

TUNING STANDARD TRIUMPHS OVER 1300 c.c.

By David Vizard



CARS
and CAR CONVERSIONS

**SPEED
SPORT
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**TUNING STANDARD TRIUMPHS
OVER 1300 cc**

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PORT
MOTOBOOKS

FIRST IMPRESSION FEBRUARY 1970

85113-029-1

Printed and Published by

SPEED AND SPORTS PUBLICATIONS LTD
Acorn House Victoria Road Acton London W.3.

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INTRODUCTION

IT would appear that over the last decade, Triumph's policy has been to introduce a car of a basically modern design-and, over a period, develop it to suit various markets. This policy has been adopted in preference to the more normal method of introducing an entirely new car for a given market.

Up to a point, this type of thinking must be commended. Since the majority of Triumphs are endowed with a respectable level of roadholding and braking power, the only items which become vastly changeable are the power units and transmissions. Since the non-sporting driver is unlikely to complain if the car has a far higher level of roadholding than he is ever likely to use, the manufacturers have, in effect, killed two birds with one stone. The two birds in question are, of course, the sporting and the nonsporting type of driver.

There is inevitably a disadvantage when going about things in this manner, and this can be easily summed up as a weight penalty. If we design a car in which a range of engines are to

be used, ones, that is, which vary considerably in size, weight and power output, we must build the rolling chassis to cater for the most powerful engine it is likely to have fitted. This means the car can be a little overweight for the smaller engines, thus resulting in a more sluggish car than is really desirable.

For an example, to take the Herald/Vitesse range of Triumph cars. The early Herald with its 948 c.c. engine and an all up weight of around 17 cwts. could hardly be considered the spriteliest of machines, rather it should be considered to be the heavy-weight amongst 1-litre vehicles. With the introduction of the 1200 the weight to capacity situation improved but was still not altogether remedied. When the 1600 Vitesse came along things took a turn for the better. Here we had a car which was not unduly heavy for its 1 1/2litre power plant. Not only did the car have-a good weight to capacity ratio, but also an engine whose output could be substantially raised to produce a relatively quick machine. But for competition purposes the 1600

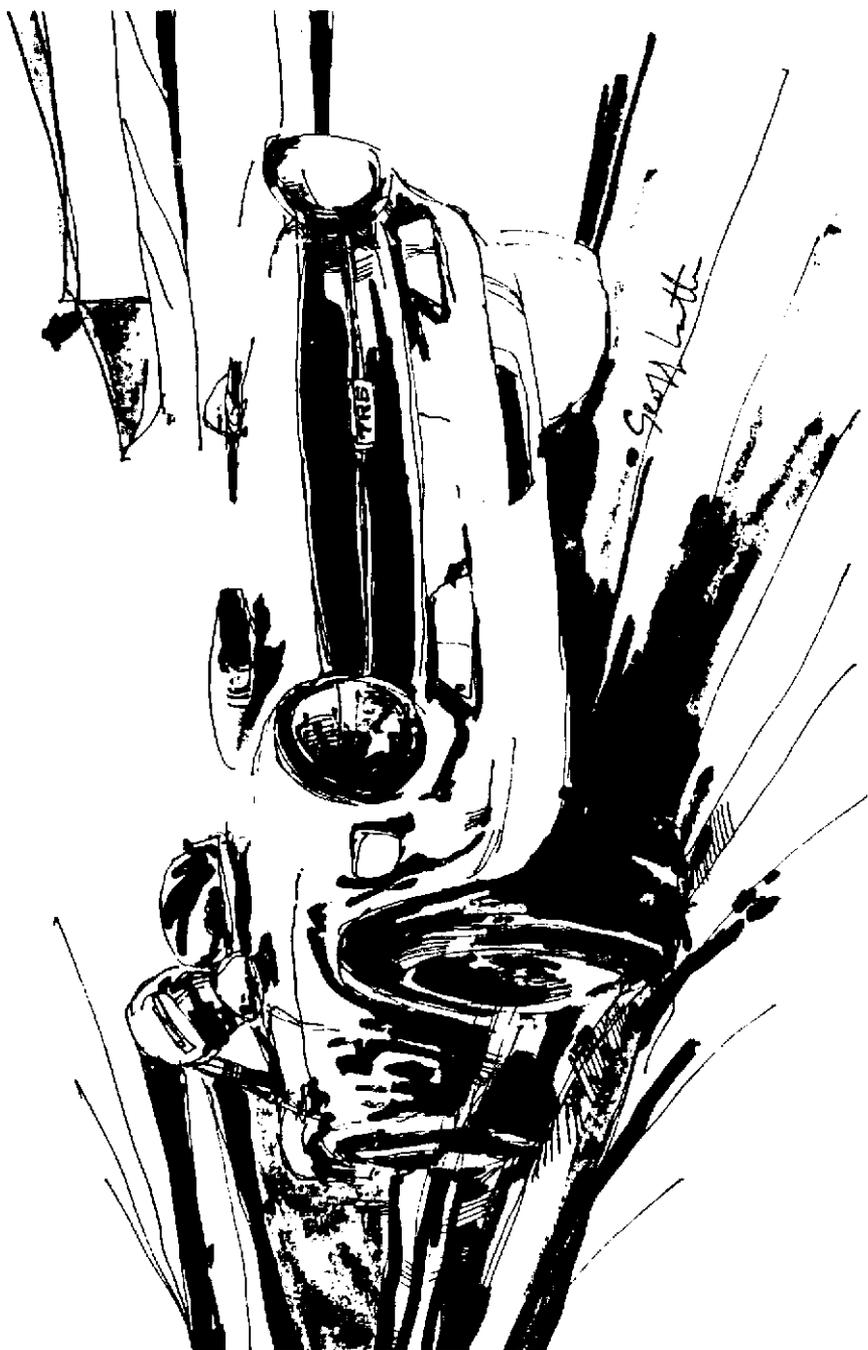
Vitesse still lacked the edge needed to make it a successful race machine. When the capacity of the Vitesse was increased to 2-litres and the G.T.6 was introduced, one would have expected the competition scene to change substantially. It would not have been unreasonable to suppose that these two cars would have dominated the competition in their respective classes, but this has not been the case. Half the reason for this could be that old ideas die hard. The thought of one of Triumph's products sweeping the field and making hay with the rest of the competition may not yet have caught on. The idea that it could be possible and that they should do so is not altogether unreasonable.

Let us look at the situation in an analytical fashion and compare, say, the 2-litre Vitesse with some of its successful competition rivals produced by other manufacturers. For our purposes a good comparison would be to see if we can achieve a machine with a similar performance to an F.V.A. engined 2-litre Escort. If this can be done, then without a shadow of doubt we can say a Vitesse can be competitive.

The F.V.A. Escort has a 2-litre engine producing a power output in the order of 220 b.h.p. The overall weight ready to race is about 15% cwt. and suspension is independent at the front but not so at the rear. The rear end on these cars is served by an extremely well located live axle. How would our hypothetical race prepared Vitesse compare with this? First off, we should find it very difficult to match or even approach the power output of the F.V.A. engine without resorting to very expensive twin overhead cam heads. Tuning Std Triumphs over 1300c.c. This inevitably limits the

number of us who could entertain the feasibility of such a project down to a very small minority. Our best plan is to ignore such a scheme and stick with the more conventional pushrod layout. If we really went to town with down draft heads, fuel injection, etc., etc., we could achieve a possible 190-195 b.h.p. from the engine, although, might I say, this figure has yet to be realized. Straight away we find we are down on the power of the F.V.A. engine by 25 - 30 b.h.p. To a certain extent this deficiency can be offset. By an intensive lightening programme involving the use of as many fibreglass components as possible, the weight of a Vitesse can be reduced to just under 15 cwt. so a little of the lost power is made up for by having slightly less weight. Even so, our power to weight ratio is still down by 8-1/2%. There is a possibility that we can offset some of the disadvantages of this lower power/ weight ratio in the suspension department.

Unlike the F.V.A. Escort, the Vitesse has an independent rear suspension. This means we have less unsprung weight to contend with which theoretically should result in better road holding. Against this must be added the fact that the Vitesse is one of the few cars which still utilizes a chassis which in all probability is not as stiff as a unit construction body. The result of this is that our carefully planned suspension geometry may not follow the exact motion it was intended to. In pointing this out I may be leading you to believe that even a fully developed suspension will be inadequate for our purposes. This is not the case. The Triumph rear end will give better results than a conventional solid axle but its superior



Geoff Lamb

ity might not be quite as much as one would expect.

Having made our comparison, it can be seen that a really competitive Triumph is not by any stretch of the imagination impossible. It may be a bold assumption, but I would predict that we are likely to see far more action in the next few years providing no great restraint is placed on the situation.

In an effort to encourage Triumph owners or would-be Triumph owners on to greater deeds in the field of competitive motoring, this book has been prepared. In the main it will give tried and proved methods of obtaining more performance. Here and there

ideas will be put forward which are although untried based on sound engineering practice.

These ideas will show a gain in power but the amount gained can only be estimated. It has also been recognized that not everyone is bent on producing an all-out racer, so all mods described will start from square one and progress from there. For those of you who are not fully acquainted with your car, it is a good idea to purchase a workshop manual for the relevant machine. This book is not intended in any way to replace an owner's manual but to supplement it for the purposes of achieving a higher performance.

OVERHAULING THE SIX CYLINDER ENGINE

SINCE the conditions of the bottom end of an engine can have a major influence on power output, we shall discuss, thoroughly and in detail, how to go about building up the bottom end with a view to extracting the maximum horsepower for a given state of tune.

If the car you are working on is second-hand and has seen a normal mileage, then it is more than likely below par. It is also noteworthy, since we are on the subject, that 99 out of 100 new engines can benefit as far as power is concerned by a careful precision rebuild. In this chapter we shall deal with the 1600, 2000, and 2500 c.c. six-cylinder engines. The 2litre and 2.2-litre engines will be dealt with in the following chapter.

Degreasing

The first move is to take the engine out of the car and degrease it with a suitable cleaning agent. Then, giving yourself plenty of working space, strip the engine down and where necessary, number components for replace-

ment in their original position. Check the cylinder bores for wear. The standard sizes of the bores are as follows:

1600
STD 2,6276-2.6287
2000
2.9405-2.9416
2500
2.9405-2.9416

In the standard bores the pistons are graded in three different sizes and the bore clearance on a new engine will be between 3.5 thou and 4.2 thou (0.00350.0042). If wear has taken place we must determine the amount of clearance which exists before we can say whether or not a re-bore is required. These engines were fitted from the factory with either solid skirt or split skirt pistons. Ideally the bore clearances which we should aim for are 0.005" for the 1600 motor, when solid skirt pistons are used, or 0.0055" when split skirt pistons are used. If the engine is to be used solely for competition work, the bore clearance should be 0.0055" with solid skirt pistons. The use of split skirt pistons

in a competition motor is not to be recommended. Also one should avoid the use of pistons with a ring below the gudgeon pin as this creates extra bore drag. On the 2000 and 2500 c.c. engines we should have a piston to bore clearance of 0.0055" for fast road work and 0.006" for competition purposes. Here again we must avoid the use of split skirt pistons or pistons which have an extra oil control ring below the gudgeon pin.

When determining the bore clearance which exists, be sure to measure the piston diameter at right angles to the gudgeon pin and at a point as shown in Fig. 1. The reason for this is that the pistons are both tapered and oval. Should the bore clearance prove excessive then ideally a rebore is called for. The best pistons to use are those supplied by Tri-

umphs. In the event of the bores having less clearance than required, some honing should be done. The bores, whether rebored or not, must be honed to a 20 micro inch finish with a cross hatch pattern at 45°. Do not attempt to use any other type of finish other than this or it will cost you a drop in power in the long run.

Piston fits

Having prepared the bores one should next ascertain whether the pistons are fit for re-use. The diameter of the skirt will have already been checked in connection with bore preparation. For reference purposes the following chart gives the graded sizes of the pistons.

The ring slot widths should be

Grade	F	G	H
1600	2.6264/2.6267	2.0968/2.6271	2.6272/2.6275
2000	2.9384/2.9388	2.9388/2.9392	2.9392/2.9396
2500	2.9380/2.9384	2.9384/2.9388	2.9388/2.9392

checked for size. If they are too tight we are likely to encounter sticking rings and if too loose, the rate of ring wear will increase and cause gas blow-by.

The ring gap can best be measured by using a ring and a feeler gauge. As new, the normal ring clearance varies for the compression rings from 0.0019" to 0.0035". For our purposes the ideal clearance is 0.002". It may be difficult to achieve this figure without sorting through a number of pistons which can make life a little difficult. Whatever the case may be, the ring gap should not be allowed to exceed 0.0035" which is the top limit of clearance as new from the factory. The oil control ring should be a slightly closer fit in its slot. The normal tolerances for the clearance are 0.0007"

to 0.0027". For our purposes we should try to achieve a clearance of 0.0015".

Before fitting the rings to the pistons, they should be gapped. The 1600 motor should have the gaps on all the rings set at 0.010". The corners of the gap when viewed from above should have a 0.005" - 0.010" radius stoned on to them. The gap on the 2000 and 2500 c.c. engines is best set at 0.012" and again a radius of 5 to 10 thou. should be stoned on to the corners of the gap.

Finally, before fitting any rings, check them for flatness on a surface plate or a piece of plate glass. If there is any detectable sign of warpage reject them. Check the fit of the gudgeon pin in its bore. If there is the slightest evidence of sloppiness

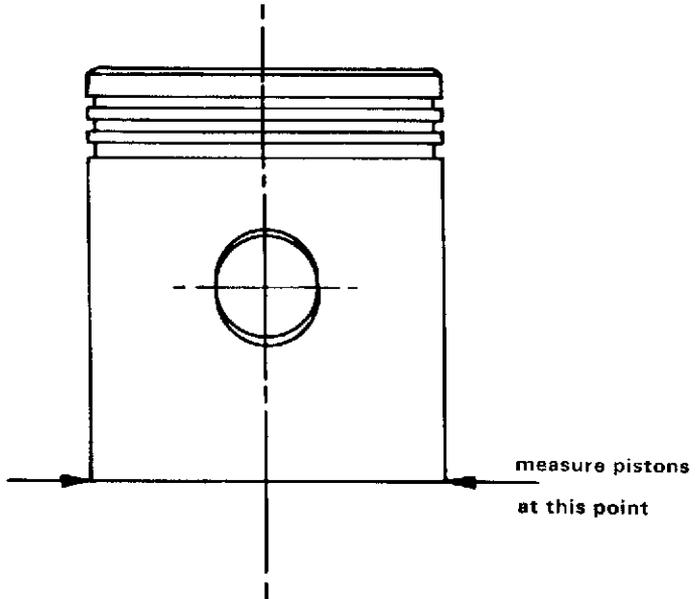


Fig. 1.

then either piston, gudgeon pin or both must be replaced.

Con-rods

We will now focus our attention on the con rods as these items can cause a profound drop in power if they are not true. Since con rods are amongst the most heavily stressed items in an engine, it is a wise move, before using them, to have them crack tested. Once they have passed any crack detection tests, checks for truth should be carried out. First off measure the bore of the big end (without shells). This must be 2.0210'-2.0215' dia. for all engines.

The little end should be a snug slide fit on the gudgeon pin, i.e. the gudgeon pin will not quite fall out under its own weight but can easily be pushed out with thumb pressure. The rods can now be checked for straightness. For our purposes the

maximum error in bend when checked across the gudgeon pin must not exceed 0.0005" and the maximum allowable twist should not be more than 0.001 ".

To make things absolutely clear as to what we are checking, refer to Fig. 2. This shows the effect of bend and twist greatly exaggerated for the sake of clarity. If you do not bother to check the rods, then you will surely pay for it in increased oil consumption and reduced power output. Those of you who have a lathe and a dial indicator, can check the rods yourselves. For those who have not, most motor machine shops possess suitable equipment to do the job. Should any of the rods prove to be out by more than 0.007", then it is a good idea to replace them with new rods. Even if you have had to resort to new rods, do not accept the fact that because they are new

they must be right. Check them to be sure they are right.

Big end bolts can be a critical area on the rod assembly. Repeated torquing up and prolonged use causes them to stretch a small amount. To avoid any unnecessary failures in this area, the bolts should only be used once. It may seem unduly expensive to replace these bolts just because they have been used once, but if a rod lets go, then the chances are it writes off a whole engine.

The cam

Moving on to the cam, we find that the standard journal clearance varies between 4.6 thou. (0.0046) and 2.6 thou. (0.0026). Ideally we want this clearance at about 3 thou. (0.003). Some blocks have cam bearings for the journals to run in and some do not. If you have one with bearings, then it is a lot easier to achieve the 3 thou. clearance since new bearings are usually on the tight side. If you have a block without bearings, these are usually the earlier ones, then bearings should be fitted. If you intend using a hot cam, then it is even more strongly recommended that you use cam bearings.

When installed, the cam must rotate freely. To put a figure on it, the cam should, when lubricated with thin oil, rotate when a torque of 1 oz. ft. is applied. If you have your clearance at 3 thou., and the cam seems stiff, the usual fault is a bent camshaft. Needless to say, it is not the done thing to use a bent camshaft. Inspect the cam lobes for wear. If there is any sign of deterioration, then replace the cam. Early 1600 Vitesses had cam followers having a diameter of 0.687". These were later replaced by cam followers of 0.800' dia. which is the

same size as fitted to the 2000 and 2500 engines. The small diameter cam followers can be a weak point in the engine, especially if a hotter cam is to be fitted. On engines with the small cam followers, it is a straightforward job to machine out the follower bores to accept the larger cam followers.

The 1600 and 2000 engines use a single row timing chain to drive the cam. The 2500 c.c. engine uses a duplex timing chain. For the sake of extra reliability on a tuned engine, one can use the duplex set up of the large engine on either of the smaller engines.

Crankshaft

The crankshaft of an engine is just about the hardest working component. Bearing this in mind, it is only fair that we give the crank every opportunity to show good service. If the crank has already seen a reasonable period of use, it will not take too kindly to all the increased loads brought about by tuning and the use of extra revs. If we hope to achieve a reasonable life from the bottom end of the engine, then we must give it a good chance to start with. This is not to say that the bottom end is in any way unreliable, but if we tune the engine we are expecting the crank to withstand loads in excess of those it was originally designed to cater for. This means that we can accept no half measures. The crank must be in A.1 condition before we can consider its use.

The 1600 and early 2000 engines up to about 1966 use virtually the same cranks. The main bearing size is 2.0005"/2.001 " dia. If the main journals are more than 0.0003' below the bottom limit, or they show

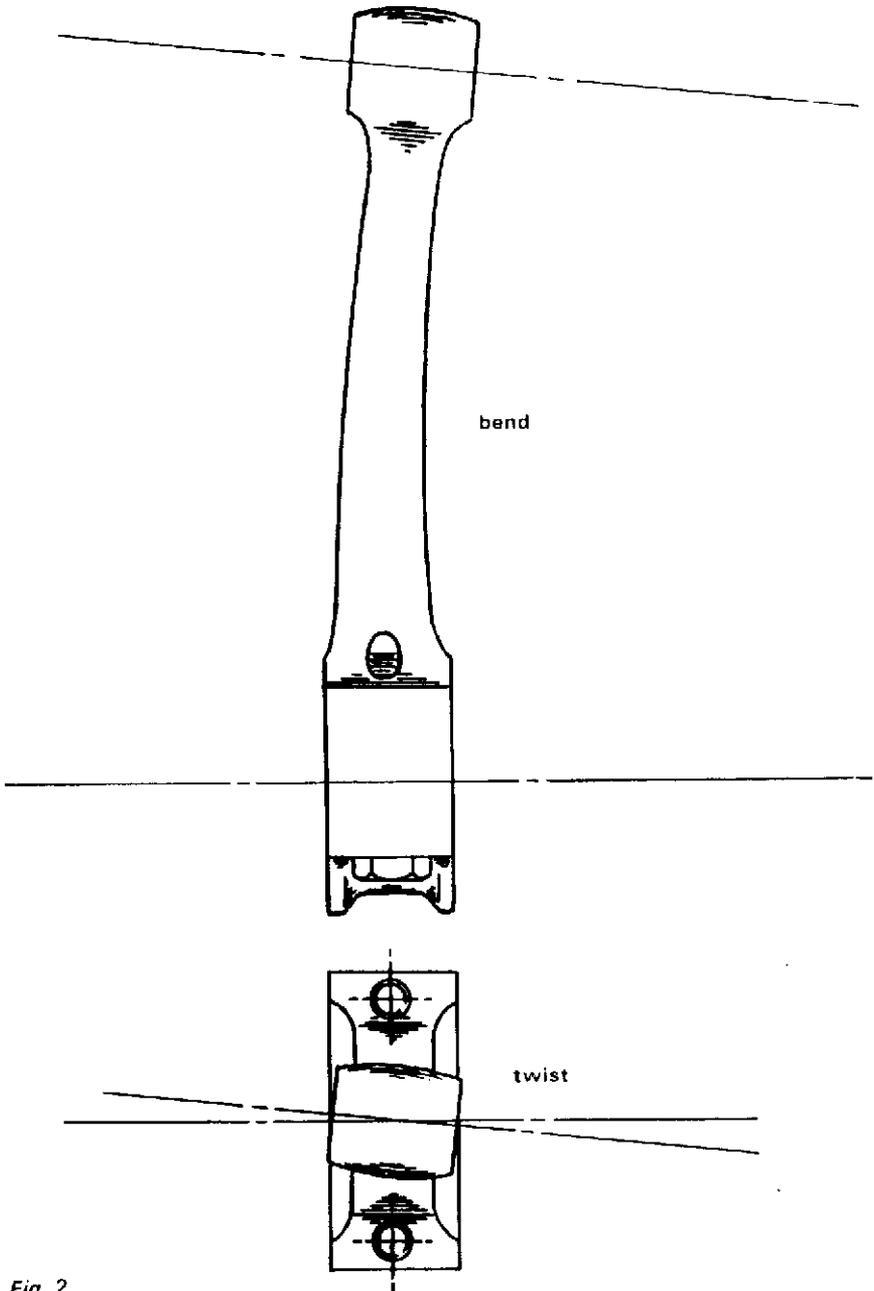


Fig. 2.

any signs of ovality or taper, then the crank should be reground. The crank pin diameter standard is 1.8750"/1.8755". Here again we should treat any discrepancy from the true size by regrounding

The cranks from the 2500 and the later 2000 c.c. engines differ inasmuch as that they are endowed with much larger main bearing journals. These are some 3/16" bigger in diameter than the lesser or earlier engines. The main journals should fall between 2.3110' and 2.3115" diam-

eter, and should the limiting tenth of a thou. below bottom limit exist, then a regrind must be done. If you are selecting a new crank or having one reground, then you should go for crank pin and journal sizes about 0.0002" below top limit. When using a reground crank, we should avoid going below 0.010' undersize if any reasonable degree of tuning is envisaged. The following list gives the ideal sizes of crankpin and journals to use with a standard crank and a 0.010" regrind.

	Big end standard	Mains standard
1600 & 2000 (up to 1966)	1.8753'	2.0008"
2500 & 2000 (after 1966)	1.8753"	2.3113'
	Big ends 0.010' O/S	Mains 0.010 " O/S
1600 & 2000	1.8653'	1.9908'
2500	1.8653'	2.3013'

The above sizes for the crank does make the assumption that the bearing shells that are used are in the middle of their size tolerance. The main bearing clearance with these cranks is between 1.2 thou. and 2 thou. (0.0012 -0.0020-). We should attempt to get an initial clearance with new shells of 1.5 thou. When the shells have run in and everything has settled down, this clearance will have opened up to about 2 thou., which is just where we want it. The big ends should also be treated in a like manner as far as clearances are concerned.

Assuming the block has been fully prepared, we can now install the crank and in doing so make certain functional checks. Fit the bearing shells into the main bearing caps and block, place the crank in situ and oil the main journals. Fit the caps and torque down the bolts to 50-55 lb. ft. for 1600 and 55-60 lb. ft. for the 2000 and 2500 engines. Check that the crank spins freely i.e. it should rotate when a

torque not exceeding 1/3 lb. ft. is applied. If the torque required to turn the crank is appreciably above this figure, then you have either a tight bearing or bearings, or a distorted crankshaft. You can determine whether the culprit is a tight bearing or not by loosening off each bearing cap in turn to see if this frees the crank. In nine cases out of ten, tight bearings can be cured by swapping the shells from one housing to another. In the odd one case out of ten, you will have to get some more shells. If you want to number your bearing clearance exactly, then I would suggest the use of plastigauge clearance gauges. These gauges allow quite an accurate measurement of the clearances existing to be obtained. If each individual bearing clearance is correct, yet the crank is still stiff, then this points towards a distorted crank. There is also a very small chance that the block could be distorted which would also cause the crank to be stiff. If the crank you are using has been reground or is a new



Geoff Shepherd and M. G. Hofman dirty their Ex-Works Group 5 Triumph 2000 during the Hafren Forest stage of the 1968 Welsh Rally.

one, the chances of the crank being out of true are very small but nevertheless should not be ignored. The most likely case of a distorted crank will arise with those which have seen use but not enough to cause them to wear outside of the tolerances required. This means a regrind will be required to rectify the inaccuracy.

Block distortion

Lastly we can on very rare occasions, get block distortion. When blocks are cast they have stresses set up in them due to differing rates of cooling when casting. Over a period this causes the casting to move i.e. small changes of shape occur. This means that if the block was machined a short period after it had been cast, then over a period of a few months it can distort itself by a number of thou. whilst the casting stresses settle down. There are two ways to overcome the effects of internal stresses in castings. The first method is to leave the castings to age, then rough machine them and then give them a further ageing period. After this period, which can be as much as a number of months, the final machining is carried out. The second method is to heat the blocks

in a furnace after they have been cast to allow the stresses to quickly settle out. Just once in a while it is possible to find you are the proud owner of a block which has not been properly stress relieved. Unfortunately this does not show up immediately but usually makes itself felt some twelve months after, or the first time the engine is completely stripped. In the unlikely event that you have a distorted block, the only remedy is to replace it.

We have now carefully prepared all the major components connected with the bottom end. Things like oil seals, timing chains, etc. should be replaced as a matter of course. The crank, rods, pistons, flywheel, clutch and crank pulley should now be balanced. This will not only make an already smooth engine smoother, but give that little extra reliability needed for high performance motoring.

Valve guides

The only other points connected with the mechanical well being of the engine are associated with the cylinder head. The wear on valve stems and guides must be checked. The valve stem sizes for the three engines are as follows:

	Inlet	Exhaust
2000 & 2500	0.3107'-0.3112' dia.	0.3100 "-0.3105' dia.
1600	0.3100'-0. 3110 dia.	0.3090"-0.3080" dia.

The bore of the inlet guides are common to all engines, this being 0.312"-0.313" dia. As you can see the clearance on the larger two engines can, on the inlet, vary between 0.0023' and 0.0008'. The clearance to achieve is between 0.001 " and 0.0012'. On the exhaust side the clearance can vary between 0.0033" and 0.0015'. For our purposes 0.002" clearance is needed. On the 1600 engine the standard clearance on the inlet valve can

be between 0.001' and 0.003". Here we need 1 to 1.2 thou. clearance (.001" 0.0012"). The standard clearance on the exhaust valve on 1600 engines is by normal standards large, this being between 3 and 5 thou. (0.003"-0.005"). One should endeavour, where possible, to select components which will set this clearance at the lower limit of 0.003".

You may wonder why the valve to

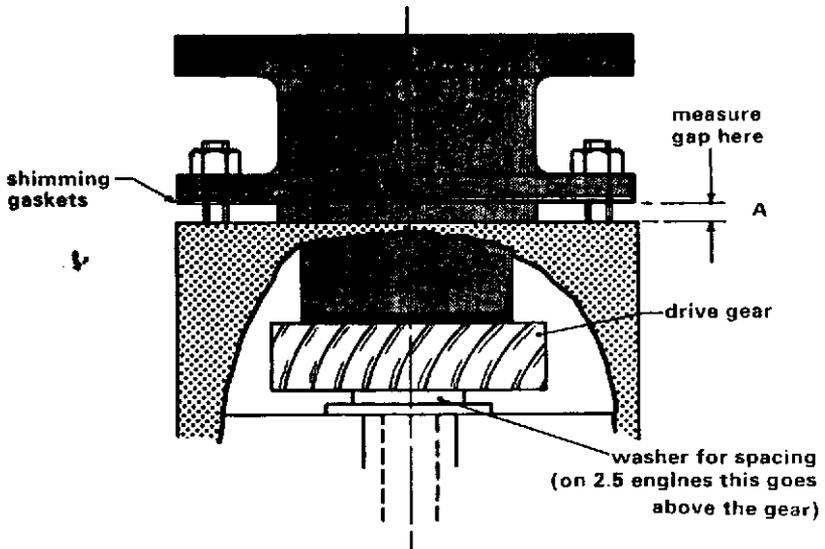


Fig. 3. Measure gap A and adjust with shimming gaskets until it is equal to the washer thickness + 0.003 in. (the clearance). Having selected the correct shimming remove the washer and assemble parts as usual.

valve guide clearance has been gone into in such detail. The main reason is that the motion of the valve must be controlled in such a manner as to allow the valve to move in a straight line only. The more clearance one has the less likelihood there is of achieving the required motion. Excess clearance will cause an increase in oil consumption, a small drop in power and an increased valve seat wear rate. The last point is especially relevant when we consider the use of radius valve seats which will be thoroughly discussed later in the book.

Before winding up our dimensional check of various engine components, we should check out the rockers and rocker shafts for wear. Normal clearance is between 1.8 thou. and 3.3 thou. (0.0018" - 0.0033") If the clearance is more than 0.005" above the upper limit, steps should be taken to remedy it. Rockers which

are too loose can set up a mode of vibration which does nothing to enhance valve timing at high revs., especially if the cam is of a hotter variety.

Assembly

We should now come to the time when we are able to put the engine together. Assemble the crank, rods and pistons into the block using engine oil on bearing and rubbing surfaces. Having assembled the engine, we now come to the test which will verify your dimensional checking. Rotate the crank assembly about fifty times. Then using a torque wrench, establish the torque required to turn the engine over. The absolute maximum torque required to turn the engines over are 20 lb. ft. for the 2500, 16 lb. ft. for the 2000, and 12 1/2 lb. ft. for the

1600. If the build up has been done carefully, these torque figures can be as much as 30% lower. If the figure is significantly higher, then something is amiss, so you should go through what you have done to locate the cause of stiffness. Excess internal friction in an engine is a sure way to lose power. 5 lb. ft. of frictional torque over and above our minimum will cost us 6.7 b.h.p. at 7000 revs. As you can see, the moral is to build up an engine which is as free as possible whilst still failing in with our component dimensional limits.

With the exception of a couple of points, the engine assembly from here is straightforward. The couple of points which we must bear in mind are connected with the accuracy and consistency of our ignition timing. The distributor is driven in a conventional manner by a skew gear coming off the camshaft. As with any gear train, backlash is necessary but on these particular engines we have a measure of control over the amount of backlash present. The total backlash which

can be present at the distributor drive shaft is not only the backlash of the skew gears but also an amount caused by the end float of the camshaft. Our camshaft end float is controlled by a thrust plate at the timing chain end of the engine. The normal cam end float is between 0.004' and 0.008". By selecting a suitable plate we must get the endfloat down to 0.004'. We can, if we are going to be ultra fussy about it, reduce this to 0.003' but we must definitely not go under this figure.

The other area which must be closely looked at is the end float of the distributor drive shaft itself. The standard clearance figures range between 3 and 7 thou. and the figure we want is 3 thou. on the nail. Fig. 3 shows the precise method to adopt for determining this clearance and setting it to the required figure. Assuming we have a perfect distributor, any erratic timing caused by excess backlash can cost around 2-3 b.h.p., so play close attention to these end float settings.

IMPROVING THE LUBRICATION

WITH any tuned or hard pressed engine, the lubrication system must be in perfect order or better still, uprated to cope with more stringent conditions.

The standard oil pump is capable of coping with the extra demands placed upon it by a modified engine if it is in perfect order. However, to ensure a good engine life, a little time and trouble spent on the standard pump will not go amiss. The work involved on the pump includes a thorough check on clearance existing and rectifying these where necessary. Fig. 4 shows an exploded view of the pump assembly just so that we know what bits we are talking about. The main object of the exercise is to build up a pump assembly having the minimum clearances between the body and the outer rotor and between the outer rotor and inner rotor. A normal value for these clearances is around 0.007'. By selecting components, we can quite easily get this figure down to 0.004' and if a great enough number of components are available this can be reduced further still. These clearance values can easily be

checked with feeler gauges as shown in Fig. 5 and 6. The only clearance that we can readily make adjustments to is the end float between the inner and outer rotor and the end plate. By machining a small amount off the body we can set the end float to 0.001" thus reducing any leakage at this point to a minimum. Fig. 7 shows how the end float clearance can be measured so as to enable the correct amount of metal to be removed.

When machining the pump body, be very careful as there is very rarely more than 0.003' to come off. Whilst machining the body, you must also make sure that the inner and outer rotor are of identical length or you will have a different end float for each component. Once all the pump components have been selected or adjusted as necessary, the pump can be assembled. Using very light oil to provide a little lubrication, check that the pump rotates smoothly and freely. Having gone to all this trouble to prepare the pump, we can expect it to be about 7-10% better than the average pump. This may not sound like a

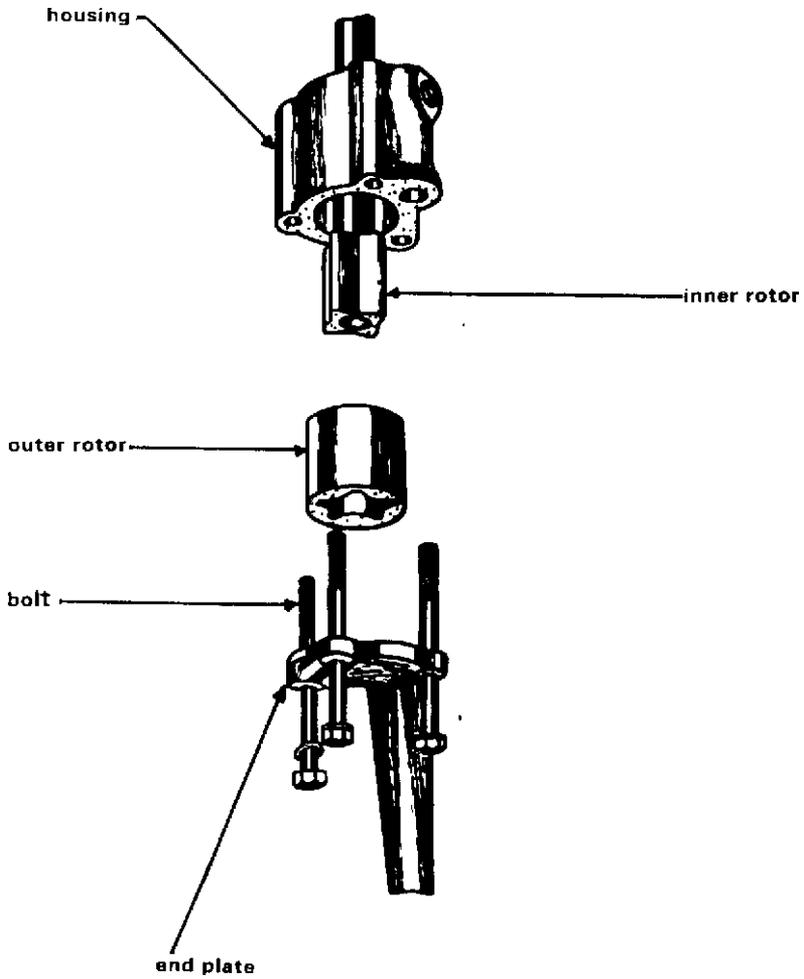


Fig. 4. Oil pump.

Fig. 5.

*Measuring clearance
between inner and outer rotors.*

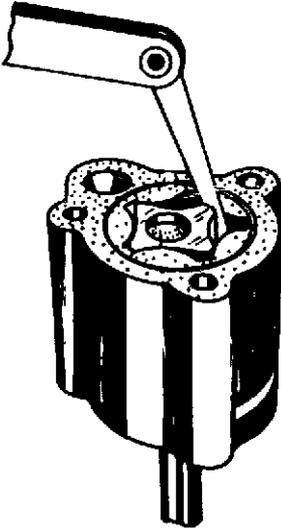


Fig. 6.

*Measuring clearance between
outer rotor and body.*

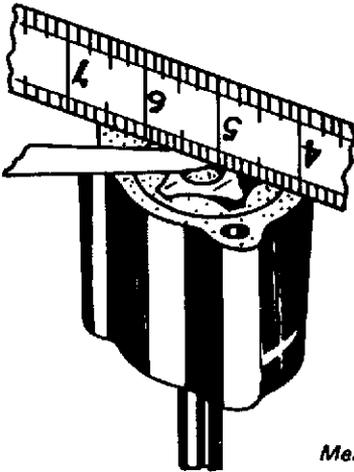
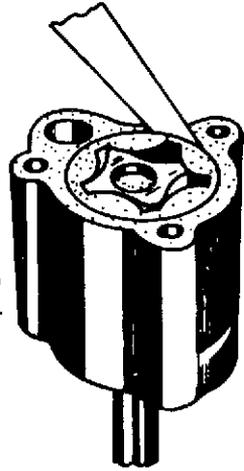


Fig. 7.

Measuring rotor end-float.

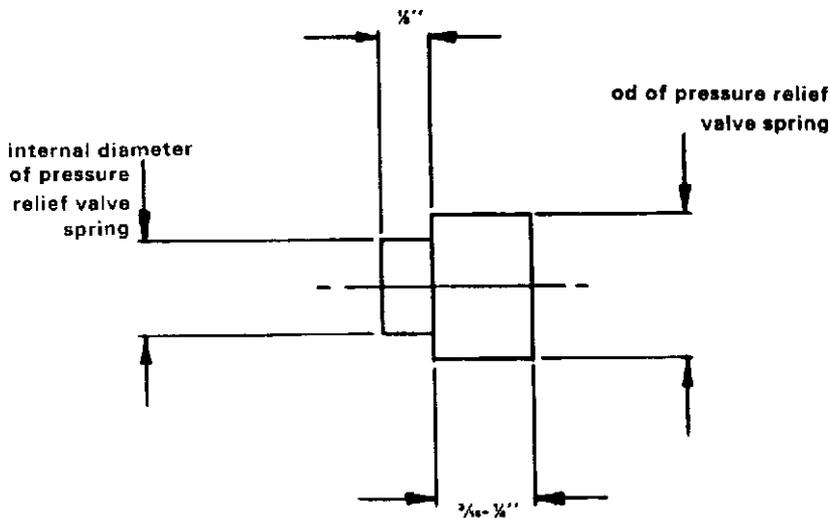


Fig. 8.

fantastic gain for all the time and trouble resorted to, but we must remember it is always the last straw that breaks the camel's back. The increase is sufficient to give us that extra margin of safety between retaining or losing the vital oil film between the highly stressed components.

Pressure release valve

Still on the subject of lubrication, suitable adjustments should be made to raise the pressure at which the pressure release valve comes into operation. This can be achieved quite

simply by increasing the spring pressure which is exerted on the pressure release valve. A spacer made up as shown in Fig. 8 and fitted under the face of the release valve spring retaining bolt will do the trick. The spacer will cause a raise of 10-12 lbs. per square inch of oil pressure before the pressure release valve comes into operation. It will be found unnecessary to so such a mod to the spring pressure unless the engine spends a great deal of its time in the upper revolution range as would an engine used in competition.

BOTTOM END PREPARATION

THE 2 and 2.2 litre four cylinder Triumph engines are well noted for their long life and ruggedness. If the engine is carefully assembled, then even in a relatively highly tuned state, reliability does not suffer to any noticeable degree providing one pays strict attention to the rev limit. We should apply the same thoroughness and attention to detail as was outlined in Chapter 1 on the six cylinder engines. To avoid unnecessary repetition of the whys and wherefores, the tolerances that we should work to will just be quoted with only limited explanation as to why.

Assuming the engine has been removed from the car and has been completely stripped, we should be in a position to carry out our dimensional checking. As good a point as any to start with is the block. A close look at the block will reveal that, unlike most engines which have pistons running directly in the block casting, the Triumph engine has removable wet liners. The definition of a wet liner is one that is directly in contact with the water, as opposed to one that is in contact solely with the cast iron of the

block. Anyway, to get back to the point, we should check the bore for wear. Since this particular engine has been out of production for some time, I should doubt if any engines which are still on standard size pistons are not in need of a rebore. The standard size of bores for the 2-litre engine and the standard piston to bore clearances are as follows:

	Standard bore	Standard clearance
Std.	3.2677"	0.0042'
+ .020"	3.2877"	"
+ .030"	3.2977"	"
+ .040"	3.3077"	"
+ .060"	3.3277"	"

For fast road work the bore clearance should be 0.005' and for racing 0.006'. When determining the piston/bore clearance, the piston should be measured level with the gudgeon pin but at 90° to it. If the piston to bore clearance exceeds 0.008", then a rebore will be required if we are to achieve optimum power. With the 2.2 litre engine the

standard bore and rebore sizes will be:-

	Standard bore	Standard clearance
Std.	3.3854"	0.0043"
+ .020"	3.4054"	"
+ .030"	3.4154"	"
+ .040"	3.4254"	"
+ .060"	3.4454"	"

For road work we want a 5.3 thou. clearance, and for racing a 6.4 thou. clearance. The bore sizes already quoted are those which give a standard clearance. Since it is easier to obtain extra clearance by making the bores slightly larger, than it is to reduce the sizes of the pistons the above bore sizes will want enlarging on. To obtain the correct clearance using new pistons, the following chart should be adhered to :

2-litre engine	Road	Racing
Std.	3.2685"	3.2695"
+ .020"	3.2885"	3.2895"
+ .030"	3.2985"	3.2995"
+ .040"	3.3085"	3.3095"
+ .060"	3.3285"	3.3295"

2.2-litre engine	Road	Racing
Std.	3.3864"	3.3875"
+ .020"	3.4064"	3.4075"
+ .030"	3.4164"	3.4175"
+ .040"	3.4264"	3.4275"
+ .060"	3.4464"	3.4475"

The bores should be finished by honing to 15-20 micro inch finish with a cross hatch pattern of 45".

As has been stated before, this particular engine has removable liners which leads us to some interesting possibilities as far as increasing its capacity is concerned. The liners from the 2.2 litre engine can be fitted to the 2litre engine, thus bringing it up to the same size as its bigger brother. Also there is a 9.25/1 piston made by Hepolite which will give a further slight increase in capacity. This piston has

a standard size of 3.4252" (87 m/m) and is available at rebore sizes of +030" and +040". The 2.2 litre liners can be bored to take this piston but the 2-litre ones cannot. If we use this piston (Hepolite part number 13958) in its +040" form we will bring the capacity up to 2239 c.c. from the original 2138 c.c. If this piston is used in its standard 87 m/m form rather than in oversize form, the capacity is 2187 c.c. For the purposes of building up a high performance road or race engine the use of this piston is to be recommended. Its advantages lie in the fact that it is quite light in weight, has no ring below the gudgeon pin and is of the solid skirt variety. The boring sizes for this piston are:

	Road	Racing
Std.	3.4262"	3.4273"
+ .030"	3.4562"	3.4573"
+ .040"	3.4662"	3.4673"

If the Hepolite piston just mentioned is not used, then it is advisable to use pistons supplied by Triumphs. There are some pistons on the market which have an extra oil ring below the gudgeon pin. The use of these is to be avoided as are pistons having split skirts.

Liners

If by some chance, your engine has already been bored to its maximum size and is yet again in need of a rebore, then it will be found necessary to fit new liners. This in itself is a simple enough job, but one which must be done properly if we are to avoid any possible leakage from joints.

First off, one should remove the old liners. It is definitely not advisable to try bashing out the liners solely with the aid of a hammer. To do the job properly without incurring any nasty side effects, one

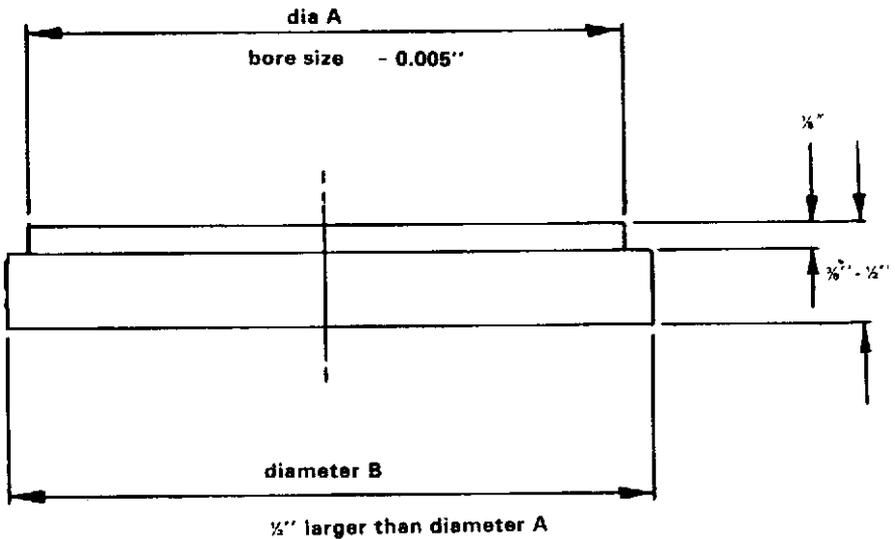


Fig. 9.

should make up a special tool to distribute the hammer blow evenly over the end of the liner. The dimensions for such a tool are shown in Fig. 9. Place the block, head face down, on some blocks of wood. Place the liner removing tool into the bore of the liner the crank end of the liner. Place a suitable stout punch i.e. 1" dia. steel bar on top of the removing tool and hit the punch with a heavy hammer. In most cases the liner will come out without undue trouble. Occasionally an obstinate liner is encountered, but the application of a little penetrating oil will ease the situation. Having removed all the liners, thoroughly clean the area on which the bottom jointing gasket seats see Fig. 10. This area must be perfectly free from foreign matter which may prevent a perfect seal being formed. Smear the new liner sealing gaskets with a non-set-

ting jointing compound. Fit the new liners in position and as soon as possible afterwards, fit the head together with an old head gasket and torque the assembly down to about 6070 lb. ft. Fitting the head effectively pulls down the liners, seating them firmly onto the gaskets. The head can now be removed so that we can check the liner protrusion above the block face. Place a straight edge across the top of each liner, and using a feeler gauge, check the gap between the straight edge and the block face see Fig. 11. This clearance should be between 0.002"0.005". In all probability the protrusion of each liner will vary between these limits. If the protruding height of each liner is the same, then you have just been lucky. In the greater likelihood that each liner height varies one to another, a little machining is called

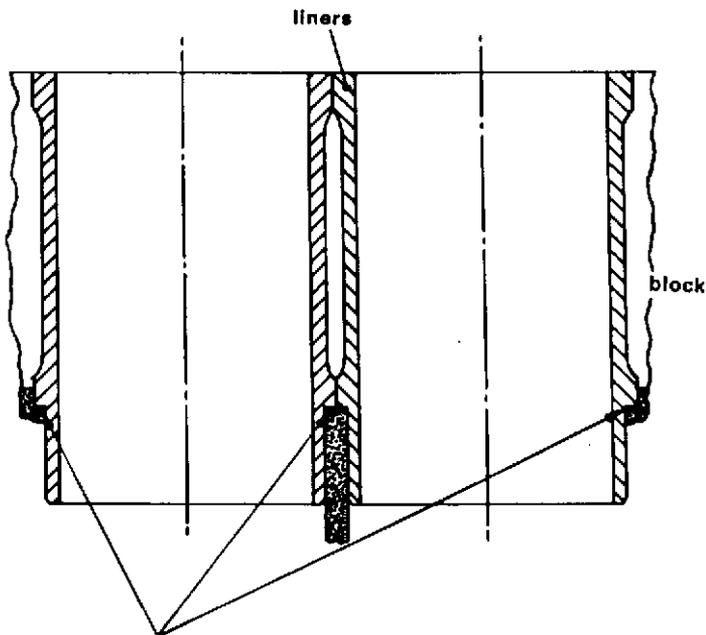


Fig. 10.

gasket area upon which liner seats must be clean to eliminate leaks

for. The top faces of the liners should be skimmed to bring them all to the same height above the block face. This height, however, must still remain within the 0.002" - 0.005" tolerance. If you should find that the protrusion height is less than 0.002", then the liners will have to be removed and a few thou. taken off the top of the block. Having done this, refit the liners as described previously, and machine the tops of the liners to bring them all to a common height, within the specified tolerance.

Rings

Having, we hope, successfully prepared the bores, we can turn our attention to the piston rings. As usual, for high performance engines, we should check the rings for flatness and for ring to groove clearance. The factory limit for ring to groove clear-

ance is 0.001 "-0.003" We definitely should not allow the clearance to exceed these figures and ideally if a selection of pistons is available, this clearance should be at 0.0015". The ring gaps for all rings should be set to 0.012"-0.015" and the corners of the gap stoned off to a 0.005"0.010" radius.

The tappets or cam followers must be a free fit in their bores but on the other hand should not be sloppy. The maximum tappet to bore clearance we can tolerate is 0.0016". Since most of the wear seems to occur on the cam followers rather than the bore, any wear problems can usually be overcome by fitting new cam followers. If a new standard cam or a high performance cam is to be fitted, the new cam followers are essential no matter what condition the original ones are in. If this is not done you have a 90% chance

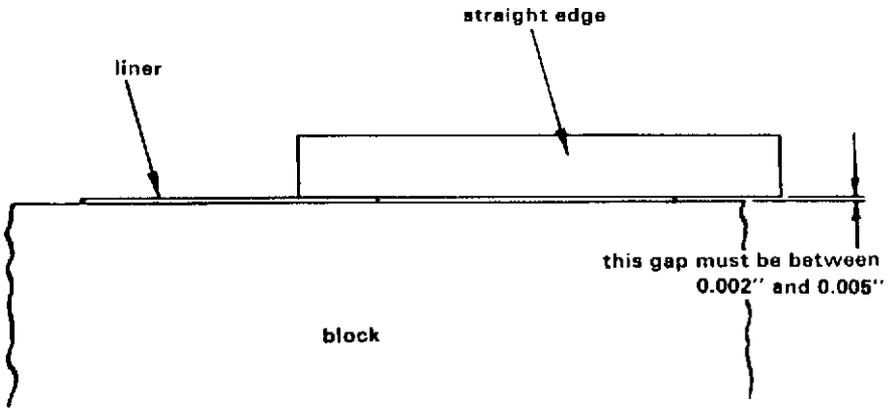


Fig. 11.

of having a cam life of less than a few thousand miles. Whilst we are on the subject of cams, the camshaft must, when fitted into the block, rotate freely and smoothly. A bearing clearance of 0.002" being ideal and 0.0037" clearance is the maximum allowable clearance for all the bearings bar the front one. The clearance on the front bearing is best around 0.003" and the maximum that is permissible is 0.005".

To avoid undue power loss, the rods must be true within fine limits. The maximum bend when checked over the length of the gudgeon pin should not exceed half a thou. (0.0005") and twist should not be greater than 0.001". Chapter 10 will clarify the conditions we are ideally seeking to minimise.

The little end of the rod must be a snug fit on the gudgeon pin. This usually means a clearance of 0.0002"/0.0004". Although these engines are not prone to rod break ages, it is a good idea to play safe and have them crack tested and as another precautionary measure, fit new big end bolts each time the big ends dismantled.

As has been said before, we are dealing with an engine which has been out of production for some time. It is therefore reasonable to assume that in most cases the majority of the useful crankshaft life has been used. This will mean that nine times out of ten a crank regrind is called for. The limits of crank journal sizes are shown below at standard size and 0.010" and 0.020" undersize.

	Big ends	Mains
Std.	2.0861 "-2.0866 "	2.4790"-2.4795"
0.010,	2.0761 "-2.0766 "	2.4690"-2.4695"
0.020'	2.0661 "-2.0666"	2.4590"-2.4595"

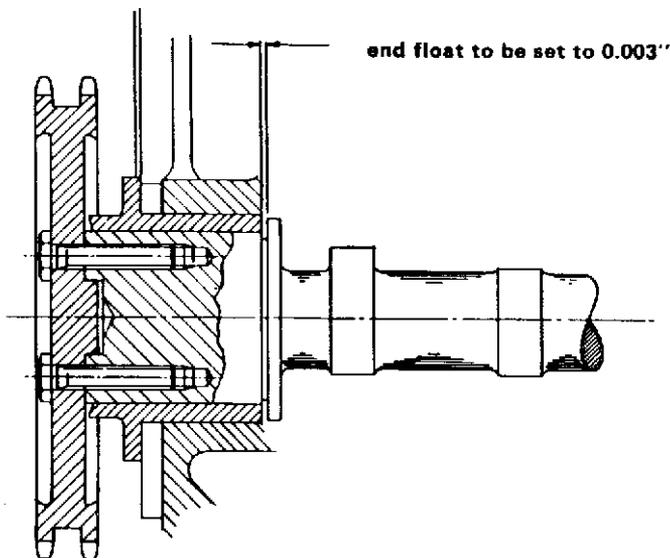


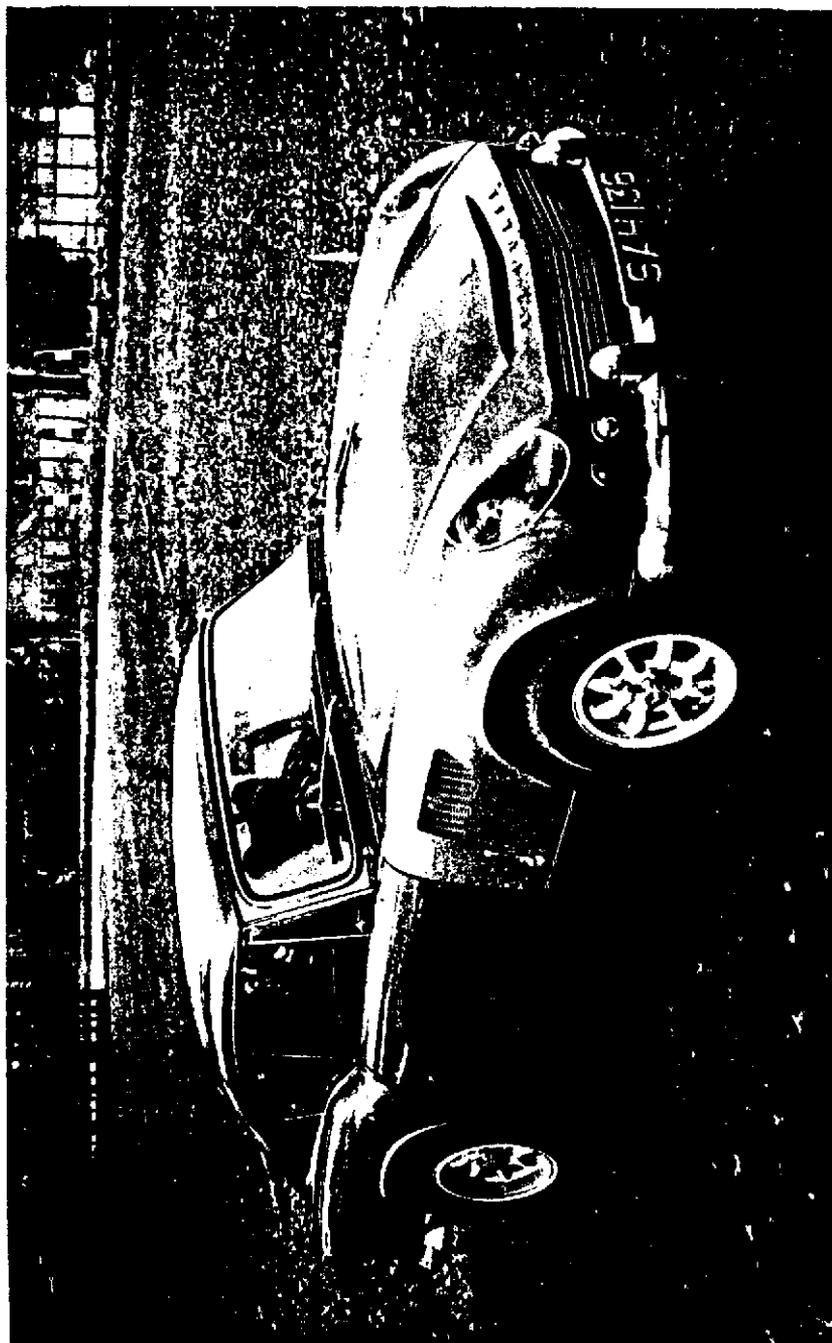
Fig. 12.

For a road going engine, a crank regrind to $-0.040''$ is passable. On the other hand, if the engine is to be used for serious competition, the $0.020''$ regrind should be regarded as maximum. It is always difficult to know where to draw the line as far as crank regrinding goes. Each step of $0.010''$ does not reduce the crank strength by a vast margin, but going from standard to $0.060''$ does. Since the crank in standard form is only safe to 6500 continuous or 7000 instantaneous, we must conserve all the crank strength possible. These engines will seemingly last for ever at 6000 r.p.m. but at 7000 the life can be measured in minutes.

Assembly

We should now be in a position to assemble the bottom half of the engine. The first move is to fit the crank and check for stiffness. The methods

of pinpointing the cause of stiffness and its remedies were outlined in Chapter 1 concerning the six cylinder engines. The fitting of pistons and connecting rods follows normal procedure. We can, at this stage, assess the turning torque of the crank/rods and piston assembly. If all is well then the assembly should rotate when a torque not exceeding 24 lb. ft. for a road engine and 20 lb. ft. for a race engine is applied. If the torque figure is significantly higher than this, then something is amiss, in which case, if you are wise, you will strip the engine to find out why. The difference between the turning torque for a road engine and a race engine, arises from the different bore clearances. Once the engine has run in, this torque figure will drop and the difference between a road and race engine will drop from 4 lb. ft. to about 2.5-3 lb. ft. This difference may sound like a trivial amount to worry



The glassfibre 'Le Mans' hood/bonnet replacement for the GT6 from SAH. Miniilites complete the lightening job.

about, but it is worth about 4 b.h.p. and 6500 revs. and with a race engine every little bit counts.

As with the six cylinder engines, we must pay close attention to the camshaft end float, and the end float of the distributor drive gear. The front cam bearing is removable and has on it a flange which controls the depth to which it can go into the block see Fig. 12. The normal cam end float is 3 to 7.5 thou. (0.003" to 0.0075"). Since we want the most accurate ignition timing possible and since cam end float effects the ignition timing, we must reduce the cam end float to a minimum. The minimum is at 2 1/2. to 3 thou. (0.0025"-0.003"). In virtually every case the existing end float will be more than we want. To reduce the end float down to the figure we need, some machining will be called for on the abutment face of the flange

on the front cam bearing. This face will need skimming on a lathe to give the end float required.

This leaves us with only the distributor drive gear end float to contend with. On a standard engine this can vary between 0.003" and 0.007". For our purposes, the 0.003' setting is required. The technique for setting this is virtually the same as for the six cylinder engines and is shown in Fig. of Chapter 11. Building up the remainder of the bottom half of the engine is a straightforward procedure. All gaskets and oil seals should be replaced during the rebuild. The timing chain should also be renewed. If all the stated tolerances have been adhered to, and a great deal of intelligent care applied, the end result will be well worthwhile as far as power and reliability are concerned.

MODIFYING THE EARLY 6 CYLINDER HEAD

THE 1600 Vitesse Mk. 1, 2-litre Vitesse Mk_1, G.T.6 and early 2000 saloons use a head which is basically common. The Mk. 11 Vitesse and G.T.6, TR5, TR6, and the 2.5 P.I. saloon use a different head. The heads, as far as fitting is concerned, are interchangeable between the early and late models. The major difference between the two basic types of head lies in the chamber design and the manifold joint face. If any head change from early to late models is contemplated, then the manifolds must be changed to suit. The later heads also sport a larger inlet valve which assists breathing. The larger valve, together with the redesigned chambers gives an increase in power of about 7%. Since cylinder heads are so expensive, one should only consider changing the head if the ultimate in power is required because the early heads can be modified to give results very nearly as good. In this chapter we will deal specifically with the early type of head. It is quite simple to determine which type of head you have by comparing your own head with the head drawing in this book. The early

heads have the manifold joint face terminating about 1 " from the edge of the casting when viewed end on. The later heads, however, have the manifold joint face terminating flush with the edge of the head casting.

We will start off with a Stage 1 type of head for the 1600 and Mk. 1, 2-litre Vitesses, Triumph 2000, early type, and the Mk. 1 G.T.6.

Removing guides

Strip the head of all parts so that you are left with the bare head casting. I personally do not like the use of a hammer to remove valve guides. If you are using a heavy hammer to remove the guides you only have to miss the guide removing drift or catch it at a slight angle to do the head an injury. By far the best way to remove guides is by using a press. Of course, if you find that none of your local motor machinists or garages have one, then the only alternative left to you is the large hammer. Either way, press or hammer, a guide removing drift will be required, the relevant dimensions of which are given in Fig. 13.

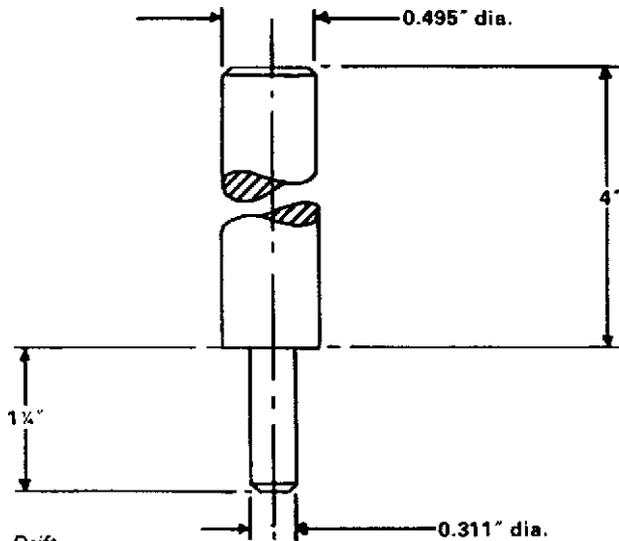
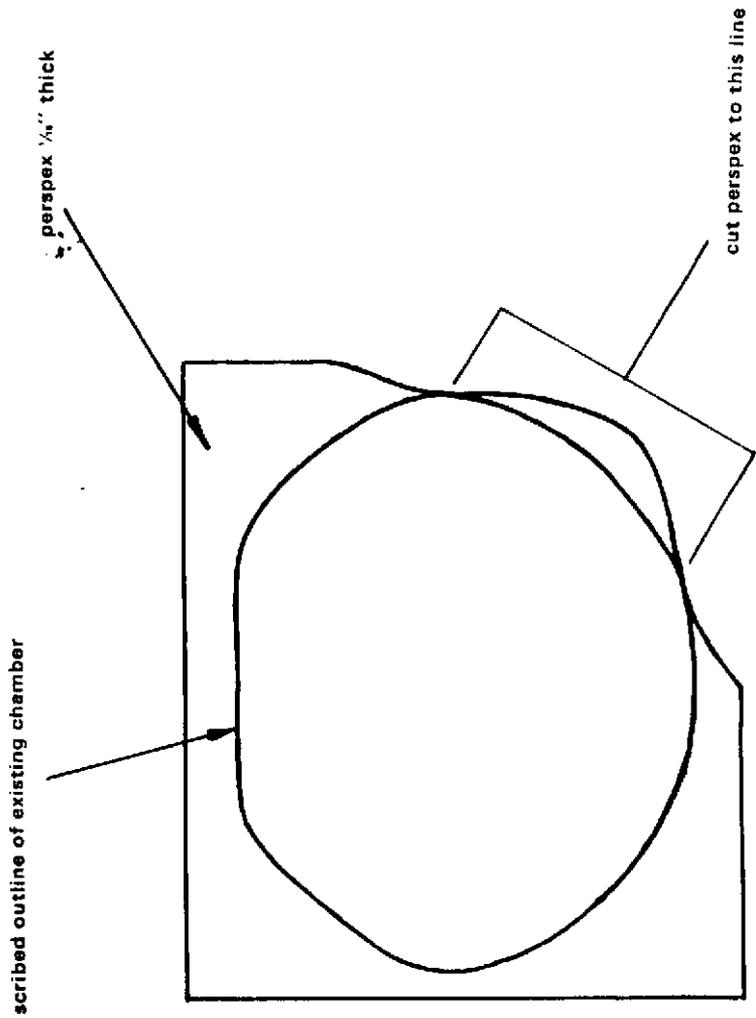


Fig. 13. Drift.

Thoroughly clean the head face using emery cloth as required, so as to achieve a surface suitable for marking on. Apply marking blue to the head face. Marking blue, by the way, is available at any hardware store dealing in engineers' tools. While you are at it, you will also need a tin of engineers' blue. For those of you not familiar with either of these compounds, marking blue is a marking agent which dries very quickly, in fact, in only a number of seconds. Engineers' blue on the other hand is a marking agent which virtually never dries out and is used to detect and show up points of contact, i.e. high spots between mating surfaces. We find the engineers' blue virtually indispensable later when lapping in valves.

At this point we will need to make up a template to accurately mark out our new chamber shape. In this particular instance, the best bet

is to make one from perspex of 1/16" thickness. The drawing of the chamber Fig. 15 is to scale so the chamber shape can be traced on to the perspex from the drawing. When tracing this shape, do so as accurately as possible, as there is very little room for error. A badly traced template and a little erroneous grinding later can cause a scrapped head. Trace out both the original shape and the new shape on the perspex and if possible, trace out the shape on both sides of the perspex to reduce to a minimum any effects of parallax due to the thickness of the material. Inspection of Fig. 15 will show that the chamber is only modified over part of its perimeter. If we now cut the perspex only along the reprofiled part of the chamber shape Fig. 14, then we can use the rest of the chamber outline, which remains unchanged, to line up our template. The chambers in



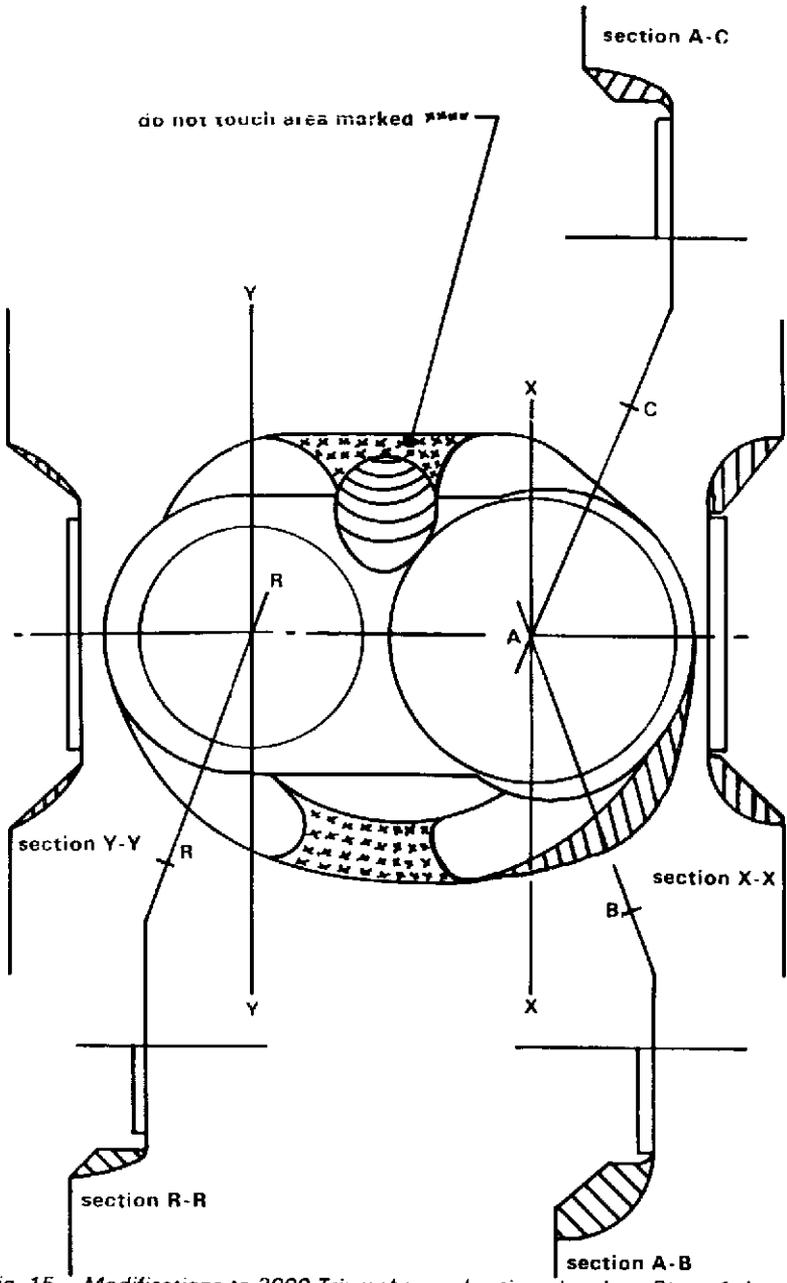


Fig. 15. Modifications to 2000 Triumphs combustion chambers Stage 1. It also applies to Mk. 1 GT6 and 2 litre Vitesses.

the head are "handed." To mark out an opposite handed chamber we merely turn the template over onto its other face. Once you have conscientiously made your template, the head marking out procedure can begin.

To do so you will need a very sharp scribe, and also one which has a long tapered point with, say, an included angle of about 30° , not a short stubby point. The only reason for this is that a short stubby point will inevitably produce a line which does not coincide with the edge of the template but is a short distance out from it. As I have mentioned previously, we have virtually no room for error on the chamber shape, or we shall infringe, if that is the word, into the domain of the gasket.

We can at last start grinding, but firstly some more words of caution. If you are the proud possessor of a steady hand you will find that on this head you have enough room to be able to avoid serious contact between the grinding wheel and valve seat. If this is the first time you have attempted any serious head work or you are not at all sure that you have a steady enough hand, then it is best to take a few precautionary measures. First off, take a couple of scrap valves, one inlet, one exhaust, and machine them across the head face sufficiently such that when they are fitted in position the valve head face is flush with the chamber roof. Next, tap the valve guides a short distance into the head, just sufficiently to locate them. By inserting the valves into the chamber we are working on, we have effectively covered the seats and protected them from accidental damage.

If you look closely at Fig. 15 showing the chamber modifications, you will see that although the chamber

outline over its greater part remains unaltered, the walls of the chambers require fairly extensive work to be done on them. Basically what we are trying to achieve is a chamber wall having no sudden changes of form or any corners formed between angled and vertical sections of the chamber. The area of the chamber wall around the spark plug, should not be touched except for polishing. Although a line is used on the drawing to depict the extremities of this area, a sharp edge should not be formed at this line. Machining the walls of these chambers requires a good eye to blend the changing curved section into the straight walls. The crossed sections should therefore be regarded as a reasonable guide as to where the wall section remains standard.

When all the grinding has been completed, the chambers should be polished to give a smooth finish. It is quite unnecessary to polish until you have a mirror finish as all the work in doing so will be completely wasted. The type of finish required is one that is smooth to the touch a good comparison for this would be a piece of china ware.

This now leaves us with the ports to tackle, and we will deal with the inlet ports first. At this stage one must decide whether or not radius valve seats are to be used. Let me warn you, it is a long and tedious task doing radius valve seats unless you are the proud possessor of some sophisticated valve seat machining gear. Doing radius valve seats by more menial methods can take 12-15 hours work, and sometimes longer. The end result of such labour is 2-4 b.h.p. more, depending on the state of tune of the engine, than conventional seats.

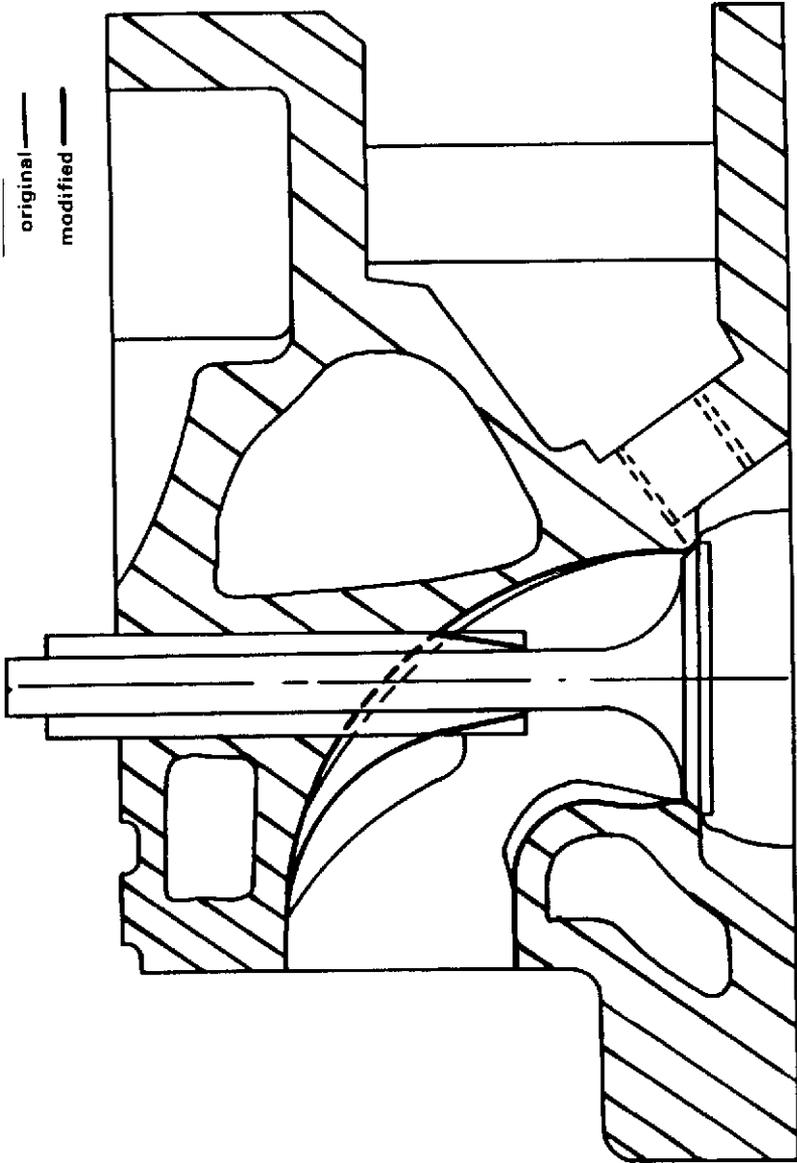


Fig. 16. Modifications to the inlet ports for Triumph 2000, Mk. 1 Vitesse and GT6s.

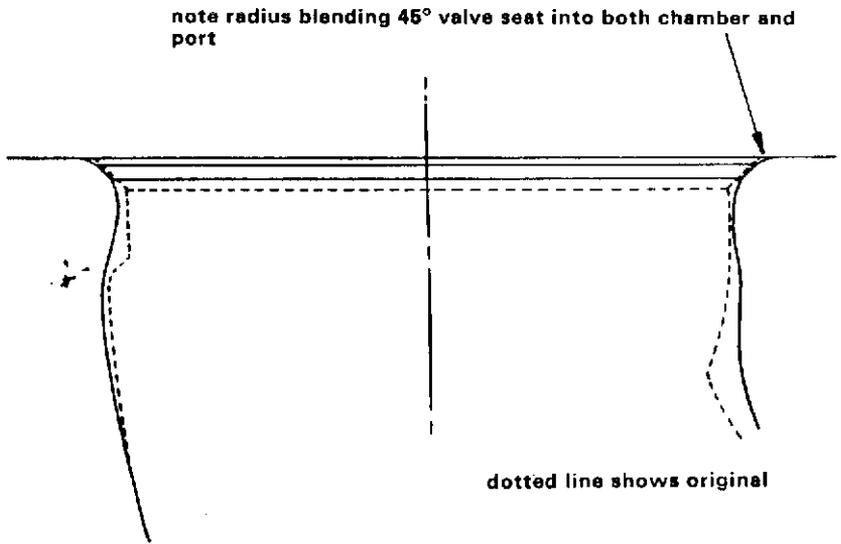


Fig. 17.

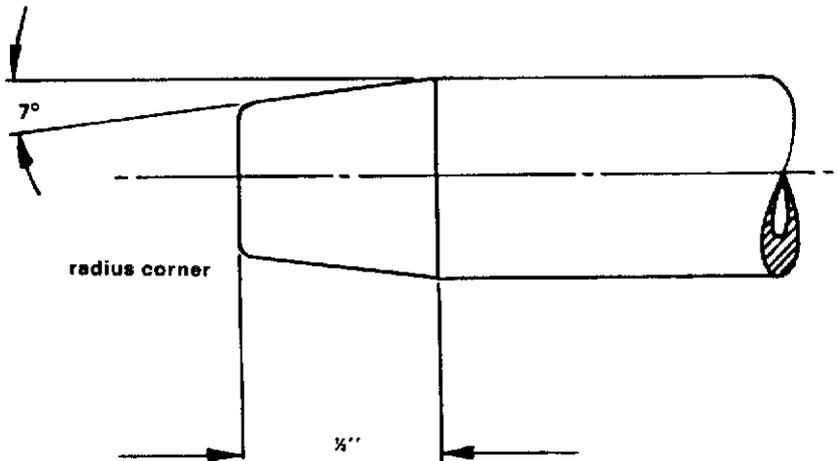


Fig. 18.

We will assume at this point that conventional seats are going to be used, and deal with the radius seats later.

Take a good look at Fig. 16 which depicts a cross-section of the head through an inlet port. A fairly large restriction is caused by the valve guide boss. It must be pointed out, however, that the drawing, by virtue of the angle the head is viewed from, does tend to exaggerate this restriction since it cannot easily be seen from this that there is a certain amount of room for airflow around the sides of the boss. The first step then, is to grind the boss as shown in Fig. 16, then profile the rest of the port as shown to within about 1/4" of the valve seat. If you look just under the valve seat about 3/16" down the port, you will see a step formed by the factory machining of the valve throat and the 'as cast' part of the port' Whether this is by design or not is debatable but nevertheless for our purposes it is somewhat irksome. We can, in most cases, when forming a venturi type port, eradicate the effects of this step. Fig 17 which shows the port in the immediate vicinity of the valve seat, shows how the port should be ground. We should not attempt any work on this part of the port until the valve guide has been fitted and the valve seat recut. When grinding the port do not increase its diameter at the manifold face end. In fact the first 3/8" of the port should be as near to standard size as possible.

Assuming that you have ground and polished the ports to within 1/4" of the valve seats, we can now fit the inlet valve guides which should be modified by turning as shown in Fig. 18. Having done so, get the valve seats recut and finish off the careful not to damage the newly cut valve seat.

Exhaust ports

We can now turn our attention to the exhaust port. As with the inlet port there is a substantial amount of metal to be removed from the valve guide boss. The rest of the grinding is straightforward reprofiling to get smooth contours, Fig. 19 shows the port shape required. With many heads we are likely to come across an irksome little step just below the valve seat. You just have to do the best you can to blend it out. If it is more than 1/16", do not try to blend it out completely as this will tend to make the area of the exhaust port a little too large at this point. As with the inlet port, we adopt the same procedure by leaving the last 1/4" or so under the valve seat until the valve guide is fitted. Before fitting the guide, it should be modified as shown in Fig. 20. When the guide has been fitted and the seat recut, the port can be finished off, leaving a valve seat 1/16" wide.

If you decide to use radius valve seats on the inlet side, then a different approach is required. To start with, you will need to make up a special radius valve seat cutting tool, the details of which are shown in Fig. 21. The seat radius we are going to use is 0.125" and to make sure the tool bit is accurately formed to cut this radius, the correct method of approach should be used. Firstly the tool bit is best made up from an old centre drill of 5/16" or so diameter. Using an off-hand tool grinder, rough out the general shape of the tool. Mount a piece of 1/4" diameter mild steel bar in a lathe or drill chuck and coat it with fine lapping paste. Rotate the bar and press the roughed out form of the radius against the

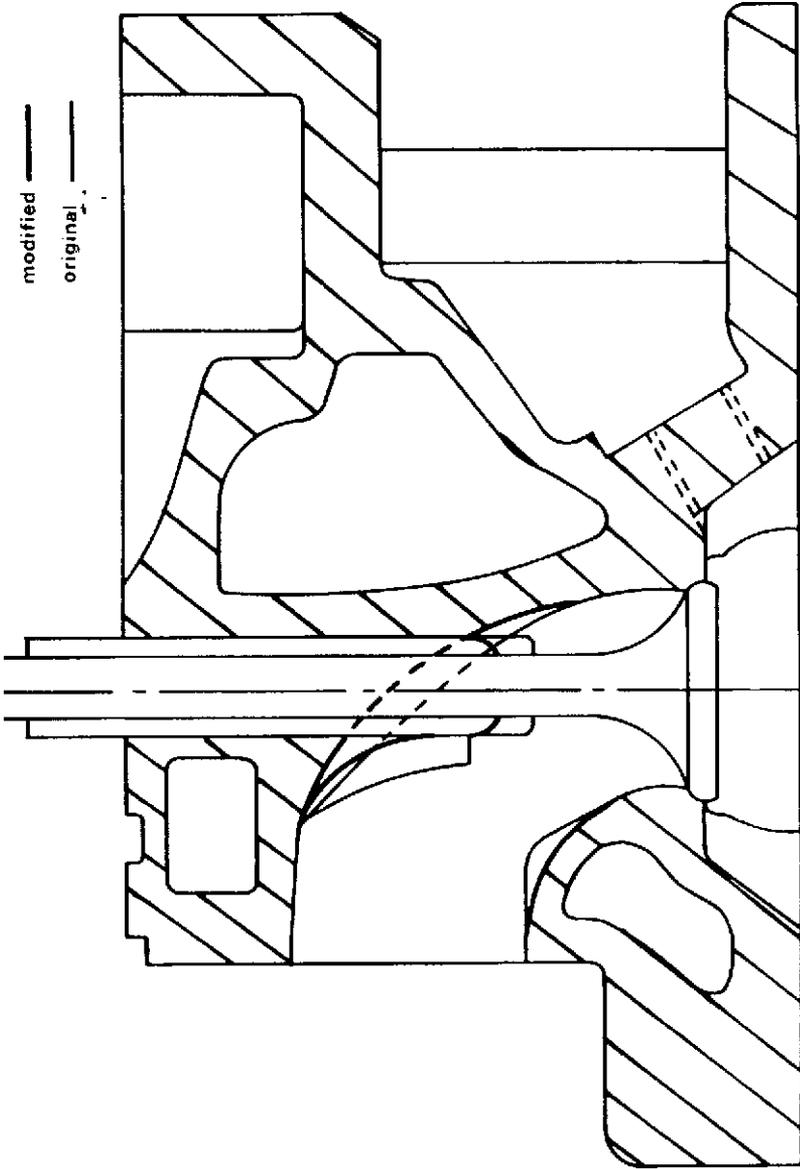


Fig. 19. Modifications to Triumph 2000 exhaust ports. Also applicable to Mk. 1 Vitesse and GT6.

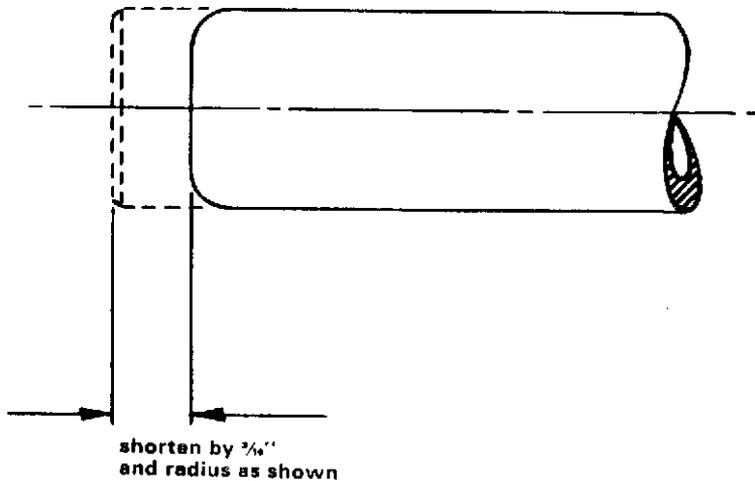


Fig. 20.

bar. Distribute the wear on the bar by moving the tool bit up and down the length of the bar. Whilst accurately forming this radius, be sure to hold to tool bit at a slight angle, approximately 10° , to the bar to give the tool a cutting clearance. Once the tool has been made up, set it in the tool holder to the dimension shown in Fig. 21. Using a pillar drill set on a slow speed to avoid tool chatter, cut the seats. Now back to the -grinder and blend the radius seat into the port as shown in Fig. 22. At this stage in the proceedings you will have to leave the head and turn your attention to the inlet valves. The idea here is to reshape the valves to assist our gas flow. It may seem strange at first sight, but tulip valves do not always give the best gas flow characteristics. To get better cylinder filling we must reduce the tulip shape to something more along the lines of the head of a nail. Fig. 23 shows the shape we should

aim for. Except for the actual seat itself this shape applies to both the conventional valve seat and the radius valve seat.

Lapping in the valves

Once the valves have been reshaped, we can start the job of lapping them onto our radius seats. The method of lapping and correcting any errors of concentricity are as follows. Using fine grinding paste, lap the valve into its seat for a short period i.e. a couple of minutes. Wipe the grinding paste from both the valve and seat and apply a thin smear of engineers' blue to the valve seat. Drop the valve back onto its seat and rotate it. If we now remove the valve, the engineers' blue will show where the valve is making contact with the seat. If the seat cutting operation was really accurate then the seat will exhibit a com

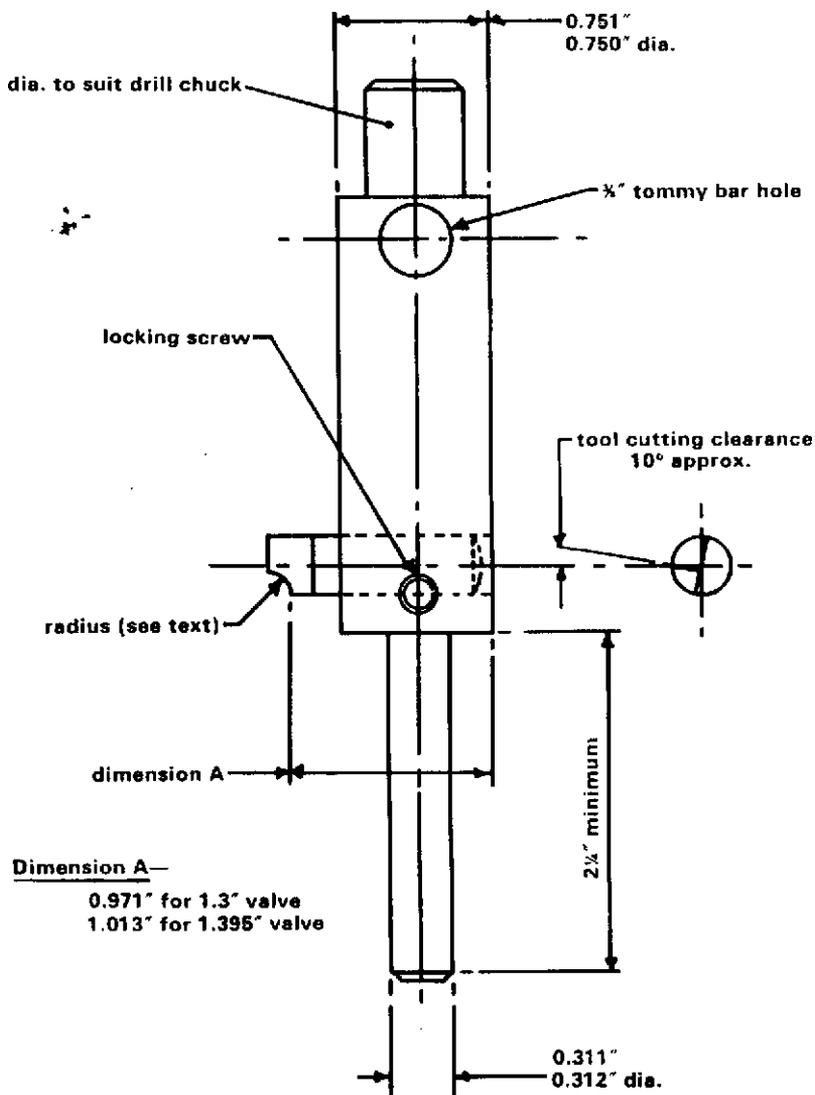


Fig. 21. Valve seat radiusing tool.

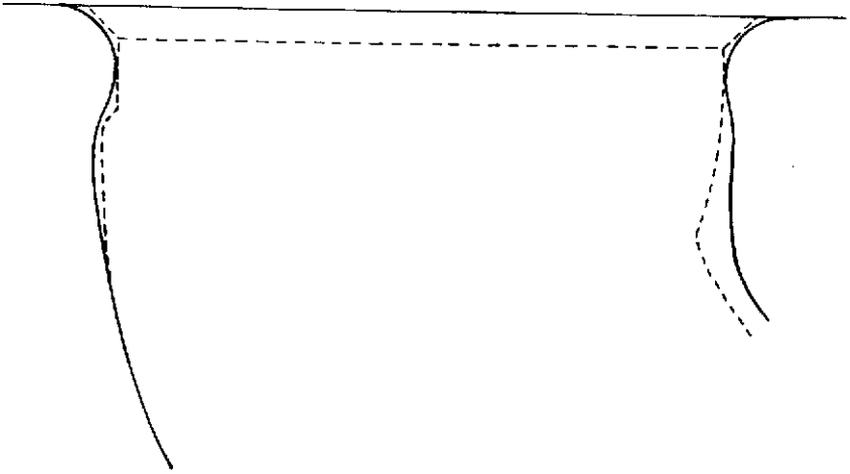


Fig. 22.

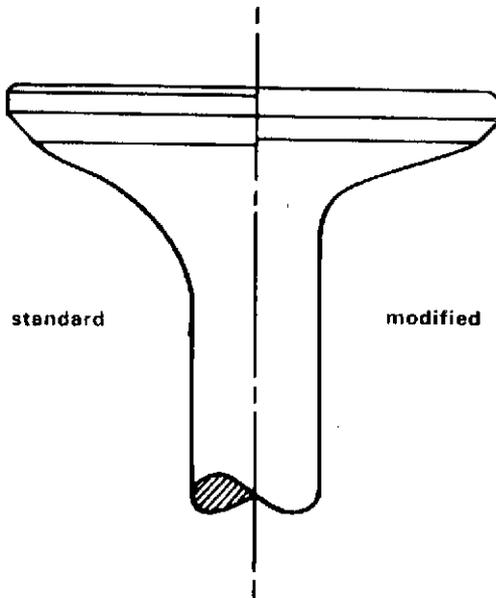
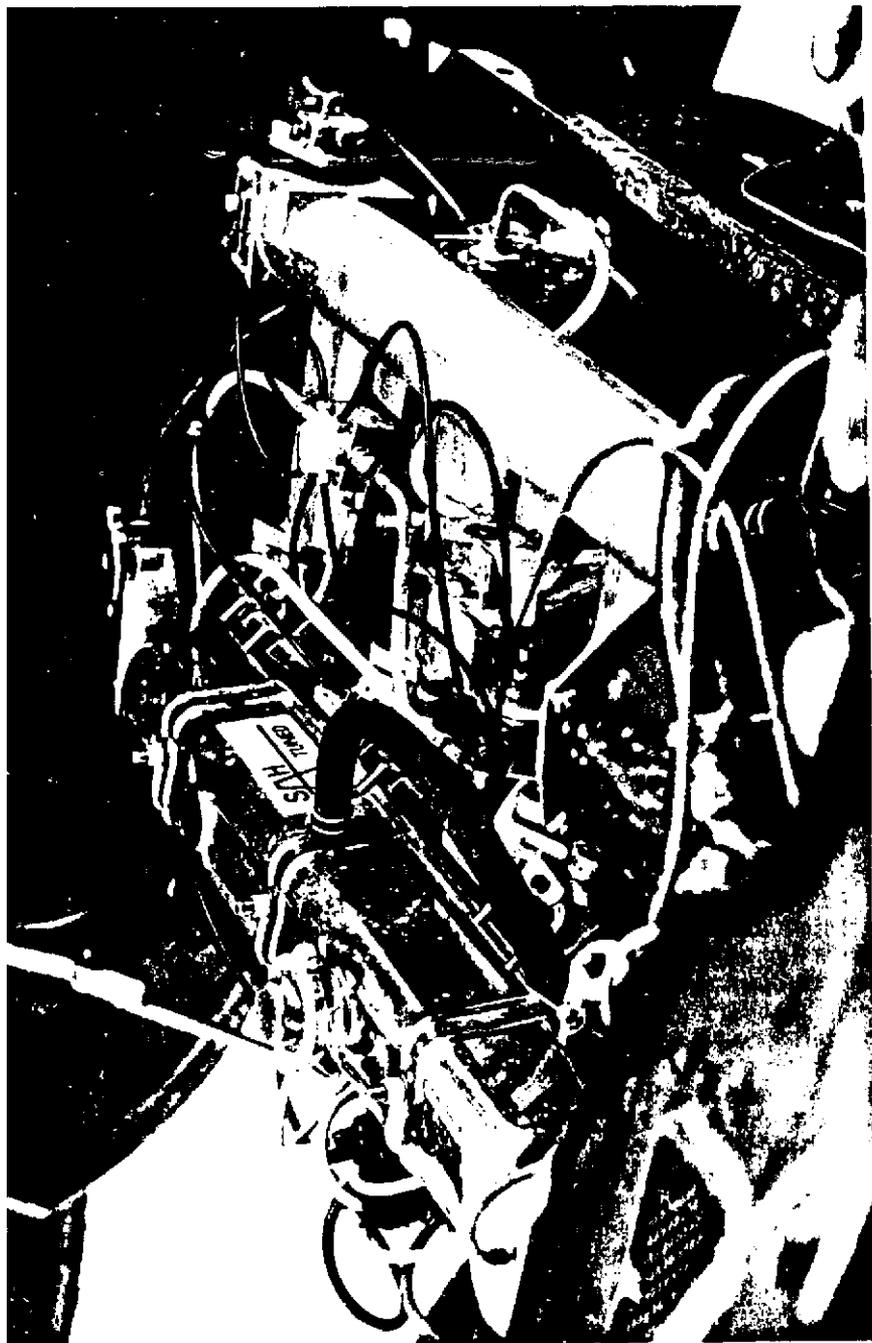


Fig. 23. Reshaped inlet valve.



A GT6 in SAH, fuel injected trim.

plete circular line. In all probability the seat will be slightly out of true and this will be shown up by a blue line over only part of its circumference. In such a case the seat will have to be dressed with the cutting tool. Using the radius cutting tool by hand, lightly cut the part of the seat showing the blue line. Having done so, lightly relap the valve and check it again with the engineers' blue. Continue the lapping, cutting and blueing until you achieve a complete circle on the seat, and a seat width of about 0.030".

The conventional 450 angle valve seats are lapped in the normal manner. It is a good idea to check the validity of the lapping with engineers' blue to make sure the job is up to scratch.

Skirning the head

The final machining operation is to skim the head to achieve the desired compression ratio. With the Triumph heads we usually have a fair margin of metal that can be taken off, thereby giving us any reasonable C.R. that we want. For road use, a 10/1 C.R. is plenty and for racing, about 11.5 will do the trick. It is not easy to quote the exact amount one should remove from the head to achieve these C.R., but the amounts will be about 0.070" and 0.085" respectively. The best way to determine what should come off the head is to fill the chambers with paraffin or petrol from a burette. If we fill the chambers with fluid to the volume required for a particular C.R., then the amount of metal left proud of the fluid

is the amount to remove. Here are two examples showing the volumes required for the chambers on the 1600 and 2-litre engines.

$C.R. = (V+C) / C$ where V = swept volume of cylinder in c.c. and C = total chamber volume in c.c.

For the 1600 engine 1600
 $V = 1600 / 6 = 266$ c.c.

Therefore $C.R. = (266 + C) / C$

Therefore $C.R. - 1 = 266 / C$

For a C. R. of 10 to 1 we have:
 $10 - 1 = 266 / C$

Therefore:

$C = 266 / 9$ which = 29.5c.c.

There is, however, approximately 4.5 c.c. in the gasket and block when the piston is at T.D.C., so this must be subtracted from our 29.5 c.c. to give us the required volume in the head. Our head volume therefore is 29.5-4.5 —25 c.c.

This volume will give us a 10 to 1 C.R. on a 1600 engine. An 11.5/1 C.R. will require a head volume of 20.8. On the 2000 c.c. engines to get 10 to 1 or 11.5/1 we will require 32.5 and 27.2 respectively.

This leaves us only the assembly of the head to do apart from any port matching operations that may be required. The methods to adopt for port matching are examined in a later chapter as is the choice of valve springs for a given application.

BIGGER VALVES FOR EARLY HEADS

FOR an even greater increase in power, the use of oversize inlet valves should be contemplated. The use of a big valve head will come into its own when other components directly related to engine power have seen attention. For this reason we will find that the use of larger valves is a waste of time unless the carburation and valve timing have been improved upon i.e. a hotter cam and better than standard carburation set-up. If the engine is to receive considerable attention in such quarters, then the use of a big valve head is to be recommended. It does, however, entail a little more than just dropping in the larger valves. As with the Stage 1 head, we have the alternative of either conventional seats or radius seats. The choice is yours, but for the highest power the radius seats must be considered the first choice.

Inlet valves

When fitting a larger inlet valve both the inlet port and the combustion chamber are modified to a slightly different shape to those used for the

Stage 1 head. This, of course, makes the assumption that you are using a valve about 1/10" larger in diameter. Let us assume at this point, that the valve to be used is 1.395" diameter, which is, incidentally, the one which is supplied by S.A.H. Accessories of Leighton Buzzard. The procedure to adopt to modify the head will only be discussed regarding those points which differ from the Stage 1 head.

First off, we will need a perspex template for the big valve chamber shape which can be traced directly from Fig. 24. The procedure from here on to complete the chambers is as for the Stage 1 head in Chapter 4.

We can now turn our attention to the inlet port. With the guides out, profile the inlet port to within 1/16" of the valve seat as shown in Fig. 24. When the port has been finishpolished to this stage, fit the guides which, incidentally, should have been previously modded as in Fig. 18. A tool will be required to bore the valve throats out to take the larger valves. If we are to use 451 conventional seats, then a tool bit

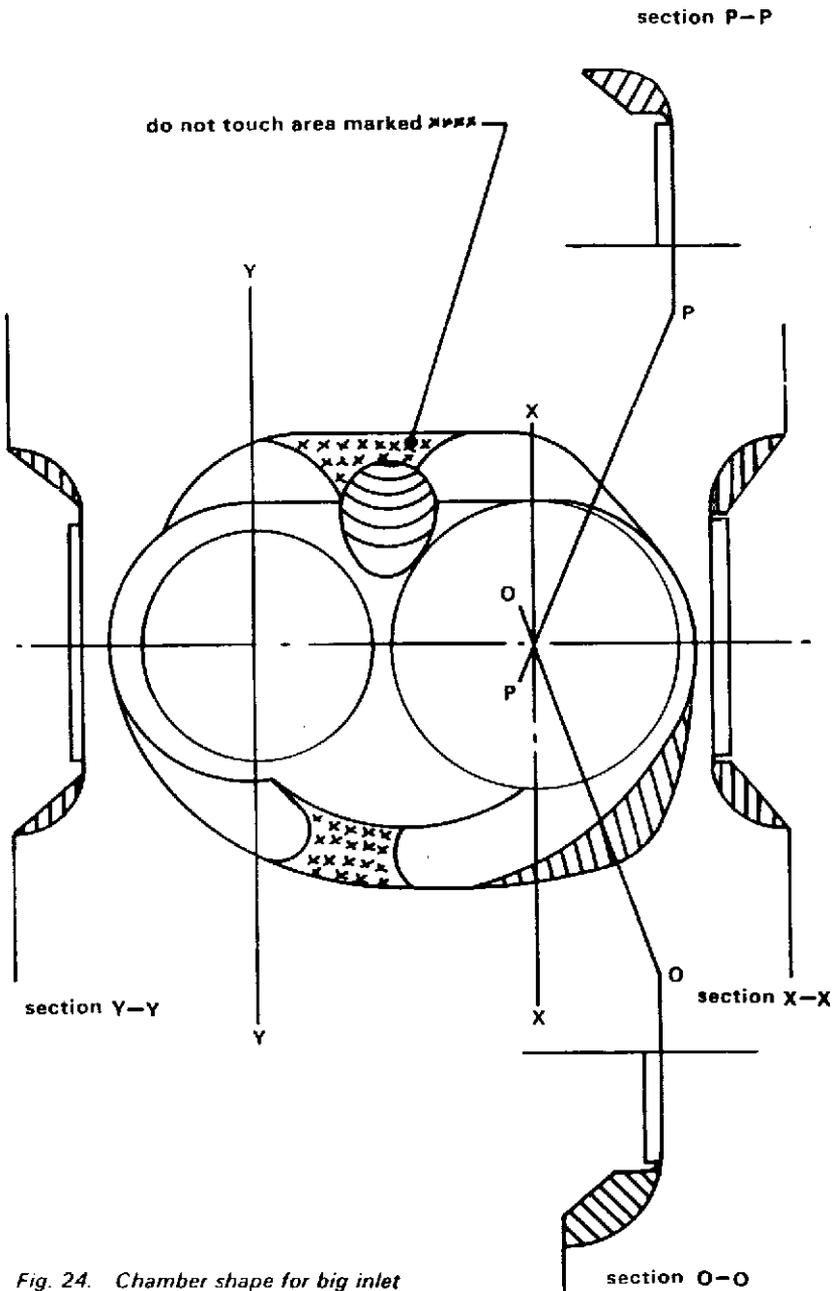


Fig. 24. Chamber shape for big inlet valves to suit 1600 and Mk. 1 2000 engine.

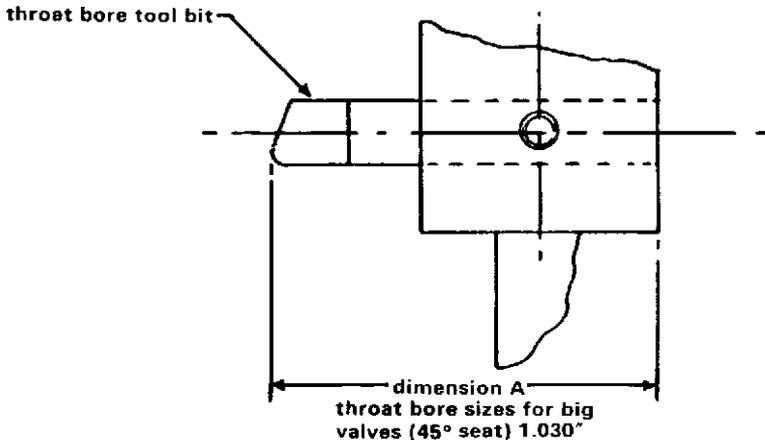


Fig. 25.

fitted into the tool holder as shown in Fig. 21. should be used. The setting dimension (A) will be 1.030". The tool setting to bore the correct size for a 1.395" diameter valve is shown in Fig. 25. This will give us a throat diameter immediately under the valve of 1.310" diameter. Boring out the throat also means we stand a very good chance of completely removing the small step which occurs just under the valve seat on the majority of heads. When the port has been bored to a depth of 1/8" to 3/16" blend out any irregularity with the finished part of the port. Cut the valve seat and lap in the valves. Now comes a tricky bit; put a small radius on the inner edge of the valve seat thus breaking the sharp corner formed by the seat at the port (Fig. 26). In doing so, this should reduce our seat width to about 0.030'.

Radiusing seats

For those of you who are brave enough to tackle radius seats, you will need the tool holder and radius cutting tool shown in Fig. 21. For this particular application, the radius of the cutting tool should be 0.140" instead of the 0.125" shown in the drawing. Our setting dimension will also be different.

Before you remove the inlet guides from the head to grind the ports, the radius seats should be roughed out. The setting dimension for roughing out the seats, dimension A in Fig. 21, will be 1.003. Once the seats have been roughed out, the guides can be removed and the ports ground to the required form as in Fig. 27 and polished. New modified guides are then fitted and the seats finished off. To do so, the radius seat cutter, dimension A, must be reset to 1.013" and carefully recut. The finishing procedure em

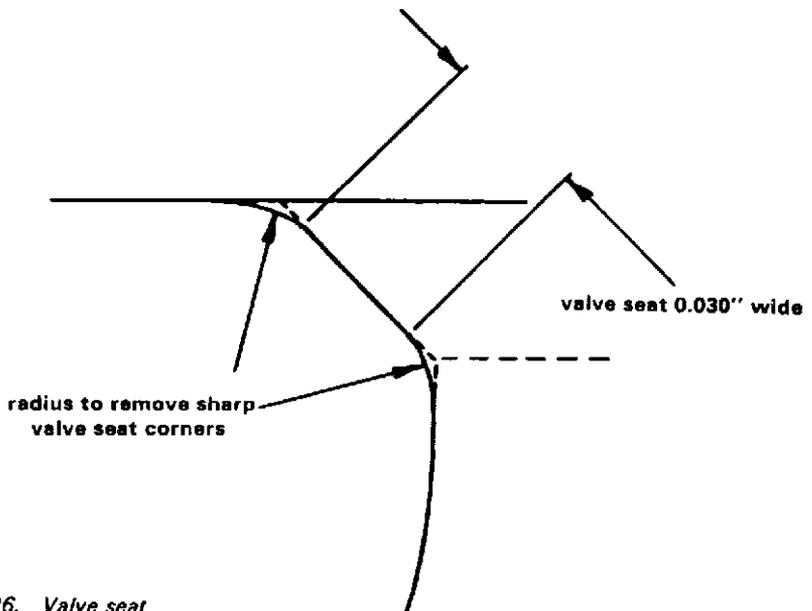


Fig. 26. Valve seat.

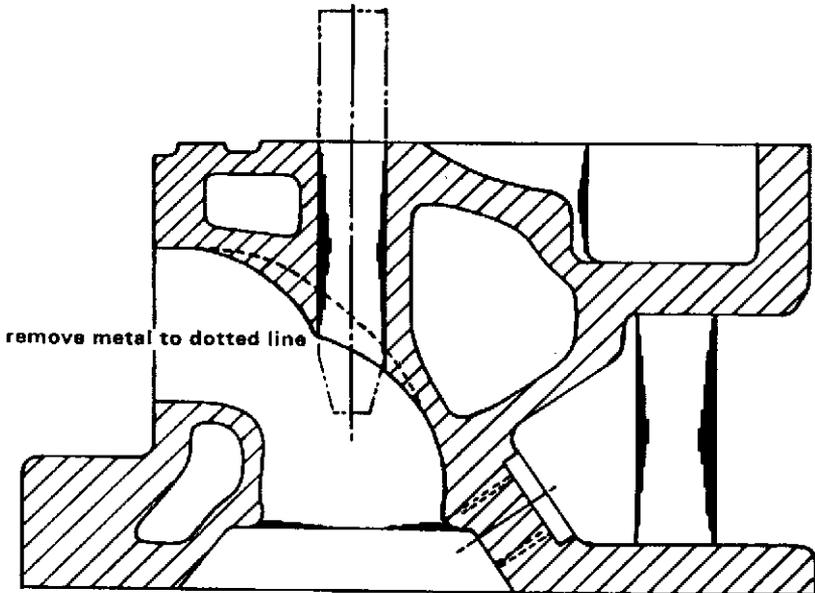
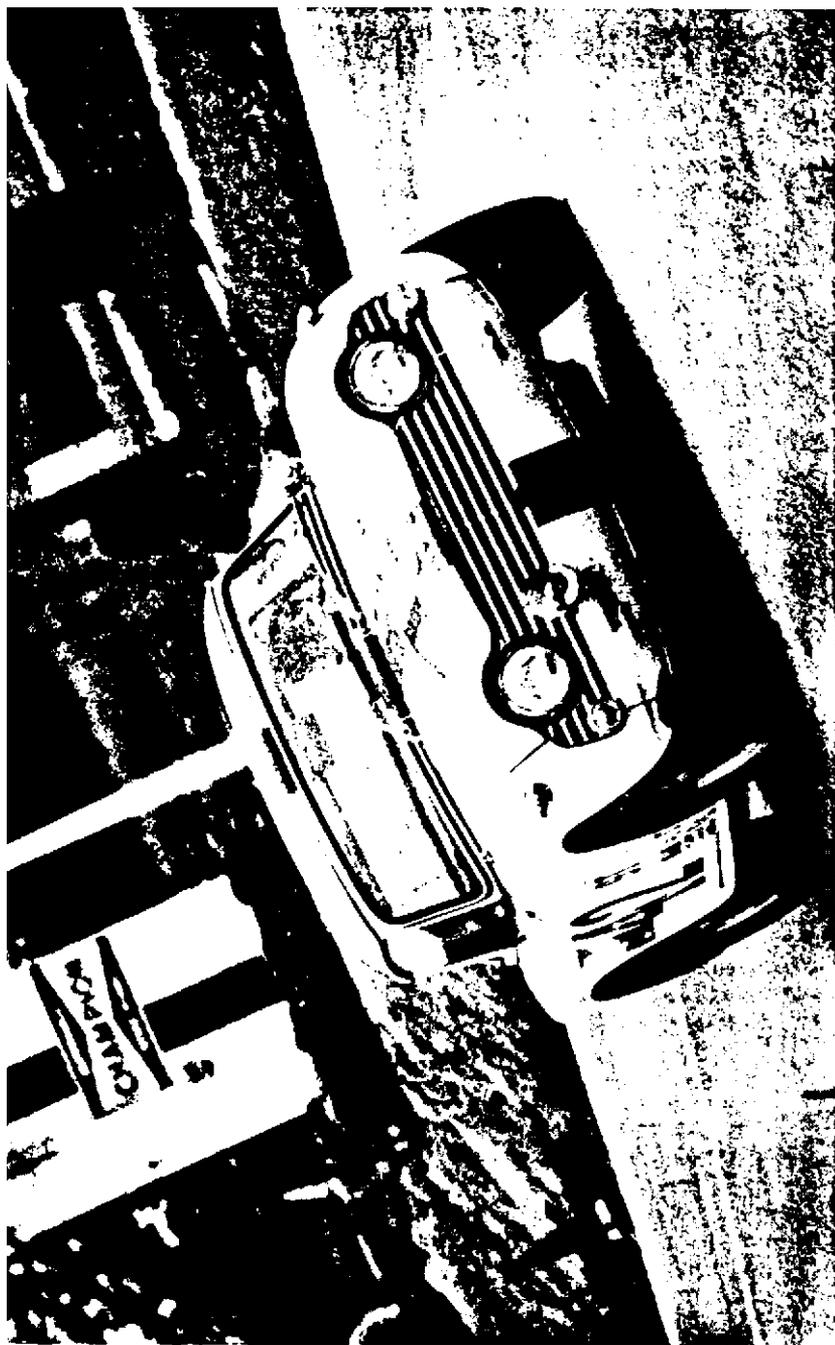


Fig. 27



A left hand drive TR4A shows Triumphs popularity in the sports car class throughout the world.

ployed on the seats is then as detailed in Chapter 4.

The grinding of the exhaust ports and the raising of the C/R remains the same as explained previously. With the big valve head it will be found that a little more is required to be skimmed from the head than for a Stage 1 head because of the greater amount of metal removed from the chamber. The extra metal to be removed is roughly going to amount to 7-10 thou. but this figure is intended as a guide only. The only sure way to get the C.R. to the required level is to use the burette and paraffin to measure the volumes and determine the correct

amount to take off.

The use of this large inlet valve type of head gives a power increase of around 7% over the standard valve size modified head. It also raises the point at which peak power occurs by about 200-250 r.p.m. on the 2000 c.c. engines and about 300-350 on 1600 engines. If used on a 1600 engine, the rest of the engine should be brought to a fairly high state of tune to bring it in line with the characteristics of the head. Even on a 2000 c.c. engine other modifications will need to be done to make full use of such a head.

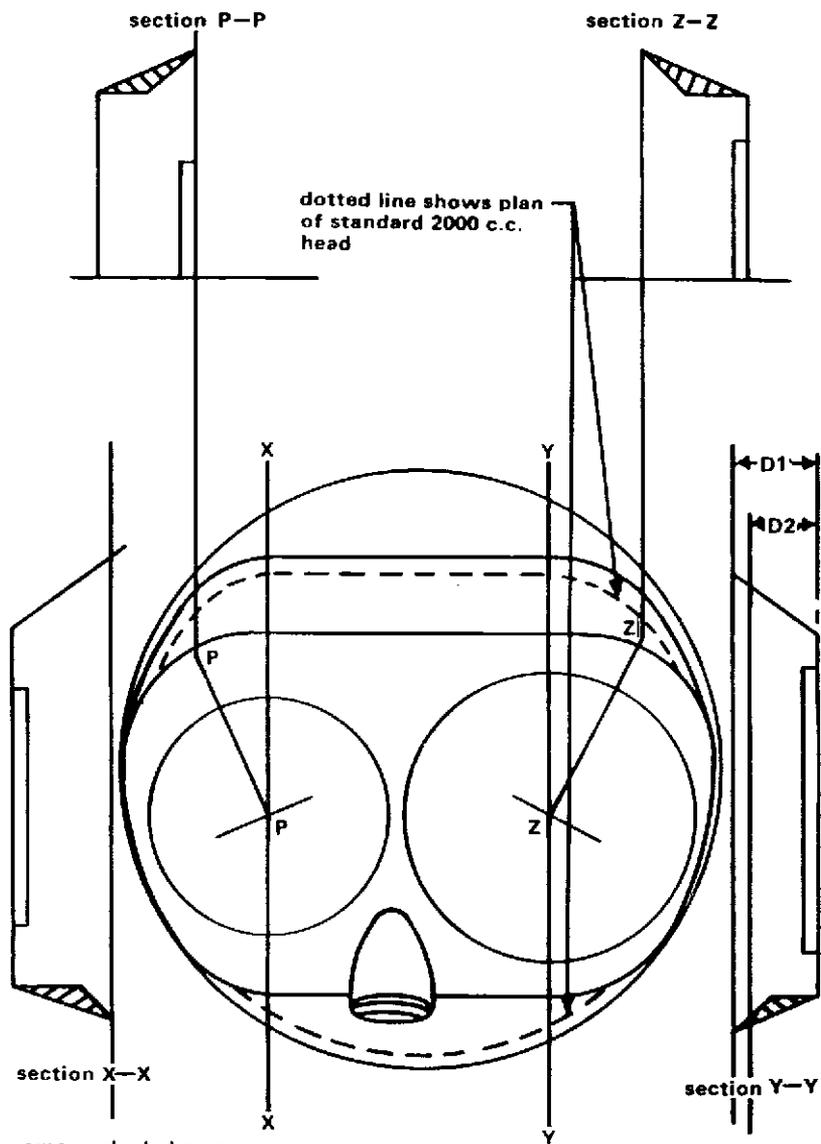
MODIFYING THE LATER 2000cc AND 2500cc HEADS

THIS chapter applies to the heads used on the T.R.5 and 6, the 2000 saloon from 1966 onwards and the Mk. 11 versions of the 2-litre Vitesse and G.T.6. Comparisons of Figs. 15 and 16 with Figs. 28, 29 and 31 will show the basic differences in these heads. The later head is generally of a superior design to the early one. The ports have a more gently curved contour to the valve than the earlier head, thus aiding gas flow. The inlet port terminates at a valve some 0.144' bigger than its predecessor which, coupled with the better port, shows a significant advantage in power. The combustion chamber in the later head is also changed for the better and to get the best from it without completely redesigning, there is little we can do to improve matters. A glance at Fig. 28 will show just how little work is required on the chambers. In plan view the chamber shape remains unchanged. The sum total of the grinding required is all performed on the chamber walls. The end result of the modifications to the chambers should be a chamber wall which changes

smoothly from a vertical to an angled wall, leaving no projecting edges in the chamber. This tidying up of the chamber shape gives us improved combustion efficiency but the gains achieved are only marginal. No doubt if we compared the difference in power gain by the slight change in profile, the net result may only be in the order of 1 A-2%. Since every engine tuner is after every little drop of power within a given budget, it can be argued that although the gain is small, the modification is justifiable. It is surely this type of attitude towards the preparation of a motor that makes the true enthusiast stand out from the rest of the crowd.

Porting

Anyway, back to the subject in hand which is now the ports. These require little more than polishing. Do not, under any circumstances, open them up, as if anything, they are slightly too large now. As with the earlier type of head, we have the presence of the small step just



D1—standard chamber depth for 2.5 litre engines

D2—standard chamber depth for 2 litre engines

Fig. 28. Chamber modifications for TR5 and 6, 2500PI, Mk. 2 GT6, later type Triumph 2000 and Mk. 2 2000 Vitesse.

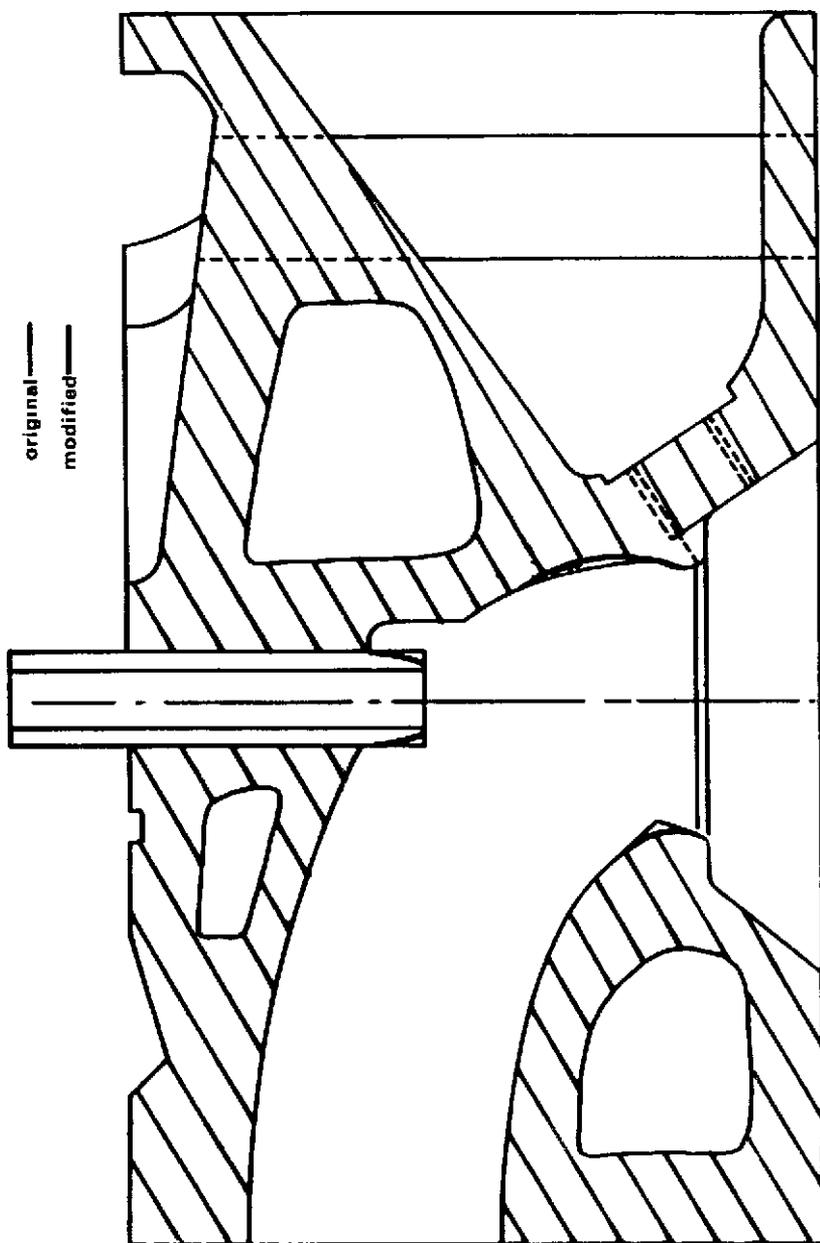


Fig. 29.

under the valve seat, the complete blending out of which is a little difficult. On some heads it will blend out, but on others it will not. It just depends on the casting. Another point which is a little undesirable but one we can do next to nothing about, is the pocket around the valve guide. It could be termed a negative boss in as much as it is a boss recessed in rather than one projecting out. The presence of this inset boss means that we cannot get a smoothly contoured port in the area either side of the guide and immediately behind it. From our point of view, the presence of this depression is a little annoying, but the Triumph Engineers in their wisdom, did not put it there for nothing. Consider the fact that the guide is a press fit in the head. If one side of the hole into which the guide is fitted is substantially longer than the other side, the forces prevailing from the press fit impart uneven loads from one side of the guide to the other. This means that the guide itself becomes bent. Also the machining of an accurate guide location hole is nearly impossible by production methods if it breaks through on an uneven surface. That explains why the depression is there but provides absolutely no remedy for it. On occasions attempts have been made to fill such depressions with plastic metal, but the practice seems to be a little unreliable. Should the plastic metal break away, it will cause serious damage to the valve and possibly the piston in that particular cylinder. Alternatively one could have the depression filled with weld to reduce any adverse effects it may have on gas flow. Here again, this is quite a tricky job as the whole head will have to be heated to

a relatively high temperature to avoid distortion. The answer to all this indecision is to leave the depression as it is unless you are bent on getting every little bit of power possible. Once the inlet port has been fully blended and polished as in Fig. 29 we can fit the inlet guide modified as in Fig. 30 and cut the valve seat. Any sharp edges left by the valve seat cutting operation should be blended out by radiusing the edges. The sharp edges of a seat can have a significant effect on the gas flow into the cylinder, especially during the period when the valve is only just off the seat. The careful blending of a valve seat into the walls of the port and the roof of the chamber can have the same effect as a cam with a faster lifting rate. The use of a cam with a higher flank acceleration rate involves higher stresses in the valve gear, whereas the seat blending has no mechanical drawbacks assuming the seat width has not been reduced to a point where it can no longer sustain the loads imposed by the valve.

Rather more work is required on the exhaust ports than that needed on the inlet. The amount of grinding required though, is still a lot less than needed on most heads. Grind the exhaust port to the contours shown in Fig. 31 and avoid taking metal from the tight corner opposite the valve guide.

One should avoid making the exhaust port any larger than is necessary to achieve a good finish. Polishing is an utter waste of time a good smooth ground finish will do the trick. With the guides machined and fitted, as in Fig. 32, cut the seat and remove any sharp corners formed. When removing the corners,

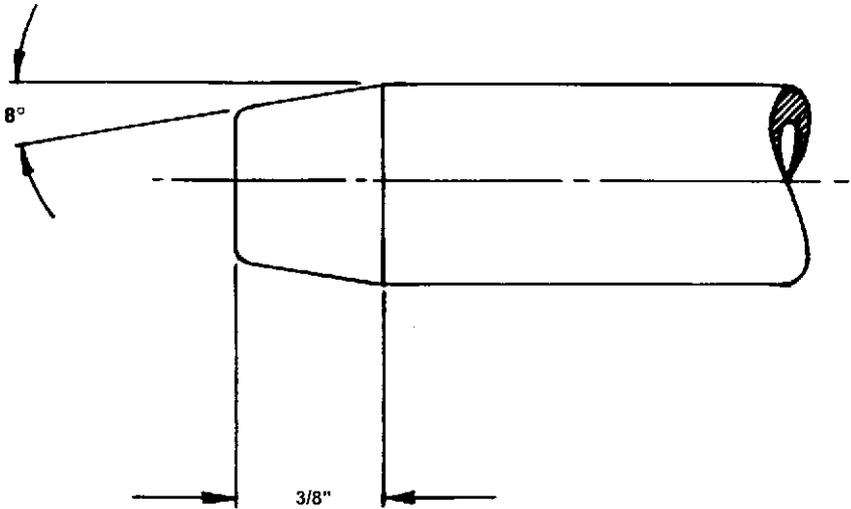


Fig. 30.

do not allow the seat width to go below $1/16$ " or short exhaust valve life will result. Apart from a bit of inlet valve reshaping, the head is virtually finished at this stage. Just to round off matters, we reprofile the inlet valve to the shape shown in Fig. 23 then give it a good polish. The important part of the valve reshaping is to make sure that the seat blends in smoothly with the rest of the valve. Any sudden changes in shape at or around the seat area cause a reduction, sometimes of substantial proportions, of the flow of gas at low valve lifts.

Radius valve seats

We will deal now with the rather more complex method of modifying this head using radius valve seats. Although the work may entail a little more skill and effort involved as compared with a straightforward modified head, the situation can be greatly

eased by farming out certain parts of the work to your local motor machinist. Even though a greater amount of effort is called for to do this head in the fashion about to be described, it is well worth it, especially if any serious form of competition is envisaged.

The only point at which we are at variance from the conventionally modified head is the area around the inlet valve seat. In all other respects the head will be as for Figs. 28, 29 and 31. It will be found very difficult, if not impossible, to use a radius valve seat with the correct radius in place of the standard seat. The reason being that the original seat has had both adjoining surfaces machined i.e. chamber roof and port throat, and has left insufficient metal to form a radius seat of the correct dimensions. The only practical way around this is to make up a valve seat insert. Apart from enabling the

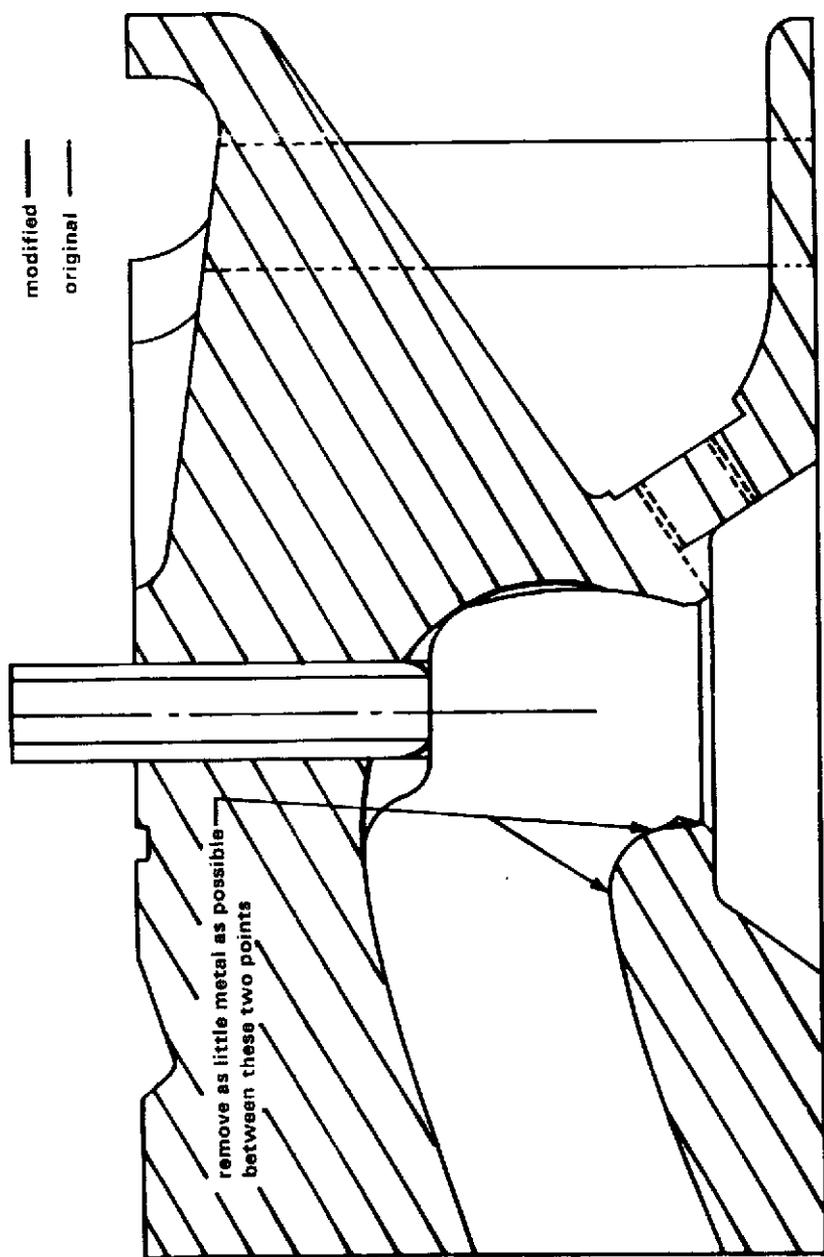


Fig. 31. Exhaust ports modified 2000 Mk. 2 and 2.5 litre engines.

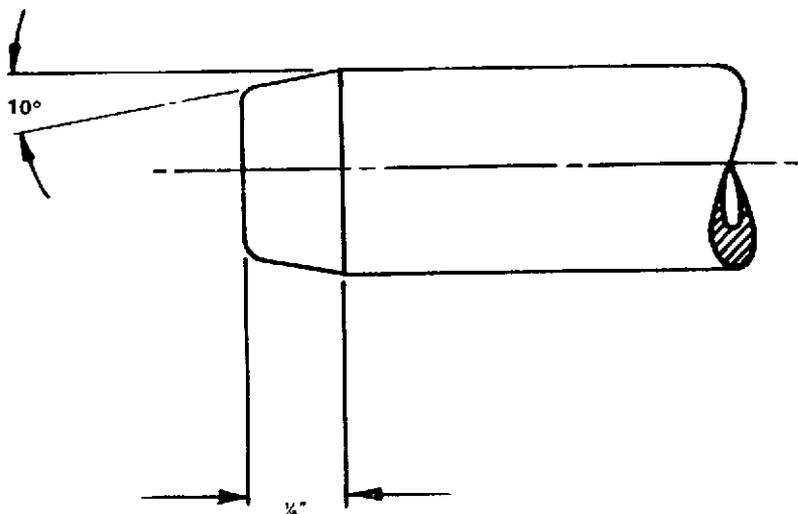


Fig. 32. Exhaust guide modification.

job to be done this has one other distinct advantage, this being that we can accurately form the area just below the valve seat. The most critical points on any head on which one is attempting to substantially improve breathing, is the area 1/2" before and 1/2" after the valve seat. since, for our purposes, there is a distinct lack of metal just before the seat, at a point we require some, the obvious answer is to put some there, hence the seat insert. To bore the head to take the inserts you will have to call on the services of a motor machine shop, and if you lack facilities, he can also make up and fit the inserts.

As you have probably guessed by now, the inserts are not going to be just a plain straightforward ring set into the head. Fig. 33 shows the profile set on the insert and one should attempt to meet the specified dimen-

sions as close as possible. The really critical one is the diameter of the radius upon which the valve seats. If this dimension is out by more than 0.005' (5 thou.) the valve will not seat in its correct position. Before fitting the inserts or in fact, machining the head to take them, the inlet port should be finished within 3/8" of the valve seat and the guides fitted. It is also of paramount importance that you have guides whose bores are on or very near bottom limit (0.312 " dia.). The size of the guides needs to be accurate for the pilot of the counterbore machine to locate when machining the head for the inserts. To get the valve seat of the inserts accurate, in relation to the guide, we must ensure that both the location diameter and the valve seat of the insert are truly concentric. Also the counterboring of the head

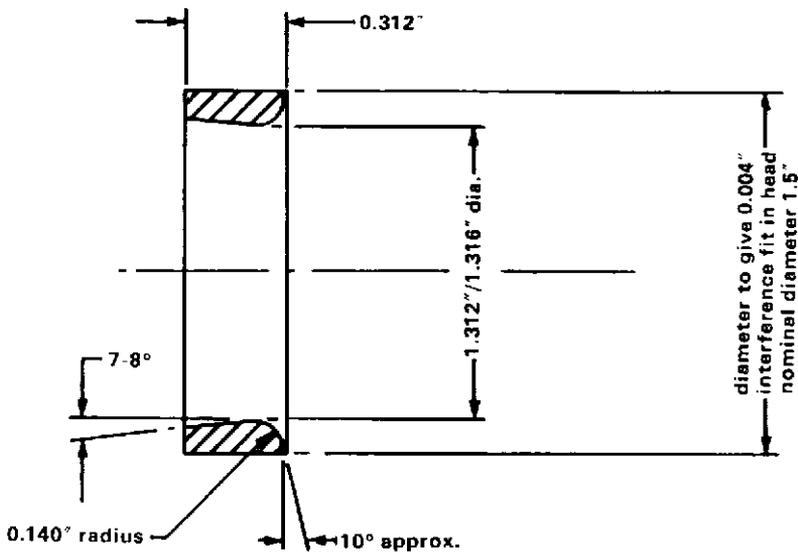


Fig. 33. Radiused valve seat insert.

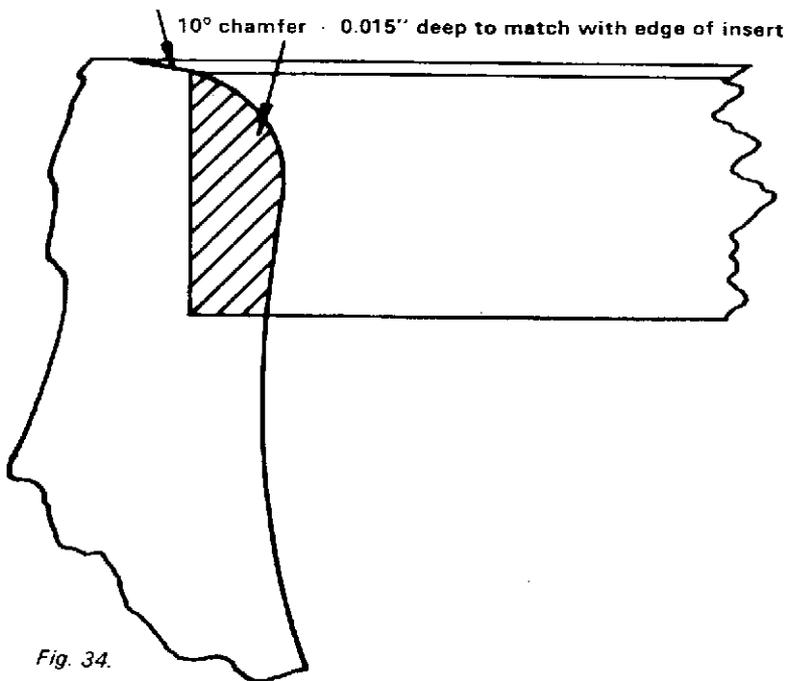
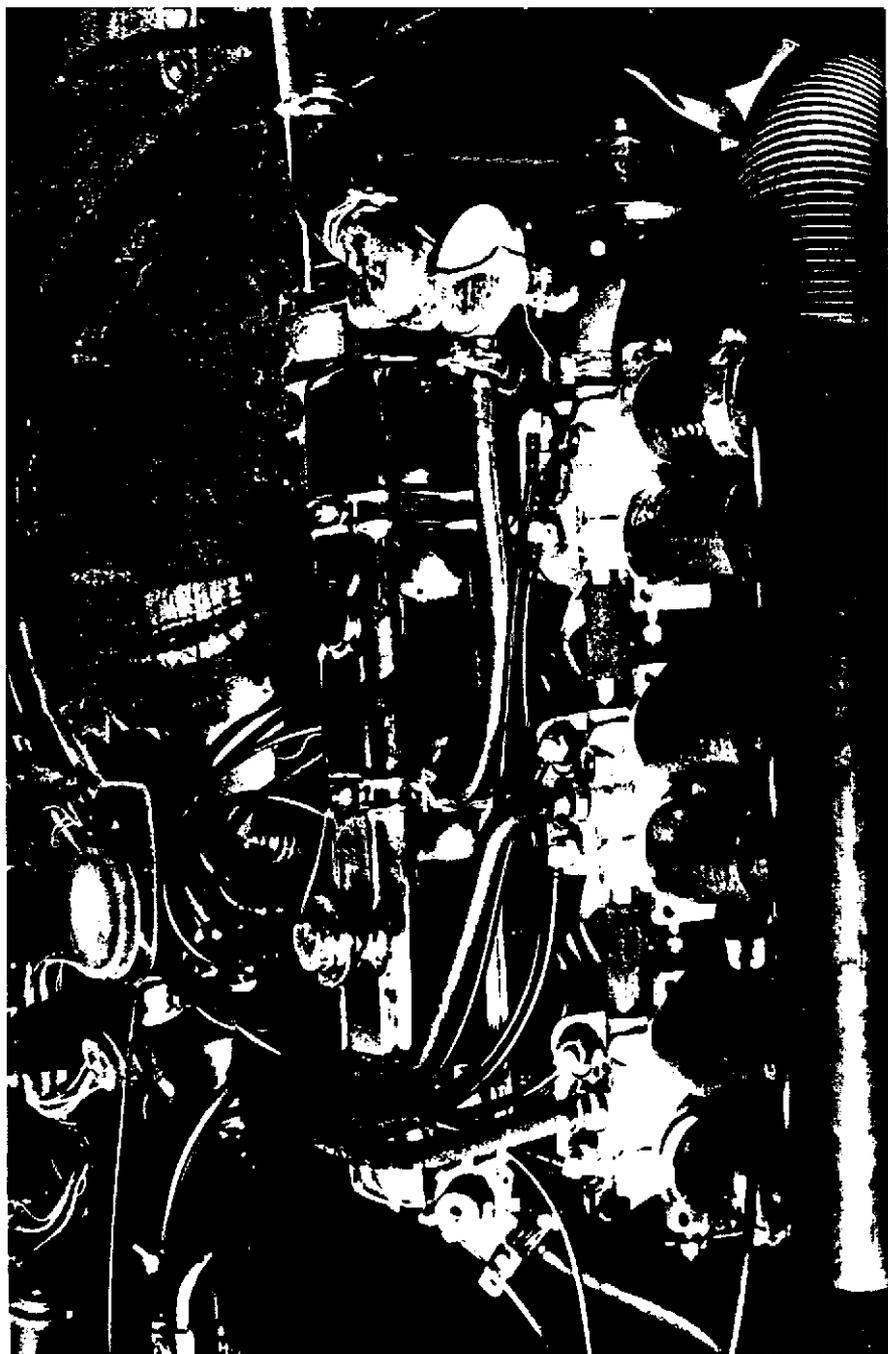


Fig. 34.



A Lucas fuel injected Triumph 2000 lump.

must be concentric within about 0.001" (1 thou.). If this is done, the valves will lap into the seats very quickly without undue deformation of the seat on the valve or insert. When lapping in the valves, engineers blue must be used to check the validity of the seat.

After the inserts have been fitted, a little blending with the grinder will be needed. The final form that should be achieved is shown in Fig. 34. The use

of this type of valve seat can add up to 5 b.h.p. on top end power but that is not the end of the story. If you are using a hotter cam, say one that comes in at about 2,500 and is all off at about 7,000, then you will find that the use of these seats will modify these figures to about 2,400-7,200. We are in effect, gaining in all directions, having increased the power, rev range and flexibility.

MODIFYING THE TR4 HEAD

THE T.R.4 head, from the home-tuner's point of view, is one that is fairly easy to modify. The head is big enough to make just about every part of the port easily accessible. The inlet port is big enough to practically climb into, and the same goes for the exhaust ports to a lesser extent. It is unfortunate from the power output point of view, that the inlet ports are a little too big, but there is very little that we can do about that. A partial solution to the problem is to avoid taking any more metal out of the ports than is required to give a good smooth finish. As with the other Triumph heads, we are plagued by the small step which inevitably exists just under the valve seat. 50% of the time this step will blend out when the ports are modified as in Fig. 35. If the step is greater than 60 or 70 thou., then you just have to live with it, or a small amount of it, even after grinding. Basically the work on the inlet port can be summed up as a little reshaping and blending and a lot of polishing to get a smooth uniform finish. We do not want a finish akin to chrome plating, as this will only serve to exaggerate the fact that the

port is too big. It will also serve to encourage fuel separation from the air because of the lower gas speed prevailing in an overly large port. The type of finish we should go for would be more along the lines of a vapour blasted or fine shot blasted finish. In fact if you have access to or know someone who can fine shot blast the inlet ports after grinding and light polishing, so much the better. If you have the port shot blasted, then the last part of the port just under the valve seat will need to be polished. The length to be polished will be about 1/4" from the valve seat down.

After fitting the guide, cut the seat and remove any sharp edges from the seat by radiusing off. In the interests of retaining a valve seat of reasonable width i.e. about 30 thou., any radius applied to the sharp edges of the seat should be small, in the order of 40 thou. or so.

Chambers

As can be seen from Fig. 36, there is a fair old chunk of cast iron to be taken out of the chambers. To

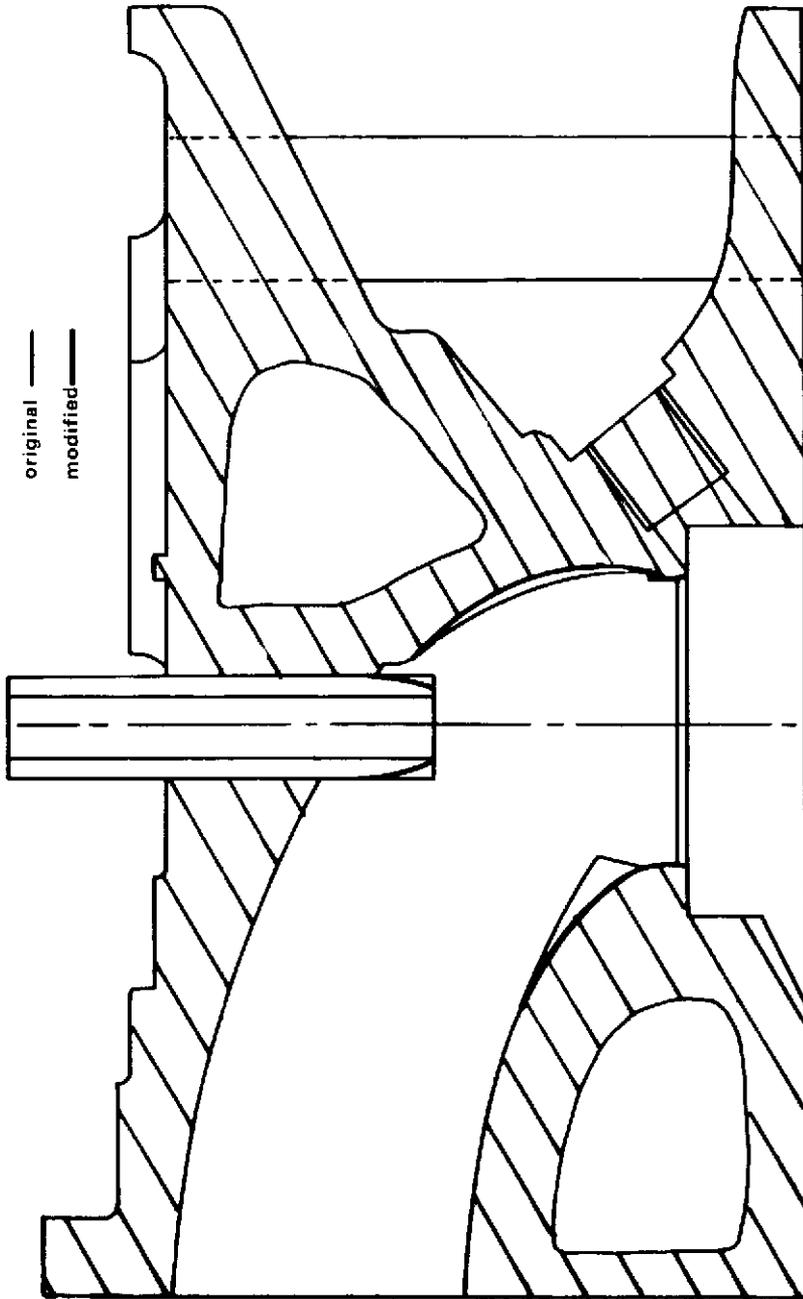


Fig. 35. Inlet port modifications to TR4 and 4A.

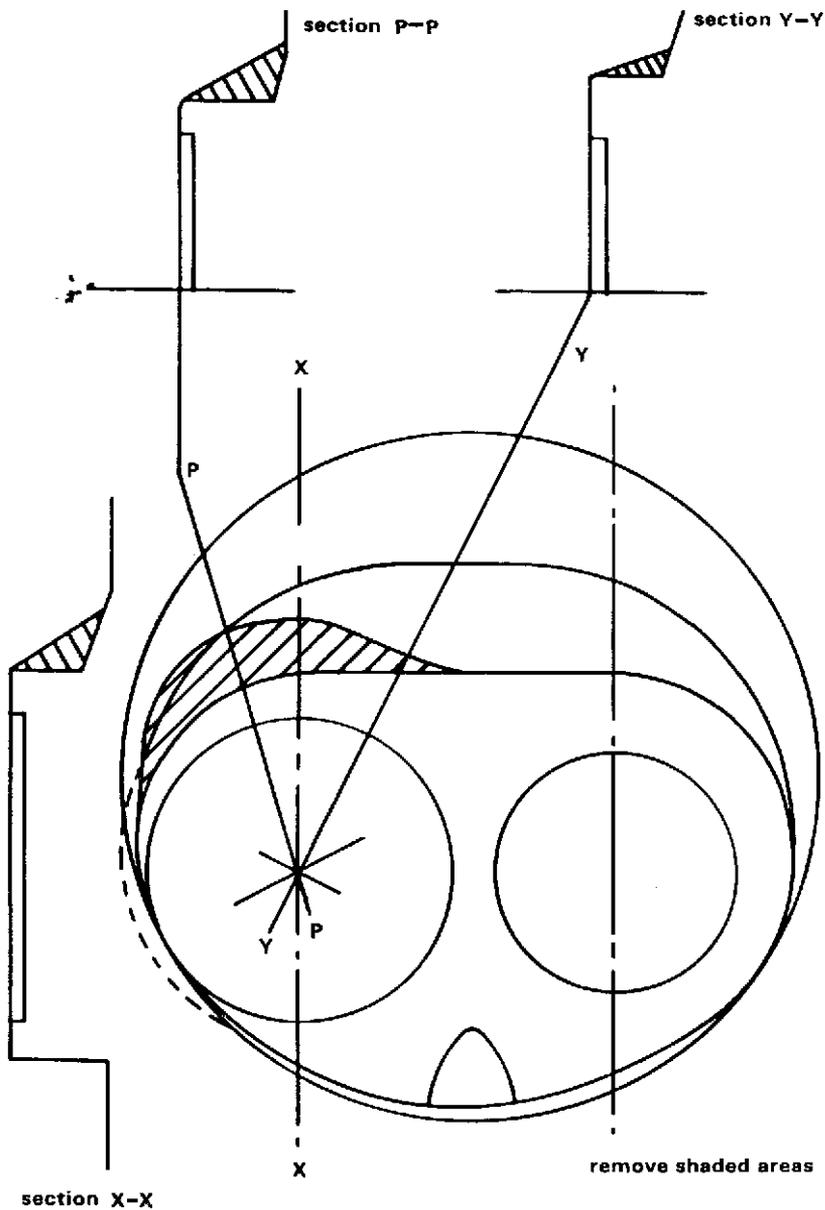


Fig. 36. Chamber modifications to TR4 and 4A.

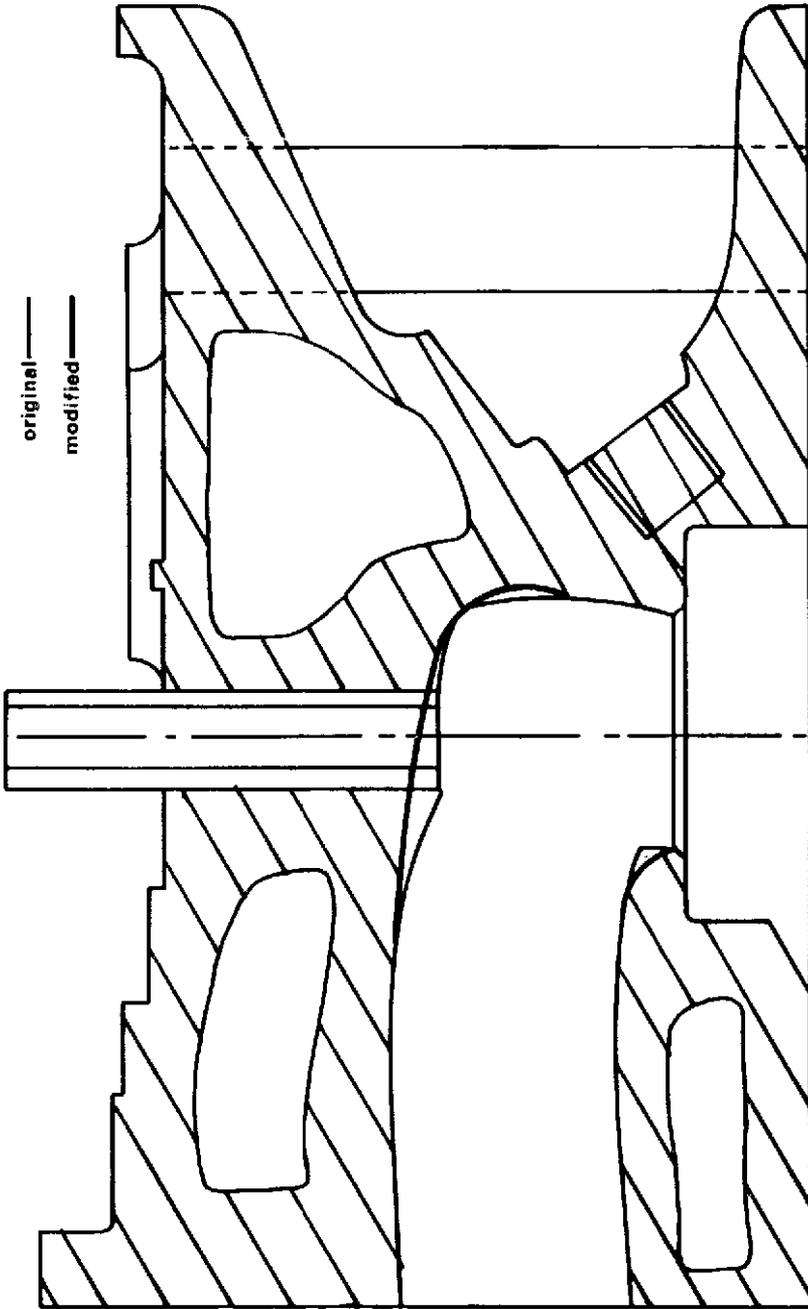


Fig. 37. Modifications to TR4 and 4A exhaust port.

enable reasonable uniformity of shape from chamber to chamber one should make up a template from which the marking out of the new shape should be done. As described in previous Chapters, the template should be made from perspex for a number of reasons, primarily though, because it is easier to make and use than most other materials. When marking out the chamber, we do come across a slight snag. Part of the squish area on these heads is not flat. Since our template is flat and will be on the head face, we will find that most of the surface on which we are marking is falling away from the template. This means that you will have to use your skill to judge by eye the line you are marking directly under the edge of the template. After marking out the chamber, it is just a case of hogging out the cast iron as shown in the drawing. One thing that is a little difficult to show in the drawing is the fact that the undercut in the chamber wall, shown dotted, blends in smoothly with the newly ground shape.

Ports

Inspection of Fig. 37 will show that there is not too much metal to come out of the exhaust ports. Unlike most of the other ports on Triumph engines, this one is best ground with the valve guide in place. This then brings the valve guide flush to the roof of the port. This practice of bringing guides flush to the port especially on the exhaust side is one which should not be practised without due consideration of the consequences. The shortening of the exhaust guide is usually accompanied by a rise in exhaust valve temperature. In this particular case the

shortening of the guide is small enough to have a negligible effect on the exhaust valve temperature. As has been said many times before, there is no point in polishing the exhaust port as the first few minutes running will completely coke up the most perfect polish. A good lump and bump free ground finish will suffice.

Apart from removing any sudden changes in section like the bump at the point the stem grinding terminates, there is little valve reshaping to do. So long as the contour change on the back face of the valve is smooth then the valve will perform its sealing function whilst presenting the least resistance to flow. As with the seat in the head, we should blend the under head face of the valve smoothly into the seat to increase the valve effectiveness at low lifts.

It is now head skimming time. So long as we bear a few points in mind, we can get the C/R to a reasonably high level. For a road car 10 to 1 should be considered about as high as we want to go. For racing, we can up this to 11 to 1 since the intervals between decokes are usually shorter. When skimming the head, we must avoid making the thickness between the head face and the undercut in the chamber adjacent to the inlet valve too thin. If we do make it too thin, this can lead to several undesirable consequences.

1. If it overlaps the bore slightly, it may cause a hot spot, leading to detonation.

2. If it does not overlap the bore but instead rests on the gasket, the pressure of the gasket caused by tightening down the head may be just sufficient to cause the edge to crack and break away. This means

we have a small but very dangerous piece of cast iron floating around in the relevant cylinder.

3. If the area of the head face directly over the pocket has been machined excessively, then when the head is fitted we will find that part of the gasket is exposed. Numerous blown gaskets will be the result of this.

To get a 10 to 1 ratio we will need a chamber volume of 55.5 c.c. Since we

have approximately 6 c.c. in the block and gasket, the volume in the head will need to be 49.5 c.c. For 11 to 1 this figure will need to be 46 c.c. To get these ratios the head when modified as in Fig. 36 will need about 70-80 thou. off. When the head has been skimmed, remove the sharp edges from the chamber and it is ready to assemble.

THE ULTIMATE HEAD

CAN we say that any of the heads so far described for the six cylinder engines is the ultimate within our limitations, imposed by using, as a starting point, a standard casting? Would the hottest head as described in Chapter 12 be good enough to produce the power required by our hypothetical Vitesse racer mentioned in the introduction? In all probability it would fall a little short of the mark, but this is merely incidental.

Do we need bigger valves? The answer to this one is no. The valve area we have, should suffice up to about 8000 r.p.m. This means that any deficiency in breathing is not due to the valve sizes employed. The ports certainly are not too small for the size of valves used. If this is the case then we are left with only three areas in which drastic improvements can be made. These being the chamber, valve seat and the entry direction of the port. If we can make substantial improvements at these points, then we could be well on the way to achieving that 195-200 b.h.p. that is needed to make a sure race-winner

Let us look at tile chamber design

first, and consider what happens when we open the valve into a chamber with a flat roof and walls well clear of the valve.

To do this we will look at tile effect of fluids in the port instead of air, since fluids exhibit the same qualities as gases except for compressibility. Fig. 38 (a) shows part of a port, a valve and a chamber. The fluid in the port is coloured differently from that which is already in the chamber thus enabling us to see the flow pattern when the valve is opened. Drawing (b) in Fig. 38 shows the flow pattern when the valve is opened, and this is characteristic of the flow pattern throughout the opening and closing cycle. It can be seen that turbulence starts right on the edge of the valve seat. This has the effect of impeding the flow of gas just before the seat and the turbulence itself is absorbing valuable kinetic energy which otherwise could be assisting flow into the cylinders.

Radius valve seats, can these help? The answer to that must be a conservative yes. By using radius

valve seats we can delay the onset of drastically turbulent flow while the valve is at low lifts. The radius seats can actually increase gas flow into the cylinders by as much as 20% at low valve lifts. As the valve lift gets higher, so the effectiveness of the radius seat drops off. By the time full lift is reached, the improvement gained by such seats is down to about 3-5%. This is by no means a small gain when you consider it has been brought about solely by a little reshaping. The basic problem remains as to how to reduce turbulence as much as possible, at least until the incoming gases are well and truly in the cylinder. Fig. 39 shows a chamber wall section which goes a long way to solving the problem. Virtually half the turbulence is eradicated until the bases are well within the cylinder. Tests on such a wall contour indicate gains as much as 6-8% on a normal engine

when compared with a flat chamber roof. Add this to the gains achieved by venturi ports and radius seats and we could well be 11 % up on a conventional headed engine.

Down-draughting

Down-draughting the inlet port should also produce a worthwhile increase in power. This could be in the order of 3-4% when used with suitable carburation or fuel injection. If we combine all these ideas, the head should look like Fig. 40.

So far everything looks quite good on paper, but what about the problems of making such a head. Down-draughting the head is reasonably straightforward for any well equipped tuning establishment. It is just a case of setting up the head in the correct position and boring a large hole in it. Having done this, a suitable size of

pipe together with some sealing compound is pressed into the hole. The valve guide locating bore is remachined to remove the obstruction at one end, caused by fitting the pipe, a bit of port blending is done and we have a down-draught head. Doing the venturi type radius valve seats is straightforward enough even if it is a delicate operation. This just leaves us with the chambers.

To get the chambers to the shape we require, it is obvious that metal will have to be put into the chambers which is, to say the least, a bit beyond the scope of most home tuners. Fortunately there are firms who specialize in welding cast iron so the problem is simply alleviated. To fill the chambers, the whole head has to be heated red hot so that stress cracks due to temperature differentials are avoided. A quick look at the relevant drawing will reveal that it is not nec-

essary to completely fill the chambers but only build up the areas around and between the valves. We will obviously have to build up more than we want to so that machining and grinding can be done to bring it back to the correct size. Another problem is going to confront us in as much as filling the chambers is going to raise the compression ratio a little higher than is really desirable. The answer to this is to use the head from the 2.5 litre engine which is identical in every respect bar chamber depth. Having filled and modified the chamber, we will find that the C/R is at a reasonable level and can be brought up to spec. by a little head skimming. With this type of chamber design C/R in the order of 12.5 to 1 would be beneficial.

With such a head, probably the best sequence in which to modify the head is: (a) weld up the

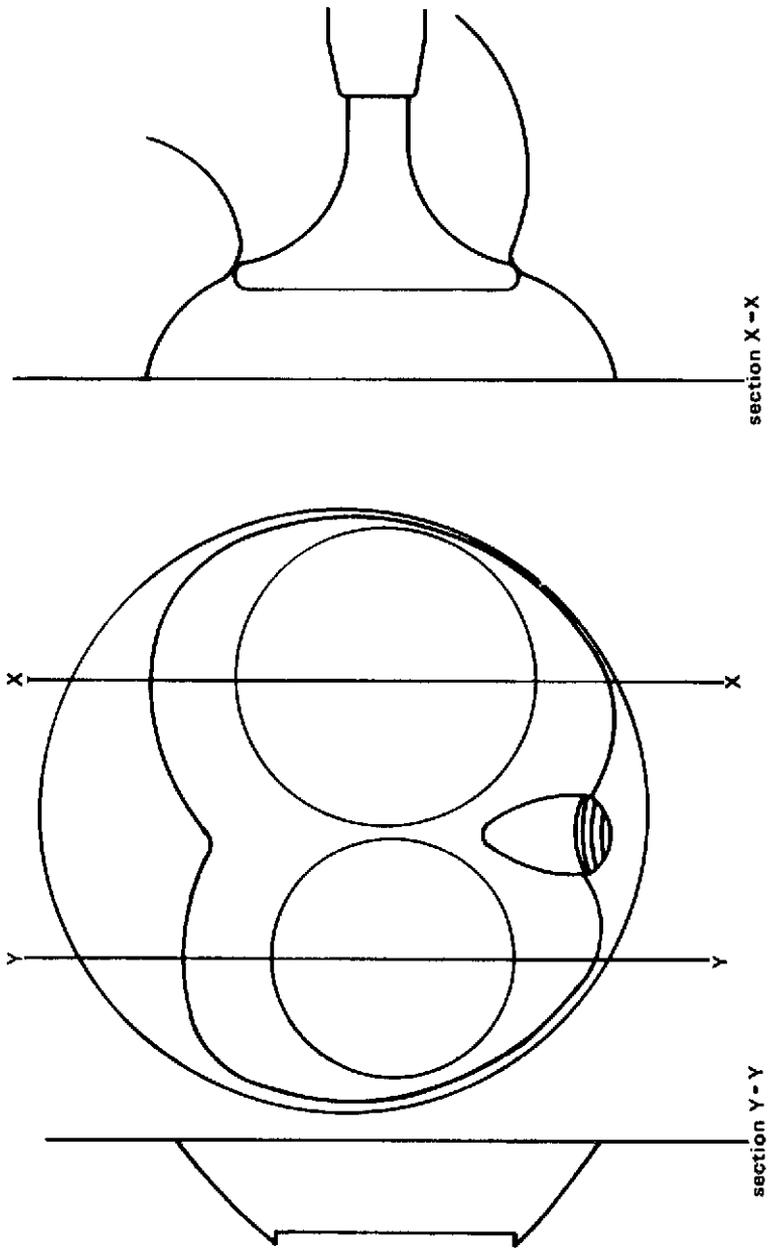


Fig. 40. Chamber design for full race head.

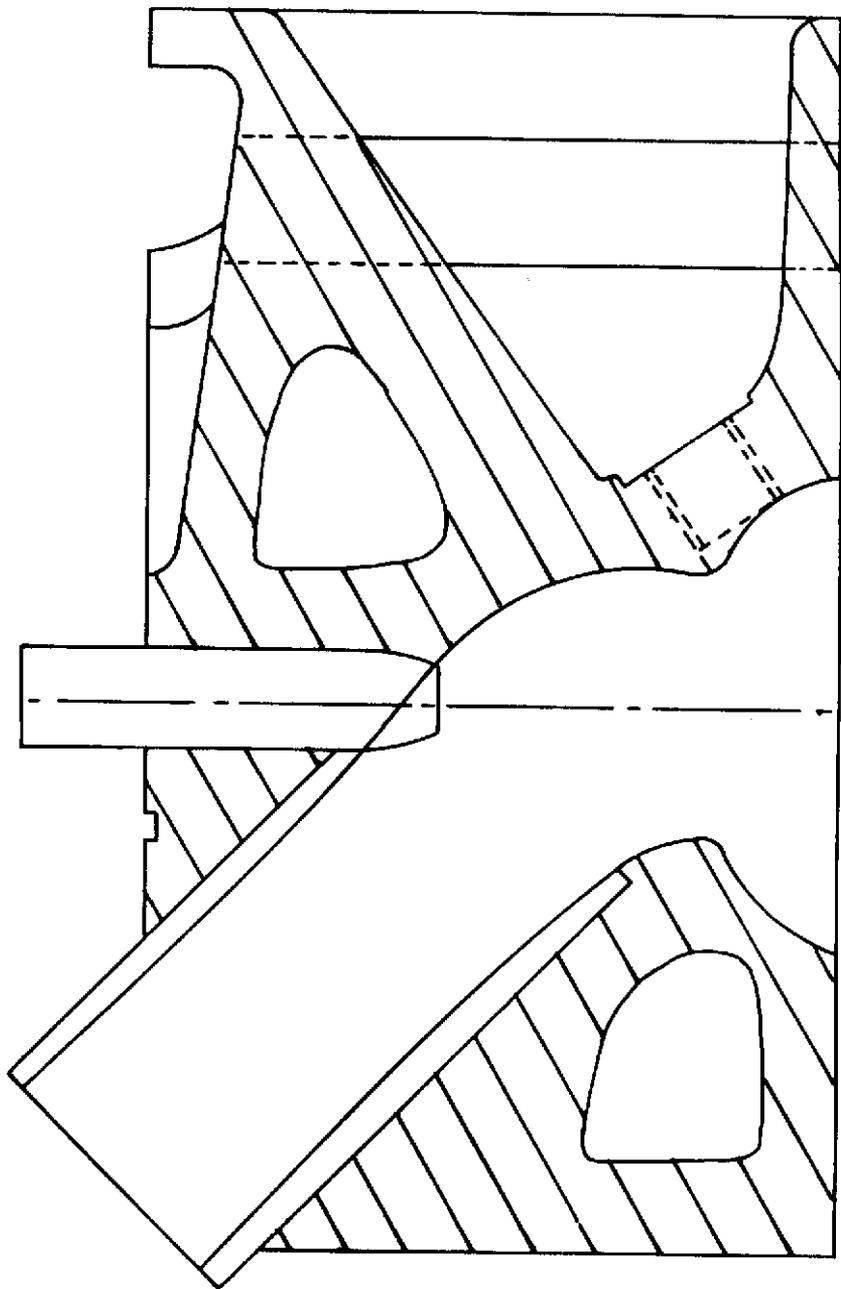


Fig. 40. Scheme for downdraught full race head.

chambers, (b) fit valve seat inserts, (c) down-draught the head and, (d) grind and polish ports and chambers.

Using this type of chamber design together with a down-draught port, a slightly tulip shaped inlet valve is likely to give us better results. The shaping of the inlet valve can have a greater influence on gas flow than is commonly suspected. Tests have indicated that there is a family relationship between chamber design and valve head profile. This relationship can be used only to give us a guide to

the valve profile as it is a little too vague to give us any concrete figures on which to base the valve profile.

This head design should go a long way to giving us the power output required of our hypothetical 2-litre Triumph racing machine. By itself, it is probably not enough to make a certain race winner. There are fortunately other areas in which substantial improvements can be made and we shall discuss these in the relevant Chapters.

PORT MATCHING AND DOWELLING

TO many people it seems that port matching is something of a mystery. It is in fact quite a simple job that needs just a little care and patience.

The simplest method, ensuring a reasonable match between the port and manifold is to match both items" to the manifold gasket. To do this, both the manifold head joint faces are cleaned and marking blue applied to each face. Two manifold studs are then fitted into the head, one at each end, and the manifold gasket located on them. We then scribe round the port apertures onto the head thus giving us a clear indication of the position the gasket will take up. Sometimes the gaskets have large clearance holes for the studs, which means that the gasket is rather a sloppy fit on the studs. This can be remedied by winding a few turns of masking tape around each of the studs, thus giving us a more stable location.

Once the head has been marked out, we set about doing the same thing to the manifold. By using some nuts and bolts the same size as the securing studs or bolts in the head, the gasket can be secured to the

manifold and the port apertures marked out. From here it is a straightforward job to cut away the metal on both the manifold and gasket up to the previously scribed line.

We have, of course, made the assumption that the ports in the gaskets are the same size as the ports required in the head. Most manifold gaskets are about $3/32$ " larger than the port sizes which in all the cases we have covered remain standard size as far as the inlet are concerned. This means that ideally the line that is scribed for working to must be $3/64$ " in from the edge of the gasket. The port size at the joint face between the manifold and head is not critical within $1/64$ " so when we say the port should not be enlarged, it should not be enlarged any more than is necessary to match it up. This method of port matching is the simplest but the accuracy obtained is limited by clearance on the manifold location studs. This clearance allows a certain amount of movement which directly affects the lining up of the ports. Although

this method is not super-accurate, it does leave things far better than they are as standard.

Dowelling

For a super-accurate port matching job, we have to be a little more sophisticated about things. Firstly we shall require some 1/8" dowel pins. These can be made up from a length of 1/8" dia. silver steel as this is very accurately sized. Silver steel incidentally, is available through most concerns dealing in engineers' tools and is commonly sold in 13" lengths. From the silver steel we make up some dowel pins about 3/8" - 7/16" long. These should then have their ends chamfered or radiused. The number of dowels required will depend on the type of manifold being fitted. A one-piece manifold will require two dowels; a two-piece manifold will require four dowels; each part of the manifold requiring two dowels to locate it.

The next step is to fit the manifold Studs in the head and bolt the manifold in place with two or three gaskets between the head and manifold. Be sure to line all the gaskets up with one another. To take up any play on the studs we can, as before, use masking tape. With the manifold bolted securely in place, drill two holes in a convenient position in each component part of the manifold. Normally the best place to drill such holes is about 1/4" away from the fixing studs. The size of drill for the hole is 0.110" and the drilling should go about 3/8" into the head casting after passing

through the manifold and gaskets. Do not remove the manifold at this stage, drill all the holes that are needed in one go. Next, using a worn 1/8" reamer, one that is about 0.001" undersize, ream out the holes through the manifold into the head. Incidentally you may find that in some cases i.e. Weber manifolds, that the drill or reamer length is insufficient to reach the base of the manifold without fouling some other part of the manifold with the drill chuck. This situation can usually be overcome by fitting the drill or reamer into a slim pin chuck and then holding the pin chuck in the drill.

Once we have reamed the dowel holes with the undersize reamer, we can remove the manifolds from the head. We now fit the dowel pins into the head. Providing you selected your reamer at 0.001" undersize, the dowels will be a light drive fit in the head. It is not a good idea to have the manifolds a light drive fit on the dowels, so using a new 1/8" reamer, ream out the manifold holes. If the job has been done right, we now find that the manifolds are a snug push fit on the dowels and there is a complete absence of sloppiness.

Since we now have a really accurate location for our manifolds and gaskets we can tackle the port matching procedure as described previously: The dowels will, however, take the place of the studs when marking out the ports to their relevant gasket apertures. The adoption of this method enables port matching to within about 0.007" when done carefully and that cannot be bad.

SPRINGS, ROCKERS, PUSHRODS AND TAPPETS

THE sole function of springs, rockers, pushrods and tappets is to convey the motion imparted by the cam to the valves. As such the valve train, including the camshaft, must be regarded as a whole. Any mechanism is bound to suffer from flexure of various components. Any flexing which is brought about by the relevant dynamic or static loads imposed, makes the job of the camshaft that much more difficult. The cam designer goes to a great deal of time and trouble to produce a cam profile which will minimize the effects of vibration, shocks and flexing in the valve train. Because of the nature of the valve train in the normal O.H.V. engine, he is fighting a one-sided battle. We must consider, when making any modifications to the valve train as a whole, whether or not we are making the situation worse as far as loads and flexure factors are concerned. The first golden rule to be applied is: Never use valve springs any stronger than are needed to do the job. If you do, it will cost you power. You must consider at what r.p.m. your peak power is likely to occur, and then select valve springs which will allow

the engine to run 10-15% over these revs. Since the camshaft is the main influencing factor controlling the point at which peak power occurs, then we should select valve springs and cam together.

If substantial improvements have been made in the breathing ability of an engine, but the cam remains standard, the situation can sometimes call for slightly stronger valve springs. This means that we cannot use the principle, at least in early stages of tune, of selecting cam and springs together, as a hard and fast rule. However, since we are dealing solely with the larger Triumph engines, we can be far more specific about things. All the Triumph cars with which we are dealing have sufficient spring pressure to control valve bounce up to high enough revs when using the standard camshaft. This even applies to engines which have had substantial improvements made in the head, inlet and exhaust side of things. Sometimes one can gain a false impression in as much as the engine feels as though it can usefully rev higher,

but quite frankly one is off the best part of the power curve and you would be better off in the next gear up. All the sixcylinder Triumph engines can rev to 6,300 or over on standard springs. Without making a cam change, it is unlikely that the point at which peak power occurs can be raised to much more than 5,800 r.p.m. This means that fitting stronger springs with a standard cam is a waste of time and serves only to reduce the power output by raising the internal friction level of the engine.

Cams and springs

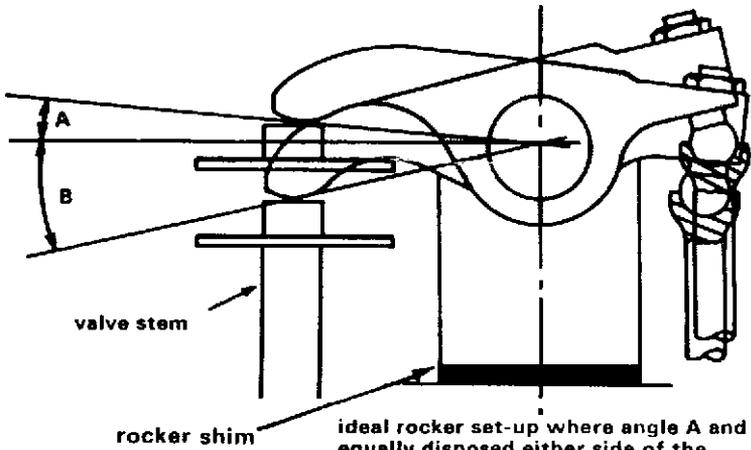
When we do change to a hotter cam, then we shall definitely need springs to go with it. Running the engine to valve bounce revs is a practice to be avoided as much as too strong a spring set-up. Valve bounce occurs when the spring pressure is insufficient to keep the cam follower in contact with the cam at the revs prevailing. This means the cam follower may leave contact with the cam just after the toe-peak lift point-of the cam and may not touch again until the heelbase circle. This sort of use imposes terrific shock loads on the components of the valve train. It can lead to premature demise of the cam followers and cam lobes. While the tappet is off the cam follower, the valve is being brought down onto its seat solely by spring pressure. Instead of being gently let down into its seat by the closing ramp of the cam, it is smashed down by the full force and speed that the spring can muster. The seat, valve and spring are not good enough shock absorbers to absorb the full energy of the rapidly moving valve, so the valve promptly bounces back off the seat. The loads and speeds involved are so high that the valve may bounce off the seat two or three times before it finally comes to rest ready to go through the next

cycle.

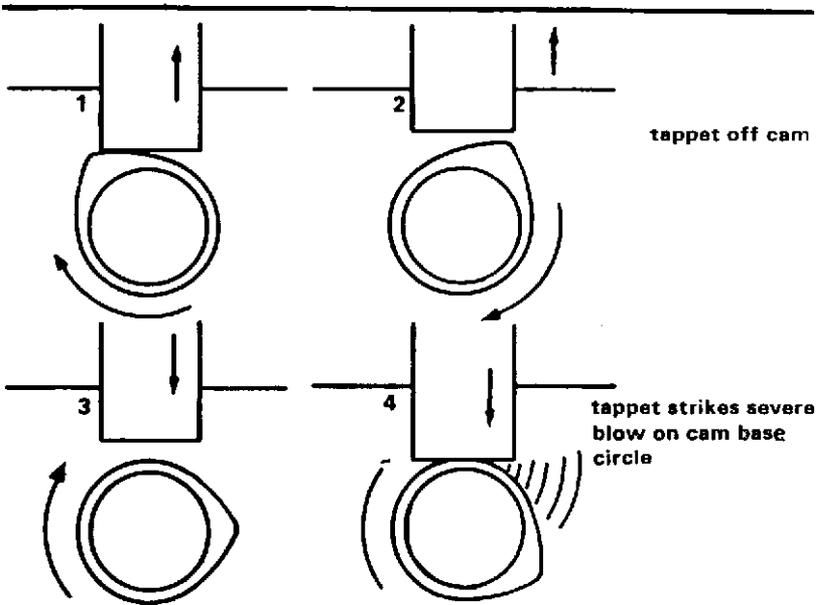
All this explanation is leading up to one thing. When buying a cam, consult the cam manufacturer as to what springs should be used. Let me quote an example. S.A. H. do five different cam grinds for the six cylinder engines, and each cam profile has a specific set of springs to go with it. The same also applies for the two cams S.A.H. do for the four cylinder engines.

Having lightly touched on the subject of springs, let us turn to the rockers. After a head has been skimmed, one can sometimes find that there is insufficient rocker adjustment left. There are two ways of overcoming this problem. We can either shorten the push rods by the same or similar amount taken off the head or raise the rocker shaft to compensate. I personally am not in favour of the second method, although it is the simplest means of solving the problem. Ideally the rocker should be horizontal when the valve is at the half lift point i.e. the rocker will swing an even amount above and below the horizontal. If we skim the head then, assuming at this point there is sufficient rocker adjustment, we will reduce the angle the rocker makes to the horizontal while the valve is still closed. When the valve opens the rocker will pass through the horizontal position before the half lift point is reached. This means the total angular displacement from the horizontal is increased. Let us go back to the situation where packing shims under the rockers is required to allow adjustment. We find when the shims are fitted, we reduce the initial rocker angle even more than with just the skimmed head. The rocker reaches the horizontal position almost as soon as the valve is

The pushrod can be considered as the weakest link of the chain in most O.H.V. engines. This is not from the point of view of strength,



ideal rocker set-up where angle A and B are equally disposed either side of the horizontal. If head is skimmed, shorter push rods will maintain the situation angles A and B become unevenly disposed either side of the horizontal when rocker shims are used



off the seat, instead of the idea point at half full lift. Also the maximum angle from the horizontal has been increased and this is undesirable for several reasons. Firstly the rubbing motion between the tip of the valve stem and the contacting rocker face is increased which in turn increases the wear rate. From this stems the fact that tappet clearances are likely to need more regular adjustment. Secondly the side forces on the valve stems causes a greater rate of valve stem and guide wear. Thirdly, any increase in friction is undesirable in any engine, especially one intended for high performance. We need not run away with the idea that shimming the rockers is totally disastrous all the points raised merely constitute small but unwanted side effects.

Unlike shimming, shortening of the pushrods does not have any unwanted side effects. By shortening the pushrod, we allow the rocker to contact the valve stem through the optimum angles i.e. an even angle of swing above and below the horizontal. The horizontal position will, of course, coincide with the half full lift valve. The shortening of the pushrods has also given us a small reduction in valve gear weight, and every little bit helps. The pushrods ideally should be shortened by the same amount of metal that has been taken off the head thus restoring the condition which prevailed originally. When the pushrods are shortened, the original form must be remachined on the end and we will, of course, shorten from the cam follower end and not the rocker end component which receives continual abuse from the camshaft, one should check that (a) it is a good fit in its bore and (b) that the face but from the point

of view of flexibility. The pushrod is simply a load bearing column. As such, it functions reasonably well, while it is dead straight. If there is the slightest bend in the pushrod the flexing it experiences during use will go up out of all proportion and the motion of the valve will bear only a vague resemblance to the motion dictated by the cam. As you have probably gathered by now, the moral is to check each pushrod for straightness.

Tubular pushrods

Although none are commercially available at the time of writing, I it would appear that there is scope for the use of tubular pushrods. It would appear that we have ample room down the pushrod holes to accommodate tubular pushrods approximately twice the diameter of the standard items. This is the sort of thing needed for our, as yet , hypothetical racer. Since a tubular pushrod of similar cross-sectional area would be far stiffer than the standard items, we could usefully employ a higher flank acceleration on the cam profile. This faster rate of lift would enable better breathing and hence more power.

Consideration of the tappets or cam followers brings us to the last component in the valve train before we get to the camshaft. The camshaft will be dealt with in a chapter of its own as it deserves more than a passing mention. There is, however, little that can be said about tappets except that they are hard working components and should be treated with respect. Since it is a which bears on the cam is not pitted or concave. If the face is at all concave, then the valve opening motion is adversely affected.

ENGINE INTERCHANGEABILITY

DROPPING in a larger motor is a simple and safe method of increasing a car's performance, but it is difficult to try to put forward every conceivable combination of bits and nearly impossible to foresee every snag that is likely to arise. However the information here should give either food for thought or a good guide as to what should or can be done with the least complications.

Vitesse 1600

We will deal with the 1600 Vitesse first as this is the most likely one to receive attention in the form of capacity increases. There appears to be no great problems which is as one would expect, in bolting the Mk. 1 2-litre Vitesse engine straight in. On the other hand fitting the Triumph 2000 engine is not quite so straightforward. The principal difference in the block arises in connection with the engine mounting points. The Mk. 1 Vitesse engines have, cast on the left hand side when viewed from the front, a large boss projecting some 2-1/2" from the main body of the block. This boss has four drilled and tapped holes

in it, one at each corner, and is the point at which the Vitesse engine mount fits.

Triumph 2000

The Triumph 2000 engine mounts are situated on the front of the engine and are in the form of a plate between the timing chain cover and the block. The early Triumph 2000 block, unlike the Vitesse block, does not have such a large boss on the side of the block. The part of the boss present has no threaded holes to permit securing of the engine mount. This means that if you should get hold of a Mk. 1 Triumph 2000 engine, you will have to (a) drill and tap the block and (b) make a metal spacer of approximately 1-1/4" thickness to make up the space between the block boss and the Vitesse engine mount. Since the boss on the block is much smaller, we must be very careful not to drill and tap the block too deeply, otherwise we shall break through into the inside.

If we resort to a later engine, that is after late 1966, then we find that

some rationalization was brought about at the factory and all blocks would appear to be the same. This means that by using the later engine, we can, without a great deal of effort, transplant any engine into any car within the range. There may be occasions when parts outside of the engine need changing to retain parts compatibility, but there does not appear to be occasion to use bits other than off the shelf Triumph parts. The biggest expense when fitting the late engine into a pre '66 car other than the engine itself, would appear to be some new manifolding because, you will remember, the later engine has a different head.

If you have the Mk. II G.T.6 or Vitesse, then it becomes unnecessary to change the whole engine to get the capacity up to 2.5 litres. The later 2-litre engines have the bigger mains journals which are, in fact, the same size as those on the TR 5 or 6. This means that we need only change the crank rods and head for those of the bigger engine.

Incidentally, whilst we are on the

subject of blocks, the Triumph 2000 engine is only a bored out version of the 1600 Vitesse engine. When the 2000 block was brought out, the bores were cast to allow for the bigger size employed. If you should try boring the 1600 out to 2000 c.c. you will just end up with a scrap block, so do not try it.

Since the engines are, in the main, bored or stroked versions of each other, we get virtually no problems associated with weight. Usually dropping in a bigger engine has certain disadvantages connected with it, such as making the car front end heavy. Here the problem does not arise since all the engines have weights within a few pounds of each other. At face value, increasing the engine size is a good way to get more performance. The cost of such a project, however, is a little on the high side in terms of b.h.p. gained per pound spent, assuming new parts are purchased. To counter this, any subsequent improvements made, result in a greater power increase on the larger engines than the smaller ones.

STRENGTHENING THE BOTTOM END FOR COMPETITION

THERE is very little we can do to increase the bottom end strength on these engines as far as component changing goes. This means that we have to resort to methods which will give the greatest reliability using the standard parts.

Tuftriding

To start with, it is a good idea, if we are to use the maximum rev potential available, to have the crank and rods tuftrided. For a very reasonable price you can get your own rods and crank tuftrided. This process has the effect, as you have probably guessed, of increasing the toughness of a steel component. By doing so we find that it is less likely to break through fatigue at sustained high revs. It also enables us to raise the rev limit before catastrophe occurs, by 5-7%. After components have been tuftrided they are covered in a grey film. This must be removed from all bearing and locating surfaces by polishing. Some "600" grit wet and dry emery paper will do the trick nicely, but be sure to remove every trace of grit before using the component.

Shot peening

Another way in which we can increase the fatigue life and the ultimate strength is by polishing or shot peening. The surface flaws and Imperfections reduce the strength of a component by a fantastic amount. If we could remove every surface flaw on a steel component down to the molecular level, the strength of the component almost doubles. Unfortunately removing flaws down to the molecular level is virtually impossible. Even if we could do so, the bombardment of air molecules is sufficient to render our perfect finish imperfect, but make no mistake about it, the polishing of the engine's vitals does make a worthwhile difference, even if the finish is imperfect. A well polished or shot peened rod can have a fatigue life easily double of that of a standard component. This is not to say that its strength is doubled. The difference between breaking and not breaking due to fatigue can be quite small. The polishing or shot peening may have only given us a 5% margin strengthwise over the standard item, but at a given r.p.m. it is enough to

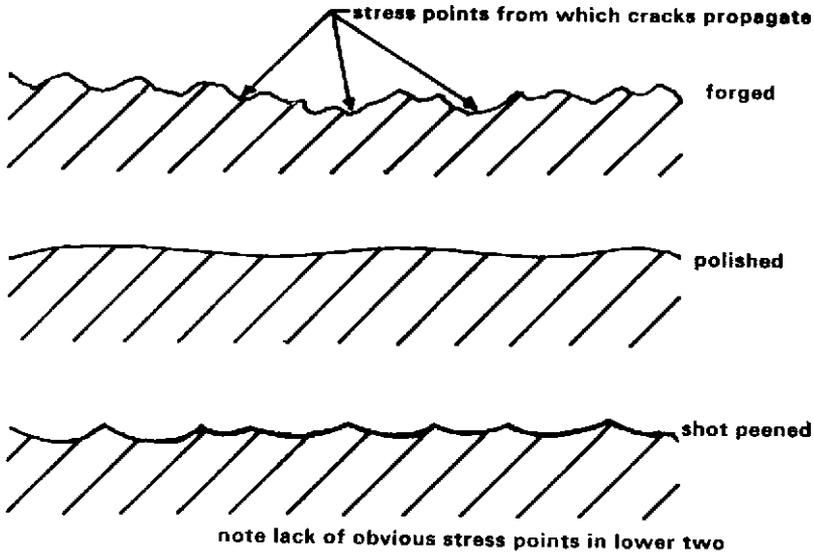


Fig. 41. Different types of surface finishes under magnification.

make the difference.

I should think we all know how polishing is done, but far less is known about shot peening. An object is shot peened by firing small glass or steel balls at the component. These balls have a relatively high surface finish. They strike the component at a high enough speed to cause a small curved indentation and in doing so, impart some of their high surface finish onto the component. This has the effect of eradicating the crevices which cause stress points. Fig. 41 shows the surface finish under magnifications of (a) a forged surface (b) a polished surface and (c) a shot peened surface. From this you can see that the shot peened surface has done as much to remove stress points as polishing, although the actual appearance of the finish is somewhat different. It is advisable to give the components a little attention before

they are shot peened. For instance one should remove all forging marks and try as much as possible to smooth out the surface all over.

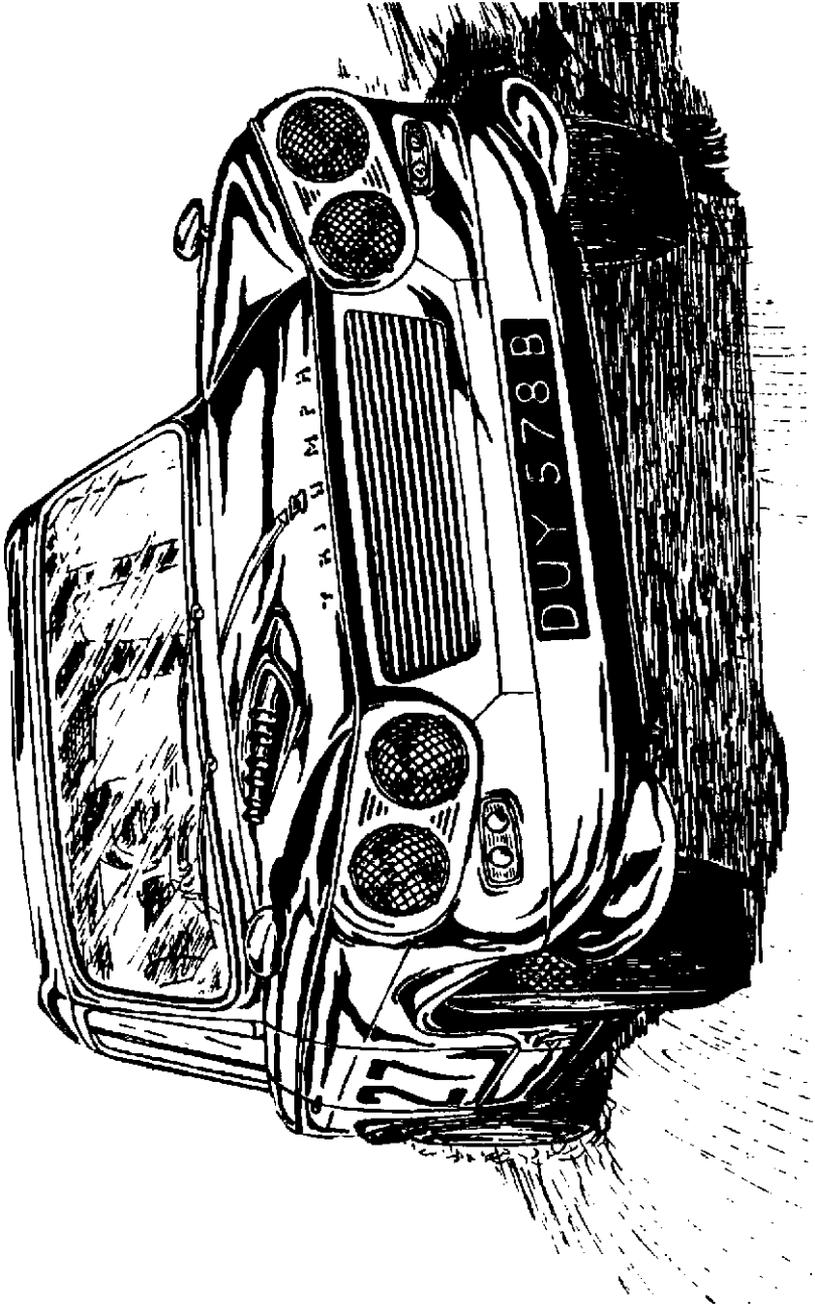
If we are going the whole hog by tuftriding and surface finishing, be it polishing or shot peening, things have to be done in a certain order. The component must be prepared prior to tuftriding by grinding off any forging marks and generally cleaning it up. If the final surface operation is polishing, the rod is best polished before tuftriding as after the processing we must avoid any metal removal. If the component is to be subsequently shot peened, then we need not be so fussy over the finish prior to tuftriding.

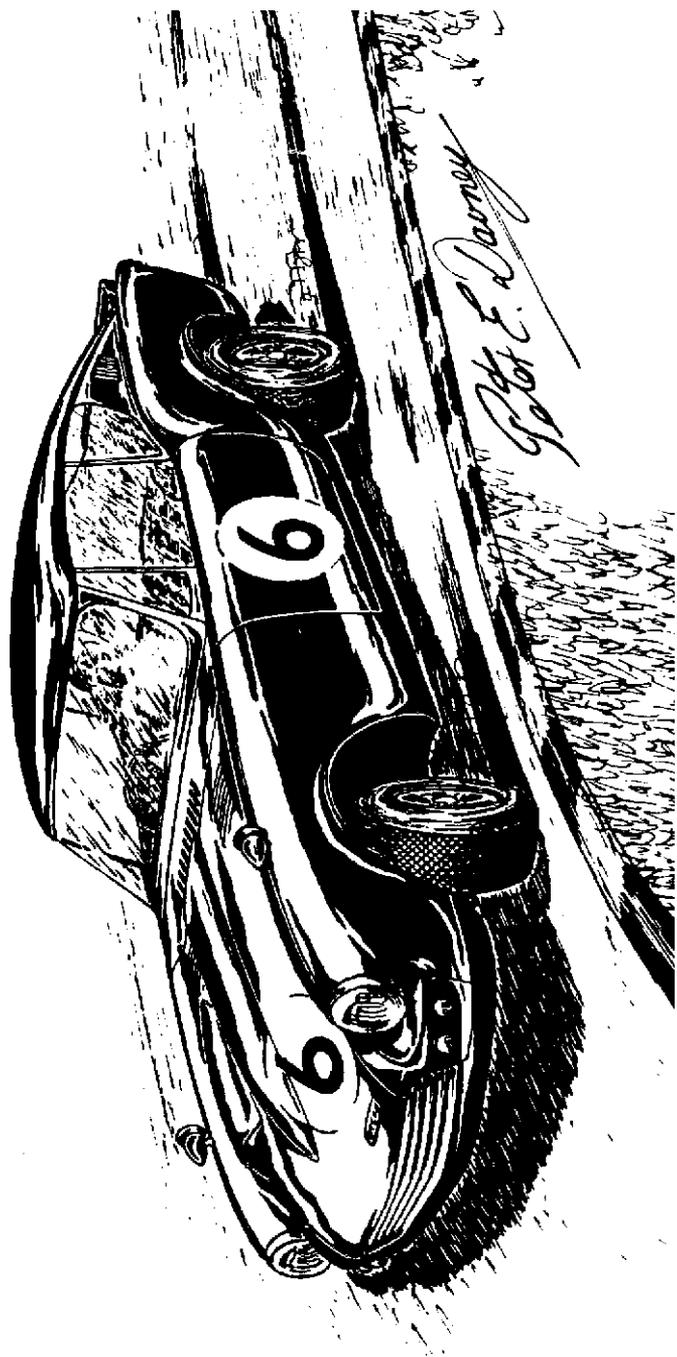
A word of warning here would not be out of place. Do not send any bits for tuftriding if at any time they have been bent and had to be straightened. The toughening process will cause them to revert back to their bent state.

After toughening, all components should be checked to see that they have not distorted as this sometimes occurs. Small amounts of distortion can be remedied by normal straightening procedures, but if too badly distorted, the relevant component should be replaced, by another toughened one. The little end bushes in the rods are always badly eroded by tuffriding and will need replacement after processing.

With the bottom ends fully prepared, we should get rev limits in the order of 8,300 for the 2-litre Mk. 11 sixes, and about 7,200 for the TR4

and 4A engines. Due consideration should be given to the 2.5 litre six, because of the longer stroke, and it is probably wise to limit them to 8,000 or just under. The early 1600 and 2-litre engines, because of their small main bearings, should not be run in excess of 8,000 revs. There are engines of similar size that will rev higher than this in safety but quite frankly these Triumph engines have enough revs for our purposes. To get a peak power which needed more than 8,000 r.p.m. we should have to resort to some very sophisticated tuning indeed!







LIGHTENING ENGINE COMPONENTS

The con-rods of an engine are usually the first items to get attention when the time comes for lightening. In the case of the Triumph rods, there is little we can do to vastly reduce the weight. Slight weight reductions will, of course, be achieved by grinding off the forging marks and generally rounding off square edges. Any weight reduction must be considered worthwhile if we are going all out on the engine, but for all normal applications it is not worth the great deal of effort involved.

Some of the Triumph pistons can be lightened by cutting away part of the skirt as in Fig. 42. One can easily identify which can be lightened and which cannot by the fact that some pistons already have this part of the skirt cut away. Those that are already cut away should be left as they are. Do not shorten the pistons as with the bore sizes quoted elsewhere in the book, the pistons will be allowed to rock too much in the bores.

Rockers

The rockers for the "sixes" can use-

fully receive some attention to reduce their moving mass. These should be reshaped as in Fig. 43 and finally polished to reduce the effects of surface stress points. When grinding, one must avoid touching the surface of the rocker which bears on the stem of the valve. Any deviation of the true curve caused by wear or otherwise, results in an error of valve motion. The valve rockers for the four cylinder engine cannot effectively be lightened any more than they are, but a good polishing session on them would certainly not go amiss.

Cam followers

The cam followers can be lightened by machining out the bore as shown in Fig. 44. The size to bore the cam followers-dimension A-varies depending on the size of cam followers in the engine. The 0.687" dia. tappet usually found in the 1600 Vitesse and the Mk. 1 2-litre engines can be bored to 0.590" dia. The larger tappet found in the later engines is 0.800" dia., and can be bored to 0.700" dia. The ones for the four cylinder

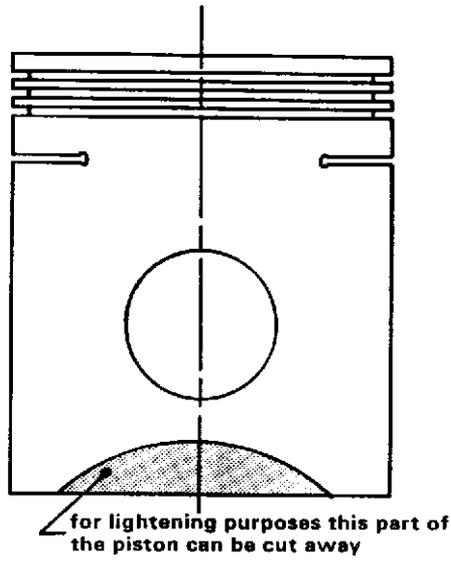


Fig. 42.

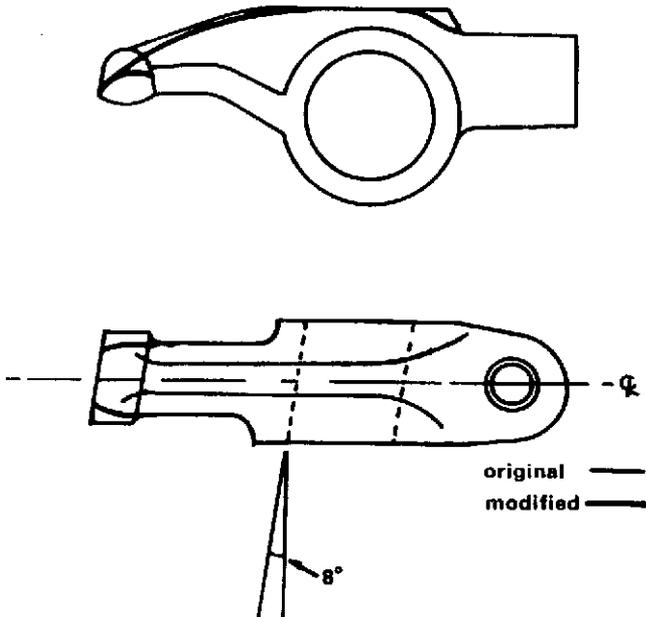


Fig. 43. Rocker lightening for 6 cylinder engines.

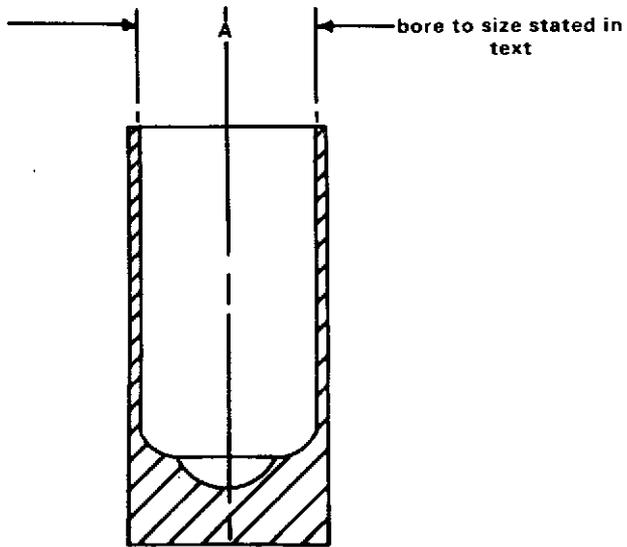
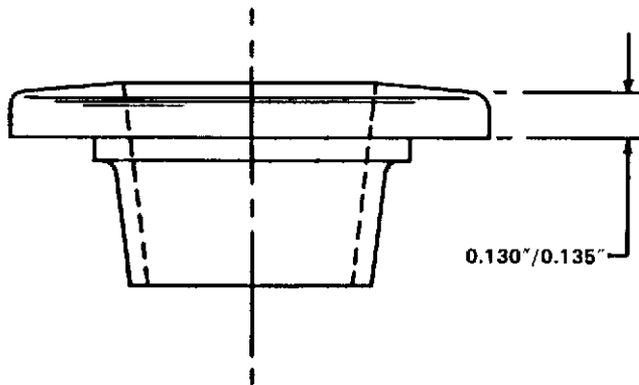


Fig. 44. Lightening cam followers.



all dimensions except those shown are as per standard

Fig. 45.

engines which are larger at 0.937" dia., can be lightened by boring to 0.840". The boring of the cam followers to these sizes will give a weight reduction in the order of 20%.

Another worthwhile weight saving can be had by reducing the thickness of the locking nut on the rocker adjusting screw to half of its original thickness. This usually results in far more of the rocker adjusting screw protruding from the locknut than is necessary. Once the tappets have been adjusted, we can see exactly how much of the rocker screw is ex-

cess. It is then shortened by an amount which leaves enough to put on a plain shank and a screwdriver slot. The advantage of lightening performed on the end of a rocker as opposed to lightening near the pivot point, is that it has the greatest effect on the reduction of reciprocating weight and is therefore most useful. Although it does not come into the category of lightening of standard components, the making up of aluminium spring retainers should be mentioned. By using aluminium instead of steel, we can cut the spring

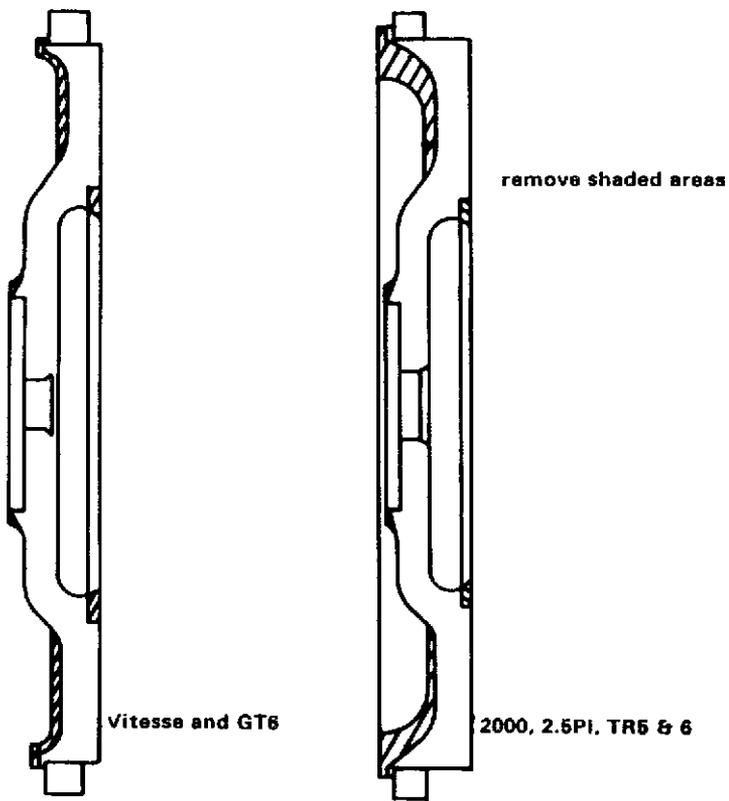


Fig. 46. Flywheel lightening.

retainer weight by 50% which is, you must admit, a fair amount. We cannot make the aluminium retainers, size for size, the same as the steel ones, as they would not be quite strong enough. To compensate for the lower ultimate strength of the aluminium, we should make the retainers thicker on the face that the outer spring beats against see Fig. 45. Only good quality high tensile aluminium alloy such L65 should be used for these retainers, this has a strength in excess of 25 tons per square inch.

Flywheel

The standard flywheel is, on the six cylinder engines, good for giving a smooth tickover, but for a tuned motor it is unnecessarily heavy. The four cylinder engines also have a heavy flywheel. For these engines, you can have one of two different flywheels depending on the type of clutch fitted, the one being heavier by far than the other. To lighten the flywheel, it is simply mounted in a large lathe and machined as shown

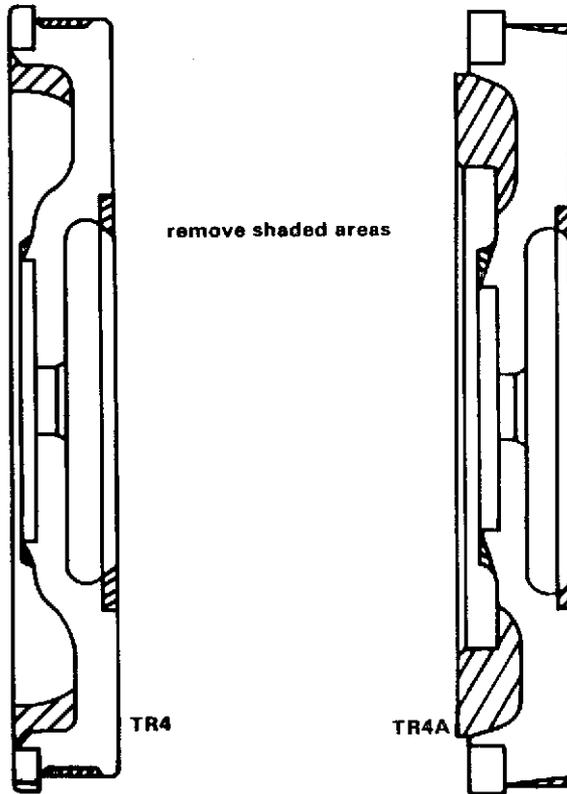


Fig. 46. Flywheel lightening.

for the relevant flywheel in Fig. 46. Practically any motor machine shop has a lathe large enough to tackle such a job, so there should be no problem getting it done. It must be pointed out that a lighter flywheel neither increases the power nor makes the car faster. It does, however, allow the car to accelerate faster because of the reduced mass which the engine has to speed up. The lower gear the car is in the greater the effect of the reduced flywheel weight. To clarify the point a little, let us look at a simple example.

Let us assume that we have reduced the effective weight of a flywheel by 10 lb. While the car is in bottom gear the engine r.p.m. to driving wheel r.p.m. is 16 to 1 i.e. the engine turns 16 revs to the wheel's one

rev. The 10 lb. flywheel weight reduction is equivalent to reducing the weight of the car by 160 lb. that is 16×10 lbs. When we change to second gear which, we will say, is about 12 to 1 overall ratio, the gain, because of the lighter flywheel, will be 12×10 lbs. which is 120 lbs. By the time we get to top gear, the effect of the lighter flywheel will only be about the same as lightening the whole car by 4×10 lbs. or 40 lbs. With the exception of one of the TR4 flywheels the effective weight saving will not be as great as 10 lbs. A more likely figure is between 6-8 lbs., but this is enough to make a noticeable difference. By way of a bonus, the lighter flywheel also enables snappy gear changing when going down the box.

HIGH PERFORMANCE CAMS

THE camshaft is a component which is often taken too much for granted. The designing of a good camshaft is probably one of the most difficult things within the internal combustion engine. For a camshaft to have a reasonable life, yet produce the required breathing characteristics for a high performance engine, the cam profile has to be carefully calculated. The profile must be such as to limit shocks and vibrations, and keep the inertia loads to a bearable level. It is only by careful control and accurate manufacture of the cam profile that the designer can come to terms and limit the undesirable elements of the cam. For this reason we should never use a cam with worn lobes as we could be imposing stresses or shocks far higher than were ever intended.

How does one go about choosing a suitable cam? Firstly we must decide at what point we need the power. For fast road work and town driving, we should use a cam that can run down fairly low revs and still give us power. This means the timing is of fairly short duration and there is only a small amount of overlap. If your

town driving is of a limited nature and the car is mostly used on open roads, we can afford to lose a little at the bottom end of the rev range to gain power at the top end. For this application we can go to a more hairy cam but still one that retains a sensible amount of flexibility. For rallying, autocrossing or for the man who has to use the daily road transport for the weekend race meeting, we can pull the stops out a fair bit and install a cam which will go between 2,500 up to 6,500 or so. For circuit work a cam giving power between 3,000 to 7,000 would be a good choice.

Not only do we have to consider the application of the car when selecting a cam, but also we must consider whether it is reasonably compatible with the rest of the engine modifications. For instance, a full race cam would be a bit of a waste of time on a 1600 Vitesse, if the standard Solex carbs were still retained. Before fitting a cam that is vastly different to the standard one, you should definitely reach a reasonable level of sophistication in other

quarters such as head, carbs and exhaust system.

With the TR4 and 4A, we cannot go too wild on cam timing because of a rev. limit, which even with a fully prepared bottom end, is only a little over 7,000 strictly speaking, we cannot really use a cam that gives a peak power any higher than about 6,500 r.p.m. This means that there is not such a vast difference between the standard cam and a full race one as there is with some cars. Since the step from standard to full race is a little less than usual, there are less cams to fill the gap between road and race profiles which means our choice is limited. Because of the relatively low rev. limit, most cam manufacturers produce only one or two profiles -from which to choose for this engine. The standard TR4 and 4A cam timing is a fairly modest 17/57-57/17 with a lift of around Y.". For better breathing up

at the top end without any appreciable drop in the lower rev range, the S.A.H. 26 cam can be fitted. This has slightly more duration than the standard cam, the timing figures being 22/62-62/22. Total. valve lift is up to 0.432". This combination of lift and timing makes the cam ideal for fast road work and the occasional competition meeting. A cam which does a very similar job to the S.A.H. item is made by Piper Cams. The Piper cam has slightly longer timing but less lift, the figures for these being 24/64-64/24 with 0.387" lift. V. W. Derrington can supply a cam which they say is designed to give the best torque characteristics in view of the relatively low maximum safe revs (7,000) of this engine. On the face of it, this cam should represent a good bet for the man who uses his race machine on the road, as tractability does not suffer too much. The timing for the Derrington cam is 30/70-

Manufacturer	Timing	Lift	No.
Piper	30/70-70/30	0.435'	1
	35/75-75/35	0.390"	2
	36/76-76/36	0.406"	3
	45/85-85/45	0.435"	4
	33/71-71/31	0.406 "	5
S.A.H.	(26) 22/62-62/22	0.388"	6
	(46) 40/60-60/40	0.398"	7
	(57) 50/70-70/50	0.425"	8
	(47) 40/70-70/40	0.428"	9
	(357) 33/71-71/33	0.395"	10
	(387) 38/72-75/35	0.382"	11
	(578) 52/78-78/52	0.410"	12
	(467) 46/74-78/42	0.385"	13
V W Derrington	30/70-70/30	0.435"	14
	35/75-75/35	0.390"	15

70/30 with a lift of 0.420". Pipers also do a cam with similar timing and lift to this one. Any cam past this stage definitely starts to chop out the low end performance, but if you intend to use the car for any form of serious competition, the revs must be sacrificed.

Racing

For racing we must go for the S.A.H. 47 cam (43/76-76/43 with a 0.480" lift). This cam must be considered as a full race cam, and if used in a road car, can be just a little tiresome, especially in traffic. The useable rev

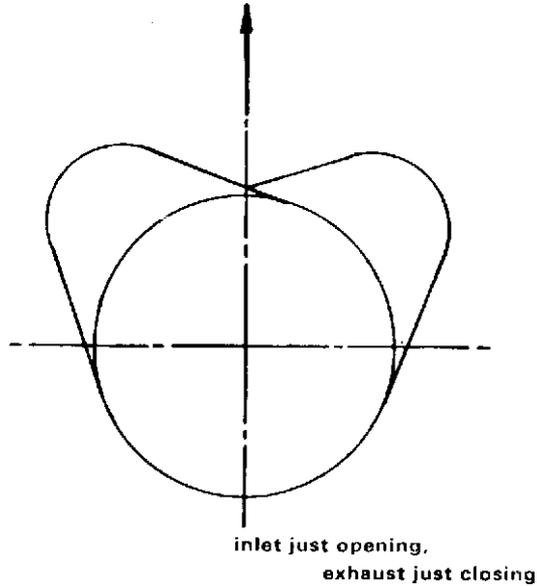


Fig. 47.

range with this cam is approximately between 3,000 and 7,000 which is the limit as far as the 4 cylinder Triumph engines are concerned.

A far wider range of cams is available for the six cylinder engines than the four cylinder ones. This wider choice allows a closer tailoring of the engine to one's needs. The easiest way to cover these cams is to list a reasonable number of them and then discuss any special merits that they might have.

For a good road cam in a 1600 or Mk. 1, 2000 motor, cam numbers 2, 6 or 15 are a good choice. The timing figures are not too drastic for road use and the lift is quite adequate for the standard valve sizes. We can lift a valve too far and there is no point in going higher than about 0.30" of the diameter of the valve. Since the standard size of valve in a 1600 or Mk. 1

2000 engine is 1.3", a quick calculation will reveal that we need a lift of 0.390". The 0.390" lift given by this cam is therefore close to the mark. It is also interesting to note that this type of cam can be used beneficially in the 2.5 litre Lucas fuel-injected engines without having to change the fuel metering cam. The S.A.H. 357 (No. 10) and 578 (No. 12) and the Piper cam (No. 5) also save this modification, which is very useful when you consider that reprofiling the fuel ram takes a good few hours on a dynamometer. The 357 cam is the sort of cam that starts running right around the 1,500 revs or so, and it is quite roadable. The 578 on the other hand, is a race profile and needs upwards of 3,000 revs just to get going. The best rev range for this cam is between 4,000 and 7,000. Cams number 1 and 14 are

also good touring cams, giving a useful increase in power without killing off a noticeable amount of power at the bottom end. To make the best use of these cams, they should be used in the Mk. 11 engines, or Mk. 1 engine with a bigger inlet valve. Cams numbers 3, 7 and 11 make a good compromise for road/competition use in the 1600, 2000, or carbureted 2.5 litre engines. Since the bigger engines bring the r.p.m., at which the cam comes in down, we can afford to go a little further on cam timing for road use before it becomes too intractable. For instance, as a road/race cam for a carbureted 2.5 litre engine, the 46/74-78/42 cam (No. 13) might be a good choice. A good example of this would be the fitment of this cam into the TR 250 which is the carbureted version of the TR 5. The remainder of the cams, including the S.A. H. 578 cam already mentioned, can be considered full race cams.

Fitting

Once you have decided on a particular cam, then it is imperative that it is fitted correctly. To fit the cam correctly it must be timed by using a degree plate and a dial indicator. If the cam is not correctly timed then we can be throwing a fair amount of power away. Just installing the cam and bringing up the timing marks is, in most cases, not good enough. To ensure that the cam will function correctly, the following procedure should be adopted when fitting it.

Using a dial indicator with its probe resting on top of the number 1 piston, establish T.D.C. When bringing the piston up to T.D.C. turn it in the normal direction of rotation and apply thumb pressure to the top of the piston to take up bearing clearances. In-

sert the camshaft so that the inlet and exhaust cams on number 4 cylinder (for 4 cylinder engines) or number 6 cylinder (for 6 cylinder engines) are rocking i.e. the inlet is just opening and the exhaust is just closing (see Fig. 47). This will mean that the cams in number 1 cylinder will be such that both valves are closed which will, of course, be the situation prevailing when number 1 is firing, and on the power stroke.

The next step is to fit the sprockets together with the timing chain. The degree plate is now bolted to the crankshaft, and a pointer fixed to the block is set on the zero degree mark of the degree plate. The situation is now as follows: we have the crank set so that number 1 piston is at T.D.C. and our datum from this is set on the degree plate. We have the cam installed in the approximate correct position within about $\pm 15^\circ$ (if your judgment of the positioning of the cam lobes is anything like, that is) and the camshaft connected to the crank. We are now in a position to run through a timing cycle on one of the cams, we will say, for instance, number 1 inlet cam. To do this we attach a long stem to the dial indicator and seat it on the cam follower of number 1 inlet cam, see Fig. 48. Assuming the engine to be still at T.D.C. on number 1 cylinder, rotate the crank in the reverse of normal rotation about 180° i.e. about B.D.C. The inlet cam follower should now be on the base circle of the cam. Set the dial indicator to read zero while the cam follower is on the base circle of the cam. Turn the crank in the normal direction of rotation. As it is turned, at some point you will see the dial indicator slowly moving round the scale indicating that the cam is now on the

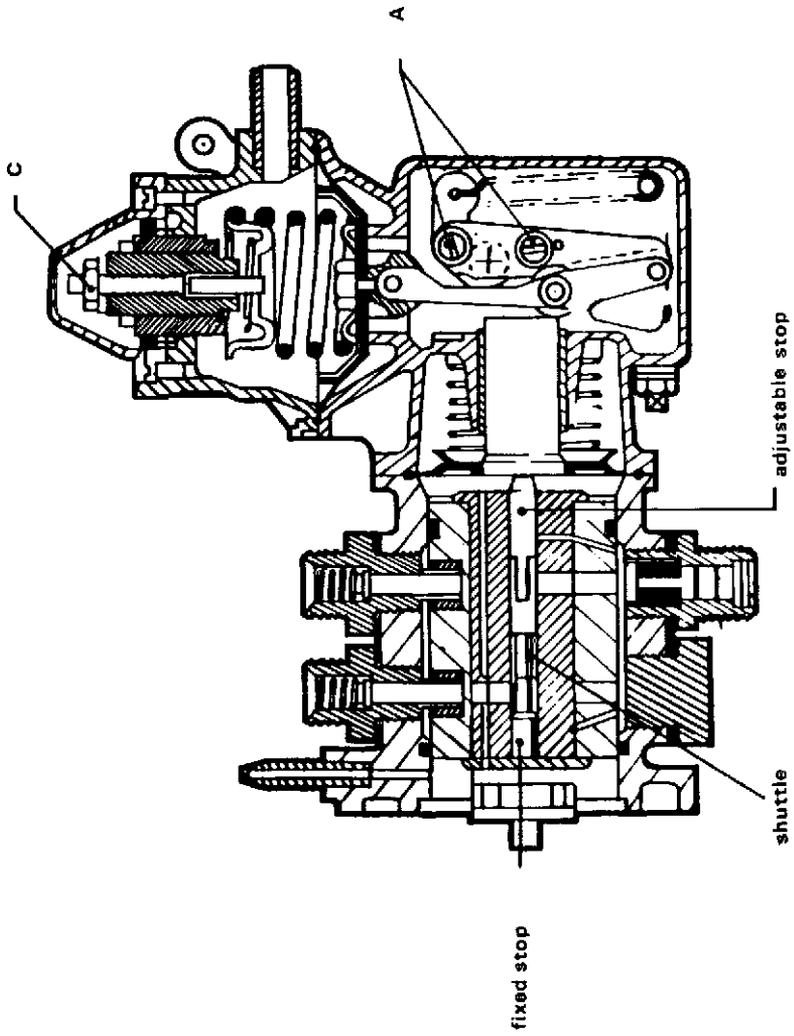


Fig. 48. Metering distributor.

clearance ramp. As the crank is turned still further, the rate of motion of the dial indicator needle will quite suddenly increase. This shows that the cam has started the lift cycle. Turn the crank back and establish the point at which the lift cycle starts, this being the point at which rapid motion of the dial indicator needle starts. This point can also be established by dividing the rocker ratio into the valve clearance. This then gives the height of the clearance ramp on the cam. This point is quite easily seen, as the rate of motion of the dial indicator needle goes up by 2 or 3 times that caused by the clearance ramp. Having established the opening point of the valve, check the number of degrees before T.D.C. that this opening point occurs, and compare it with the quoted cam timing figures. By comparing the actual figures existing, with the correct figures, we can determine the exact error present. Once the error has been noted, turn the engine back to T.D.C. on number 1 cylinder and remove the timing plate. Remove the sprockets and chain to make the necessary timing corrections. If the cam is too far advanced i.e. the inlet is opening early, the cam will need to be turned in the opposite direction to normal rotation i.e. anti-clockwise when viewed from the sprocket end of the engine. If it is too far retarded, then turn the cam clockwise. Move the cam sprocket in the chain around the relevant number of teeth (usually one or two) to line it up with the fixing holes in the cam and the keyway on the crank, and refit it. Recheck the timing. By this time the error should be no more than one tooth of the cam sprocket.

If you look closely at the cam sprocket, you will see four securing holes which are equally spaced but offset from the centreline of the sprocket teeth. By juggling the

sprocket around, we can make timing corrections down to a quarter of a tooth pitch. This means that we can get our timing accurate to a little over $\pm 2^\circ$ at the crank. Assuming we are at the point where the difference in the required timing, and the timing existing on the engine is no more than one tooth, i.e. 8.6 cam degrees of 17.2 crank degrees, we can adapt the following procedure to correct it :

Error at crank	Error at sprocket
17.2 ^o	1 tooth (8.6 ^o)

Remedy

Remove sprockets and timing chain, unmesh the sprocket and turn it one tooth in the relevant direction within the chain. Turn the cam a similar amount and refit sprockets and chain.

<i>Error at crank</i>	<i>Error at sprocket</i>
12.9 ^o	tooth (6.45 ^o)

Remedy

Remove sprockets and timing chain. Turn the cam sprocket back to front and rotate it through 90°. Refit it into the chain and offer up to engine. Turn the cam to align up holes in sprocket. Refit the assembly.

<i>Error at crank</i>	<i>Error at sprocket</i>
8.6 ^o	1/2 tooth (4.3 ^o)

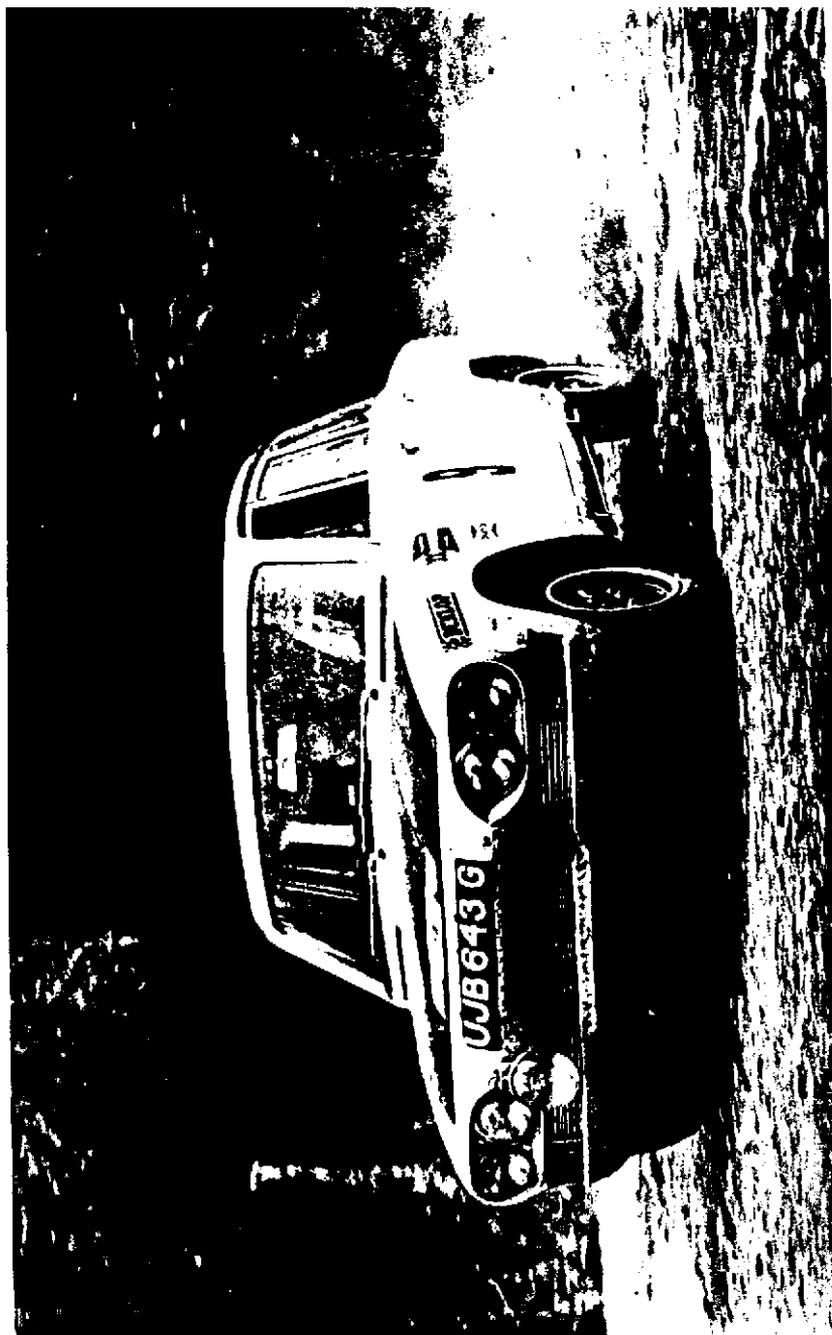
Remedy

Remove sprockets and timing chain. Unmesh the sprocket from the chain and turn it 90° then remesh it. Offer it up to the engine, turn the cam to line up the fixing holes and refit the chain and sprockets.

<i>Error at crank</i>	<i>Error at sprocket</i>
4.3 ^o	1/4 tooth (2.15 ^o)

Remedy

Remove sprockets and timing chain. Turn the sprocket back to front.



A little bit sideways—Brian Colcheth and John Syer storm the Glenurquhart Special Stage during the 1969 Scottish Rally.

Refit it into the chain and offer up to the engine. Turn the cam to align fixing holes and refit chain and sprockets.

Looking at the chart you can see that our incremental adjustment at the crank is 4.3° . The maximum error from the required timing with such steps of adjustment will be half of 4.3° . This means you must adjust to the nearest quarter tooth to the required timing. The maximum final error will occur when the error we are adjusting for, falls exactly half way between a quarter tooth pitch, i.e. 2.15° . If you have a choice of two settings which

are equally spaced either side of the desired setting, then choose the one most retarded of the two as a delay in inlet valve closure is better than an earlier opening. Once the cam is correctly installed, it is wise to put on two of your own timing marks so that you will not have to repeat the cam setting when the time for rebuilding comes. Incidentally, one can obtain a cam timing plate from S.A.H. or a 360° protractor can be adapted for such a purpose, A dial indicator can be purchased at shops dealing in engineers' tools and is a worthwhile investment to any engine builder.

CARBURATION FUEL INJECTION AND MANIFOLDING

THE Solex carburettors fitted to the early 1600 Vitesses do not allow the engine to breathe as well as it could do even in standard form. If, for some reason or another, you have to stick with them, then a few hints may be useful.

If the head has been modified, the main jet size will need to be increased by about 0.3 to 0.5 of a m/m. Secondly, the very first Vitesses had fixed jet carbs on the accelerator pump. On the later 1600 cars this pump was discarded to assist hot starting. The performance was seemingly unchanged by such a move. If you have the early model fitted with accelerator pumps, then an improvement can possibly be gained by discarding them. By far the best method is to discard the carbs completely and fit the later twin Strombergs as these have a greater breathing capacity and allow more mods to the engine before they become inadequate. The twin Strombergs were fitted to the last of the 1600 Vitesses and the Mk. 1 2-litre Vitesses and G.T.6's. If you can get the carb and manifold setup from any of these cars for the early 1600,

it can be bolted straight on. If the carbs are off one of the 2-litre engines, they will, of course, need reneedling. In fact if anything has been done to the engine, the needles will inevitably need changing to some which will give the correct fuel/air characteristics through the rev. range. It is very difficult if not totally impossible, to state what needle will be required since it will depend on what other components have been modified. If we allowed for all combinations of various parts, we could end up with a needle list big enough to make a small book. At this stage we need not worry too much about this as there are ways and means of sorting carburation to get it spot on which will be gone into in a later Chapter.

The next stage of carburation improvement is to fit triple S.U.'s which are available through V.W. Derrington. These give vastly improved breathing over the Solex carbs and are distinctly better than the twin Strombergs, not, might I add, through any deficiencies with the Stromberg carb., but from the fact that three S.U.'s are better than

two Strombergs.

By using triple S.U.'s instead of the standard set-up on carbureted 1600 or 2000 engines, we can expect a power increase of 15%-20%. As is usual with a good S.U. set-up, the fuel consumption does not go up out of all proportion when compared to the power gained. In fact, driven at normal road speeds, triple carbs seem to give better consumption figures than an engine with standard carbs.

Next step up the ladder as far as carburation is concerned, is to use triple Weber carbs. For most purposes on the 2000 engine, 40 DCOE Webers work out very nicely. We can consider the use of Weber carbs right from square one but they will come into their own more on an engine which has seen mods in other quarters. For an all out competition engine, 42 DCOE Webers will give a small gain in power over the 40's something between 2-3 b.h.p. being gained by their use. As soon as we start making vast improvements on the inlet side of things on the Triumph 6's, the inadequacy of the exhaust system becomes apparent. To get the full benefit of a good set of triple carbs one should give serious thought to a change in exhaust manifolds. This will allow the full potential of the carbs to be gained in the middle and upper rev. ranges. For road use all tail pipes of a high efficiency exhaust manifold can be fed into one silencer so long as it has a reasonable bore. Better results however, can be gained by using two tail pipes, each one connected to three of the cylinders. There is little to be gained by messing about with pipe lengths on an exhaust system which has to have silencers fitted. One should concentrate on reducing back-pressure by selecting a pair of good

straight-through silencers. On the other hand if the car is to see use where open exhaust can be used, then our exhaust pipe length can be made to work for us. The hotter the cam the more critical the exhaust pipe length becomes. With a really hairy cam, the correct exhaust length can result in 10 b.h.p. gain at the wheels on a 2-litre car and that amount of power is not to be sniffed at. Unless one has facilities to make up manifolds complete, there is little one can do about the primary lengths of the exhaust system. So long as the primary pipes are of a reasonable length, which is the case for most exhaust manifolds sold by tuning firms, we can advantageously juggle the secondary or tail pipe length. This length is going to depend on the cam used and the length of the primary pipes, if a proprietary exhaust system is used. Adjustment of the pipe length, for each group of three cylinders, should be done on a dynamometer. As a rough guide, you will find that the length will be around the 30" mark 1 6" for each of the two tail pipes. The diameter of the tail pipes used, needs to be in the order of 1 7/8" - 2". If you can have an exhaust system specially made up for use with a full race cam, then the following sizes should be used: 1 3/8" bore primary pipe x 36' long into 2" dia. tail pipe 24' long numbers 1, 2 and 3 primary pipes feeding into one tail pipe and numbers 4, 5 and 6 feeding into the other. The tail pipe length may need to be finally set on a dynamometer because of the various types of full race cams available but in the main, will not need changing by more than about 3'. It should be pointed out that even a 3' change in tail pipe length can make several horsepower difference.

Fuel injection

We have talked of carburation changes and exhaust manifolds and the time has come to look at fuel injection. We are likely to come across two types of fuel injection on Triumph cars. The Lucas type fitted as standard to the T.R.5 and 6 and the 2,500 P.I. and the Tecalemit/Jackson-system which is, of course, not a standard fitment. Both systems rely on a profiled cam to give the correct fuel metering throughout the rev. range. If the engine characteristics are changed, the profile of the fuel cam will have to be altered to suit. Usually the largest single influencing factor is the type of camshaft fitted. As pointed out before, you can buy a camshaft which is designed to go with the standard fuel cam on those engines having Lucas fuel injection. When fitting such a cam to the Lucas injected engine, the overall mixture may have to be adjusted but the standard fuel cam can remain. Unless you happen to be an expert on either type of fuel injection, the setting up of injectors is best left to an expert. This setting up should be done on a dynamometer in conjunction with a mixture analyser. With this sort of equipment, one can determine and try a variety of fuel cam profiles to enable the fuel injection on a tuned engine to show up at its best. If the car you are going to tune has not already got fuel injection and you decide to have it fitted, then in all probability you will go for the Tecalemit/Jackson equipment. If this is the case, then S.A.H. have a variety of fuel cams to suit different stages of tune for the six cylinder Triumphs. This being the case, one should not experience undue problems with the T.J.

system as far as metering accuracy goes. Even if a fuel cam is selected which is near enough right for the job in hand one should still set up the system on a dynamometer for the best results.

To make sure you are not put off injectors by their seemingly complex nature, let us look at the advantages of their use. First off you get more power. Just how much more really depends on the state of tune of the engine. If the engine is in a relatively mild state of tune, we may only gain 3-5 b.h.p. over a really good set of carbs like Webers, for instance. On the other hand if the engine is fully tuned, the gain over carbs can be 8-10 b.h.p. Some mods to the standard barrels or intakes can raise this even further. The power gain is not the only advantage of injection. We can also get better flexibility and a slight widening of the power curve which will enhance the driving characteristics of the car. With either the T.J. or the Lucas injectors, one has definitely got the edge on carbs but either of these systems can, with good effect, be modified to give even better results.

Before delving into the ins and outs of fuel injection systems, a couple of points should be considered. The Lucas system is built for road use and as such any mods for racing will in all probability have to be carried out by the owner or some tuning firm willing to tackle the job. The T.J. system is on the other hand built for both road and racing, the racing injector being of a slightly different form. Since T.J. are very much connected with competition, one can get a fuel cam specially made up to suit any engine in any state of tune which is using their injectors.

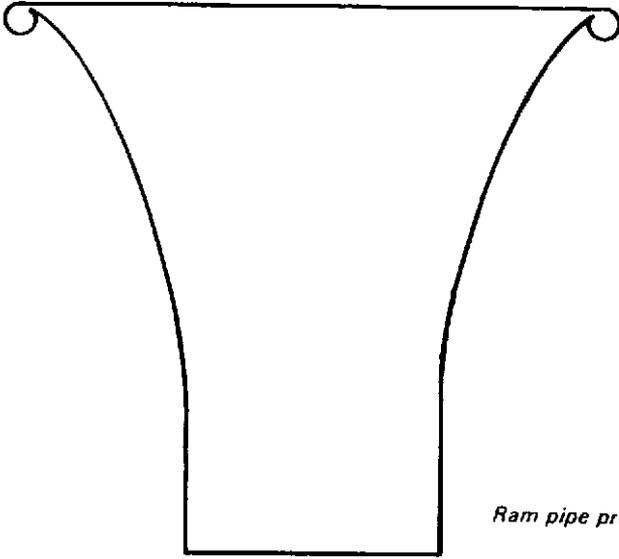


Fig. 49.

Ram pipe profile.

This now only leaves us with a fuel cam problem for the Lucas injectors. If you do not fancy trying to develop your own fuel cam then any tuning done to the engine will have to be compatible with the standard fuel cam. Even if this is done one will have to make overall adjustments to the mixture. If the overall breathing ability of the engine has been improved, the mixture will in all probability need to be made richer by altering the basic slope of the fuel cam. This is achieved by loosening the two screws 'A' in Fig. 48 and moving the contact point of the roller on the cam away from the plunger. A small amount of movement gives a relatively large change in mixture so do not be too rash, only make adjustments a few thou. at a time. Moving the fuel cam changes the mixture at full throttle conditions and part throttle. If the part

throttle conditions are still found to be out, adjustments can be made by altering the screw denoted by 'C' in Fig. 48. Undoing the screw will richen the mixture, and tightening it will give a weaker mixture. We may also find a condition at full throttle and high revs. where the shuttle which moves back and forth between the two stops does not displace sufficient fuel, causing a weak mixture. If this situation occurs, and it is not likely, except on the most highly tuned engines, the remedy is simple. Both the fixed and moving stops are shortened by removing metal from their plain ends. We cannot, however, continue to apply this method ad lib since too much shortening will cause the shuttle to close off the fuel ports in its sleeve and defeat the object of the exercise. Again any adjustments should be made by removing metal

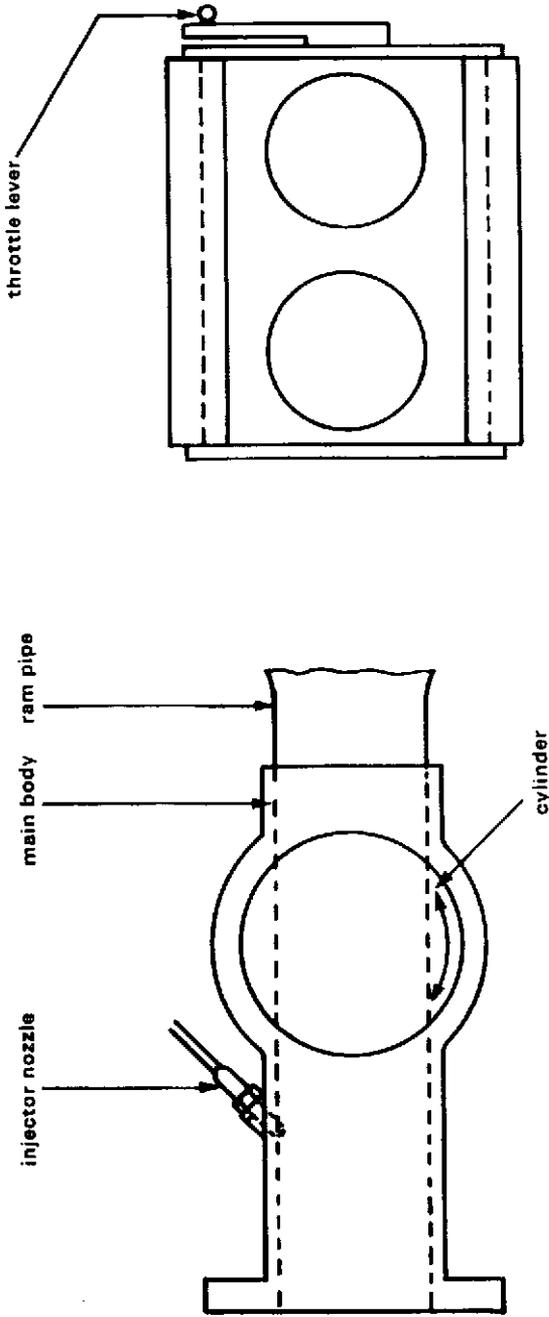


Fig. 50. This cylinder is free to rotate in a similar fashion to a normal butterfly spindle. When the cross holes in the cylinder line up with the bore in the main body a clear unobstructed passage is allowed. The whole set-up can be likened to one enormous gas tap as far as the working principle of throttle operation is concerned.

at a few thou. at a time.

If one is using injectors without air filters one can obtain a useful increase in power by fitting ram pipes to the injector intakes. These items will have to be specially made up since there are no proprietary ones available. The form these ram pipes should take are as shown in Fig. 49. These will give a power increase around the 5-6 b.h.p. mark, so they represent a fairly worthwhile modification. Of course, if we really want to get fussy over things we can get a bit more power by completely remaking the injector bodies and at the same time, do away with the conventional butterfly. Fig. 50 gives a suggested layout of an injector body suitable for use with the downdraughted head shown in Fig. 40. This injector body with the downdraughted head and a suitably hairy cam could conceivably give the power output required of our hypothetical 2-litre racing engine.

To round this off a little something should be said about carburation and manifolding for the four cylinder engines. The standard carburation for the TR4 and 4A is either a pair of 1-3/4" S.U.s or Stromberg C.D.s. The standard carbs can be used on engines tuned to give 135-140 b.h.p. but they are on the limit. The fitting of bigger carbs of the S.U. or Stromberg variety is not the complete answer. Since we have an engine with four inlet ports we may as well make full use of them. As such the fitting of a

pair of DCOE 45 Weber carbs is to be highly recommended. On an engine which is already giving 135-140 b.h.p. they will add to this another 10 b.h.p. The standard exhaust manifold does tend to make power reductions over about 4,500 which are of sizeable proportions. The fitting of an extractor manifold such as those items supplied by V. W. Derrington or S.A.H. can result in about 7-8 b.h.p. gain. The TR4A exhaust system has two silencers fitted, the exhaust pipe splitting into two separate pipes about half way down the car. If possible, this should be replaced by a single pipe system using one large straightthrough silencer. An exhaust pipe diameter of approximately 2" should be used from the junction at the manifold right through to the silencer and on out to the tail pipe. If open exhausts are to be used and the engine is equipped with a full race cam, Webers, etc., a tail pipe of about 40-45" is required, the length being measured from the end of the extractor manifold to the end of the tail pipe. The diameter of the tail pipe is best around 2-1/4" which in most cases will mean a step up from the manifold size. This can be done by sleeving the manifold pipe up to accept the tail pipe. To prevent things dropping apart the sleeves should be welded to the manifold and the tail pipe clamped to the sleeves. If you weld the whole lot together you are likely to experience great difficulty removing the manifold from the car.

ASSESSING THE RESULTS

WE have up to this point discussed various ways and means of modifying the engines, but very little has been said as to the power increase given by any one or a combination of modifications. The power outputs quoted by the factory are rarely if ever, shown in practice. In most cases the factory quote figures gained under conditions which could never exist in the car, the net result of which is a power unit which is often about 10% down on the factory quoted figures. Consequently we often find that a certain amount of modifying is necessary before we can achieve the factory quoted power output. The amount below the quoted power output varies from one make of car to another.

It would appear that in the main, Triumph figures are within a few per cent of the quoted figures, indeed, if anything, the early 2000 engine was under quoted which is quite unusual. To avoid further beating about the bush, let us go to some power output figures for completely bog standard engines which have been carefully built up and then set up whilst on the dynamometer.

Starting at the top of the list and working downwards, the power output at a number of successive stages will be given in the order in which they should be done, if tuning is done in stages.

Starting with the TR4 and 4A, the first modification should be a re-

TR4 &4A	2000 c.c.	98 b.h.p.
“ “	2138 c.c.	102 b.h.p.
1600 Vitesse	1596 c.c.	70 b.h.p.
2000 Mk. 1	1998 c.c.	100 b.h.p.
G.T.6 Mk. 1	1998 c.c.	100 b.h.p.
Triumph 2000 Mk. 1	1998 c.c.	100 b.h.p.
2000 Vitesse Mk. 1(1998 C.c.	105 b.h.p.
G.T.6 , Mk. 11	1998 C.c.	105 b.h.p.
Triumph 2000 Mk. 11	1998 c.c.	105 b.h.p.
2.5 P.1, saloon	2498 c.c.	135 b.h.p.
TR5 Et 6	2498 c.c.	135 b.h.p.

worked head. This will put the power up from 98 b.h.p. to 108 b.h.p. The addition of an extractor exhaust manifold will raise that to 114 b.h.p., still with silencer. A half-race cam will raise this still further to about 120 b.h.p. and a fullrace one, to about 124 b.h.p. With a fullrace cam, the silencer has quite a profound effect and the removal of the silencer plus the substitution of the correct length of pipe will put the power up by another 8 b.h.p., bringing it to 132 b.h.p. At this point the standard carbs are definitely choking the engine, and the fitting of 45 DCOE Webers is called for. The Webers at this stage will put up the power to 142 b.h.p. These figures are for the 2-litre engine. By using the 2.2 litre engine with the largest piston available (87 m/m), the power output can be raised by a further 9-10 b.h.p., bringing it to a little over 150 b.h.p.

With the 1600 Vitesse engine fitted with the Solex carbs one should consider the changing of the carbs as one of the first modifications. A modified head fitted to a 1600 engine with Solex carbs will only give about 4-5 b.h.p. increase. On the other hand a modded head fitted to a 1600 engine which is using twin Strombergs will give around 8-9 b.h.p. increase. The fitting of triple S.U.s will give about 10 b.h.p. increase on top of the amount given by a Stage 1 head, bringing the power up to 88-90 b.h.p. At this stage a good exhaust system becomes essential if any further power increase is required and its fitment will, under ideal conditions, push up the power to about 95 b.h.p. Fitting a half-race cam at this stage will bring the power to about 100 b.h.p. The replacement

of a Stage 1 head by a Stage 11 head will show around the 105 b.h.p. mark as a result. If we go really mad and fit all the hottest goodies going, such as triple Webers, Stage 11 head, full race cam, etc., we should expect the power to be in the region of 130 b.h.p. The Mk. 1 2-litre engines will show the same basic increases as the 1600 engine, but because of its greater capacity, the power will be up by 1520% at each stage. On 2-litre engines, using fuel injection instead of carbs, the power can be brought up to 160 b.h.p. plus, when suitable cams, heads and exhaust are used, even without resorting to super heads and very sophisticated injector bodies, etc., etc. If we do go to such exotic means for gaining power than 190 b.h.p. plus should be possible.

With the 2.5 litre engine, the first step is to modify the head. This will result in a gain of about 8 b.h.p. Since all the 2.5 litre engines with the exception of the TR250 have fuel injection, the fitting of a good extractor manifold should be considered next. For the TR250 some improvements in carburation are needed before going to the exhaust side of things. From this point on, any further increase on 2.5 litre engines should be done by changing the cam. By selecting a cam suitable for use with fuel injection, the power output can be raised to 165-170 b.h.p. and still retain reasonable road going characteristics. A full race cam, open exhausts of tuned length and other competition items can give up to 180 b.h.p., but by this stage the tractability in the lower rev. range has completely gone.

TRANSMISSIONS

THE TR4 clutch is of the spring type, and as such is capable of handling up to 140 b.h.p. Beyond this one is likely to experience a fairly short clutch life before clutch slip starts to set in. The TR4A clutch is capable of taking more power than the early TR4 clutch, and is of the more modern diaphragm type. The TR4A clutch can be used in the TR4 but it means swopping a few more components other than the clutch itself. The securing holes in the flywheel are different for the spring clutch as compared to the diaphragm unit. This can be remedied either by redrilling the TR4 flywheel to accept the diaphragm clutch or purchasing a TR4A flywheel which will fit directly to the crankshaft of the earlier engine. Incidentally, the TR4 flywheel is a lot lighter than the 4A flywheel in standard form. After lightening, both flywheels will weigh the same. When changing from the earlier clutch to the later one, the appropriate clutch plate, thrust bearing and thrust bearing carrier sleeve must be used, all these parts being as fitted to the TR4A. Since the diaphragm clutch can hold out against 150 b.h.p. we should not

experience too much trouble in this quarter

The 1600 Vitesse spring clutch can be treated in a similar manner-TR4 to 4A swop. We can use the relevant parts from the 2-litre Vitesse to convert to a diaphragm clutch. The standard spring clutch on the 1600 is all right for power outputs up to 105-110 b.h.p. or so. After this, the use of the diaphragm clutch is to be advised. The 2-litre Vitesse and G.T.6 clutches appear to be quite capable of transmitting 165 b.h.p. plus, so little attention is required in this department. With a highly tuned Triumph 2000 or a 2000 with the 2.5 litre engine in it, some trouble with the clutch may arise. This can be easily cured by using the 2.5 P.I. clutch as it is some 20% stronger and should suffice to about 180 b.h.p.

Ratios

Gearbox ratios can be all important on a car using a tuned engine. Since it would appear that very little is available in the way of special gears for any of the Triumph cars,

it is a good job that the standard ratios are pretty close. For instance the 2-litre Vitesse and G.T.6's have the following ratios: 1st 2.65/1, 2nd 1.78/1, 3rd 1.25/1, top 1/1 O.D. top 0.8/1. These, by road going standards are pretty close. For racing one could do with ratios a little closer but the standard ones do come quite close to the mark. The early 1600 Vitesse has a lower first gear than the later ones at 2.93/1 instead of 2.65/1. It would appear to be a little awkward to adapt the later gears into the earlier box so the easiest route is to use the later box complete. The cost, time and trouble of converting the earlier box is just not worth it since practically all the internals have to be replaced and also some machining is required here and there. With the Triumph 2000 and 2.5 P.I., we can get slightly closer gearbox ratios by using the TR4, 5 or 6 components.

The saloon ratios are: 1st 3.3/1, 2nd 2.1/1, 3rd 1.386/1, top 1/1, whereas the TR ratios are 1st 3.139/1, 2nd 2.01/1, 3rd 1.325/1 and top 1/1. One can either swop the gearbox and tailshaft assembly complete as a unit or swop the TR gears into the 2000 or 2.5 P.I. box. If just the gears are swopped over, then one should retain the original 2000 or 2.5 P.I. gearbox input shaft as the spline on these differ to the TR ones. If a complete TR gearbox is being installed, the stripping of the gearbox to fit the saloon input shaft can be avoided by using the TR clutch plate since the splining in this will obviously match the TR box input shaft. If this is done then a check should be made to see that the input spigot bearing is the right size. Also if any difficulty is experienced through misalignment of bolt holes in the bell housing (which is in one piece with the gearbox on 2000s, 2.5 P.I.s and

TRs) the back plate on the engine should be swopped for the one relevant to the gearbox.

Although the TR ratios are not as close -s those found in the 2-litre Vitesse and G.T.6, they are well supplemented by overdrives. The saloons have an overdrive effective on the top two gears and the TRs on the top three. For most purposes then, we will have an adequate selection of gears. Overdrives, however, tend to be a little unreliable for racing and some benefit may be gained by using the Vitesse or G.T.6 box in the big saloons or the TR sports cars. This will give far closer ratios without resorting to the use of overdrives. To do this one will need to use the G.T.6 or Vitesse gearbox, tailshaft, gear change extension, bell housing and clutch release mechanism. On the 2000 and 2.5 saloons the clutch to use is the 2.5 P.I. one. The difference in height from the clutch face to the spring tips between this clutch and the G.T.6 unit is insufficient to cause any undue bother. The same also goes for the TRs but here we should check the spigot bearing clearance for a running fit. The tailshaft of the G.T.6/Vitesse box is shorter than that of big saloons and the TRs. The prop shaft will need to be lengthened to compensate.

Final drive

Having juggled the gearbox or gearbox ratios to suit our requirements we should select a compatible final drive ratio, bearing in mind the sort of usage the vehicle is to be put to. Starting again with the TR4, 5 and 6, we find that standard final drive ratios are available at 3.7/1 and 4.1/1. For final drive

TRIUMPH 2000 & 2.5 PI. 3.45 FINAL DRIVE

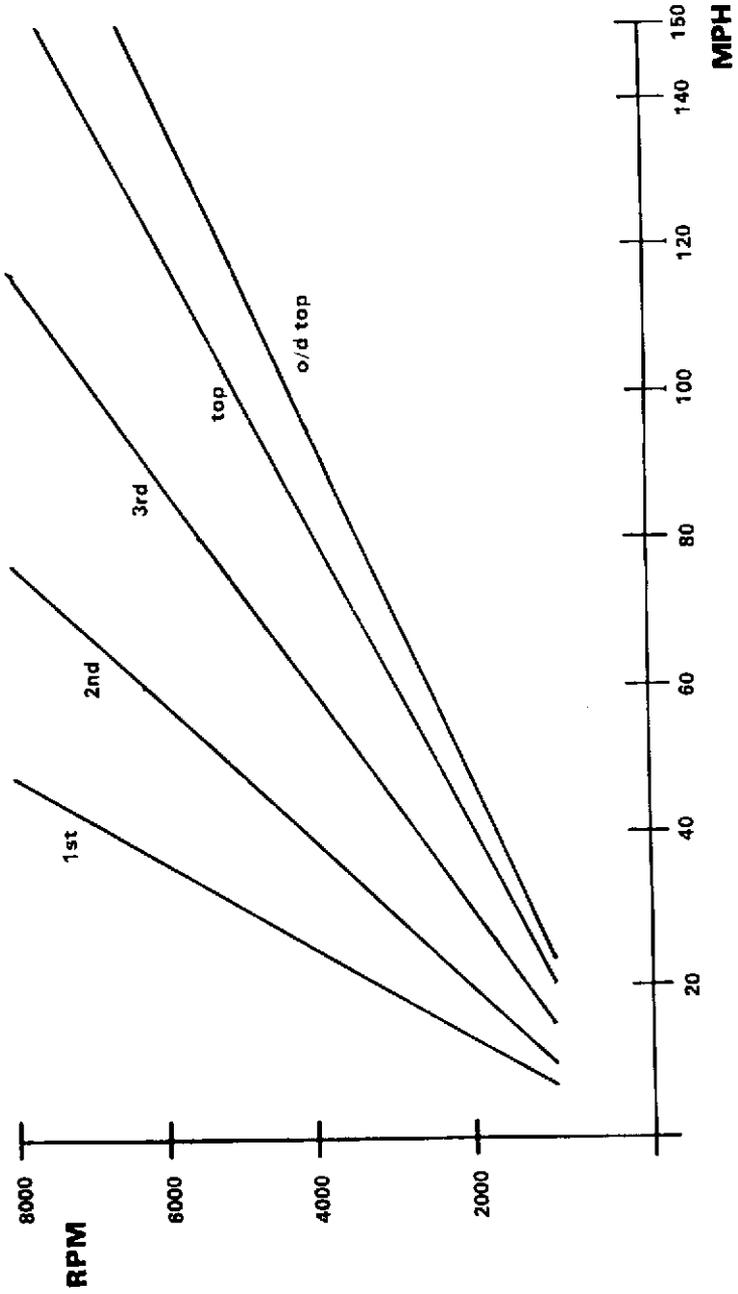
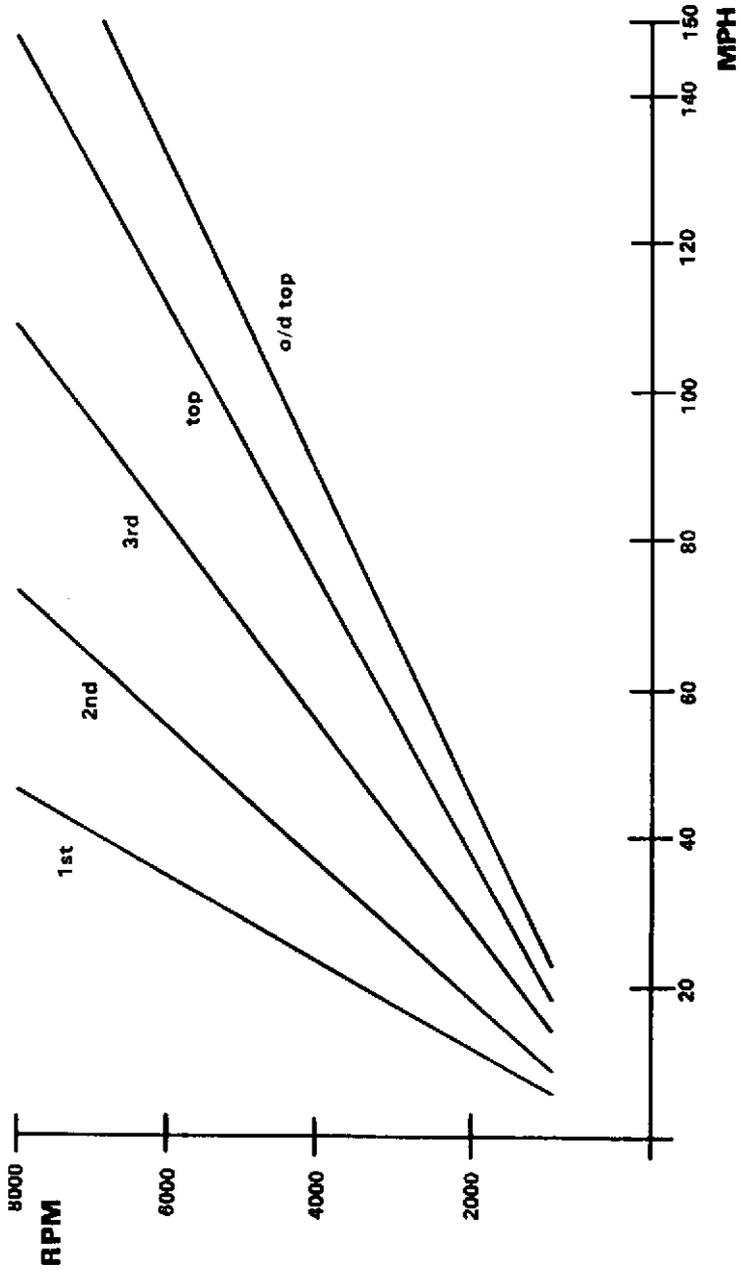
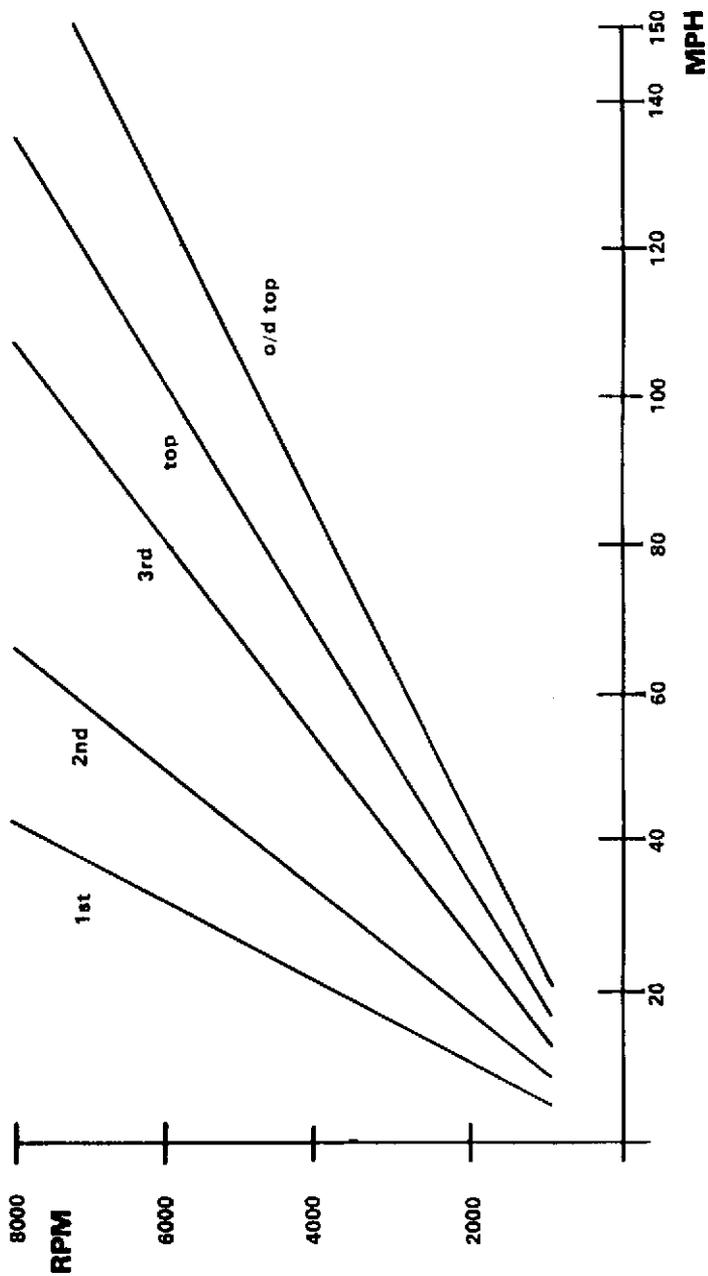


Fig. 51.

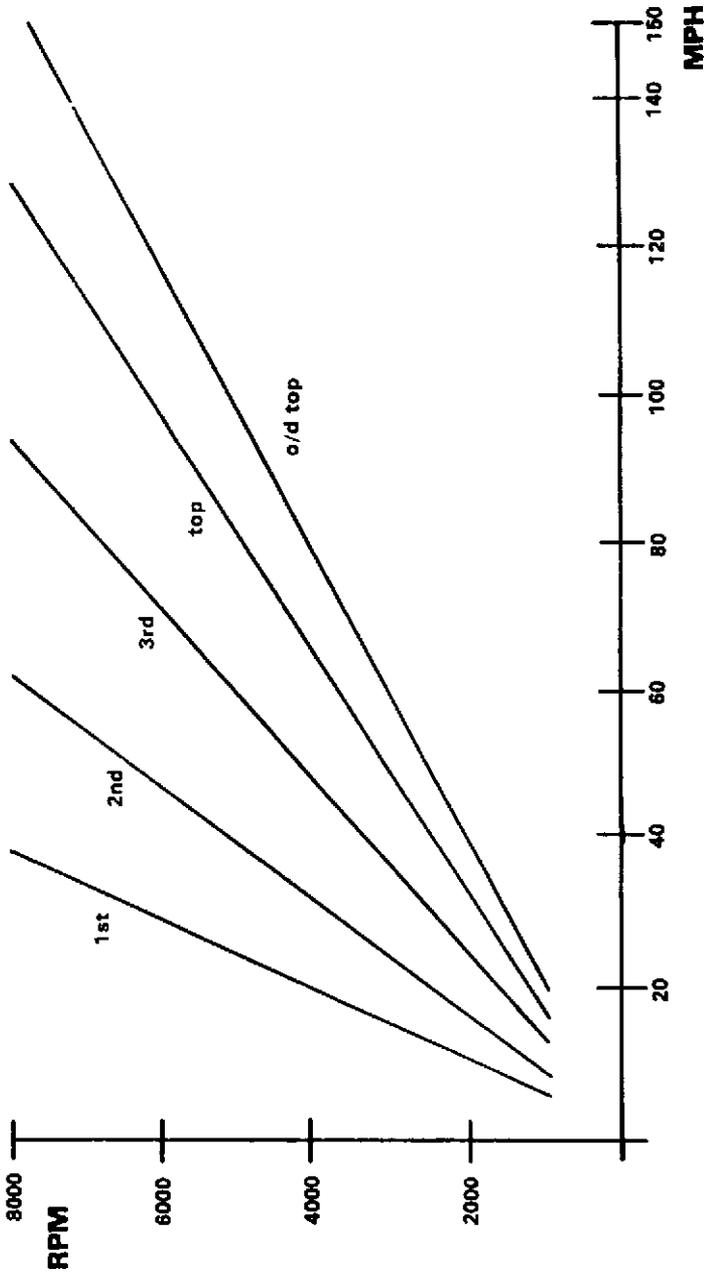
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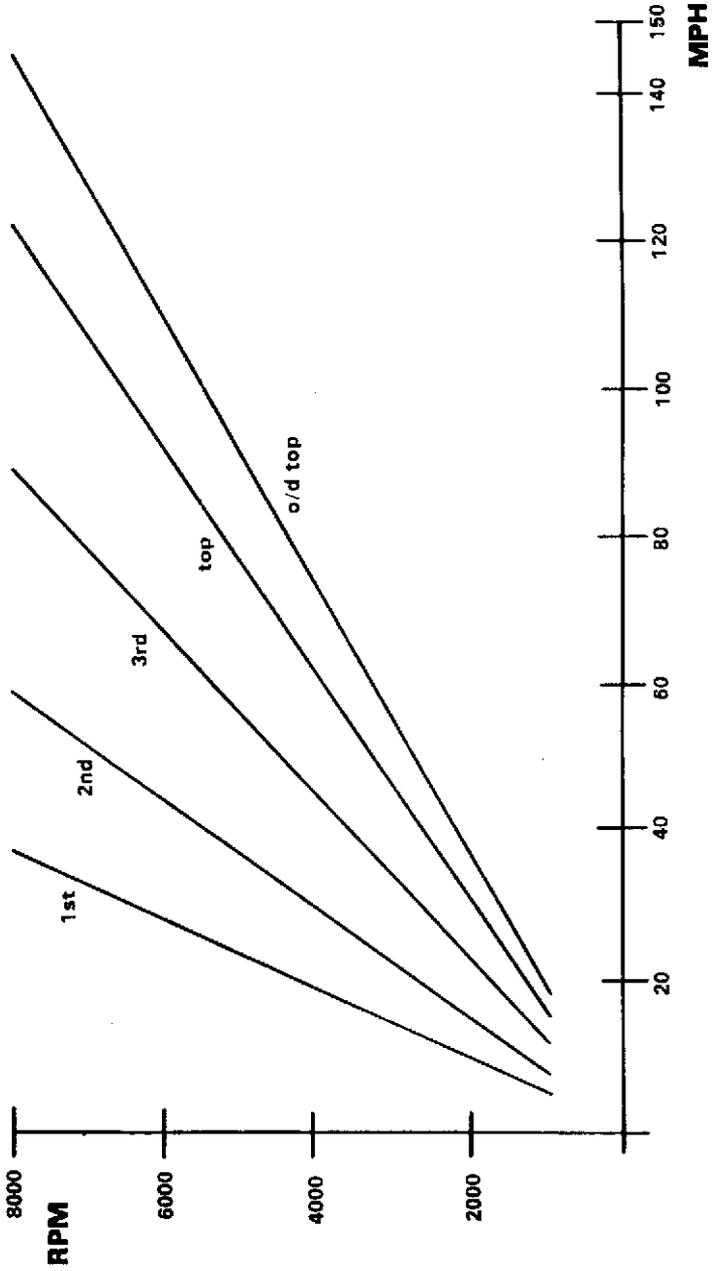
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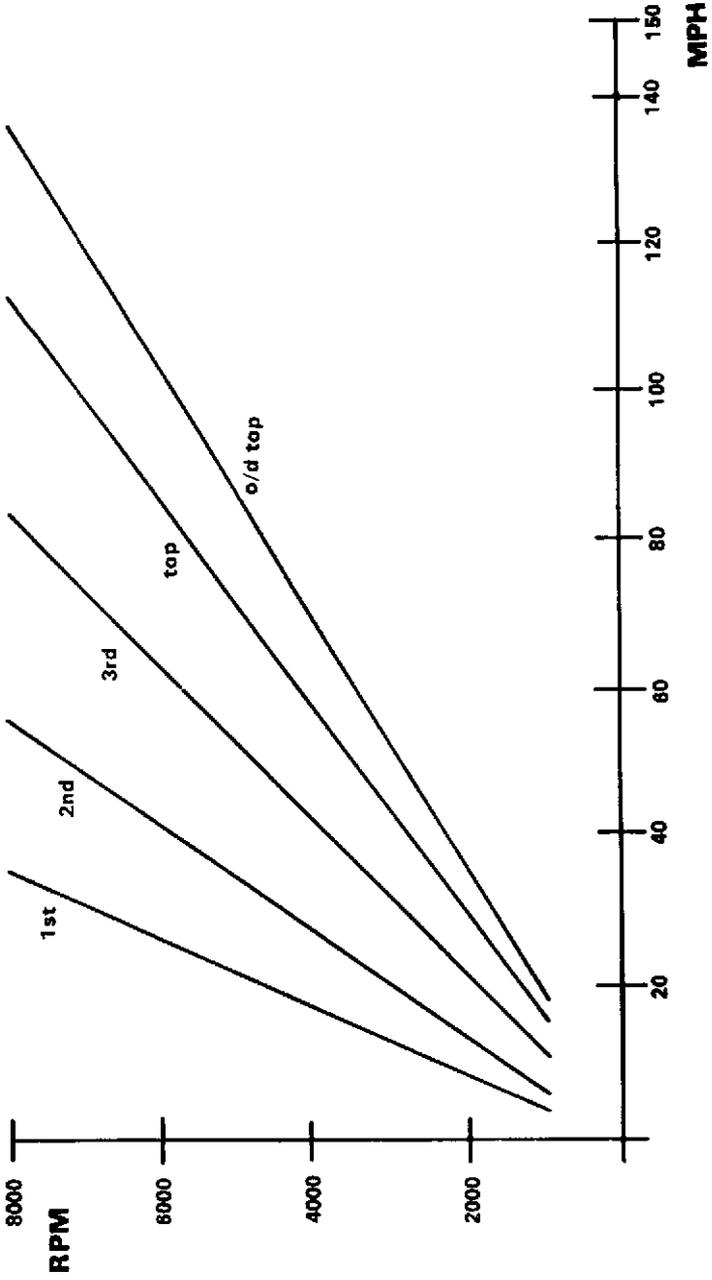
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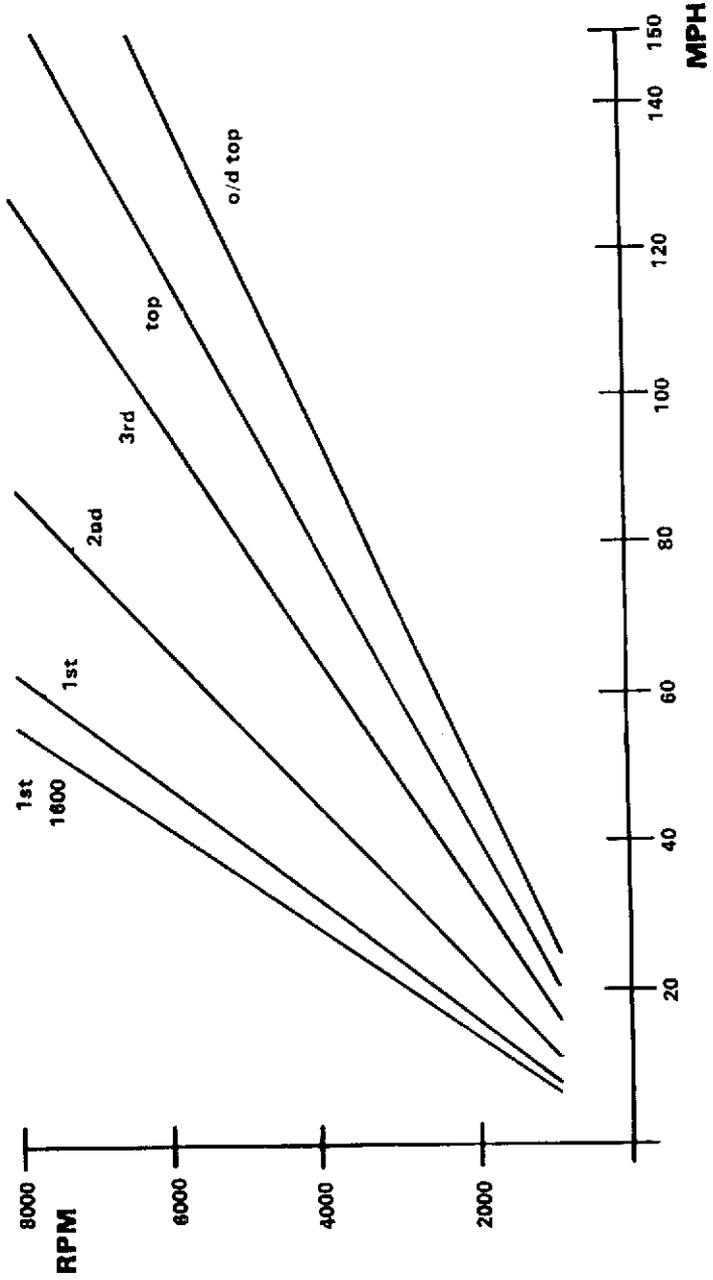
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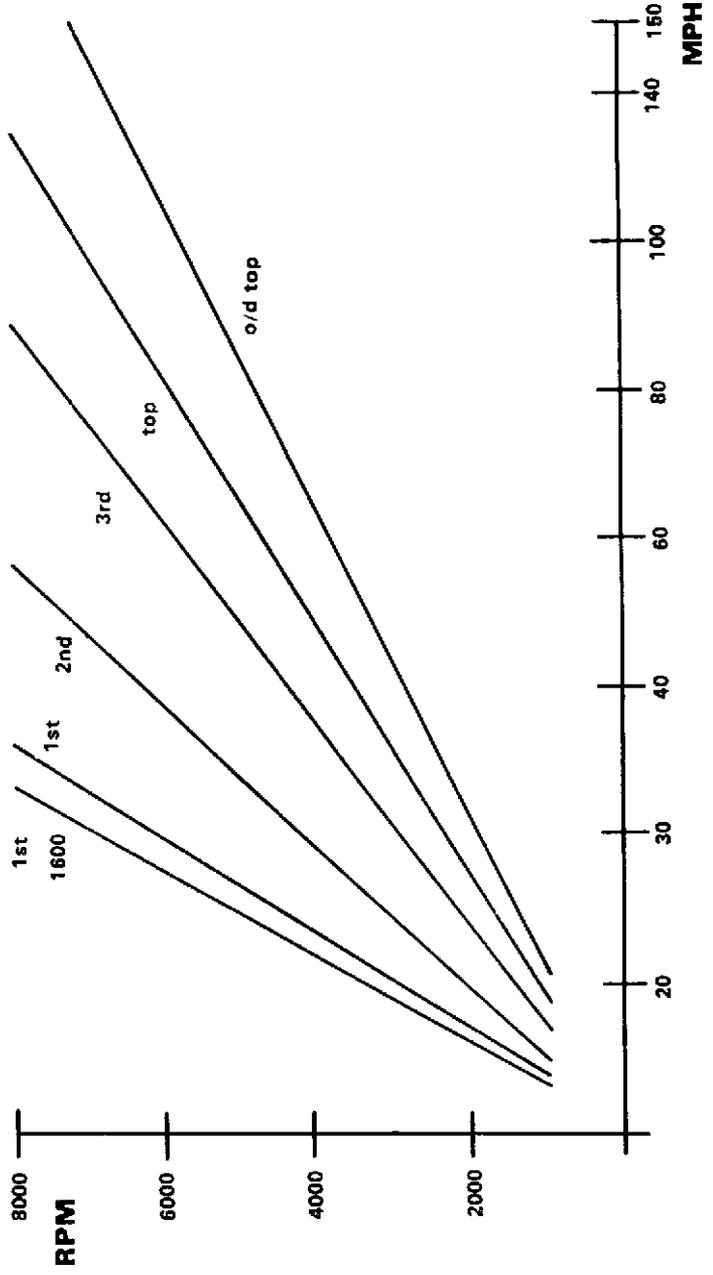
TRIUMPH 2000 & 2.5 PI. 4.875 FINAL DRIVE



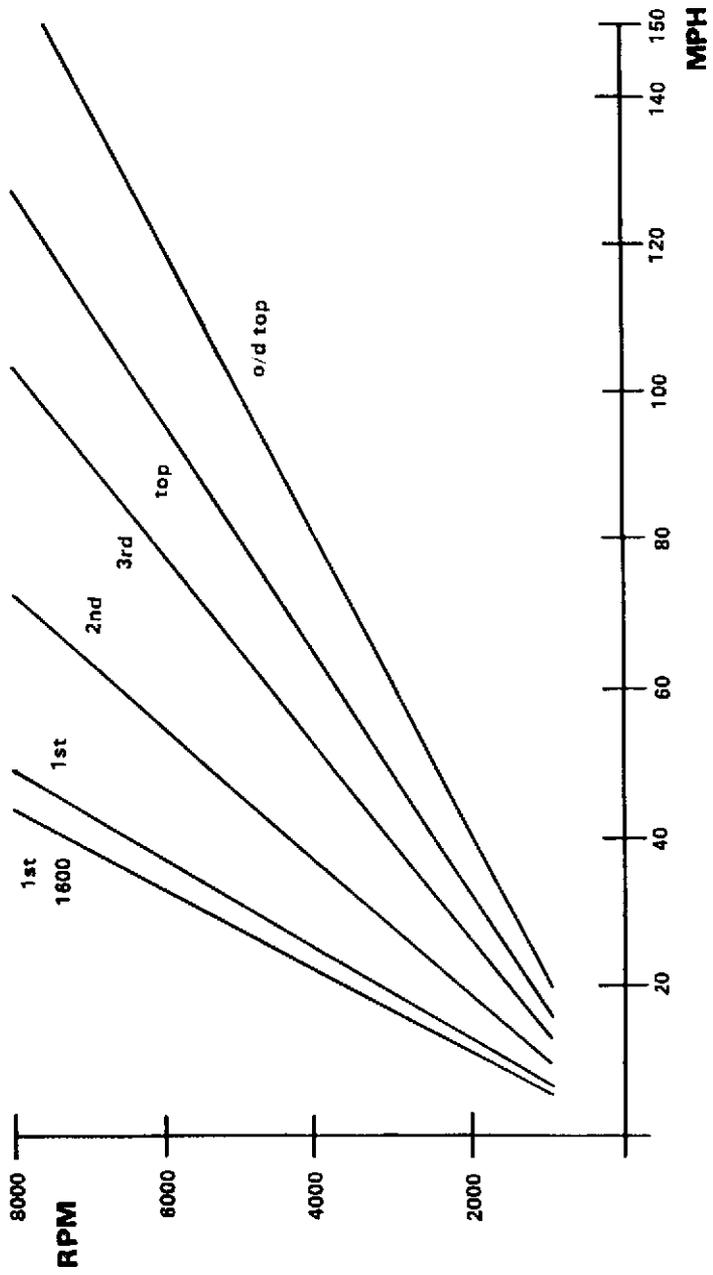
VITESSE 1600, 2 LITRE & GT6. 3.27 FINAL DRIVE



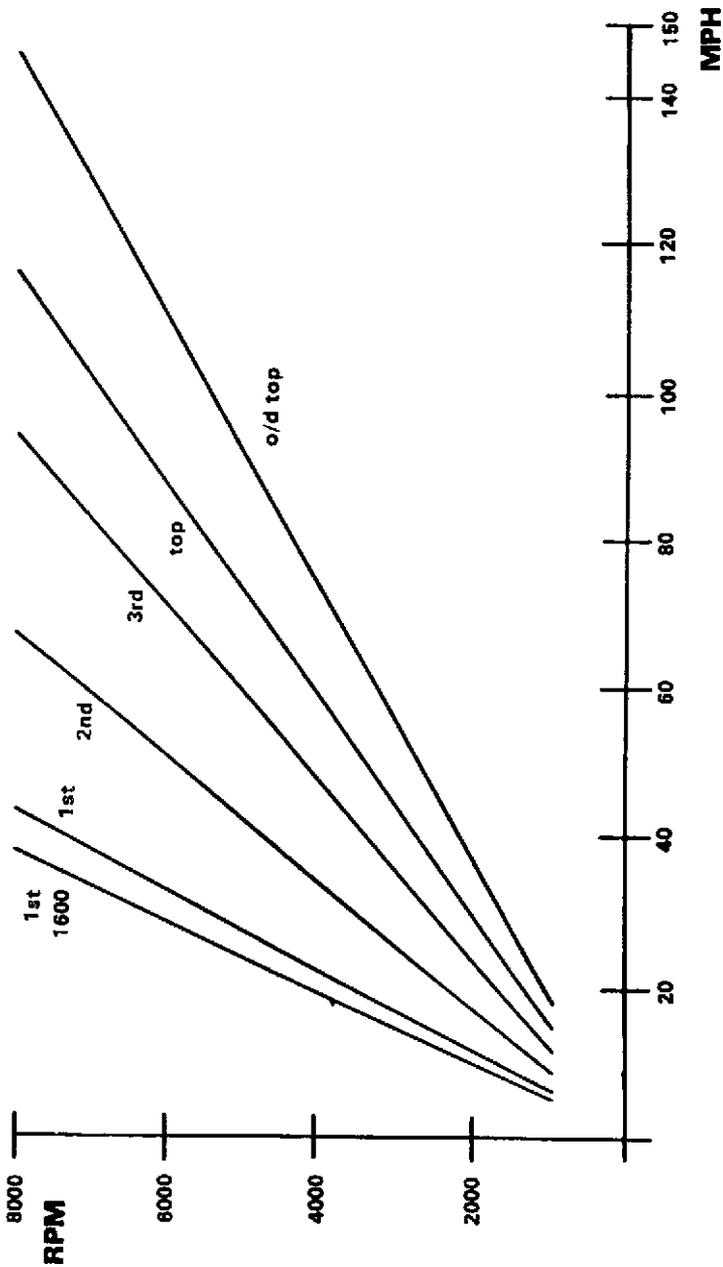
VITESSE 1600, 2 LITRE 6 GT6, 3.89 FINAL DRIVE



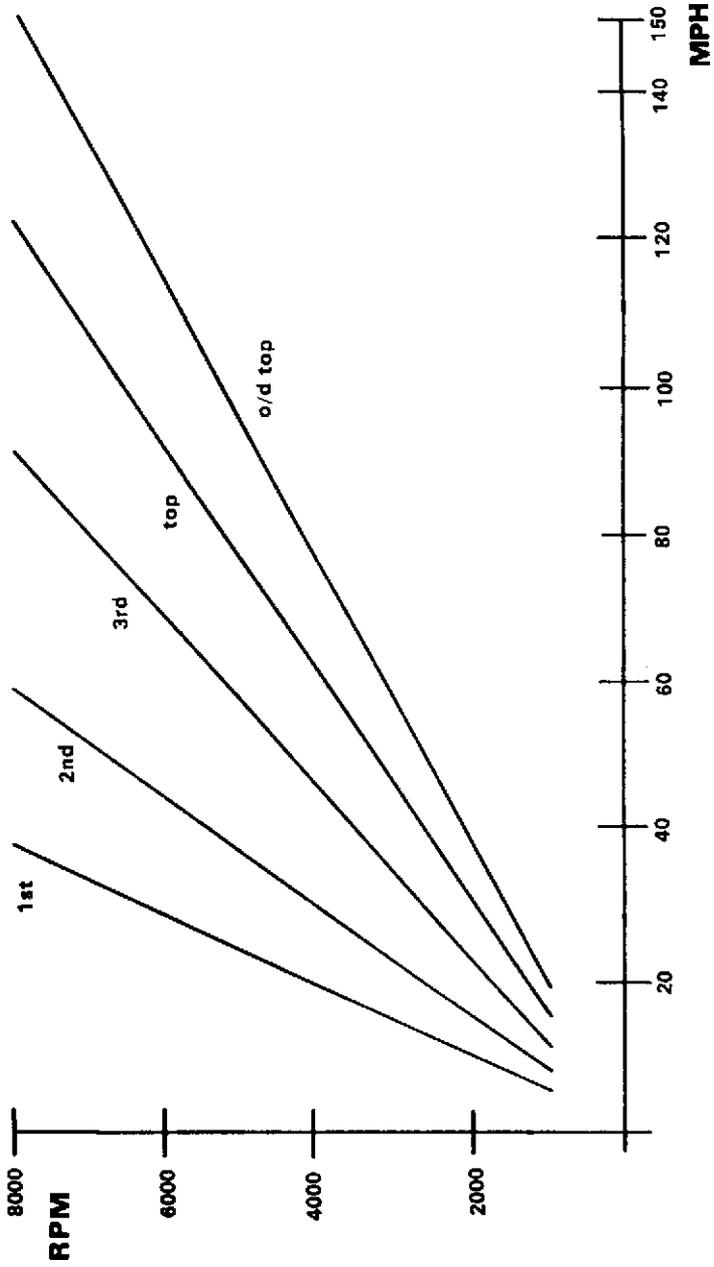
VITESSE 1600, 2 LITRE 6 GT6. 4.11 FINAL DRIVE



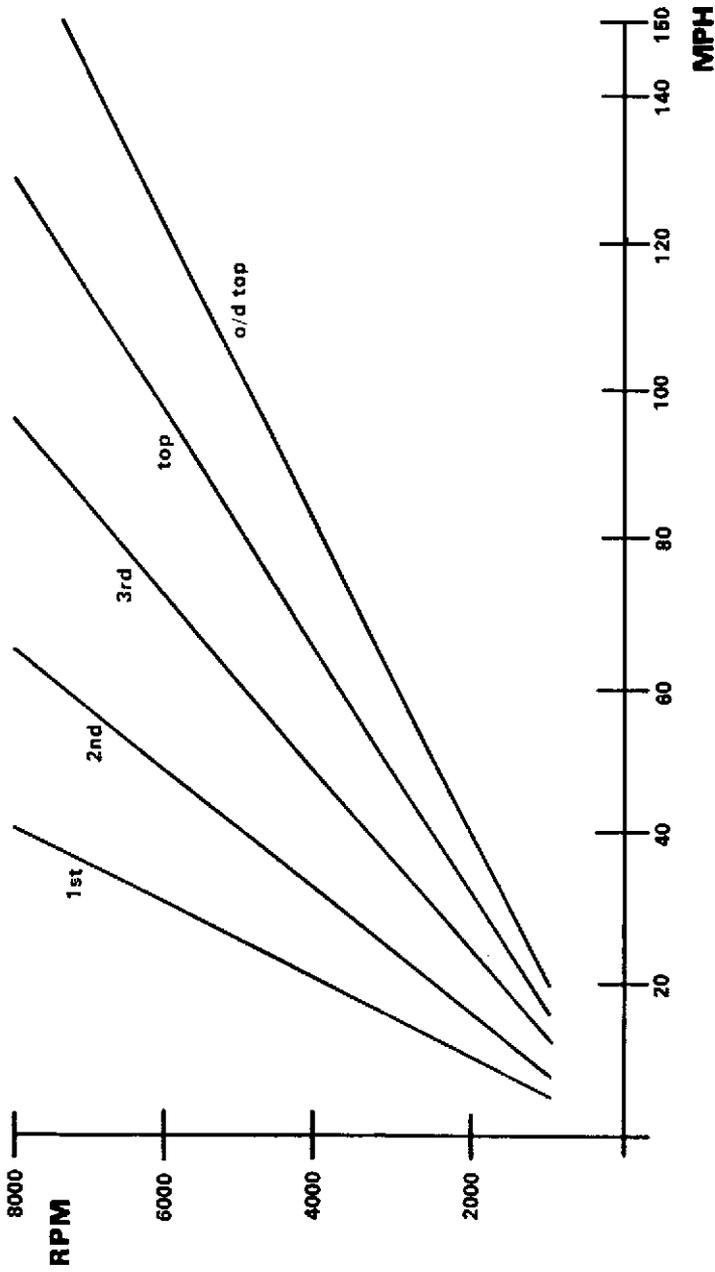
VITESSE 1600, 2 LITRE 6 GT6. 4.55 FINAL DRIVE



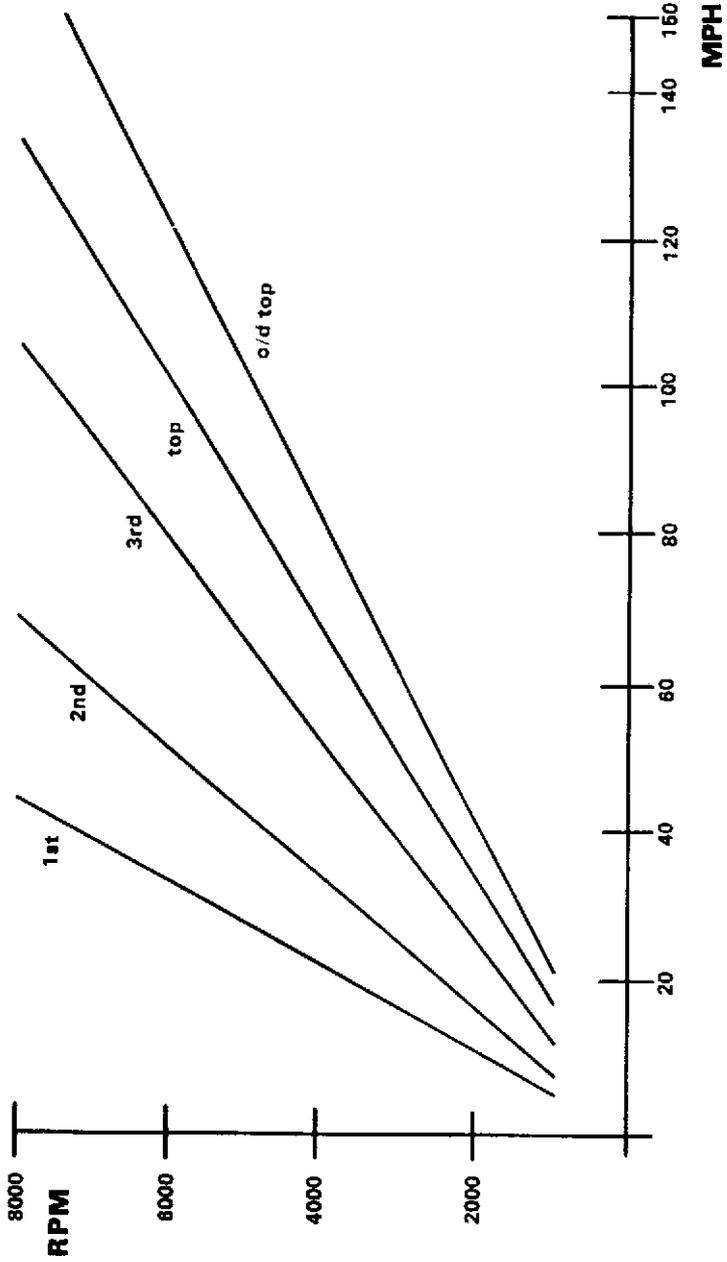
TR4, 4A, 5 & 6. 4.875 FINAL DRIVE



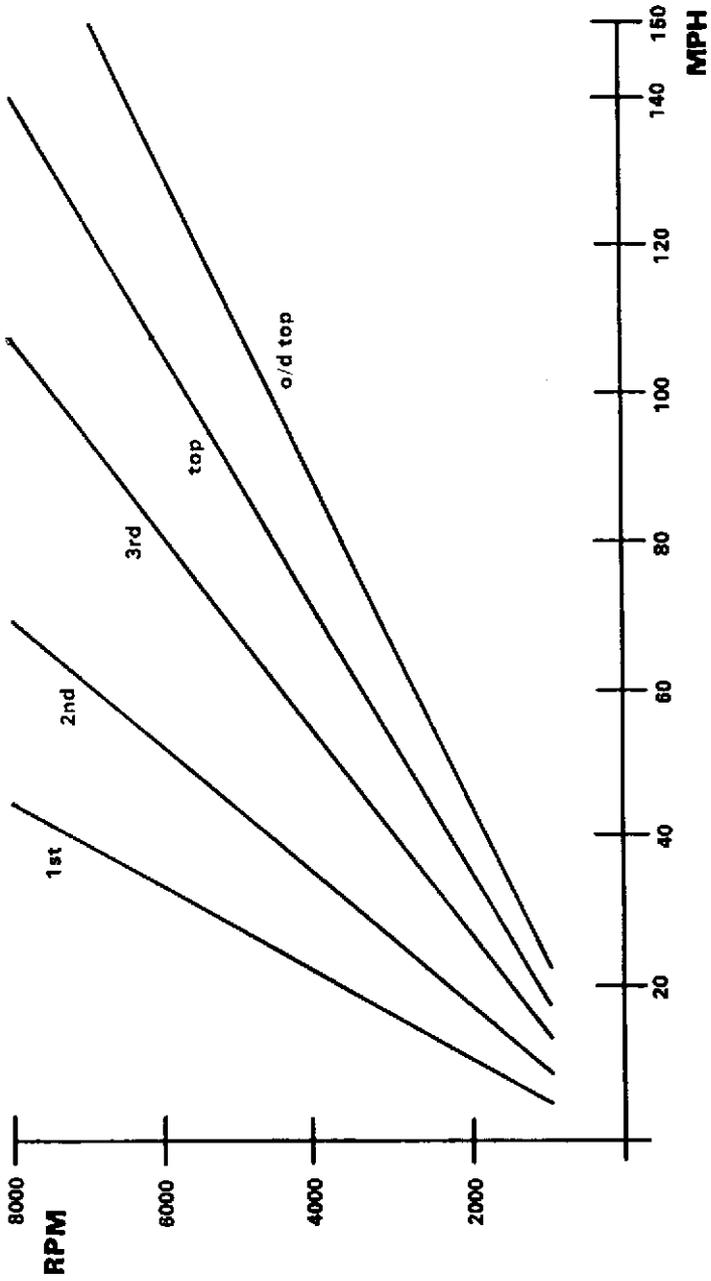
TR4, 4A, 5 & 6, 4.55 FINAL DRIVE



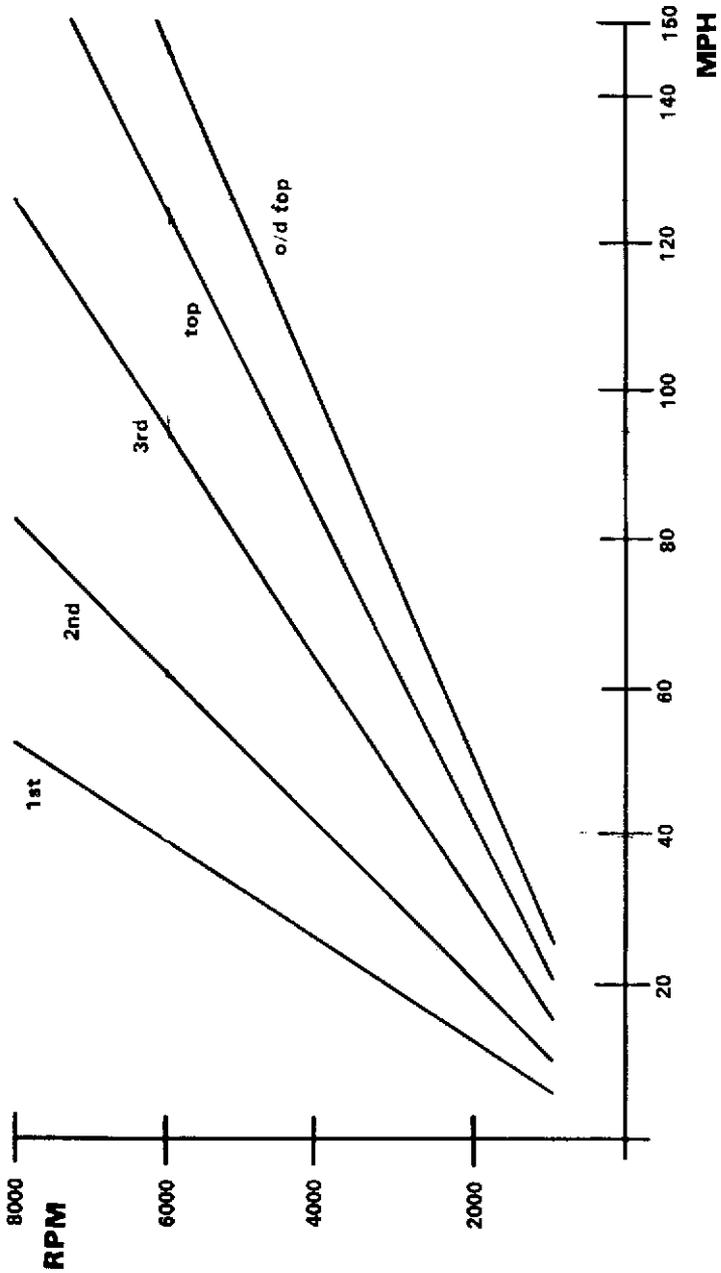
TRA, 4A, 5 & 6. 4.3 FINAL DRIVE



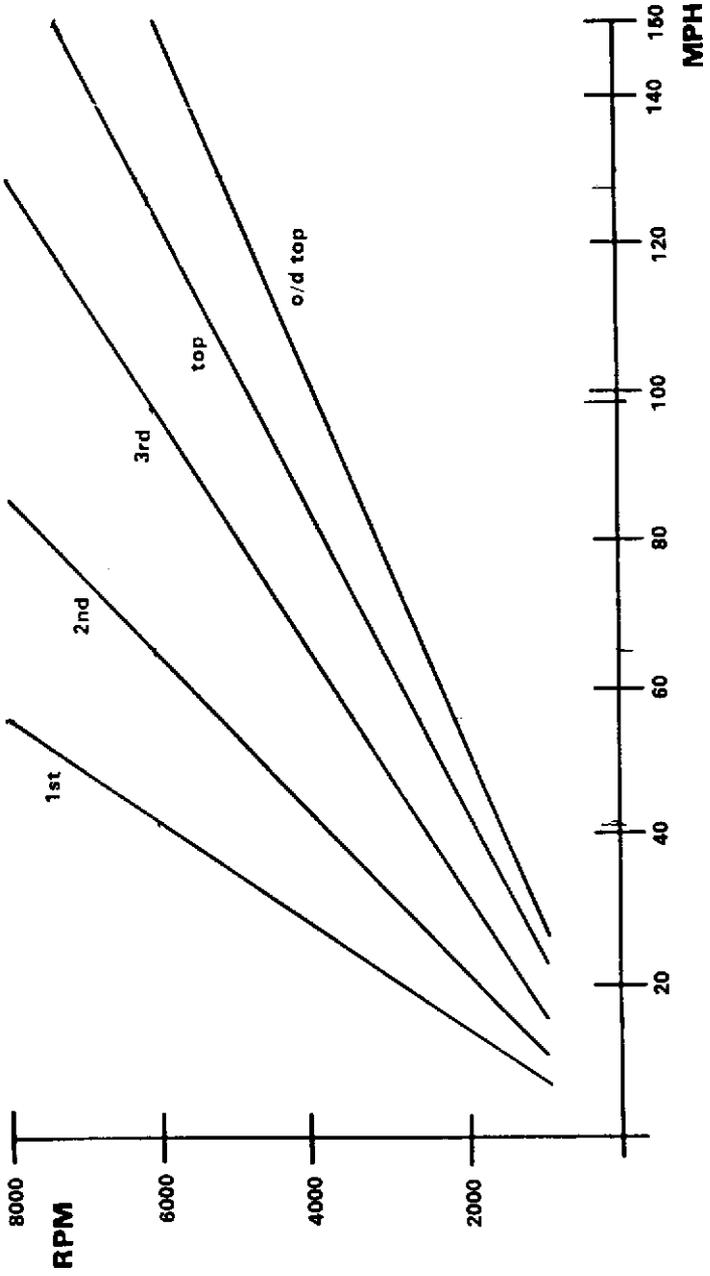
TR4, 4A, 5 & 6. 4.1 FINAL DRIVE

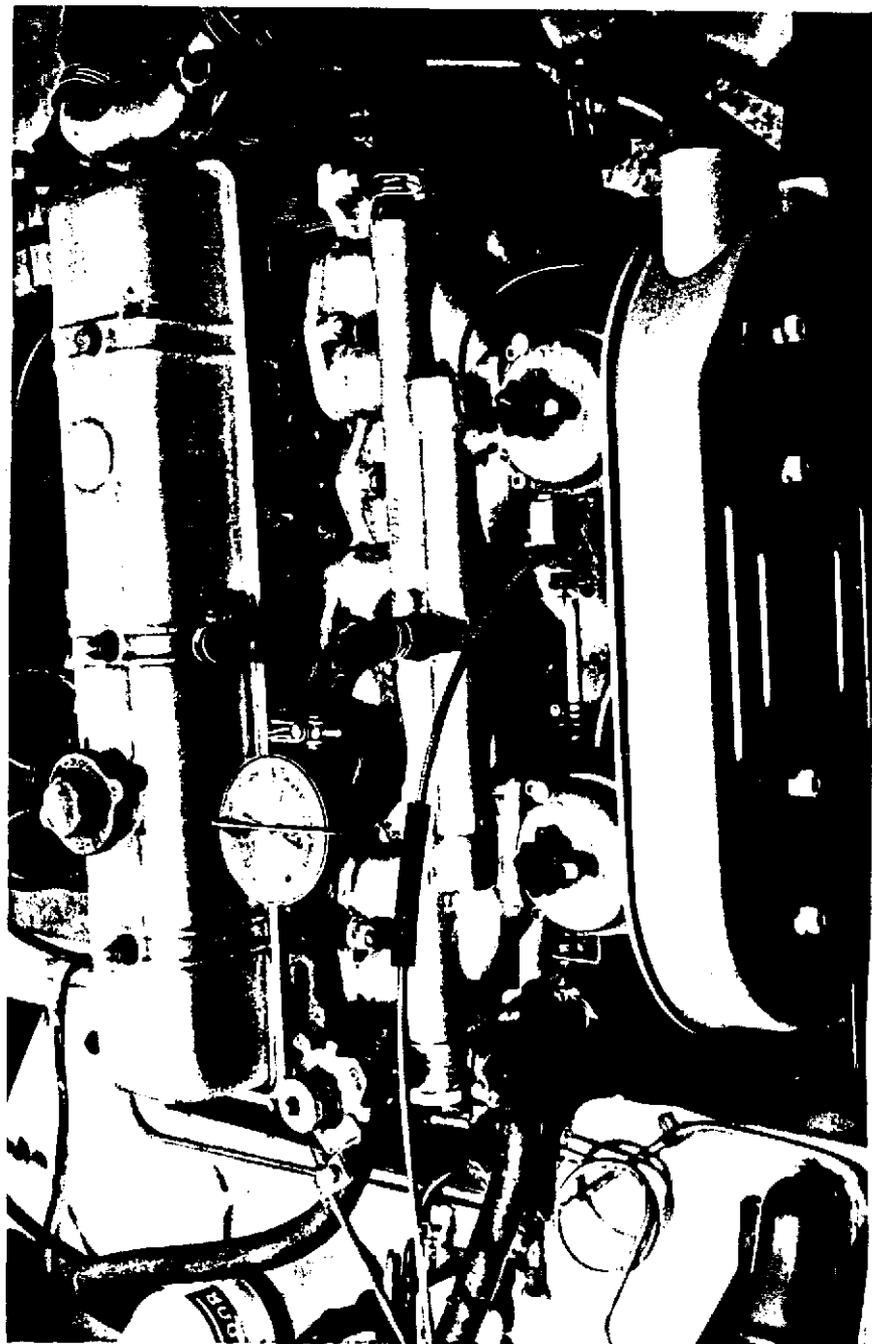


TR4, 4A, 5 & 6, 3.7 FINAL DRIVE

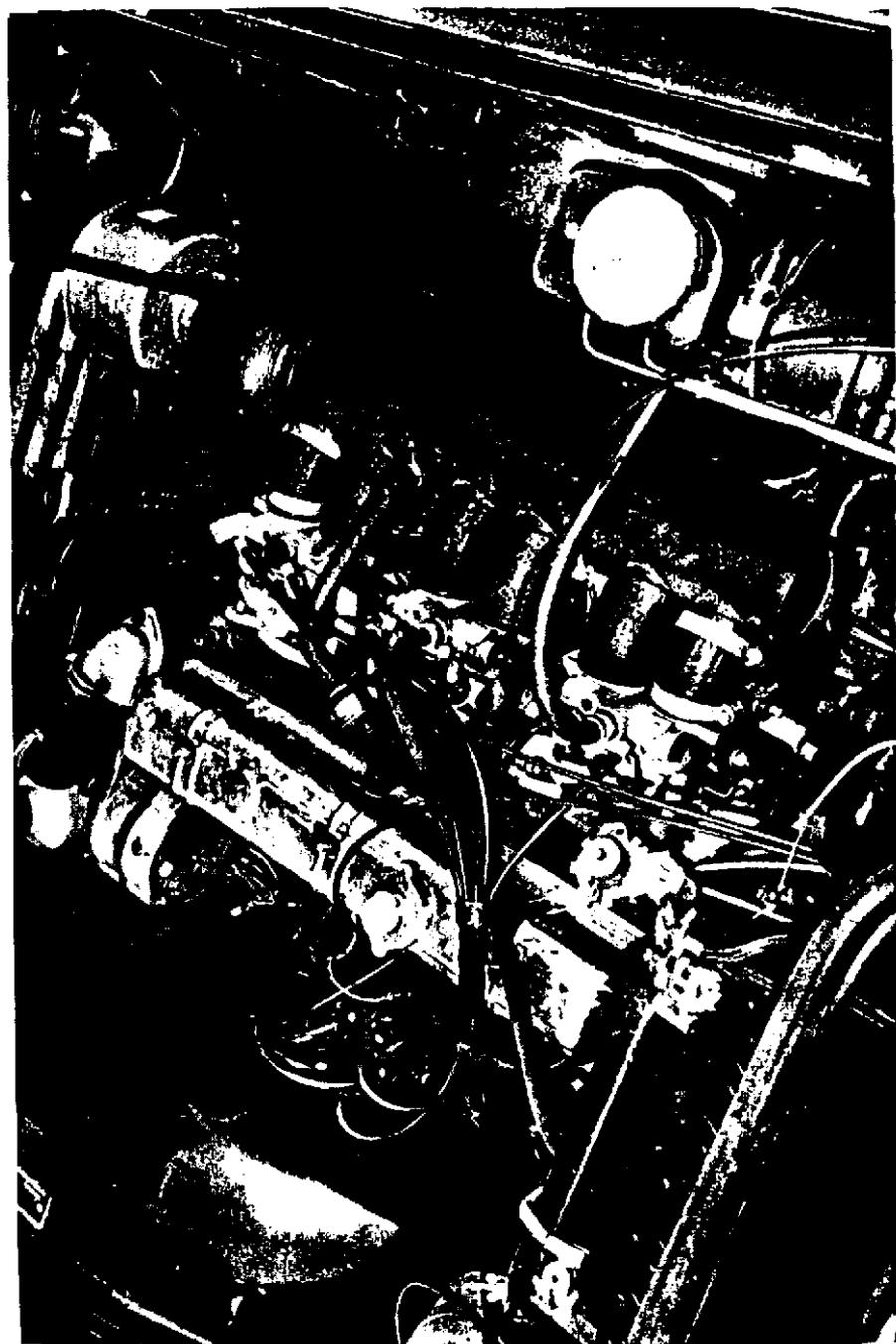


TR4, 4A, 5 & 6. 3.45 FINAL DRIVE





Twin Stramberg on the GT6 power unit.



Under bonnet view of an unmodified TR5 Pi

ratios other than these we can go to S.A.H. and they can supply ratios at 3.45/1, 4.3/1, 4.55/1 and 4.875/1. Since the 2000 saloon and 2.5 P.I. use basically the same final drive unit, all these alternative ratios will apply to these cars as well as the TR sports cars. When selecting a final drive for the TR sports cars or the big saloons, we must carefully consider what the final application of the car is to be. If a high cruising speed is required without excessive engine revs then a ratio higher than standard should be selected. If maximum top speed is required, one should stick to the standard ratio nine times out of ten. The only occasions which warrant the use of a higher final drive for all out top speed is when the engine has been made to produce a good deal more power without any appreciable increase in the r.p.m. at which it produces this power. Coupled to this we must bear in mind that both the saloons and sports cars are fitted with overdrives. This means that for top speed, the overdrive ratio is plenty high enough. If, by some chance, you have a car without an overdrive, then if the engine has had considerable modifications at the top end and a cam designed for torque, then it can conceivably benefit on top speed by using a higher final drive. The only drawback to this is that it will reduce the acceleration. The best solution to retain acceleration and get top speed is to fit the overdrive unit if it is not already fitted. By dropping the final drive ratio to the next lowest, the reduction in top speed is only marginal but the acceleration is better. For the best acceleration for the saloons we should go for the lowest final drive ratio or the one above the lowest, e.g. 4.875 or 4.55. With the 2000, the

4.875 will probably be found to be the best. The 2.5 P.I. should be just right on the 4.55. When using these low final drive ratios it is advisable to use the closer ratios of the TR sports cars in the saloon. If this is not done, one can end up with too much torque at the rear wheels in bottom gear. Excessive rear wheel torque just manifests itself in the form of excessive wheelspin which is not really the best method of getting off the line. If racing tyres are being used instead of road tyres, excessive wheelspin is far less likely to occur, due to the better grip of the racing boots. If all the relevant factors are taken into account, then the selection of a final drive ratio can become a little complex. Probably the easiest way to ascertain which final drive ratio will do the job is to look at the graph, relating engine speeds to road speeds in various gears, Fig. 51. This should give us a good insight as to what is going on. Other factors which will contribute to our selection of a final drive ratio, for maximum acceleration, are (a) the weight of the car, (b) the revs at which peak power occurs, usually influenced mostly by the type of cam fitted, (c) the spacing of the gearbox ratios, and (d) the amount of grip we have at the back wheels. We can apply some basic logic which will assist us to select the final drive, although it will be by no means foolproof. As a starting point we can assume that if no other conditions exist which are detrimental to accelerations such as too much wheelspin - then the lowest final drive ratio will give us the best acceleration. So at this point we arbitrarily select the lowest ratio available. However, much we may surmise, other factors do bring

to bear a considerable amount of influence, so we will not take these into account.

Firstly, if road tyres are being used, we have less grip available than is possible by using racing tyres. The extra torque at the wheels is therefore being wasted by lack of sufficient grip and this will make bottom gear less useful than it possibly could be. The answer is to go up one, maybe two ratios from the lowest. Against this must be weighed the effects of a closer ratio box. If ratios as close as those fitted in the Vitesse or G.T.6 are used, then we must consider going down on final drive ratio to compensate for the higher bottom gear. Since the Vitesse/G.T.6 bottom gear is nearly as high as a 2000 or 2.5 P. I. second gear, one would anticipate going down one or possibly two ratios from the ratio which had last been arrived at due to other considerations.

If the car is lightened considerably, then we can again come to the stage where it is possible to have the rear end gears at too low a ratio. With the big Triumph saloons and the TR sports cars we can say approximately that each 10-15% reduction in weight will call for a final drive ratio one ratio higher than the lowest available. We cannot make this too much of a hard and fast rule because it does depend where the weight has been removed. If the reduction in weight means that both front and rear wheels become less loaded by an even amount, then the aforementioned approximation will hold fairly true. If the lightening, instead of being 50/50 front and rear, is 70/30 front and rear, the effect on the final drive ratio will be less so this must be taken into account.

The last point to be considered is

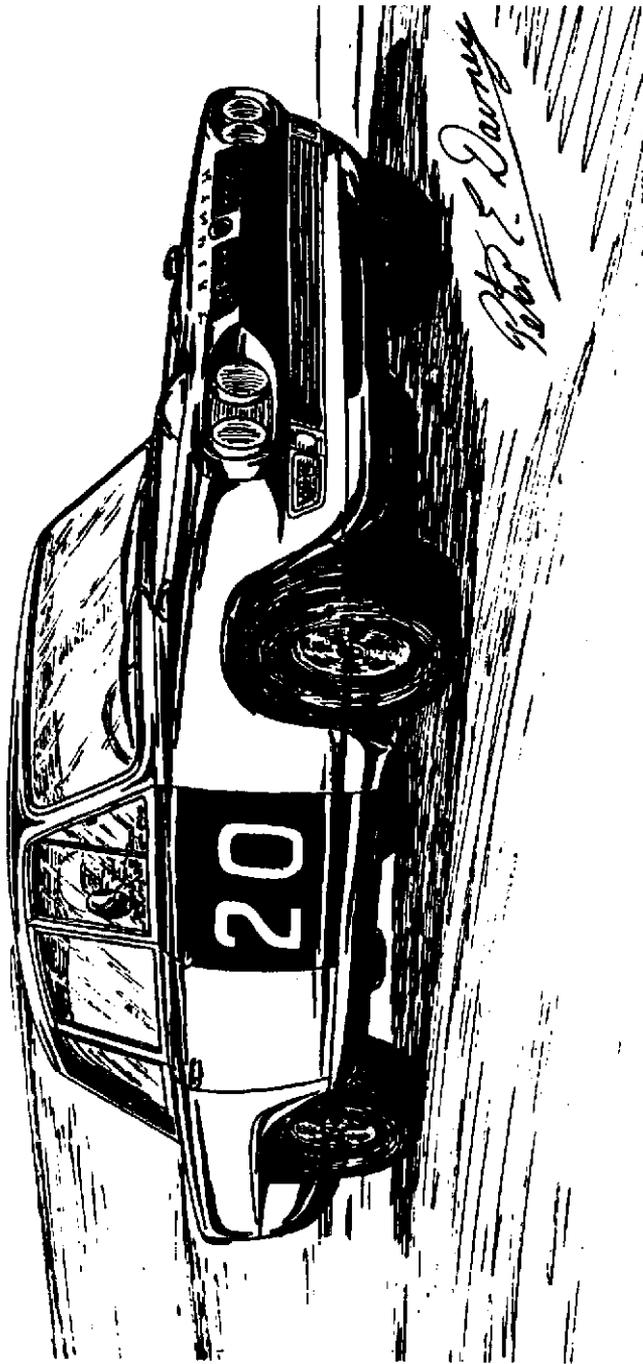
the effect of the camshaft, since this has a distinct bearing on the width of the power band. A cam up to about half race spec. can usually run one ratio higher than a full race cam. Strictly speaking it is better to say that the full race cam should be run together with a lower final drive than a half race cam, primarily because it enables one to make better use of improved top end power. Superimposed on top of all the aforementioned considerations is the fact that the car must have sufficient top speed with the final drive ratio selected. If the car is to be used for competition, the best plan is to find out from other competitors using a car with a similar power to weight as your own, what speeds they attain. You then select your final drive to give you a top speed somewhere in the region of 10-15% in excess of this. The graphs given in this chapter should help in your choice of final drive ratio. The technique to use is as follows: first establish the r.p.m. at which peak power occurs. Multiply the peak power r.p.m. by 1.1, i.e. increase it by 10%. Using this as datum r.p.m., find the various speeds given by various final drive ratios relevant to your car on the graphs given. Select the ratio that most closely gives you the top speed required. If you find your selected speed falls half way between two ratios, the best bet is to start off with the highest one. Selecting a final drive for the road is less difficult than for track purposes. You need only decide on what top speed will suffice, assuming the engine has enough power to reach it, then look through the graphs until you find the ratio which will give you that speed at an r.p.m. 10-15% over peak power r.p.m. If you are gearing for all out top

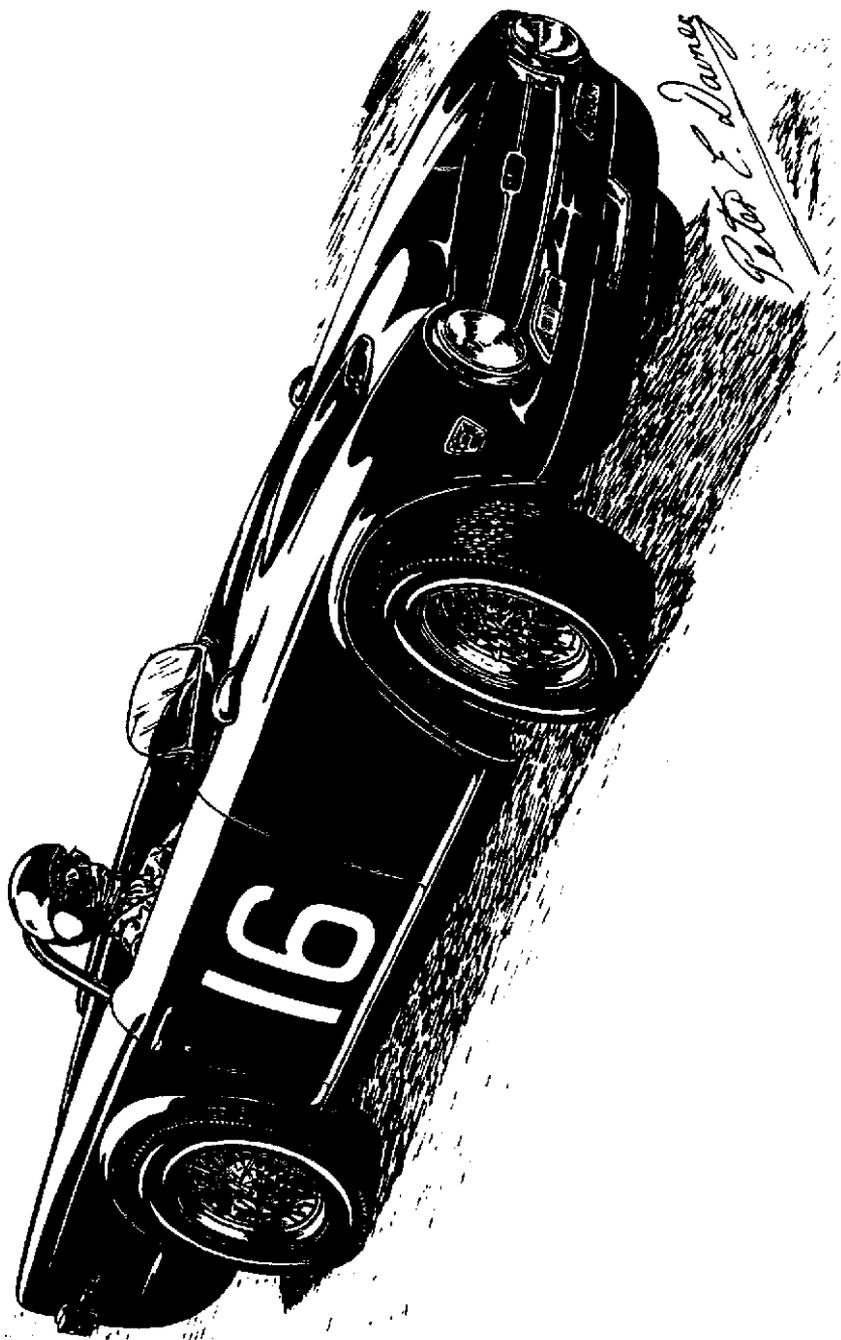
speed, do not think that the highest final drive will necessarily give you this. The final drive ratio for top speed must be equated to engine power output. The higher the final drive, the higher the power, or to be more precise, the torque output of the engine to pull it. If the engine is not putting out the power required, overgearing will serve only to reduce the top speed. If you are gearing for top speed, then you need to know exactly how much power is required to reach a given speed. If your engine is developing the power required to reach your target speed, then you select a final drive which will allow the peak power r.p.m. to coincide with the top speed. Practice shows in most cases that unless engine torque has increased appreciably then the highest top speed is usually achieved with the standard final drive or one either side of it.

Having dug into some basic theory on final drives, I think it best that we should come back to reality and discuss just what is available for the G.T.6 and Vitesse in the way of rear end gears. The standard ratios for the

G.T.6 and Vitesse are 3.89/1 or 3.27/1. To augment these two, S.A.H. can supply ratios at 4.11 and 4.55. For most purposes, this range of final drives is adequate and you should find one of the four suits.

Having got the engine to put out a respectable amount of power it is essential that we get it down to the road. To help power transmission to the Toad a limited slip differential is a really useful item to have. Should there be appreciably less grip at one driving wheel than the other, the power will get transmitted through to the wheel with the grip. The result of fitting such a device is a better take off from rest, due to less wheelspin. It also allows the throttle to be applied harder and sooner when coming out of bends, especially tight ones. For any form of competition work, a limited slip differential must be considered a vital piece of equipment. Limited slip differentials for the cars in question can be obtained from S.A.H. who, as you have probably guessed by now are one of the biggest suppliers of Triumph tuning equipment.







SUSPENSIONS AND BRAKES

THE pleasure and ease of driving any particular car can be made or marred by its roadholding or braking qualities. Should either of these departments be seriously deficient, then it is possible to have a downright dangerous car. All the Triumph cars with which we are dealing have roadholding and braking compatible with the power output of engines in standard or mildly modified form. The braking department is usually up to a far higher level than the roadholding, and in consequence quite an amount of extra performance can be handled by the brakes before any deficiencies start to make themselves felt. If you find that your particular car seems to lack the roadholding or braking power you think it should have, do not immediately jump to the conclusion that it needs to be modified to effect a cure. Brakes and suspension are prone to wear just as any other working components. Any wear in the suspension, steering and brakes can bring about a big reduction in the handling qualities, so before any modifications are considered, make sure that wear is

not the culprit for below par performance.

If the car is to be used for a specific purpose and you definitely want to modify the suspension or brakes, then any worn parts can be replaced during the modification process. As such, we will run through the procedure for modifications to each car which will either improve roadholding or braking make the car handle in the fashion you want it to.

TR4

The first move on suspension modifications for the TR4 is to fit an antiroll bar, which can be acquired from V. W. Derrington or S.A. H. This gives a marked reduction in roll during cornering, it increases the maximum cornering ability and on top of this, gives the driver a greater confidence as the car feels more sure footed.

The next step from here is to fit firmer shock absorbers.

A good combination to use would be the Spax adjustable on the front with the Armstrong Firmaride on the back, but of course this is only

		Front	Rear
Armstrong	Adjustaride	-	-
"	Firmaride	-	R-7999/-DAS9
Koni	Special D	80-1005	80-1108
Spax	Adjustable	S164/209	-

personal preference and you may have your own idea as to what should be used. Armstrongs do, apart from the shock absorber listed above, make a special range which are available only through S.A.H. These shocks are available as adjustable or fixed for the front and fixed for the rear, which is as listed above.

Wider wheels can improve the roadholding but for road use with road tyres we should not overdo things. For the front 5 1/2.-6J wheels will work out just right. On the rear 6-6 1/2 are needed. Although alloy wheels such as 'Minilite' are the best, they are quite expensive. As an alternative, one can use widened steel wheels which works out far cheaper. A firm which caters for widened steel wheels and produces a very fine finished product is Weller Racing Wheels, Edenbridge, Kent.

If the suspension is to be modified for racing or rallying, we will need to go a little further. To start with, stiffer springs are needed and maybe some alterations to the ride height. Firstly on the TR4 two types of front springs are fitted. One is short and the other is long. The short ones were used on the earlier TR4 which has a packing block above the spring (see Fig. 52). The later ones did not have this packing block and the distance lost was made up for by a longer spring. The short spring (free length of 9.75") has a rate of 310 lb. per inch. The long spring (free length of 11.1 ") has a rate of 312 lb. per inch. A competition front spring which is available through

S.A.H. has a rate of 380 lb. per inch and a free length of 9.2" which means it is suitable as a straight swop on the earlier TR4. To use the competition springs on the later models we have to reintroduce the aluminium packing piece. If the competition springs are used, additional lowering can be achieved by machining the aluminium packing piece. These can be machined top and bottom to achieve the desired ride height (see Fig. 53). The amount to be taken off the blocks is not in direct proportion to the amount of lowering. The ratio between metal removed from the blocks to the change in ride height is about 5/8:1 i.e. 5/8" off the blocks will lower the ride height by about 1 ". When refitting the blocks do not refit the top rubber packing piece. If the ride height needs to be increased as would be the case for rallying, then this can be simply accomplished. On the early models, with the packing piece, fit the later long spring. On the later models, with the long spring, fit the packing piece. In each case this will raise the ride height of the front by about 1 3/4 - 2" If this is too much, then the packing pieces can be machined to make the necessary adjustments. If the S.A.H. competition spring is used which is 9.2" long, shorter than either standard spring, then the ride height must be adjusted by fitting an additional packing pie - That is, two packing pieces will be needed at the top of each spring, the standard one plus one to make up for the shorter length of spring.

top rubber packing piece
do not refit when lowering
by machining ally packing blocks

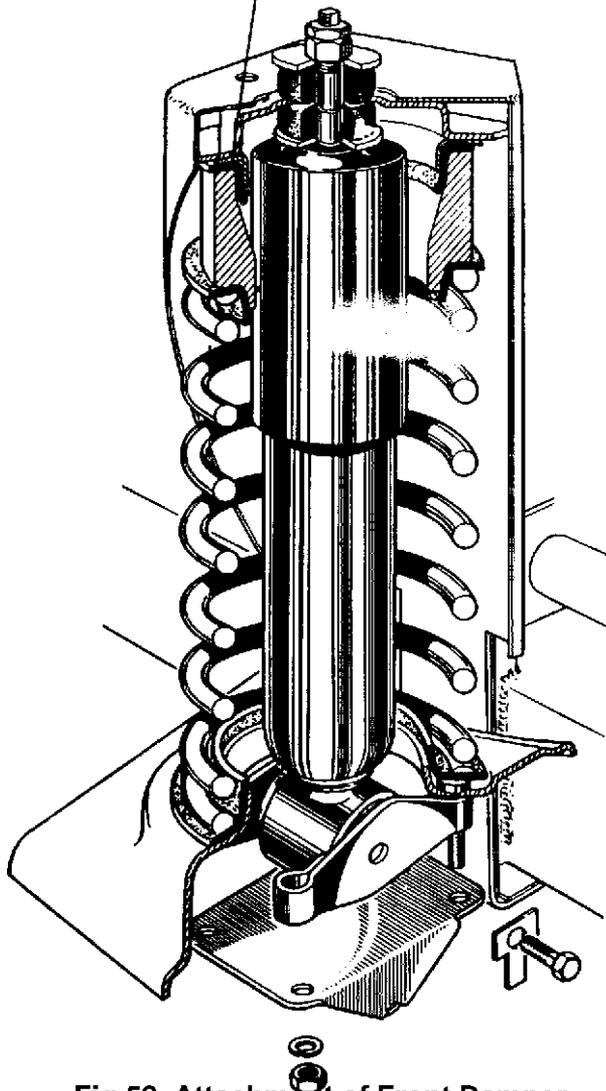


Fig 52. Attachment of Front Damper

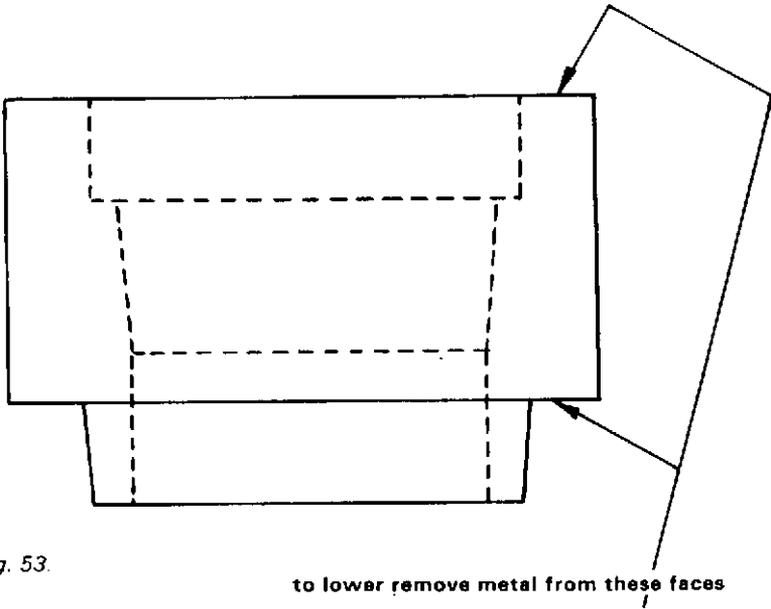


Fig. 53.

to lower remove metal from these faces

Again, if the ride height is excessive, adjustments can be made by machining one of the packing pieces.

The camber angle of the front wheels may, under certain circumstances, be found to be excessive as would be the case if either the ride height is increased or if the car is to be used with wide racing tyres. The standard camber angle is 2° positive although this could be as much as $3-3.5^{\circ}$ if the ride height has been increased. If the car has been lowered, the camber angle will probably be down to $1-1.5^{\circ}$ positive. If the ride height has been raised, for rallying, then the camber angle should be reduced to $0.5^{\circ} - 1^{\circ}$ plus when laden. Laden weight is equivalent to two passengers 150 lb. each. If the car has been lowered for racing and is using wide racing tyres, then the camber angle should be set at $0^{\circ} - 0.5^{\circ}$ nega-

tive. To reduce the positive camber you will need to make up new pivot plates for the bottom wishbone of the suspension. Fig. 54 shows exactly which plates need to be remade. For each degree of positive camber reduction required, move the pivot bolt hole in the plate 0.170" outwards.

Early TR4's had a 0° caster angle whilst later ones have a 3° caster angle. You can tell which is which by the top wishbone. If it is a one-piece item made up of sheet steel in a rectangular box section form with the top king pin or steering knuckle joint stuck on its outer edge, then the car has 0° caster. On the other hand if the top wishbone is a two piece affair with the top king pin or steering knuckle clamped between the outer ends then the car has 3° of caster. The high speed

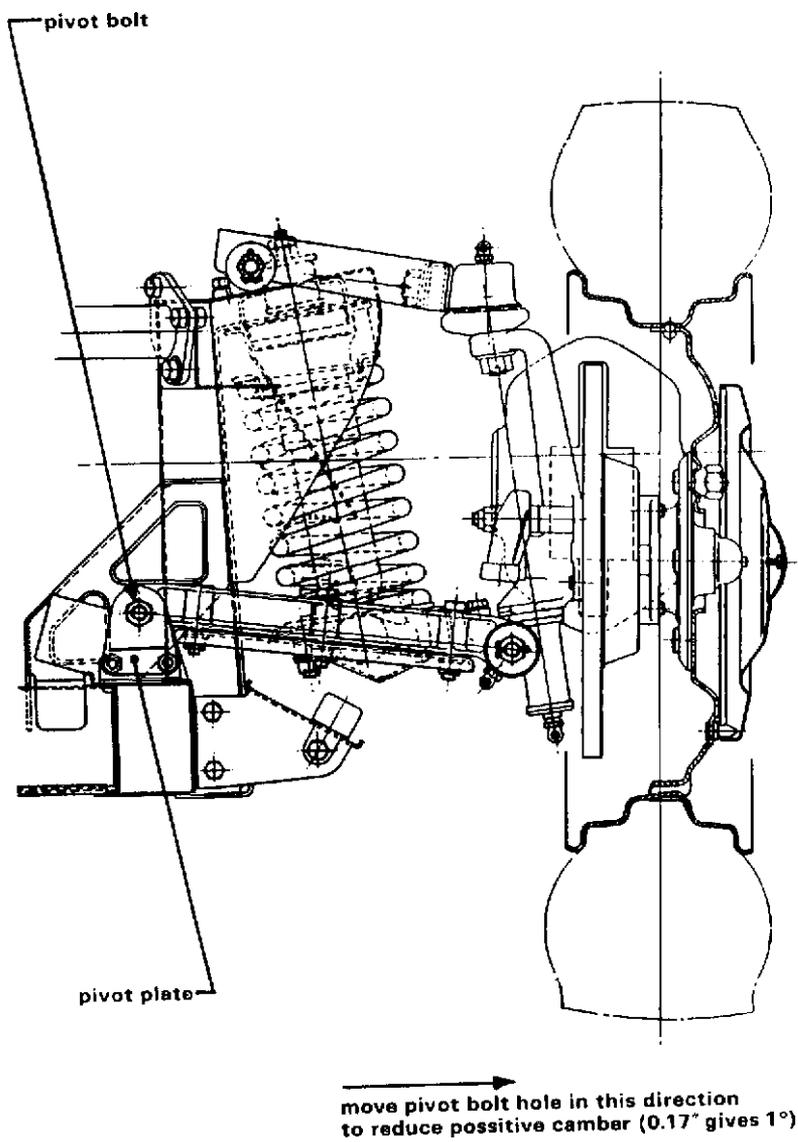


Fig. 54. Front suspension TR4.

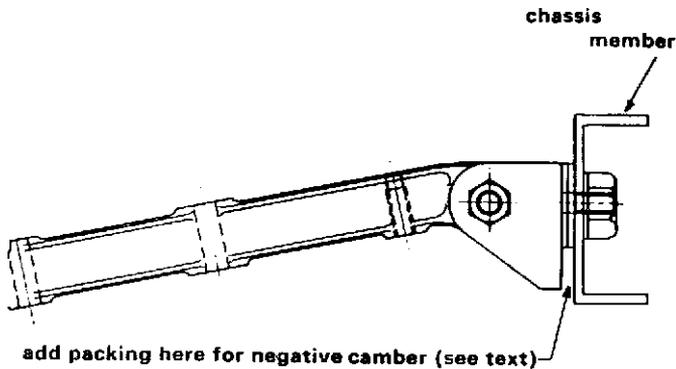


Fig. 55. Front suspension TR4A.

handling will be found to be better with the 3' of steering caster. The early models can be converted by fitting the later parts. The parts needed are: (a) two new top wishbones and knuckles, (b) tie rod lever i.e. the lever attached to the hubs to accept the steering motion imparted by the tie rods, (c) new bottom king pin trunion or lower trunion bracket as it is sometimes called.

Rear suspension

Having flogged our way through the front suspension, we had better take a look at the rear suspension. The first thing that you will find apparent if you look at enough TR4's is that some have a large aluminium block between the axle and spring, and some

have not. It is, in fact, the early ones which do not have the spacer. The object of the spacer on the later cars was to reduce the effect of roll steer and it is very advisable to incorporate this spacer, especially if the front end has been decambered. If this is not done, the oversteer, when cornering hard, may prove excessive. To convert the early rear suspension to the later one, you need to acquire the later recambered springs (part no. 209964) together with the spacer blocks and larger U-bolts. The only drawback to using the spacers is that with a fairly highly tuned engine, axle tramp is likely to occur. If the stronger rear spring supplied by S.A.H. is used, then the possibility of axle tramp is eliminated. The S.A.H. spring must be used in conjunction with the spacing

blocks or the ride height will be about 3.5° too high. For rallying the rear end can be made higher by shortening the spacing blocks although there is a limit to how much the standard blocks can be shortened due to them having an oval hole in the middle. Since there is nothing very complex in the nature of the blocks, it is a straightforward job to make some shorter ones that will give the required increase in ride height. Lowering is obviously the reverse of raising and can be achieved by increasing the thickness or height of the spacing block.

As far as braking goes, we find that the standard braking can handle quite a bit more go. For competition purposes, a brake booster plus competition linings will give the performance necessary.

TR4A

All that has been said about the later type TR4 front suspension applies to the TR4A with a few exceptions.

Firstly the standard camber angle is between '1/2° plus-1/2° minus, and if the car is lowered it will end up around 1° negative which is just about right for a circuit racing set-up using the stiffer front springs and racing tyres. If the car's ride height has been raised, the camber angle will be about 1° positive and this should be reset to 0°. This can be done by adding packing pieces where the bottom suspension wishbone bolts to the chassis, see Fig. 55. The TR4A rear suspension does, of course, differ vastly from the TR4 in as much as it is independent. S.A.H. offer two springs which will lower and stiffen the rear end. One is 1/2" lower and the other 1" lower. In its lowered state the camber angle of the rear wheels is about 1.5 - 2°

which is just a little too much for use with racing tyres, although it is all right on the road with road tyres.

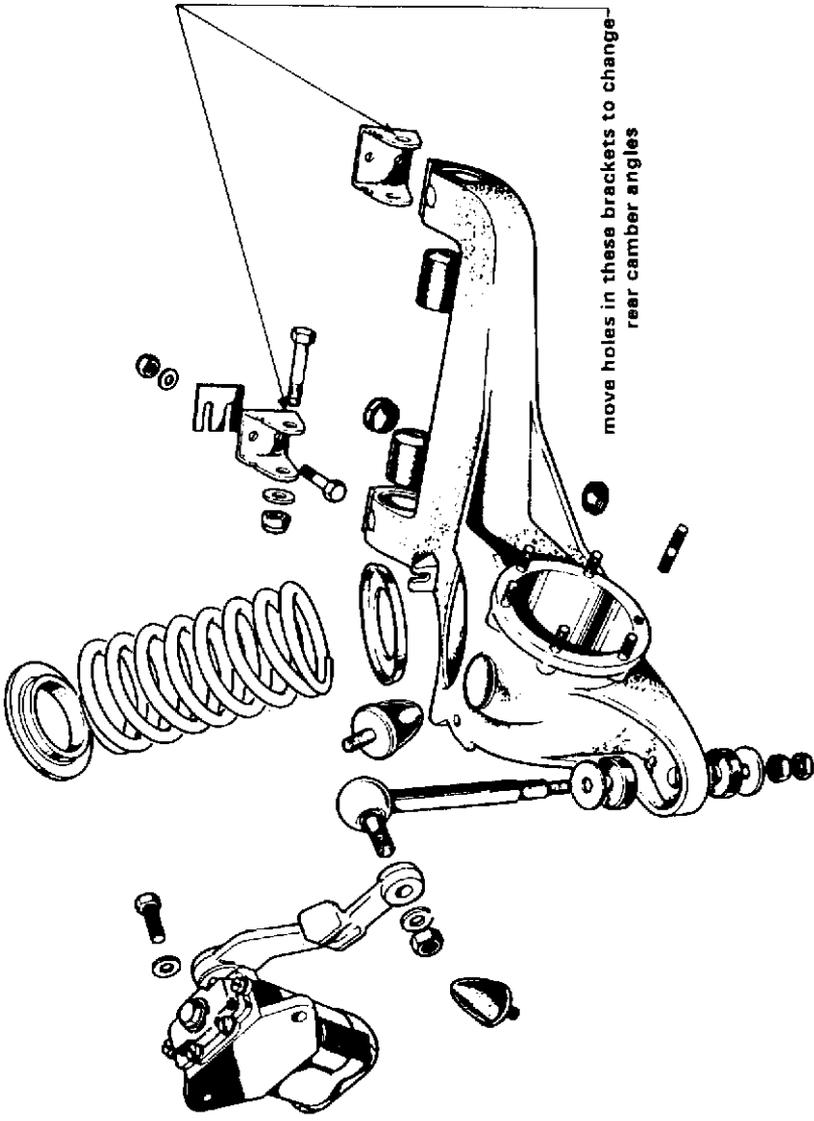
To reduce the rear wheel camber it will be necessary to move the holes in the fulcrum brackets which secure the suspension member to the chassis, see Fig. 56. The inner end of the member will have to be moved up and the outer end down, to reduce the camber. This means moving the holes of the inner bracket down and the holes of the outer bracket up. It depends on how much the car is lowered as to whether both brackets will need altering. If the excess camber is no more than 1° - you should aim for 0- 0.5° negative with racing tyres - we need only alter the outer bracket. The holes in the bracket have to be moved about 1/8" for each degree of camber change. The easiest way to do this is to fill in the original holes with weld, file the weld flush with the surrounding metal and redrill them in the new position.

For increasing the ride height for rallying, one can obtain special adaptors from S.A.H. but these can only be used with the standard springs.

As with the TR4, the braking department needs only minimal attention (assuming it is in perfect working order). If a servo is not already fitted, then it is advisable to fit one. This, together with harder linings, will do the trick. The shock absorbers will be different to the TR4 and the Armstrong R-8686/1 - DAS10 will be found to be suitable for the TR4A rear end. Check on rear roll bar.

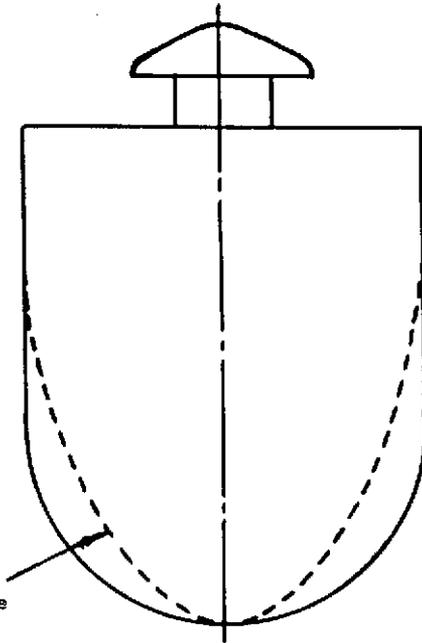
TR5 and 6

The TR5 and 6 can be treated as for the 4A. The only major difference is that the TR6 is fitted with an antiroll bar as standard. The brakes can



move holes in these brackets to change rear camber angles

Fig. 56. Exploded rear suspension.



reshape to this line

Fig. 57.

also be treated in the same manner. One thing that should be mentioned in connection with wheel rim widths, and this is applicable to all independently sprung TRs in connection with racing tyres. Rim widths should be about 7" for the front and about 8" for the rear. If we go too wide with wheels and tyres the suspension set up becomes hypercritical and as a result both handling and roadholding can suffer.

2000 and 2.5 P.I. saloons

The front suspension of both these cars needs very little doing to them. The first modification considered should be the fitment of an anti-roll bar. This will bring about a considerable increase in roll stiffness but should be ideally used in conjunction with a rear anti-roll bar. Incidentally both front and rear antiroll bars are

available from S.A.H. For further improvements the ride should be stiffened up by fitting harder shocks such as the Armstrong model AS 1514/5 or Koni 82P-134a. For road work a pair of 6" wide wheels on the front end will increase the road holding further.

Lowering the front end can present a problem in as much as there are no shorter springs readily available. The only way round this is to shorten the existing springs. The standard spring specification is 129.5 lb. per inch, a free length of 13.23" with 6.5 working coils. If we cut off one working coil, i.e. shorten the spring by 2.03", the ride height will be reduced by 1.78" and the spring rate will increase to 136 lb. per inch. Of course if you do not want to lower it as much as 1.75" then cut less off the spring. Over the range we are concerned with, the car is lowered by approximately

7/8 of the amount the spring is shortened by. If the car is lowered by 1.75", which is suitable for track work but a little too much for the road, the bump stops should be made more progressive. This can be done by reshaping them as shown in Fig. 57.

The camber and caster angles on the front suspension are just right for high speed work and should not be altered.

Should anyone decide to race one of these saloons, I would suggest the front wheel width with racing tyres should be about 7-7.5". All the extra offset on the wheels will need to be outwards and a thin spacer, about 1/8" thick, may have to be used to stop racing tyres fouling the bottom spring retainer on the suspension strut.

As mentioned before, the rear suspension should firstly have an antiroll bar fitted in conjunction with the front. Stiffer shockers can be fitted to the back and the following shocks will give the desired results. (Spax No. 164/2 16; Koni +80C-1 68, and Armstrong adjustaride A2933). The Koni unit needs special brackets in order that fitment can be carried out, so order these with the shocks. Stiffer rear springs are available from S.A.H. to stiffen up the rear end which are just right for rally work where the near original ride height is retained. For use where the car needs to be lowered, we are slightly better off using the standard rear springs shortened. To lower the rear end 1-7/8" you will need to reduce the working number of coils from 10.5 to 8.5, i.e. shorten the spring by 2.36". This means the ratio of spring shortening to lowering is again about 7/8 to 1. By reducing the number of working coils to 8.5, the spring rate will go up from 260 lb. per inch to 320 lb. per inch.

When the car is lowered, the camber angle on the back may be excessive. With road tyres 1-1.5° negative is OK but on racing boots 1/2 - 3/4° is required. The same method is used as on the rear suspension of the independent TRs for reducing excess negative camber. If racing tyres are to be used on the rear, then rim widths around 7.5 - 8" would seem appropriate. For road use 6 - 6.5" rim widths should be used.

Before winding up the suspension side of things on these cars (2000 and 2.5 P.I.) a little word of warning. For road use, if we are to avoid continual bottoming on the bump stops when carrying four passengers, lowering should be limited to about 1" front and rear. For racing when only one occupant is carried and the roads encountered, as on a circuit, are relatively smooth, the lowering can be carried out to a greater degree, up to say 1.75".

Vitesse and G.T.6 Mk. I and II

Before delving into the "ins" and "outs" of the suspension modifications, it should be pointed out that the rear suspension on the Mk. II G.T.6 and Vitesse differs quite appreciably from the Mk. I versions.

The Mk. I version uses the swing axle arrangement, which means as the wheel moves up and down so the camber angle of the wheel exhibits a proportional angular change, i.e. if the axle swings through 2° the camber angle changes 2°.

The Mk. II version uses a double wishbone type of rear suspension where the spring which is in the same position as the Mk. I version acts as the top link. The end result of the suspension change is that the degree of

camber change with respect to suspension travel is far less on the Mk. II cars than the Mk. I cars. Fig. 58 will show clearly the difference between the two types of suspension.

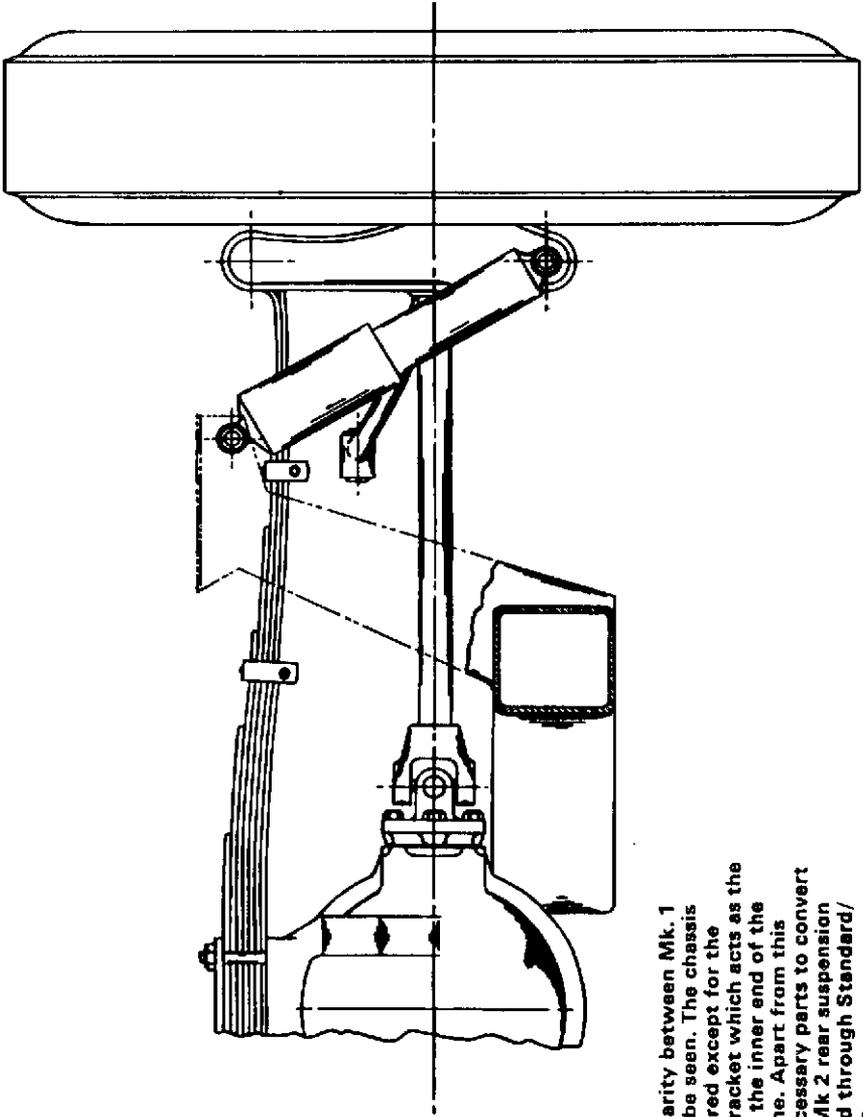
Since the general train of thought has got onto new suspensions, we will discuss the modifications to the Mk. I rear ends. The biggest drawback to swing axles is the fact that they tend to allow the wheels to tuck under when fast cornering is being indulged in. This tucking under completely spoils the handling of the car and it is one of the first things to give attention to when modding the suspension. There are two ways in which this problem can be overcome. Firstly one can fit the S.A.H. recambered rear spring which sets the static negative camber at 3° - 5° instead of the 1° to 0° found standard. This gives a vast improvement in the high speed handling qualities of the car and is fine when road tyres are being used. If racing tyres are going to be used, then 3° - 5° negative is far too much. The other alternative is to use the Speedwell camber compensator. When this is fitted it acts as a stiff but nevertheless flexible bottom link and effectively stops the wheels tucking under during hard cornering. If a camber compensator is used then the camber angle of the wheels is best left unchanged from the standard 1° - 0° .

If lowering of the rear end is contemplated, then the fitting of the S.A.H. recambered spring accomplishes recambering of the back wheels plus a degree of lowering, about 1"-1.5". If we are to lower the rear end without getting too much negative camber then life gets a little more complex. We could, by packing the spring up in the middle, lower the rear end (this would achieve almost

the same thing as a recambered rear spring) but we should end up with too much negative on the back. Really the only solution would appear to be to move the whole axle assembly up relative to the chassis. The easiest method to do this would be to remove the rubber bushes and cut off the small towers on the final drive front mounting plate (see Fig. 59). A corresponding change will have to be made to the rear final drive mounting by moving the holes in the chassis upwards by the same amount as the front of the final drive (see Fig. 60). All this effort should result in about 1.25" of lowering. If a camber compensator is then used with this set-up, we have a suspension which is usable for track work or road usage. Remember that we must not have excessive negative camber for use with racing tyres. The static laden negative camber should be no more than 1° .

For track or high speed road use, stiffer rear shock absorbers should be fitted. Any of the following type of shockers can be fitted to the rear end with good advantage; Armstrong Adjustaride A.2612, Spax 164/211 or Koni 80C-1389.

For use on the road 5.5" or 6" wheel rim widths are best and radial tyres should be used. Cross-ply tyres are prone to an early breakaway due to the large camber angle changes which can occur. If the rear suspension has been modified there will not be anything like as much camber change. Do not, under any circumstances, use cross-ply tyres on the back and radials on the front-it turns the car into a mobile death trap. If you are going to use racing tyres, which are definitely not



The basic similarity between Mk. 1 and Mk. 2 can be seen. The chassis remains unaltered except for the addition of a bracket which acts as the pivot point for the inner end of the lower wish bone. Apart from this bracket the necessary parts to convert from Mk 1 to Mk 2 rear suspension can be obtained through Standard/Triumph agents.

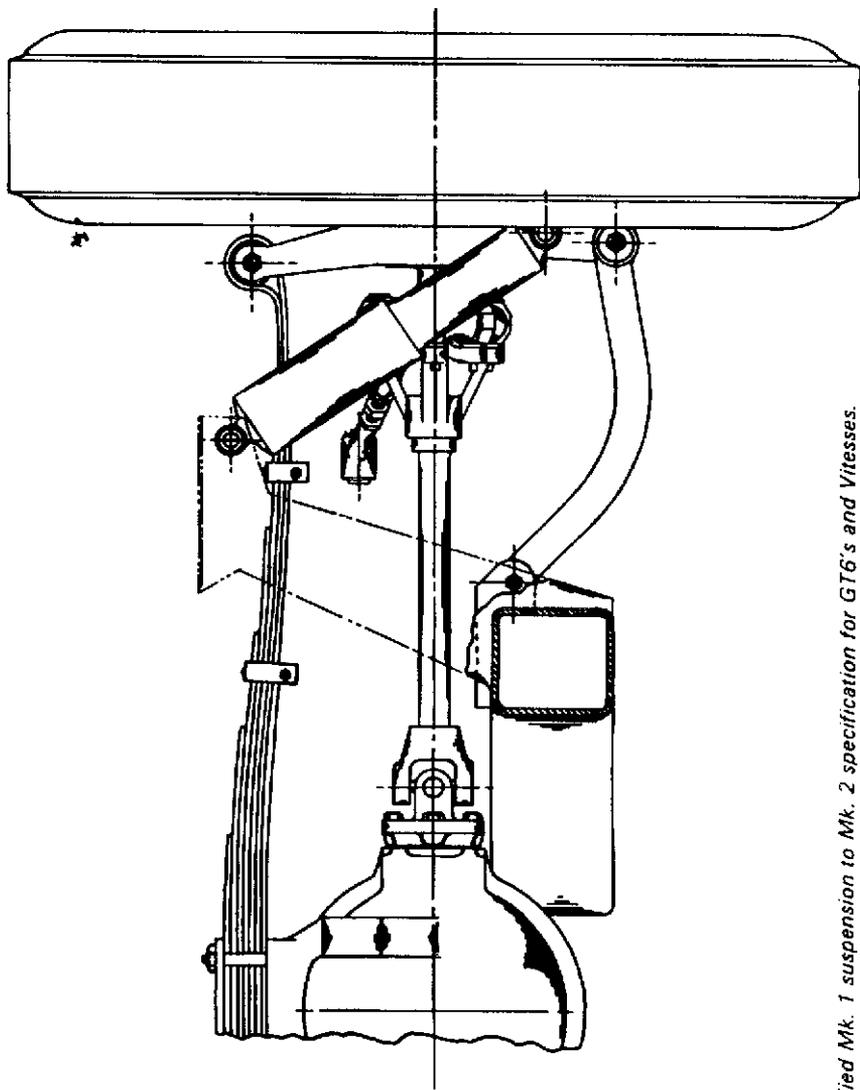


Fig. 58. Modified Mk. 1 suspension to Mk. 2 specification for GT6's and Viteses.

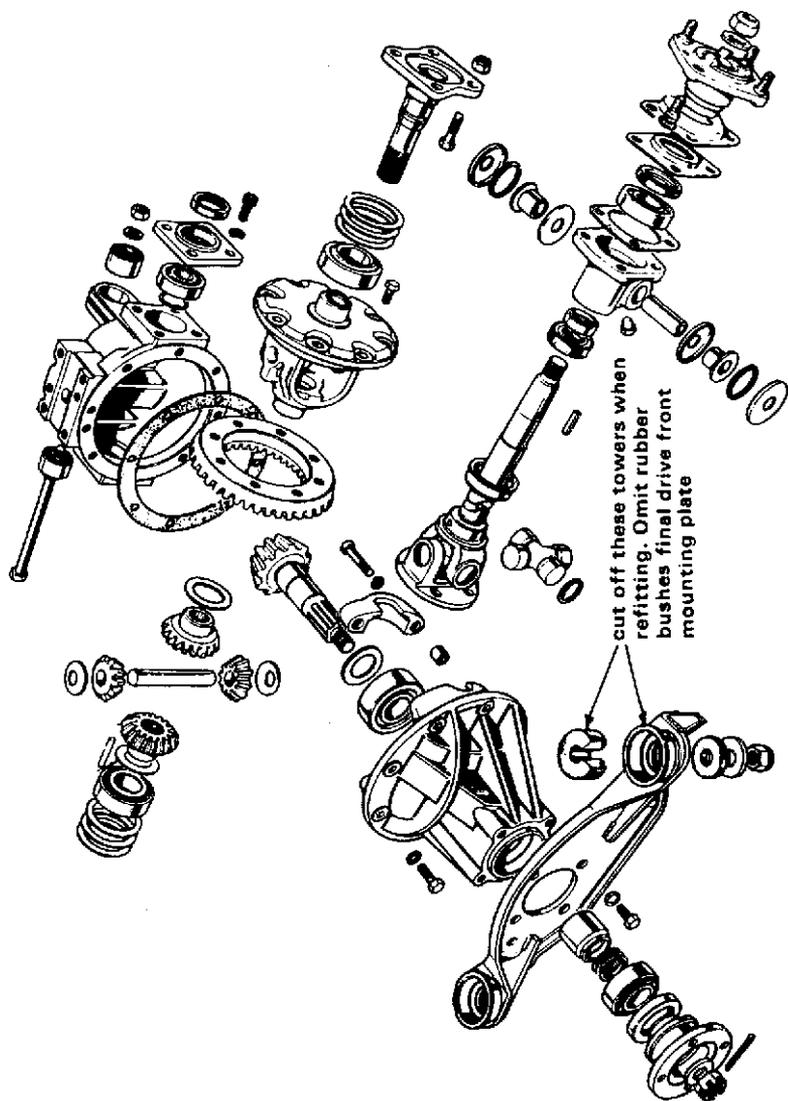


Fig. 59. Rear axle details for Viteses and GT6 Mk. 1 and 2s.

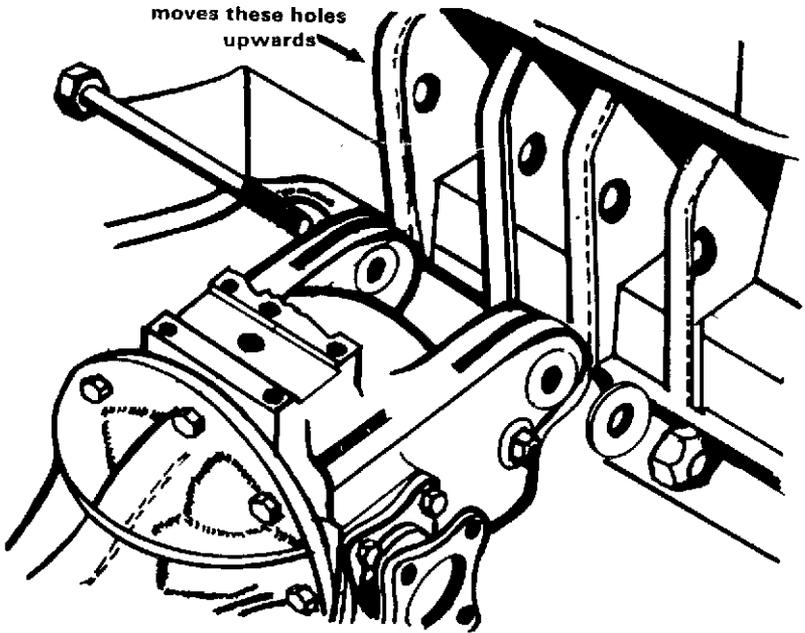


Fig. 60. Raising the differential on the Vitesses and GT6 Mk. 1 and 2s.

recommended unless the rear suspension is fully modified, rim width in the order of 7.5-8" will be required.

The ultimate rear suspension modification for the Mk. I G.T.6 and Vitesse is to convert it to the Mk. 11 spec. This can be done by purchasing the wheel hub carrier assembly, drive shafts, rubber doughnuts and bottom wishbones. A bracket needs to be welded on to the chassis to accept the pivot point for the lower wishbone. If this is done, the pivot point hole in the bracket needs to be in such a position to give 1° negative camber in the static laden condition.

This brings us very nicely on to the Mk. 11 rear suspension. Unlike the early suspension this does not suffer from the rear wheel tucking under, so no problems arise from this quarter. The first step which should be under-

taken to improve the handling is the fitting of harder shocks. (Armstrong R62-4181 or Adjustaride A.22236, Spax 164/211).

To lower the rear end we can either use a recambered spring or fit a packing block under the spring at the clamping point. With the rear end lowered by 1-1.5" you should find the negative camber is just right at about 1°. If racing tyres are to be used, then you should attempt to get the 1° negative as near as possible. If it is more or less than this, the pivot point of the lower wishbone should be moved in or out to compensate. This will have to be done by firstly elongating the hole to establish the correct position. Having done this, the excess part of the hole should be filled with weld then tidied up with a file. The under

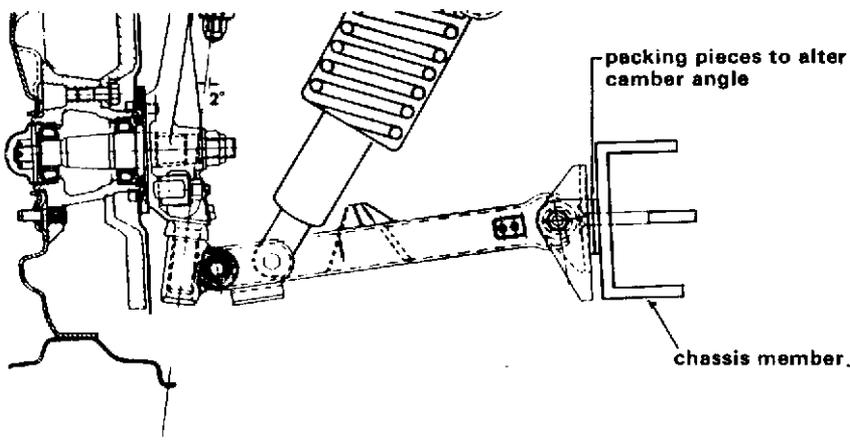


Fig. 61. Right-hand front suspension of Vitesses and GT6 Mk. 1 and 2s.

steer or oversteer characteristics can be influenced to a degree by adjusting the tie rods. There is no selling in particular that will give the best roadholding but it does allow the driver to set up the handling within limits to his own requirements. By shortening the tie rod adjuster shorter, the handling will tend towards understeer and by lengthening the tie rod, the tendency will be towards oversteer. By adjusting the tie rods we are altering the toe in on the back wheels. The range over which adjustments can be made for the purposes of setting up the handling are from $2/32$ " to 0 " toe in.

For road use, wheel rim widths of 5.5-6" are advisable while track use will warrant wheels of about 8" fitted with suitable racing tyres.

The first modification to the front suspension is the fitting of stiffer shock absorbers. (Koni 80H-1 388, Spax S164/210, Armstrong Adjustaride SA 2697). This enhances the stability and sure footedness when high speed cornering is indulged in. Going one step further, shorter front springs (available from S.A.H.) can be

fitted and these will reduce the ride height by about 1.5-2". The increased stiffness of these springs together with the reduction in centre of gravity height brings about a useful reduction in roll during hard cornering. They also bring about an increase in front wheel camber of about 1° , a degree more towards negative that is.

The standard front and camber angle is about $2-2.5^{\circ}$ positive in the static laden condition, i.e. carrying two 150 lb. passengers. The shortened spring will reduce this to $1-1.5^{\circ}$. For road use we can reset this to 0° by inserting packing pieces between the bottom wishbone pivot bracket and chassis (see Fig. 60). For circuit work the ideal camber angle is $0.5^{\circ}-1^{\circ}$ negative, this being right when the car is shod with racing tyres. Wheel rim widths for the front for road work will be the same as the rear at 5.5 to 6". For racing 7-7 1/2" will do the job nicely.

As far as brakes go, if you have a servo and harder linings they will adequately perform the job expected of them.

BODYWORK MODIFICATIONS

THE main subject to be dealt with in this chapter is the use of lightweight body components. The use of fibreglass replacement parts can bring about a useful reduction in weight and hence increase the power to weight ratio. In some instances there are glass components available which make a quite staggering reduction in weight.

Chris Williams (fibreglass components) is a specialist in fibreglass parts for Triumphs. One of his pet projects is a complete G.T.6 bodyshell in glass. This includes wings, doors, bonnet, boot, roof, undertray, bulkhead, etc., in fact all the original pressed steel parts are remade in fibreglass. The net result is a weight reduction of over 2.5 cwt. If you couple this to a racing set-up, having only the bare necessities, the overall weight reduction is just about 4.5 cwt. This gives us a percentage weight reduction of about 30% and that has the same result as 30% more power as far as acceleration is concerned.

Lightening

Although we cannot make such

large weight reductions on the other Triumph models, the weight reductions that are effected play a part in increasing acceleration. To give an idea of how even a small weight reduction can pay off, let us look at an example. Let us say that we fit a glass fibre bonnet on a TR4 and in doing so, lose 30 lb. or thereabouts. The fact that the 30 lb. lost weight is up the front end means that our weight distribution changes, giving more bias to the rear end. This, in conjunction with the decrease in weight, means that we will theoretically reach 80 m.p.h. about 0.1 second quicker. This small time difference might not sound like much, but it does mean a distance difference of something in the order of 12 feet. To put it another way, if you raced two identical cars with the exception that one had a glass fibre bonnet and the other one did not, the lightened one would be about 12 feet in front of the other by the time they reached 80 m.p.h.

To be more practical it would be a good idea to see just what is available for the cars in question for lightening purposes.

As mentioned before, Chris Williams does a complete G.T.6 body shell in fibreglass. Apart from this he can also supply bonnets or boots for the TR4, 5, 6 and the Vitesse bonnet unit. S.A.H. do a very smart Le Mans Spitfire type front end for the G.T.6. E.V.A. are also in the market for Vitesse and TR glassfibre components. The Triumph 2000 and 2.5 P.I. are not catered for as such, but both Chris Williams and E.V.A. Fibreglass Products will undertake the manufacture of oneoff components if approached.

The flaring of wheel arches often presents problems unless you happen to be an ace panel beater. If you do not feel up to flaring your own wheel arches, then quite frankly the best bet is to get an expert to do the job. The appearance of an otherwise immaculately prepared car can be completely spoiled by badly flared arches. Funnily enough, expert panel beaters can make short work of flaring wheel arches, and the job does not come as expensive as it would at first seem.

For competition work the use of

perspex windows is a great weight saver. You can save 30-35 lb. on a Vitesse by replacing the windows with perspex. Most regulations of course demand that the windscreen is of the laminated glass type so no change can be made there.

If rallying is contemplated then the underside of the car will need adequate protection. S.A.H. supply a range of sump guards and under shields for these cars. This piece of equipment is very necessary if we want to avoid leaving half the innards of the car on some projecting rock. For rallying, all the petrol pipes, brake pipes and wiring should run inside the car and not outside.

If your car is to be used for competition, then remember that nine tenths of the battle lies in the preparation of the car. Preparation does not necessarily mean large financial requirements. In most cases it just means careful, thorough and meticulous workmanship. This should apply to the whole car, engine, transmission, brakes, suspension and bodywork. The end result of painstaking work will be a very fast and reliable motor car.

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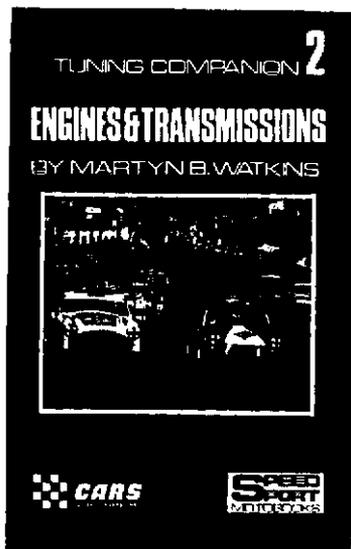
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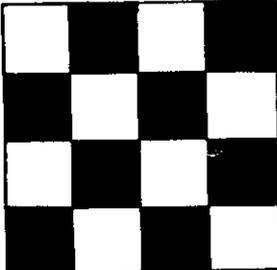
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