and its drive gear on the camshaft as well as increasing torsional stress on the oil pump drive shaft. Because such an oil pump is of additional benefit only at low engine speeds on an Original Equipment-specification block, it will do little for any engine other than one whose oiling system has been comprehensively modified to suit a high performance specification. Should you choose to pursue this objective, it would be prudent to install a bronze spindle gear to preclude the rapid wear that attends such applications and to preclude breakage at high engine speeds. This can be obtained from Cambridge Motorsports.

Remove any and all burrs that you can find in the pump body and make sure that the pressure regulating valve operates freely. It should have a satin finish chrome plating on it to prevent galling. Be sure to lap it into its seat to ensure idle pressure and to flush out the lapping compound completely before reassembly. The relief valve spring should have a free length of 3 inches. Using your factory service manual, check the clearance tolerances on the rotor and make sure that the passageways in the body and delivery arm have no sudden steps or angles to inhibit oil flow. These can often be removed with a Dremel tool and a polishing bit. Doing so should eliminate the risk of a loss of oil pressure resulting from cavitation at operating speeds up to 6,800 RPM.

The Special Tuning Manual mentions, amongst other modifications, machining an extra feed port into the bottom end cover of the oil pump to improve flow. Today's replacement pumps already incorporate some of these modifications, but do not include the extra feed port. Some specialist suppliers offer pumps fully modified with the extra feed port according to the Special Tuning Manual specifications for use in engines that attain very high engine speeds. The disadvantage of this modification is that when the engine is shut off the extra feed port then becomes a drainage passage. Oil that is inside the pump body flows back into the sump. At each cold startup, it will require an extra second or two for oil pressure to build up. In addition, after every oil change it will take longer to build up oil pressure (about 20-30 seconds or more) because draining the oil sump exposes the oil pickup, and this helps drain the oil out of the pump through the extra port. While this is not a problem on a racing engine that will be disassembled and inspected several times during a season, on a street driven car it can contribute to severely shortening the life of the journals of the crankshaft as well as that of the engine bearings. Unlike racers at a track, few owners of street-driven cars will be willing to go through the procedure of repriming the oil pump every time that they want to start their engines.

Be aware that two different diameter oil strainers (105mm and 135mm) were used to protect the oil pump, the larger of the two being the more desirable due to its larger strainer area. When mounting it to the oil pickup extension of the oil pump, take care to ensure that its top surface is flat against its gasket and is well sealed so that no air leak can occur. Under normal operating conditions this area is below the level of the oil, but under hard cornering it can become exposed to the air, resulting in air bubbles being pumped into the bearings and in consequent hammering of the bearing surfaces. If you have a tendency to push the car very hard through curves and turns, this later oil strainer was introduced on and is common to all five-main-bearing versions of the engine. If you have a tendency to push the car hard through curves, have a baffle plate welded into the sump pan to prevent oil surge and thus ensure a ready supply of oil for the pump. A blueprint for a sump baffle plate can be found on page 457 of the Bentley manual. If you do not have access to the means to create your own baffle plate, one may be purchased from Cambridge Motorsports. They have two versions available, one compatible with the 105mm oil strainer of the 18G and 18GA Series engines and the other with the 135mm oil strainer of the 18GB through all 18V Series engines.

It is possible to install the larger capacity 12H3541 oil sump of the 18GA through 18GK engines onto a 18V engine to take advantage of its 50% larger oil capacity (9 pints Vs 6 pints). Although the earlier oil sump has a bulge at its rear to allow for drainage from a slot in the earlier rear main bearing cap, its lip will match the flange of the later engine. Both oil sumps have the same bolt hole pattern and use the same gasket. However, the later 18V sump is not usable with the earlier 18GA through 18GK engines due to its lack of the bulge at the rear.

While the oil supply created by an Original-Equipment oil pump and oiling passages in the block is adequate for use within the normal operating speeds of a stock-output engine, if an increased-output engine is called upon to operate at higher than normal engine speeds or under heavier loadings, such as when a Piper BP285 camshaft is installed or the engine is modified to Big Bore specifications, it becomes prudent to modify the oiling system. This is due to the fact that the oil flow from the front main bearing supplies the number one cylinder's connecting rod big end bearing, oil flow from the rear main bearing supplies the number four cylinder's connecting rod big end bearing, and the flow from the center main bearing supplies both of the connecting rod big end bearings for cylinders number two and three. The oil passages from the main oil gallery to the main bearings are all the same diameter, thus for the same oil pressure they all have the same flow capacity. However, the center main bearing has almost twice the flow requirement because it is oiling three bearings (the center main bearing and two connecting rod big end bearings) as opposed to only two bearings for each of the front and rear cylinders (one main bearing and one connecting rod big end bearing).

To compensate for this, open up the oil passage from the pump to the oil outlet at the rear of the block to 1/2" (.500"), the same size as the outlet on the oil pump. A special 1/2" Internal Diameter oil feed line using -10 Aeroquip adapter fittings will need to be custom-fabricated to enable the increased oil supply to flow efficiently to the oilfilter stand. The oil passage to the center main bearing will then need to be enlarged from its original 5/16" (.3125") to 11/32" (.34375") diameter and the main crankshaft journals #2 and #4 cross drilled and center grooved. This grooving should be accomplished by grinding rather that by turning on a lathe to prevent the creation of stress risers that could result in breakage of the journal. The journals for the connecting rods cross should then be drilled 110 degrees back from Top Dead Center with the drilled passage intersecting the original oil passage to prevent lubrication failure resulting from centrifugal forces at high engine speeds. Remember that whenever any journal is drilled it will need to be chamfered and reground afterwards. The crankshaft should then be hardened. With these modifications, a high volume oil pump becomes useful as the extra flow through the bearings provides additional cooling under conditions of high load and sustained high engine speeds and you should be able to reliably run the engine to 7,000 RPM. However, if you desire higher operating speeds than 6,500 RPM, you will have to fit rocker arms which run on needle bearings as the standard bushings will fail. Cambridge Motorsport offers these items as roller rocker arms in either the Original Equipment 1.426:1 or 1.625:1 high lift ratios with the option of either central or offset oil feed. Both types are located by tubular steel spacers to prevent the rocker arms from "walking" at high engine speeds.

Be aware that there are essentially three types of bearings available to suppo