

RING AND PINION INTERCHANGES FOR THE MG TD/TF

Third Edition

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PREFACE TO THE THIRD EDITION

The VMG will no longer distribute this booklet. Instead, it will be available from Moss Motors Ltd. An addendum will be added to cover our experiences of the past year and a half but other than that the text will remain basically unchanged.
Brea, California May, 1977

PREFACE TO THE SECOND EDITION

A year has now passed since the first edition of this booklet appeared at the November VMG Swapmeet. In the intervening year a considerable number of MGA ring and pinions have been fitted into T series rear ends. In fact, exchange rear ends fitted with 4.300/1 gear sets have become an "off the shelf" item at one of the local T series garage/spares houses. This revision will include the accumulated experiences of the past year and a few necessary corrections in the text. The national speed limit of 55 mph is now adhered to reasonably well in the slow lanes on the L. A. freeways. A T series with a 4.300/1 rear end can be motored along at that speed in an almost dead silent manner. Though no one here has yet bothered to measure the reduction in fuel consumption achieved by this combination of lowered gear ratio and road speed, it is probably considerable. Since the first edition appeared I have made two separate attempts to have the measurements checked against original factory drawings. Neither met with much success. The interchange has, however, been made many times and all is well.
Brea, California November, 1975

PREFACE TO THE FIRST EDITION

This booklet is dedicated to the members of the Vintage MG Club of Southern California -- a group of devoted T series owners of unbounded enthusiasm and friendliness. Dean Woodyatt initially discovered that the MGA to T interchange was technically feasible and to my knowledge owns the first TD in which the 4.300/1 conversion was ever made. Many club members participated in the project of converting several rear axles in order to develop the procedures described here. The overall crew consisted of: Jerry Austin, Jim Bigler, Dale Mastell, Don Kosup, Pete Thelander, John Marsh and Laura Cederstrand. The typist was Joan Cederstrand. -
Brea, California November, 1974

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I. INTRODUCTION

The ring and pinion gears in the rear axle determine how many revolutions the engine must make to turn the rear wheels around once. The engine in a TD in 4th gear is required to make 5.125 revolutions for each turn of the rear wheels. When the car has proceeded 10 miles down the road, each piston has travelled a total of 4.7 miles up and down in its cylinder, a sobering thought.

This pamphlet will describe how the ring and pinion fitted into the TD or TF rear end may be changed to an alternate T series, MGA, or early MGB ring and pinion. The effects of this interchange on the performance of the car will be examined in detail. While it was originally planned as a short article to accompany a club slide show on the same subject, this text has now far outgrown its original boundaries. Much of the information that was intended for the talk during the slide show is now included in anticipation that it will prove useful to T series owners everywhere. Perhaps it will someday form a chapter in a book on the T series MG, but that is for the future.

Section II will summarize the performance data and hence provide a basis for choice among the five possible rear axle gear ratios.

Section III will describe the measurements and subsequent adjustments necessary to fit these different gear sets. The most general case will be considered. This is the case in which the rear end is to be built up from an odd group of T series castings and fitted with an MGA or early MGB ring and pinion. In this general case the factory instructions for the MG TD/TF are of little aid, since they presuppose that the original set of three castings (main casting, cover casting and differential carrier) are still together as that group originally assembled by the factory. Even if the castings from two TD/TF rear axles have not become mixed, the fitting instructions in the factory workshop manual are only applicable to the fitting of T-series ring and pinions. These T series gear sets are now very scarce and unavailable in the desirable 4.555/1 ratio.

Section IV will deal with other aspects of the rear end not immediately concerned with the ring and pinion interchange.

II. CHOICE OF A REAR AXLE RATIO

The rear axle gear ratio fitted by the factory achieves the maximum possible top speed from the T series engine and chassis combination. The engine speed at which the peak power occurs, 5200 rpm, is also that rpm required to drive the chassis at that road speed, 77 mph, which absorbs exactly the peak power available. *Installation of a rear end gear ratio substantially lower than the factory choice of 5.125/1 for the TD and 4.875/1 for the TF must necessarily reduce 4th gear performance.* The decrease in 4th gear hill climbing ability will be greater than the reduction in the top speed or the reduction in acceleration. This curious phenomenon will be examined later. Though these performance losses are inevitable, they are not completely undesirable since the design criteria for maximum possible road speed are directly opposed to the design criteria for engine longevity. The factory selected ratios are eminently suitable for either racing or stump pulling. They are, however, grossly unsuitable for cross country motoring! I will categorically state that operation of any T series engine at a steady 4500 rpm (this is 64.8 mph with the original 5.125/1 rear end) amounts to little more than an exercise in engine destruction. At about 10,000 miles a newly installed set of rod bearings will have developed fatigue cracks, a newly installed timing chain will have become noticeably slack, and the seating of new exhaust valves will have generally deteriorated to the point where a gentle "pist" is audible from the exhaust pipe. I have seen this occur over 20 years without exception to every T series that has crossed the southwestern United States at normal slow lane highway speeds with its engine flogging 5.125/1 rear end. When they were available, a change to the alternate 4.555/1 ring and pinion made it just possible to drive a TD or TF across country without also destroying the engine. You were still a slow car in the slow lane but at least you didn't feel you ought to ship an engine ahead just to drive across country and back. This ratio provides 65 mph at 4000 rpm and is the axle ratio I have run for the past 15 years in my TD. It is also that ratio that gives the TD and TF the same overall gearing as was supplied with the TC.

My TD has run on freeways and back roads from Canada to Florida and crossed the United States many times at a steady 4000 rpm. With this gearing, fatigue cracks generally appear in the rod bearings at about 30,000 miles, while the main bearings (center main first) begin to crack at about 45,000 miles -- a considerable improvement. This dictates a change of the main bearings with every other rod bearing change, provided the automobile is driven more gently toward the end

of the second set of rod bearings. An even lower rev. limit for continuous operation probably produces a corresponding increase in engine life. Here I have no accumulated T series experience to draw on but will have to wait several years for the results of the now numerous 4.300/1 interchanges to become apparent.

Note that by bearing life I do not mean the service life to catastrophic bearing failure but rather only to the appearance of fatigue cracks. Though shell bearings will continue to operate a considerable mileage after the initial fatigue cracks appear, the prudent enthusiast replaces them before they fail, not after they have failed and have galled the crankshaft. Similar observations can be made concerning the service life of all the moving parts in the engine and the drive train. While some components are relatively unaffected, the service life of most of the parts is intimately related to how hard the vehicle is driven and to the rear axle ratio fitted.

The 4.555/1 and 4.300/1 both have my unrestricted recommendation for general use. The 4.555/1 is perhaps a little more suited for twisting country roads, while the 4.300/1 is absolutely without peer for cross country travel.

The 4.555/1 is both a TD/TF and MGA ratio. Though the stocks of the T series version are probably exhausted everywhere in the world, there may still be a few MGA versions available. The MGA version (88G.284) was sold as a competition option. The 4.300/1 and 4.100/1 are the only new gear sets still available in large quantity. The 4.555/1, 4.300/1 and the 4.100/1 rear axle ratios have all been fitted to TD/TF series cars owned by members of the Vintage MG Club of Southern California. The owners of the 4.555/1 and 4.300/1 conversions are all universally pleased with the resulting performance of their automobiles. The driver of the single 4.100/1 conversion feels that while it is satisfactory for level roads (he does much freeway driving), it is too low for hilly country. A 3.909/1 is probably too low for a 1250 cm³ engine under all road conditions.

Fortunately for the T series MG owner there are sufficient technical data available so that the performance of a T series may be calculated in advance of the fitting of any alternate rear axle ratio. I will consider the fitting of a 4.300/1 ratio into an unmodified TD and strive to make the calculations so lucid that the most non-mathematical T series owner will follow. A change to a different tire size, rear axle gear ratio, chassis (TD to TF), or degree of engine tune will necessitate a recalculation since the performance of each combination is different.

Table I lists all the gear ratios available for fitting into the TD/TF rear end. These rear axle ratios are expressed as the ratios of the numbers of teeth

TABLE 1

REAR RATIO CHOICES

Rear Axle Gear Ratio	Miles/hour-1000 rpm* (4th gear)	BLMC Part No.
TD-IF Ratios		
8/41 (5.125/1)	14.40	AAA 1090
8/39 (4.875/1)	15.14	AAA 1087
9/41 (4.555/1)	16.20	AAA 1092
MGA Ratios		
9/41 (4.555/1)	16.20	88G. 284
10/43 (4.300/1)	17.16	88G. 283
10/41 (4.100/1)	18.00	ATB 7240
11/43 (3.909/1)	18.88	BTB 653

* These road speeds are calculated on the basis of tires which make 813 revolutions per mile. *77.93 / hr*

on the pairs of gears, i.e., 8/41 indicates 8 teeth on the pinion gear and 41 teeth on the ring gear. The numbers in parentheses were obtained by dividing the number of ring gear teeth by the number of pinion gear teeth, i.e., $41 \div 8 = 5.125$. This figure gives the exact number of rotations the propeller shaft must make for each turn of the rear wheels. The 5.125 number is generally written as 5.125/1, which signifies that 5.125 turns are required of the pinion for each 1 turn of the ring gear. The second column gives the mph of the chassis (in 4th gear) for each 1000 rpm of the engine. This was calculated on the basis of tires which make 813 revolutions per mile (Michelin 165 x 15, ZX). If your MG is fitted with tires of a significantly different size you must then recalculate all the figures given in this column. You may calculate the mph for each 1000 rpm as follows:

$$\frac{(1000 \text{ rev/min})(60 \text{ min/hr})}{(5.125)(813 \text{ rev/mile})} = 14.400 \text{ miles/hr-1000 rpm}$$

enter your rear axle ratio here \curvearrowright \curvearrowleft enter your tire constant here

Figures 1 and 2 provide, in graphical form, the remainder of the basic information needed for the rear axle calculations. Figure 1 gives engine horsepower and torque curves, while Figure 2 gives the rate at which energy is consumed by the chassis and running gear at different road speeds. This is the sum of the frictional losses in the drive train, the tire hysteresis losses, and the aerodynamic losses arising from the motion of the body through the air. The TF and MGA energy consumption curves are the only ones I have been able to locate in MG literature.

Brake Horsepower (B.H.P.) and Torque
as a Function of Engine Speed (R.P.M.)

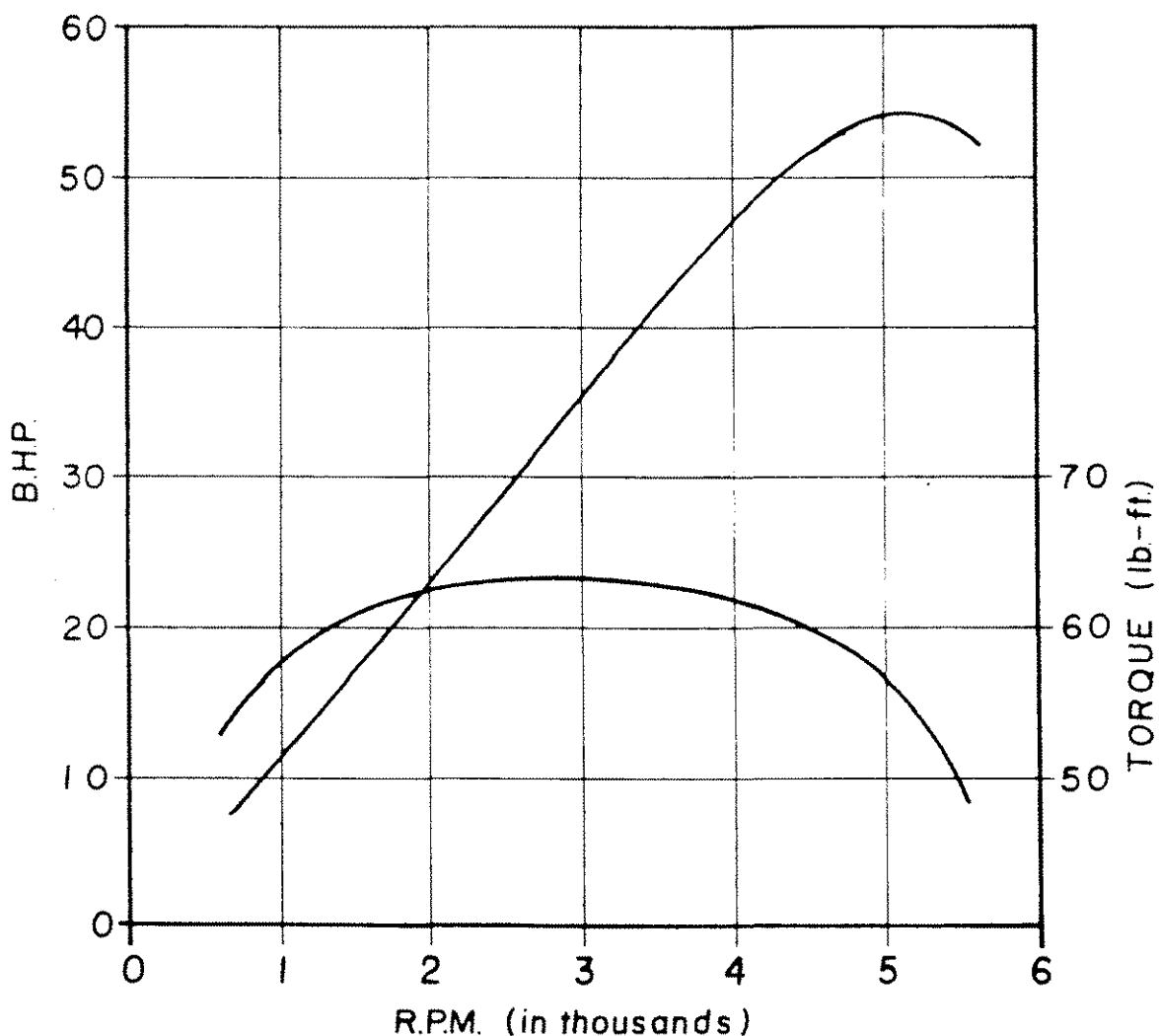


Figure 1. These curves are for a stock 1250 cm³ XPAG engine. They are taken from pages 146 and 147 of *Maintaining the Breed*, John W. Thornley, Motor Racing Publications Ltd., London, England.

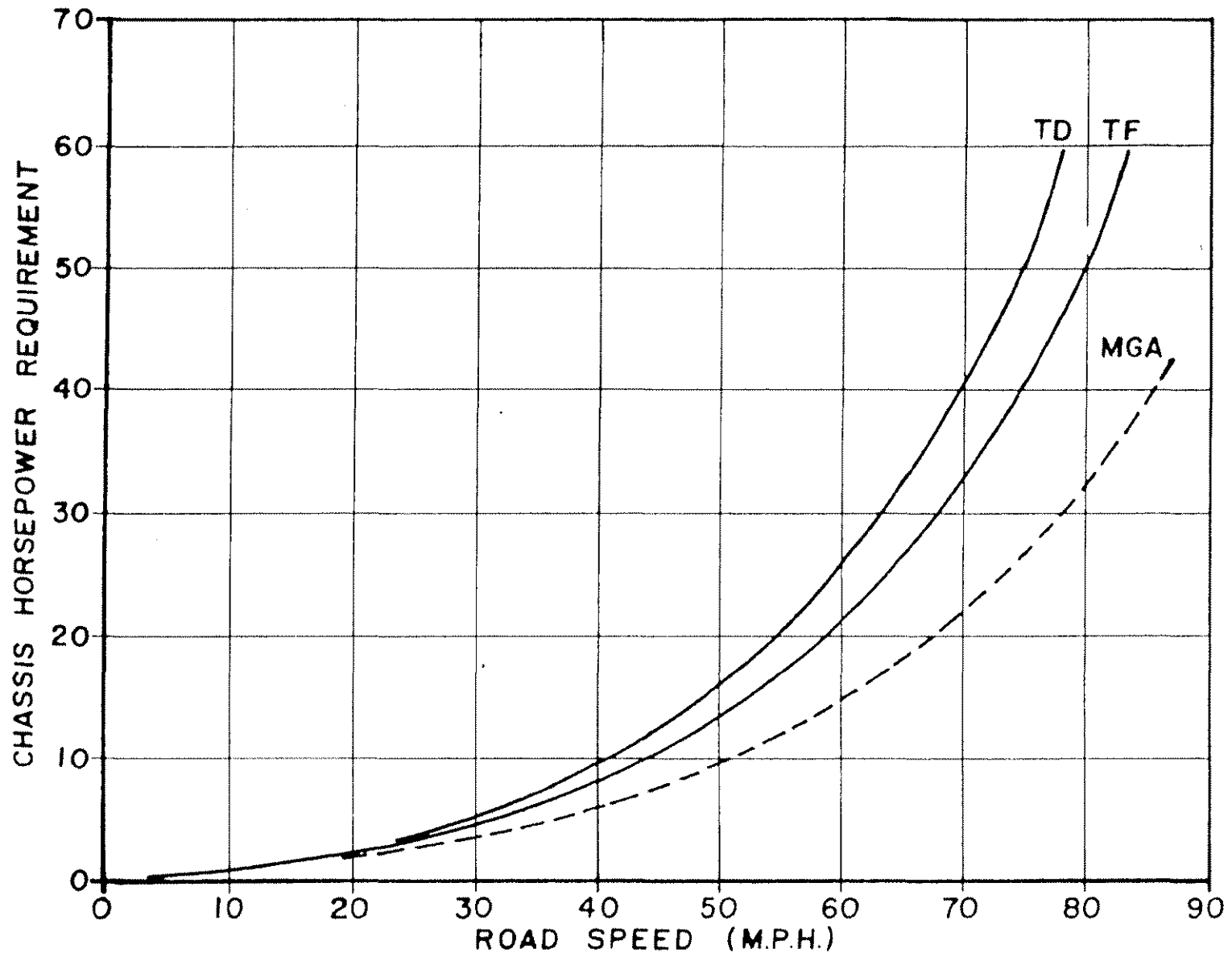


Figure 2. These curves show the horsepower consumed (aerodynamic, drive train and tire losses) by a TD, TF and MGA as a function of road speed. The TD curve is speculative. The TF and MGA curves are taken from page 155 of *Maintaining the Breed*, John W. Thornley, Motor Racing Publications Ltd., London, England.

As a result, the curve drawn for the TD in Figure 2 is my *estimate* of where the TD curve probably lies. With the TD curve drawn in this position, the calculated results agree reasonably well with road tests published back in the 1950s. This completes all the basic information needed -- the resulting performance of a stock MG TD can now be calculated (or reasonably estimated) for any possible rear end ratio. For a TF or an engine in a different degree of tune the horsepower and torque curves must be replotted.



Let us make the most simple calculation first: using the mph per 1000 rpm information from Table 1, the 4th gear road speed of this MG is to be calculated for all possible rear axle gear ratios. The family of straight lines in Figure 3 displays these road speeds as a function of engine speed. The heavy vertical line at 4000 rpm marks the rev. limit for cross country operation.

Figure 4 is more involved and presents much more information. The heavy curve (taken from Figure 2) shows the horsepower required to drive a TD at any road speed. The lighter curves show the maximum horsepower available from the engine at all road speeds in 4th gear. Each of these lighter curves is for a different rear axle ratio. These lighter curves were constructed from data given in Figures 1 and 3. For each road speed and rear axle ratio, a corresponding 4th gear engine speed (rpm) was determined from Figure 3. The maximum horsepower (B.H.P.) available at this engine speed was then determined from Figure 1 using the curve for the standard engine. If your engine is modified, as most are, you must use the B.H.P. curve for your particular engine and then replot Figures 1 and 4 so that they describe your MG.

The maximum possible road speed for a stock TD with each rear axle ratio is now given by the points of intersection between the chassis horsepower requirement curve and each of the curves showing the maximum engine horsepower available. The TD can never operate beyond this intersection point since the chassis then requires more horsepower than the engine can develop (at that road speed with that particular rear end ratio). Observe that the maximum road speed is not significantly reduced by these lower rear axle ratios since the intersection points fall on the steeply rising portion of the chassis horsepower requirement curve. All possible modes of 4th gear operation MUST lie within the area bounded by the chassis horsepower requirement curve and the engine horsepower availability curve. The vertical distances within this bounded area show the horsepower remaining for hill climbing

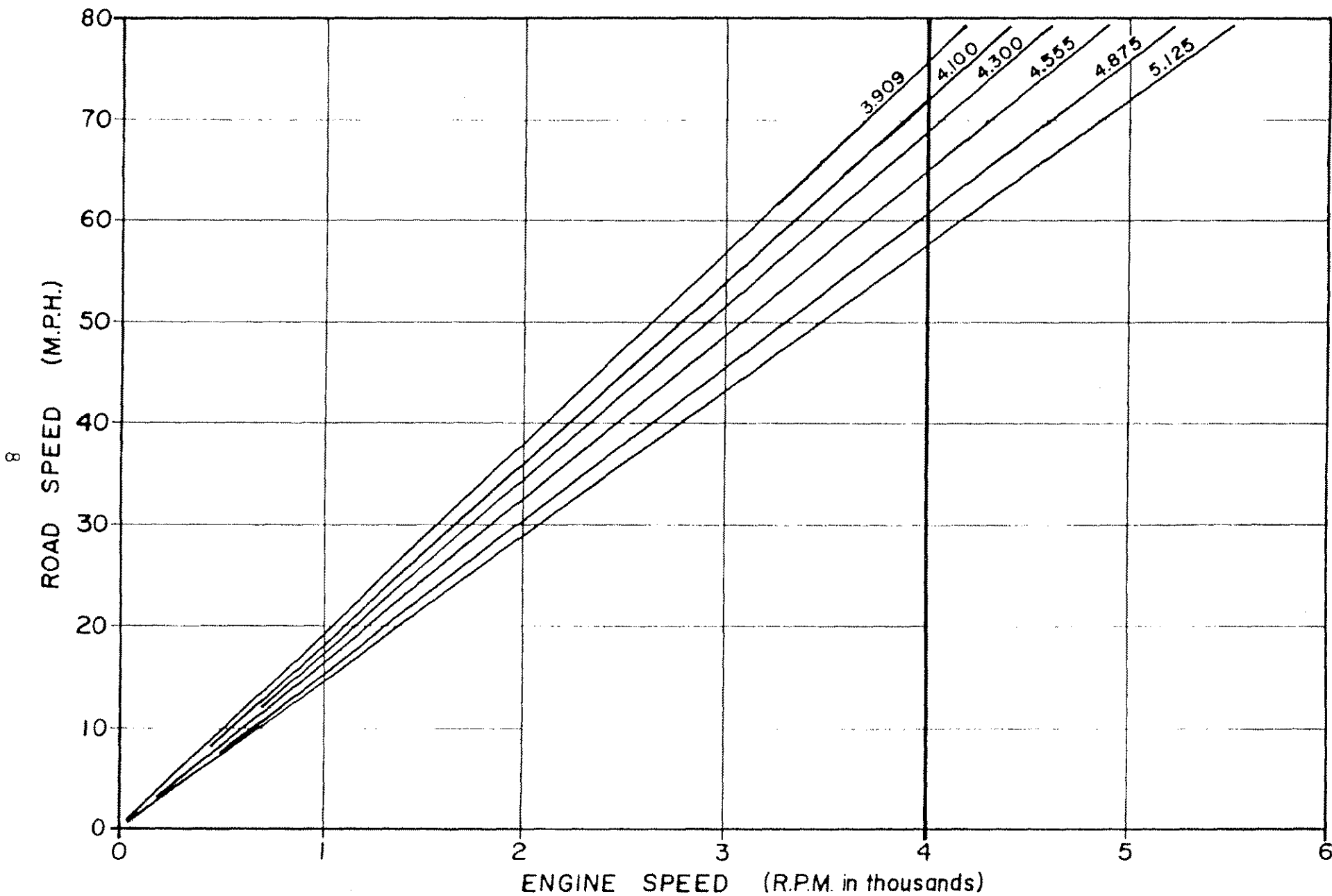


Figure 3. These curves give the road speed of a TD or TF as a function of engine speed. Each curve is for a different rear axle ratio. The car is assumed to be fitted with tires which make 813 revolutions/mile.

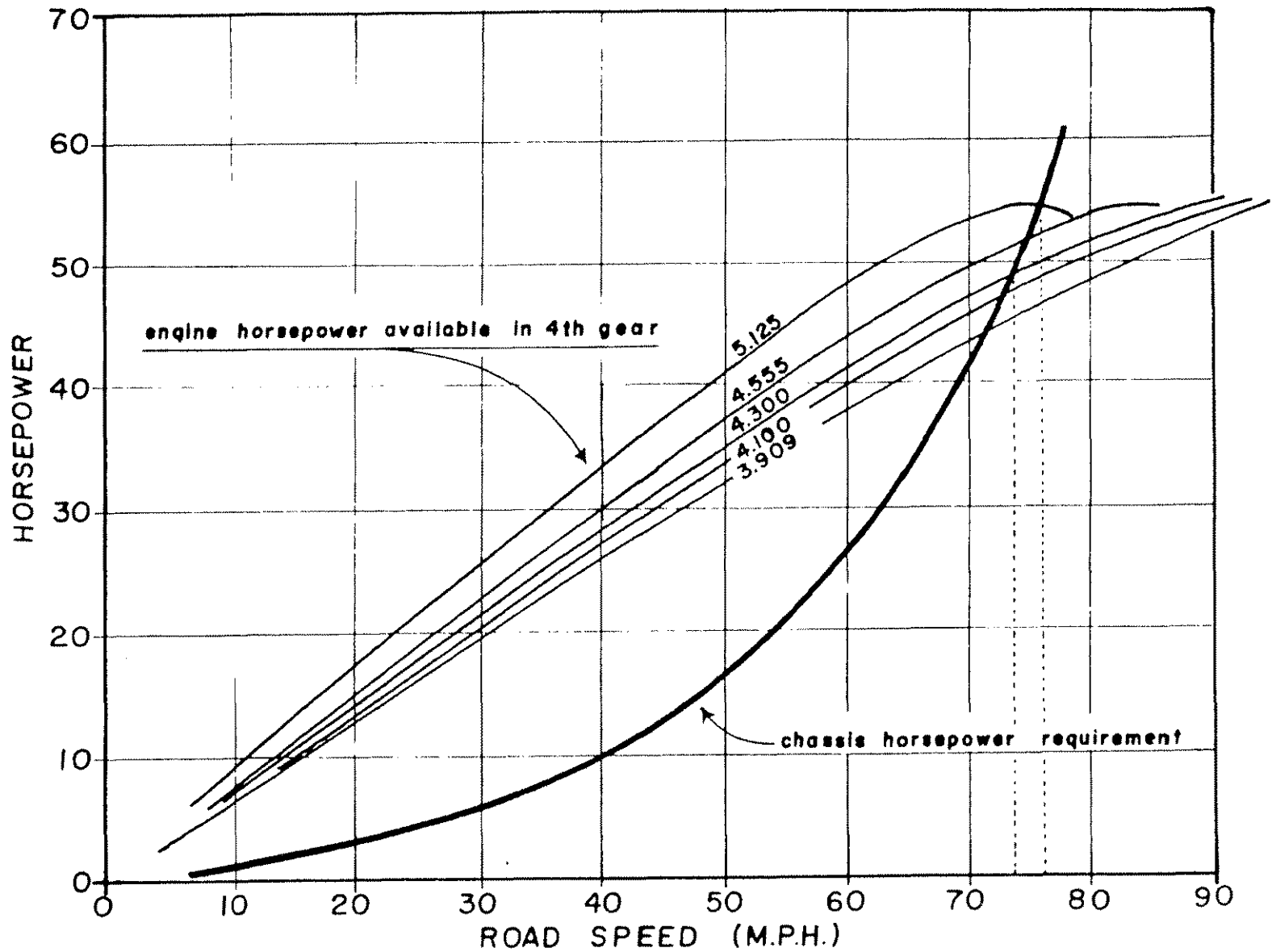


Figure 4. The five lighter curves show the maximum horsepower available from the engine as a function of road speed for five rear axle ratios. The heavy curve shows the horsepower consumed by a TD chassis. The dotted lines show that large changes in rear axle ratios make small changes in top speed.

and acceleration. This surplus horsepower, at a particular road speed, is the difference between the horsepower that the chassis and drive train consume and the maximum horsepower the engine has available. Again, this surplus horsepower information is obtained by noting the vertical distances between the two curves. The upper curve is the horsepower available with a particular rear axle ratio and the lower curve is the chassis horsepower requirement. The drop in horsepower when converting to a lower final drive ratio will be noticed most when climbing a hill in 4th gear. The hill climbing ability of a car varies directly with the surplus horsepower. (A 20% decrease in surplus horsepower at a particular rpm will decrease the slope of the hill that can be climbed at that rpm by 20%).

The 4th gear acceleration is, however, less reduced by a drop in horsepower than the 4th gear hill climbing ability. The ability to accelerate varies as the *square root* of the horsepower available (reducing the surplus horsepower available to 1/4 its original value reduces the acceleration by only 1/2, or, said in the other direction, it takes four times the horsepower to double the acceleration).

Figure 5 summarizes the hill climbing and acceleration information for the 4.300/1 interchange. I have compared the 4.300/1 rear end to the 5.125/1 rear end and plotted the performance decrease as a percent decrease from the performance of the standard 5.125/1. These curves were generated from the surplus horsepower information given in Figure 4.

Though this concludes the curves that will be plotted, there are, however, several additional curves that a mathematically inclined enthusiast may wish to plot. Plot the surplus horsepower for the 4.300/1 and 5.125/1 rear ends for each of the gear box ratios as a function of road speed. You will then note that the 4.300/1 raises third gear up to a very useful operating range. You can now use third gear on the highway for the first time! If you plot the torque instead of the horsepower in Figure 4 then the relative acceleration can be measured directly from the distance between these two curves. You eliminate the square root botheration by plotting the torque directly but then you have first to reduce all the horsepower curves to torque curves. I have always thought it was six of one and half a dozen of the other. Another interesting graph to plot is the rear axle torque as a function of road speed for each gear ratio in the gear box. The three intersection points of these four curves provide the unique shift points for maximum acceleration. All of these intersection points will occur at very high revs. They are of some use for racing but that is about all -- they all occur close to 6000 rpm. This is just a little too strenuous for normal motoring. After any three alter-

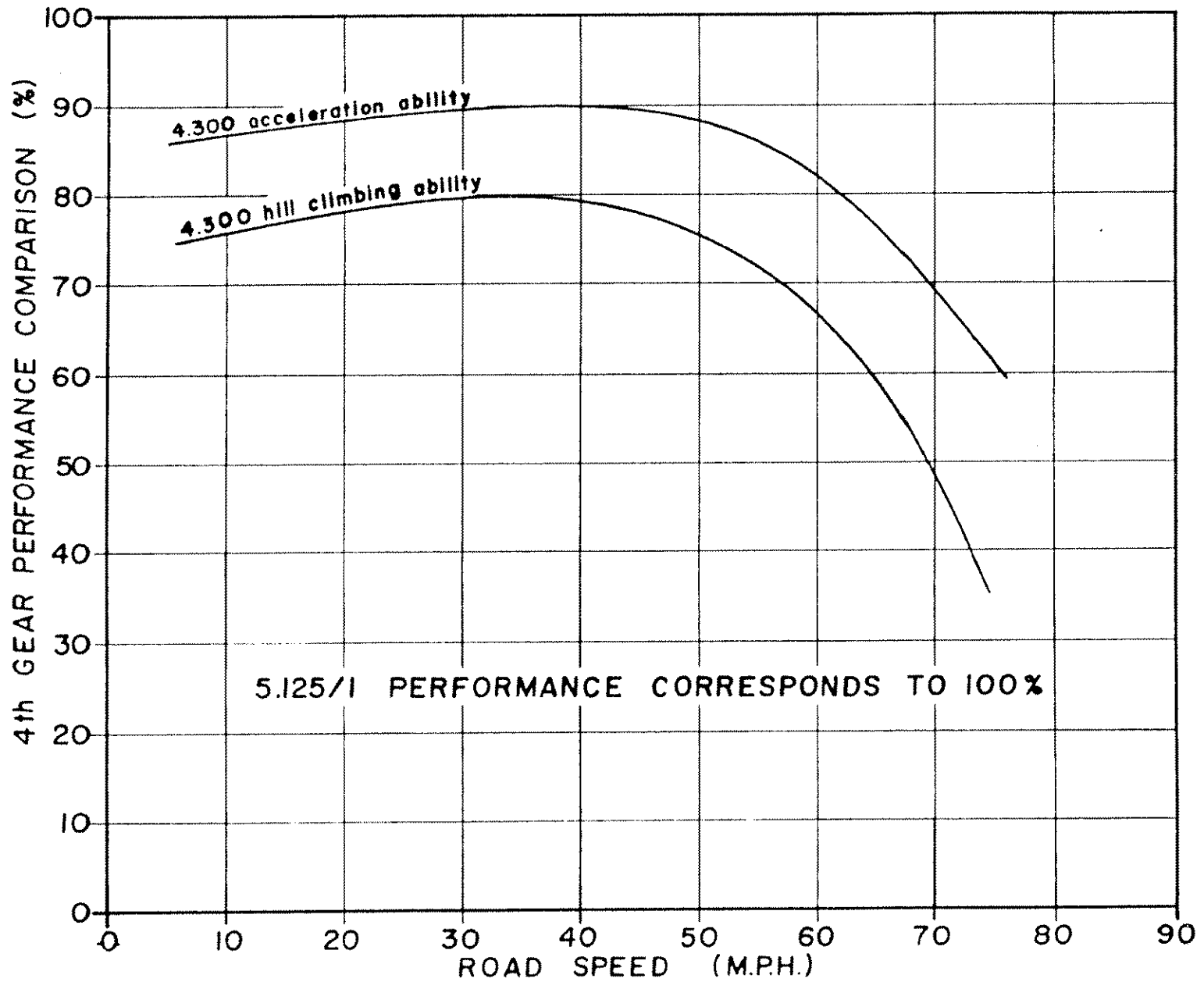


Figure 5. These two curves show the percent decrease in 4th gear performance that results from a change of a 5.125 to a 4.300. Were these curves plotted for 3rd gear instead of 4th, the advantage would be in favor of the 4.300.

nate shift points have been chosen, a numerical integration can be performed and the actual acceleration times for any normal motoring type of speed change (0-60, 30-50 mph, etc.) may be calculated.

As a final thought, if you have any inclination to ever change your rear axle ratio, buy the appropriate MGA parts now while they are still available. An MGA rear end currently (1974) costs about \$25 in western junk yards. New 4.300/1 gear sets from BLMC cost about \$100. When stocks of new gear sets are exhausted, BLMC will not make more since the factory has discontinued parts manufacture for both the MGA and the early MGB. Hence, if you want new parts, order them soon.

III. FITTING THE RING AND PINION

The fitting of an MGA or early MGB ring and pinion into a TD/TF rear axle housing is not a Saturday afternoon job for the casual mechanic. To accomplish the interchange in a straightforward manner requires an assortment of machinist's tools in addition to special tooling whose fabrication cost several hundred dollars. If, however, you have much time and insufficient money you can substitute your labor for some of the tooling. This approach will be tedious and circuitous but it does work. There are four, and only four, measurements and subsequent adjustments necessary to fit a ring and pinion into any rear end. Where possible, several approaches to each of these four steps will be given. Any measuring method that results in locating gears and bearings properly is satisfactory.

In the order these four steps are to be accomplished, these steps are:

- A. Location of the Pinion.
- B. Adjusting the Pinion Bearing Preload.
- C. Location of the Ring Gear.
- D. Adjustment of Differential Carrier Bearing Preload.

A. LOCATION OF THE PINION

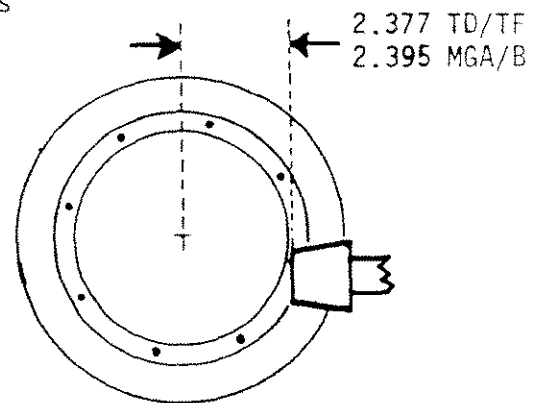
There are four methods^{*} for locating the pinion in the main casting. The first three involve mechanical measurements which must be made very carefully (± 0.0005 inch) since the locating tolerance on the pinion is ± 0.001 inch. The first and easiest method involves the use of the factory tooling (18G.281). This tool (originally \$22.61) is no longer available from BLMC and unfortunately few MG owners bought it when it was available. The tooling was, however, not made by BLMC but

* For a fifth method see ADDENDUM.

by an independent English tool company. The manufacturer was V.L. Churchill & Co., Ltd., P.O. Box No. 3, London Road, Daventry, Northants. There may yet be a chance that the tool could be obtained on special order from Churchill -- someone should explore this possibility and perhaps arrange for group purchase of a small production run. I own this particular tool and would be pleased to make the appropriate pinion location measurement for anyone who might arrive at my home with their main rear axle casting (please call first). The description of the use of factory tooling for fitting the TD/TF pinion will be omitted since its use is covered in the T series workshop manual.

The fitting of the MGA or early MGB pinion will require a modification in the procedure to be used with the factory tooling. The head of the dummy pinion supplied with the factory fixture is exactly the same thickness (1.375 inch) as the TD/TF pinion it represents. The MGA and early MGB pinion heads are thicker. They are 1.385 to 1.386 inch thick with most of those so far measured being 1.386 inch. When the factory fixture is used to locate the MGA pinion, the thickness of the pinion spacer must be *reduced* by 0.011 inch ($1.386 - 1.375 = 0.011$ inch) below that value dictated by the factory fixture. This will then locate the head of the MGA pinion in the same plane as the original T series pinion. [When you read further, you will be instructed to move the MGA pinion an additional 0.018 inch. Hence, when using the factory T tooling on an MGA or early MGB pinion, reduce the pinion spacer a total of 0.029 inch ($0.011 + 0.018 = 0.029$ inch) less than the thickness stipulated by measurement with the factory fixture.]

What does one do when the factory tooling is not available? The second approach is to make up tooling functionally identical to the factory tooling. As closely as I can measure, the factory tooling locates the head of the TD/TF pinion 2.377 ± 0.0003 inch perpendicularly away from the axis of the ring gear. Hence, devise some method of locating the hole that accepts the differential carrier bearing and then from its axis locate a plane 2.377 inches distant. This is the plane that the head of a TD/TF pinion must just touch. If you are fitting an MGA or an early MGB pinion, the perpendicular distance from the head of the pinion to the axis of the ring gear is to be 2.395 inch. This second method requires some first class tool design and careful machine work.



This 2.395 inch dimension was determined as follows: A very very early morning transatlantic telephone conversation with Mr. Donald Welch of V. L. Churchill & Co., Ltd. disclosed that the height of the step used to zero the dial indicator in the MGA pinion setting tool is 20.828 mm (0.8200 inch). Mr. Welch pulled the engineering drawing and read this dimension directly from the print. If you will examine your MGA shop manual you will discern that the 0.8200 inch dimension is the perpendicular distance from the plane containing the head of the pinion to the bore in which the MGA carrier bearings are clamped. The outside diameter (O. D.) of the MGA carrier bearing is 80.000 mm or 3.1496 inch. The radius of this carrier bearing is accordingly $3.1496/2 = 1.5748$ inch. With the bearing radius now known, the distance from the head of the MGA pinion to the axis of rotation of the MGA ring gear is arrived at by adding these two distances, $0.8200 + 1.5748 = 2.3948$ inch. You will probably need to examine the drawings and pictures in the MGA workshop manual to clearly visualize this geometry from this brief explanation.

The consequence of this information is that you must set the MGA or early MGB pinion 0.018 inch ($2.395 - 2.377 = 0.018$ inch) further away from the ring gear axis than the original T series pinion that you removed from your housing.

A third and certainly simpler method is to use an existing T series rear end as part of the tooling. Remove the differential carrier but leave the original pinion undisturbed. Devise a right angle plate that locates in the main casting and then measure the perpendicular distance from the angle plate to the head of the existing pinion. It doesn't matter what this distance is as long as you know what it is. You will probably need to drill and ream two extra holes in the main casting to locate the angle plate with dowel pins so that its location is reproducible. The distance to the pinion can be measured with either an inside micrometer or a dial indicator permanently attached to the angle plate. You should measure this distance with a precision of ± 0.0005 inch. Remove and replace the angle plate once or twice so that you demonstrate that the same number (± 0.0005 inch or closer) is reproduced each time. With this distance written down, replace the original pinion with the new pinion and its new associated inner bearing. Fit any pinion spacer, and omit the pinion oil seal and preload spacer between the two tapered roller bearings. Place the propeller shaft flange on the pinion shaft and draw the pinion nut down to a tightness such that it requires 11 - 13 pound-inches of torque to rotate the pinion. This is to ensure that the pinion measurement is made with the inner bearing in the deflected position it will later assume in actual operation.

If you do not own an appropriate torque wrench you may improvise an equivalent with a piece of heavy string and an inexpensive (\$3) spring balance. Secure one end of the string to one of the bolt holes in the drive flange, wrap several turns of the string around the flange, and then pull. Be sure to pull the other end of the string with the spring balance at a right angle to the pinion shaft. The torque is the reading on the spring balance (in pounds) multiplied by the radius (in inches) of that portion of the flange around which the string is wrapped. Disregard the initially high torque required to start the pinion rotating and consider only the lower reading attained during a slow steady pull on the string. Set the torque to 11 - 13 pound-inches by tightening the flange nut, and then measure the distance from the angle plate to the head of the new pinion. Aha, this is a different distance! Now one simply calculates that thickness of pinion spacer required to bring the new TD/TF pinion to the same location as the original TD/TF pinion. In the case of fitting the MGA pinion, calculate the thickness of spacer required to locate the MGA pinion 0.018 inch further away from the axis of the ring gear than the original TD/TF pinion. When fitting one of the MGA series ratios I would be inclined to purchase the thick MGA spacer, ATB7064, (0.126 inch to 0.112 inch by 0.002 inch increments, ATB7064 through ATB7071) and then take it to a grinding shop and have it ground to the required thickness. The TD/TF pinion spacing washer will not fit the MGA pinion since these TD/TF washers are made for the 1.125 inch diameter TD/TF shaft and are not large enough to slip on over the 1.250 inch diameter MGA shaft (If necessary the TD/TF spacers could be internally ground to fit). In the event the pinion washer is just a few thousandths too thick, it may be hand ground to the required thickness on a surface plate with a sheet of 600 wet or dry carborundum paper, and some very light oil or kerosene. If necessary, a thick piece of plate glass can substitute for the surface plate. Place the abrasive paper, abrasive side up, on the flat surface and work over the moistened 600 wet or dry in a figure eight motion. After every three figure eights remove your fingers and rotate the spacer a little so that you don't press at the same place all the time. You must grind the spacer as uniformly as possible. Be sure to measure the spacer frequently since the actual grinding time is only going to take a minute. If the spacer is too thin, add a homemade shim to bring it up to the required thickness.

The fourth method of pinion location (guess method) requires no fixtures. It consists of assembling the rear end with the teeth of the pinion gear lightly and uniformly smeared with a heavy oil mixed with some brightly colored pigment. A mixture of red lead and 90 wt. oil works although I would prefer a tube of Prussian

Blue, obtainable at any artist's supply store. A tube of this oil paint lasts for years and is readily storable. In any event, both of these dye systems will work satisfactorily. With the pinion teeth very lightly coated, rotate the pinion in an assembled rear end and then disassemble the rear end to examine the contact marks made on the two gears. The tooth patterns are interpreted and adjustments made in the pinion location. Drawings of the correct and incorrect patterns are found in most general automotive texts. As improbable as it seems, the guess system does work as there is a functional TD rear end in our T series chapter that was set up in this manner. The disadvantage to this approach is only that the rear end must be assembled and disassembled each time a spacer change is made and that the pattern is generated by *both* pinion and ring gear locations. As a consequence, the dye pattern takes a little thought to interpret since you must distinguish between pinion and ring gear displacements on the pattern. Every time the rear axle is reassembled with another group of spacers to generate a new dye pattern make sure that the pinion bearing preload is correct so that the pinion is in its operating location.

If you have the time available, there is another and more elegant method of generating these tooth contact patterns. The pinion and ring gear are cleaned thoroughly with solvent and then their teeth are "painted" with an aqueous solution of copper sulfate. The copper ions displace the iron on any iron surface with which the solution of copper sulfate comes in contact. In this manner, a monomolecular layer of copper is deposited on the surface of the gear teeth. The rear axle is reassembled, fitted to the car, and filled with oil. Driving the car around for a few miles immediately wears the copper off and generates the tooth contact pattern characteristic of loaded conditions. Unfortunately, one also has to remove the rear axle, disassemble, and then clean up the gears before the pattern can be seen. This is a monumental amount of work even though it is the standard engineering method for determining loaded tooth patterns. Any T owner who survives this procedure should certainly be awarded the "order of the octagon".

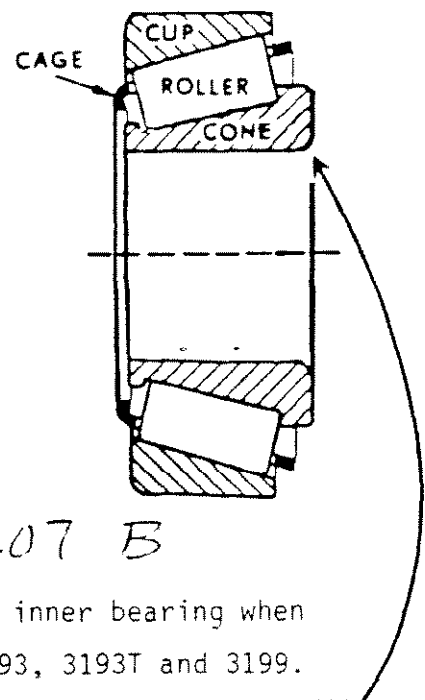
As a secondary complication to the pinion location problem, note that the measurements may require slight alteration in accordance with the production variation numbers found on the head of the pinion. Read and reread the rear axle sections in both the T series and MGA workshop manuals until you understand how the factory wants these production variations accommodated. A zero or no number

denotes location of the pinion in accordance with the factory jig, while a + number instructs you to increase the distance (thinner spacer) between the head of the pinion and the axis of the ring gear by the number of thousandths noted on the head of the pinion gear. A - number denotes a decrease in the nominal distance (thicker spacer). Hence, if you are going to make an angle plate and measure from an original undisturbed pinion, be sure to observe its pinion head markings and make the necessary mathematical corrections in determining the plane where the head of the new pinion is located. The new pinion will also have a production variation and hence, if not zero, you are required to locate the new pinion the noted distance from the nominal location.

If the tapered roller bearings you remove from your T series and MGA rear ends show any signs of failure, then install all new bearings. To fit the MGA ring and pinion, you will need to make up a mixed set of pinion bearings in which the cone of the inner bearing is the corresponding MGA cone and the pre-load spacer between the bearings is the MGA spacer. The rest of the bearing assembly will utilize stock TD/TF components.

The Timken numbers for the bearings are:

	Fitting of T Series Gears	Fitting of MGA/B Gears
Outer Bearing	cone 15100S	15100S
	cup 15250X	15250X
Inner Bearing	cone 3198	3188
	cup 3120	3120



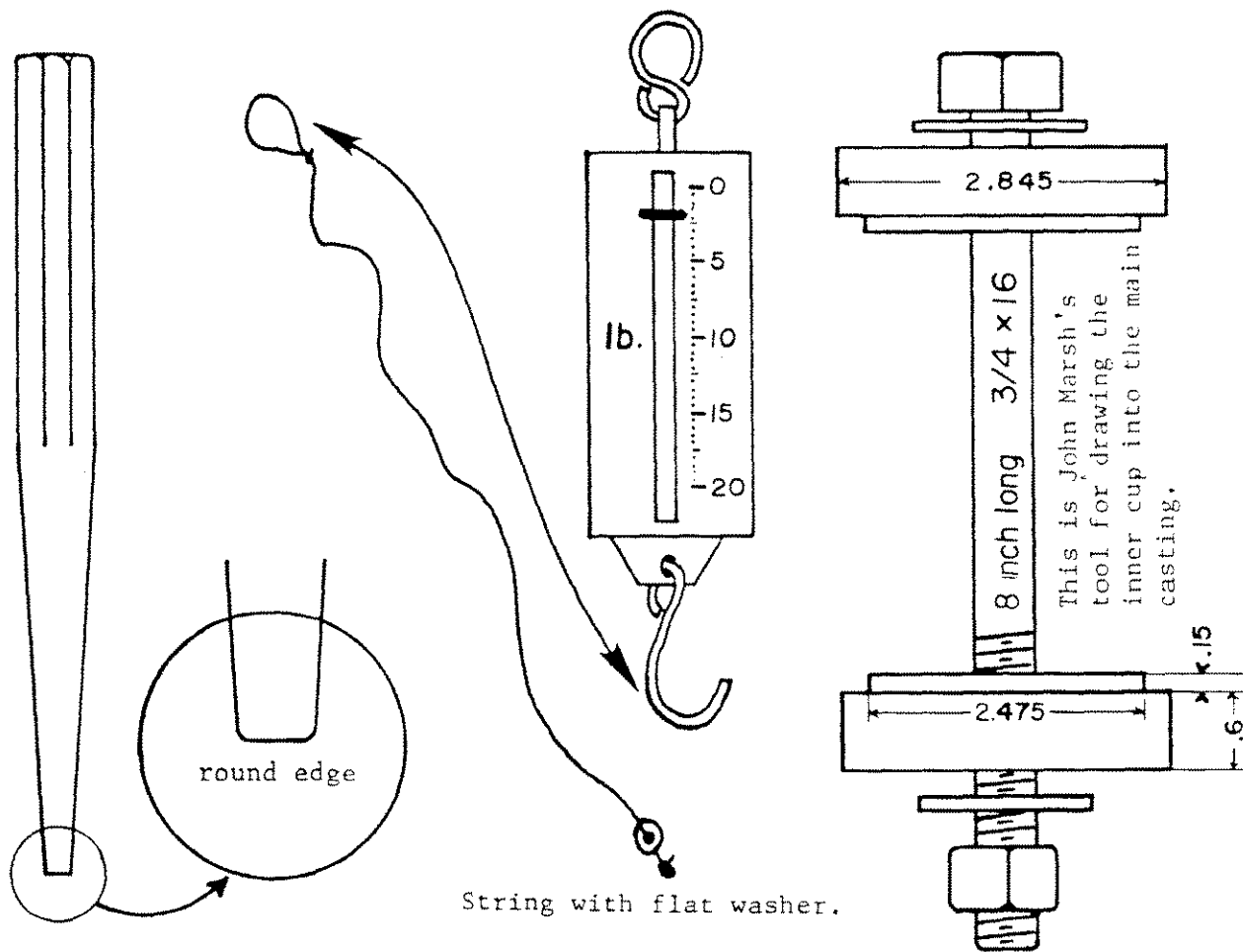
They are "off the shelf" bearings at large bearing supply houses at prices generally below those of BLMC.

*new departure 20207
carrier bearing SKF 7207 B*

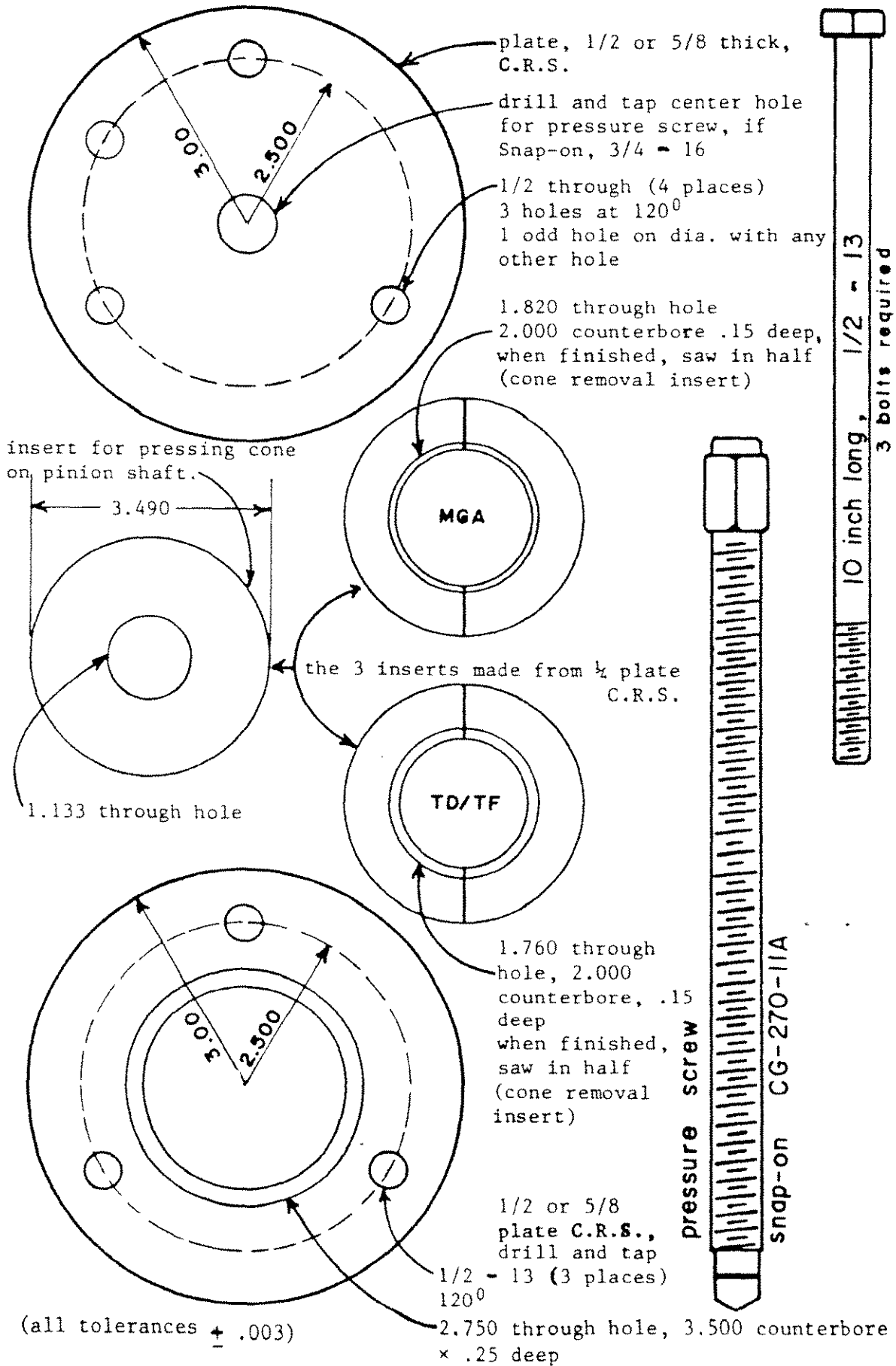
Timken lists five cones that could be used to make up the inner bearing when fitting an MGA pinion. These cones are: 3188, 3188S, 3193, 3193T and 3199. The difference is only the radius on the back face of the cone. The five radii are: .03, .06, .14, .06 and .09 inch respectively. The 3188 is the one generally stocked according to my local bearing house.

Another problem to be solved in the ring and pinion exchange is the removal of the bearings, the cones from the pinion shaft and the cups from the main casting. The cones and cups are all interference fits of 0.001 inch with their respective mountings and hence require considerable force to remove. (The bores are always 0.001 inch too small for what is to be pressed into them and hence they "interfere"). The two cups that are pressed into the main casting can be removed with a pair of 6 and 10 inch drifts. When doing so be sure, however, to file and polish a smooth rounded edge on the business end of the drift in the place of its original sharp (90°) edge. The sharp edge will dig into the casting badly when driving out the cups. Observe that the factory has thoughtfully provided two internal reliefs in the main casting for positioning the drifts to drive out the cups.

Removal of the cone from the pinion shaft for a spacer or bearing inter-change is a very difficult task without a puller. You might be successful with an offset drift, but I personally wouldn't attempt it. Drawings for the puller that was made up for this task are included with the tool drawings.



Pinion Bearing (cone) removal and replacement tool



B. ADJUSTING THE PINION BEARING PRELOAD

The pinion bearings are preloaded (squeezed together) to increase their rigidity and hence reduce the deflection of the pinion under load. This is a desirable end since the pinion will then maintain its position more closely under all operating conditions. The preloading amounts to several thousand pounds and is controlled by the spacer between the two bearings. It is measured indirectly by measurement of the torque required to rotate the pinion. We will obtain the correct preloading by shortening the preload spacer (in small increments) until the torque reading increases to its desired value.

To withdraw the pinion, the pinion nut must first be removed. When you try to loosen the pinion nut it is obvious that a little problem exists -- the pinion turns as the nut is turned. The rotating flange can be grabbed with a large pair of vise grip pliers if you can endure the resulting mutilation of the flange. A more satisfactory solution is to purchase the factory flange wrench (18G.34A). Alternately, make a flange wrench by drilling two appropriate holes in one end of a length of heavy steel angle so that the angle can be bolted to the flange. Upon removal of the nut, the flange can sometimes be tapped off (gently) but more properly it should be withdrawn with a puller. A puller of suitable size is available at all large automotive supply stores -- it is essential for other jobs about the car, so go out and buy one if you do not already own one. With the pinion flange removed the pinion shaft still doesn't want to come out. This is due to the 0.001 inch interference fit between the pinion shaft and the cone of the outer bearing. The pinion shaft can be driven out with a soft metal (copper, lead or brass) mallet, even though every bearing manufacturer in the world advises against this bludgeon method. Unless you are careful in locating the blows of your mallet you might damage the threads on the pinion shaft -- take care. A more satisfactory removal procedure would be to construct your own press in the form of a large wooden rectangle to drop around the main casting. A screw plate will need to be made up and attached to one end of the wooden rectangle. This plate then serves to locate a pressure screw that is used to push out the pinion shaft. An appropriate 3/4 inch diameter screw can be purchased separately as the replacement screw sold for a gear puller; for example, Snap-on pressure screw No. CG-270-11A.

The 2.4 inch tubular spacer between the two tapered roller bearings controls the bearing preload by determining the degree of crush that occurs when the pinion nut is tightened. The fitting of the MGA gear set will require the use of the MGA

pinion flange (ATB 7059) and preload spacer (1B 7240). The MGA shaft has eleven splines instead of the six of the TD/TF. It is also a little larger in diameter at the pinion end than the TD/TF shaft. The MGA preload spacer is slightly longer than is required for the T series application and thus must be shortened. Assemble the MGA pinion in the T housing with the MGA preload spacer and measure any end play with a dial indicator. You will probably observe an end play (movement) of between 0.012 and 0.015 inch. Add an additional 0.004 inch to your measurements to allow for the seating of the rollers and then have this sum (end play plus 0.004 inch) ground off the length of the preload spacer. Go to a grinding shop (not a small general machine shop) since the ends of the spacer MUST be parallel (0.0001 inch/inch). Reassemble the pinion with the shortened spacer, retighten the pinion nut to 140 lb-ft, and now measure the torque required to rotate the pinion.

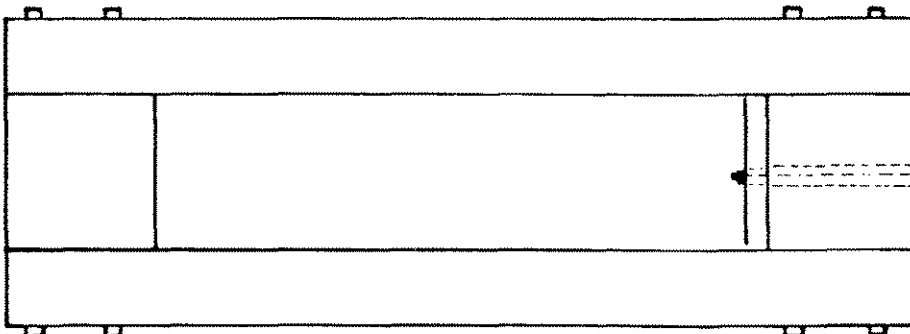
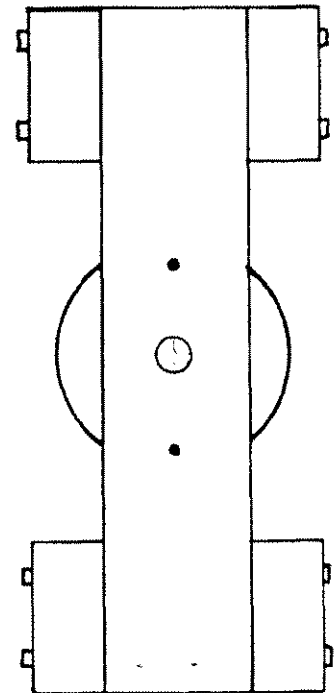
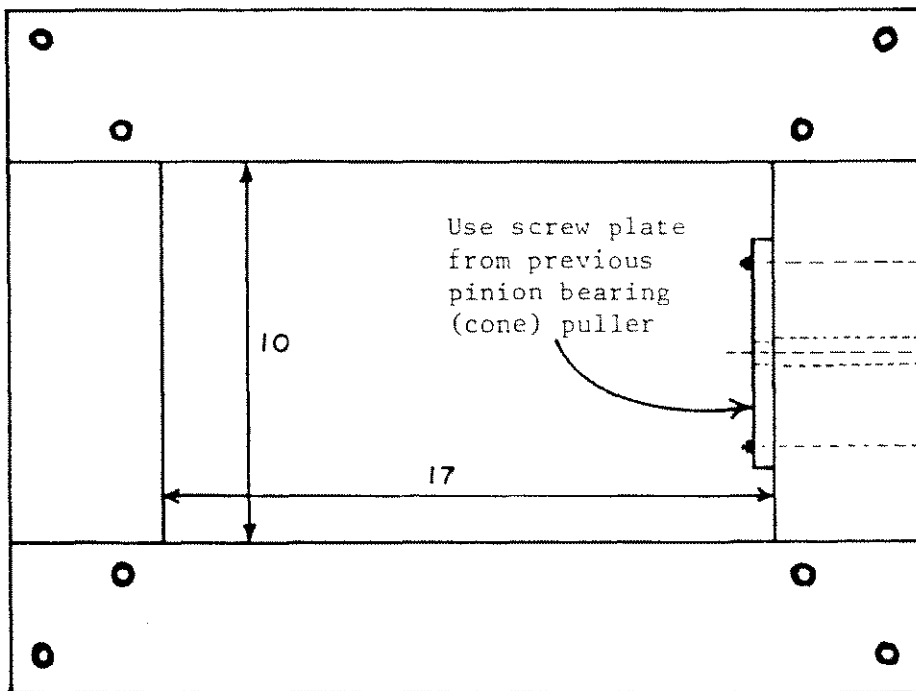
The torque is measured with either a torque wrench or the spring balance and piece of string. A torque reading of 1 to 3 lb-in will be obtained for this first measurement. Since this value is too low, the pinion structure must again be disassembled and this time an additional 0.002 inch (no more) ground from the length of the preload spacer. Reassemble again and measure the torque required for steady rotation of the pinion. The goal is a torque value of 11 to 13 lb-in. Continue shortening the spacer until you attain the required reading. Do not shorten the spacer by a large amount after you obtain your first measurement as a change of only 0.001 inch in length makes a large difference in torque (about 5 lb-in).

After installing the pinion each time, rotate it a few times by hand before making a torque measurement to make sure that the rollers have seated. Also be sure the pinion nut is firmly tightened (140 lb-ft) so that the two cones are drawn firmly against the preload spacer. After the pinion nut has been tightened several times it should be discarded since its threads will be badly distorted. Buy half a dozen nuts so you can use a fresh one for final assembly. If the recommended value for the rotating torque is exceeded, then you must shim your excessively shortened spacer, or start over with a new spacer. Ready made shims for this purpose are still available from the MGA (ATB 102 is 0.004, ATB 103 is 0.006, ATB 104 is 0.008, ATB 105 is 0.010, ATB 106 is 0.012, ATB 107 is 0.020 and ATB 108 is 0.030 inch). If necessary, cut out the required shim(s) from steel shim stock. Small assortments of steel shim stock are available from large tool supply houses (about \$6.00). A particularly nice assortment (0.001 to 0.015 inch) that is probably sold all over the country is No. AF-12 made by Precision Steel Warehouse, Inc., Downers Grove, Illinois. You will need such an assortment later since you must

also cut out shims for locating the ring gear.

Some rear ends that have been set up with mixed used bearings give erratic torque readings. If you run into this problem, you may solve it by lubricating the bearing with a few drops of hypoid oil during your torque measurement. You will then obtain very consistent torque readings that have dropped about 25 per cent. Allow for this oil film by reducing the desired torque reading from 11 to 13 lb-in down to about 8 to 10 lb-in. Remember that you are really interested in attaining a suitable compressive load on the bearings and are only using the torque as a means of indirectly indicating when the preloading is correct.

When you are ready for final assembly, be sure to install the pinion oil seal. Install it in such a manner that any oil pressure from inside the housing tends to



This press is framed from 2 x 4's and 4 x 4's that are bolted together.

pinion washer
.125 tk
1.575 O.D.
~ 3/4 I.D.

press the lip of the seal more snugly against the shaft. Be sure to smear the lip of the oil seal with hypoid oil so it is lubricated when the car starts. If the lip starts out dry it may heat up and melt its sharp edge, thereby ruining its sealing ability. I would leave the sheet metal guard (1G.7439) on the pinion flange as was provided with the MGA. This guard helps keep grit away from the oil seal and hence can prolong its life -- this is a change that is well worth retrofitting to a TD/TF. Some T series castings will accept the MGA guard directly, others will require the flange to be machined a little to move the guard away from the main casting.

*-carrier bearing
SKF 7207 B*

C. LOCATION OF THE RING GEAR

The location of the ring gear is determined by the required backlash. This measurement will be made by holding the pinion still and measuring the backlash present when the ring gear is rocked back and forth.

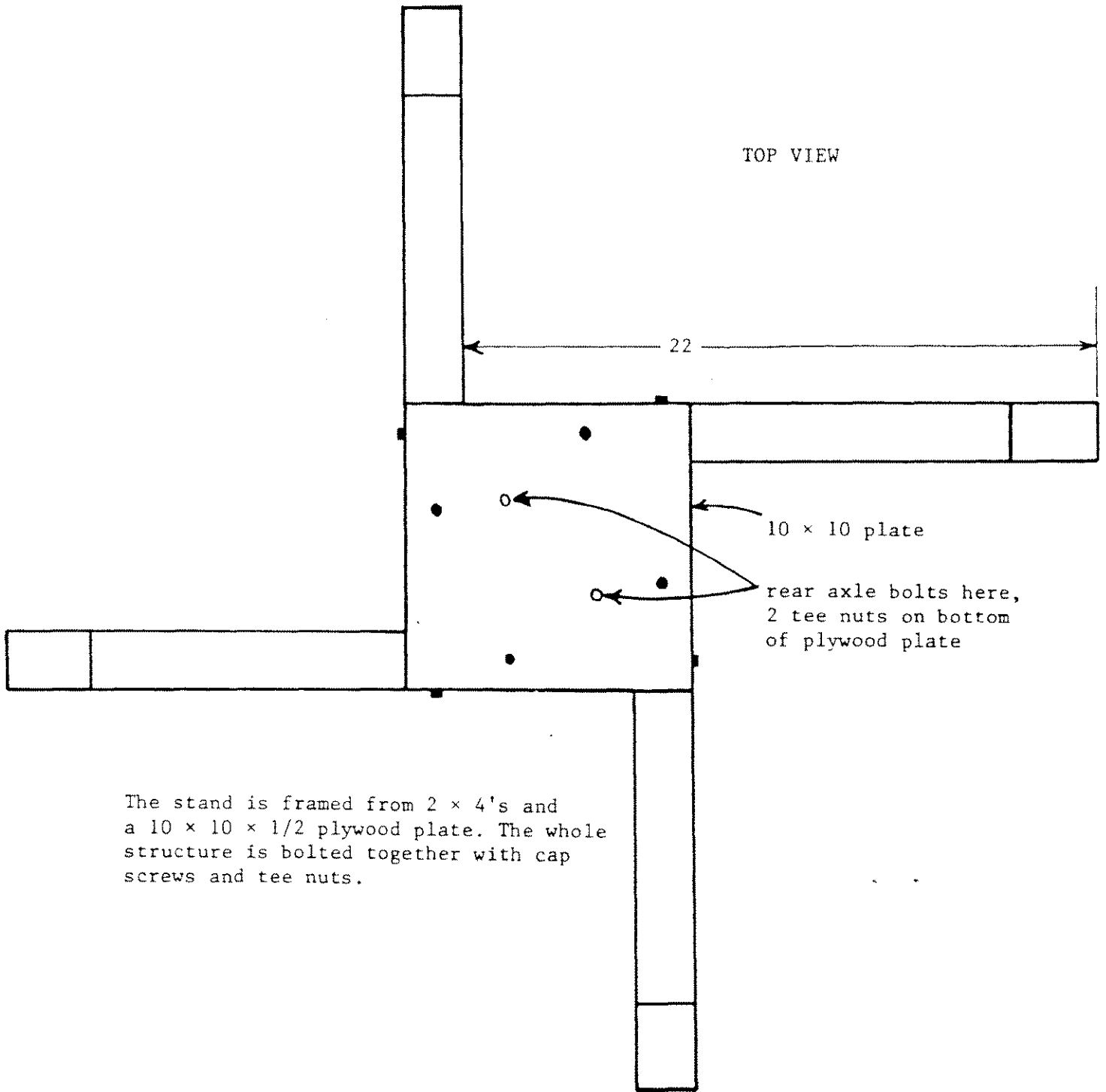
Locating the ring gear is much simpler and faster than locating the pinion. The MGA, early MGB, and TD/TF ring gears are "bolt on interchanges". Before you place the ring gear on the carrier make sure the surfaces are scrupulously clean and free of any burrs. The ring gear must run true (maximum runout 0.002 inch) as shown by a dial indicator touching the back of the slowly rotating ring gear. This measurement is generally made on a surface table with a set of large V blocks, and a magnetic base for the dial indicator. Admittedly this is a little more equipment than most T enthusiasts normally have in their garages but you can make up a functional equivalent from two wooden blocks if you own a dial indicator. Fortunately, the runout measurement can be omitted at this time if you are picky about assembly cleanliness. It can be checked later by other means, but you will have to wait until after the rear axle is completely assembled. The 3/4 inch long bolts that secure the ring gear are the same on the later T series, MGA, and early MGB (3/8 x 20 U.N.F.). The early TDs use 3/8 x 24 B.S.F. These bolts are hardened steel and should not be substituted for with the usual hardware store bolt. Secure the ring gear to the differential carrier, omitting one bolt, and bend up the appropriate ends on the lock tabs. The factory recommends that the earlier 3/8 x 24 bolts be tightened to 450 lb-ins, the later 3/8 x 20 bolts to 540 lb-ins. I would decrease these torque settings by 10% to stay a little further away from the elastic limit of the bolt. Place the differential carrier in position with any carrier spacer in the main casting. Place the cover casting in position on the main casting but omit the carrier spacer from the cover side for the present. The assembled

castings must be in a vertical position bolted to the wooden stand that was previously made. Someone (wife) might hold the rear axle assembly vertical if no stand was made, but she sure will tire and become bored after a while. The backlash is determined by the location of the ring gear. Moving the ring gear toward the pinion reduces the backlash while moving the ring gear away from the pinion increases the backlash. The minimum operating value for each gear set is etched on the back of the ring gear in the case of the MGA/B gears. There is no backlash number etched on the T series gears. For either gear set the backlash must be set at a minimum of 0.006 inch under operating conditions. Since preloading the differential carrier bearings (next section, D) will reduce the backlash by 0.002 inch, the backlash at this time must be set at a minimum of 0.008 inch. Hence, set the minimum backlash to the number on the back of the ring gear or 0.008 inch, whichever number is greater. There are two methods of measuring the backlash -- both will probably work satisfactorily although I personally have only used the first method.

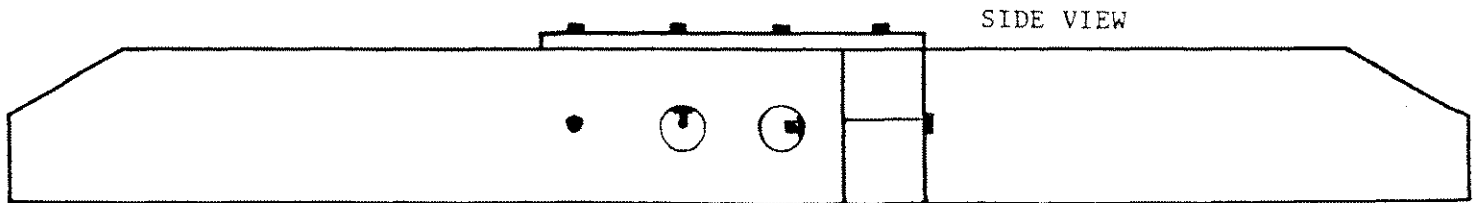
1. The first, most satisfactory, method is to screw a short threaded rod with locknut in the open bolt hole in the ring gear so that the rod extends down below the teeth of the ring gear (most hardware stores carry 3/8 x 20 threaded rod). This rod must have an indent drilled in its side and must be turned so that the indent faces the oil drain hole in the main casting. The push rod of a dial indicator is then inserted through the oil drain hole and then picks up the indent on the rod. The dial indicator will need a long extension so its push rod can reach in far enough to pick up the indent. The ring gear must also be turned so that the threaded rod is located on the ring gear approximately opposite the pinion. The push rod will then make a right angle with an imaginary diameter line across the ring gear. This is to avoid a cosine relation that would otherwise reduce the reading obtained. Since it is impossible to reach down through the upper axle tube and grab the differential carrier, make a fork that will go all the way through the axle tube, through the splined hole in the spider gear, and then engage with the cross shaft in the differential carrier. This fork serves to grab the differential carrier. Hence, with the pinion gear held still, the fork can then rock the ring gear back and forth so that the dial indicator can read the backlash between the two gears.

2. If you do not have a dial indicator available, the less direct method of measuring the backlash is to lock the ring gear so it cannot rotate and then rotate the pinion gear back and forth to measure an angular backlash. The preload-

Stand for Rear Axle

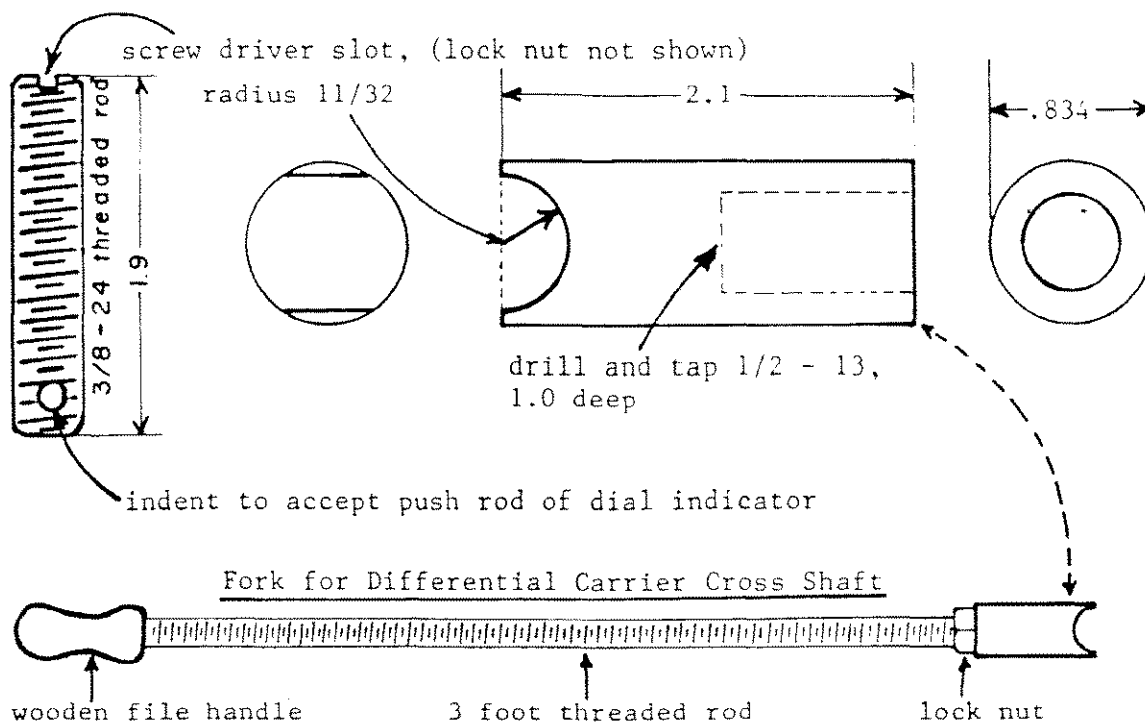


The stand is framed from 2 × 4's and a 10 × 10 × 1/2 plywood plate. The whole structure is bolted together with cap screws and tee nuts.



ing that was set on the pinion shaft interferes with the "feel" of rocking the pinion. Hence, slacken the pinion nut if you have trouble. Do not, however, loosen the pinion nut so much that you introduce play in the pinion shaft. I read a pinion rotation of about 1.5 degrees between the clicks made when the gear teeth touch. This corresponds to the minimum 0.008 inch backlash at the gear teeth. The angle can be estimated with an inexpensive plastic protractor held up against the pinion flange, but believe me, this second method is very difficult.

All the MGA to T series conversions I know of have had insufficient backlash when the ring gear was first installed. They all required the addition of a 0.010 to 0.015 inch shim to move the ring gear away from the pinion so that even the minimum 0.008 inch backlash was attained. You will probably have to fit an initial shim of about 0.010 inch just to obtain a measurable backlash reading. Once the first shim has been installed and its associated backlash has been measured, the thickness of a second and final shim can be calculated. An axial movement of 0.001 inch in the ring gear produces very close to a 0.001 inch change in the backlash. The homemade shim should always be fitted between the carrier bearing and the thick (about 0.5 inch) carrier spacer. If the O.D. of the homemade shim closely fits the axle tube, as it should, then were the shim inadvertently placed between the carrier spacer and the axle tube, the shim might pucker since the seat in the axle tube does not have a square inside corner. Notice the enormous chamfer on the original T series carrier spacers.



If you have any doubt about your backlash measurement, increase the backlash to a greater value. If it is too tight, you will ruin the gear set. If it is too loose, you may hear a little click when you change from acceleration to deceleration using the engine as a brake. Typical backlash values for American engineering practice range from 0.012 to 0.025 inch. Hence, if you insist on setting up to the minimum English value of 0.006 inch, you will have removed all possible play and have left just enough clearance for ring gear wobble and thermal expansion -- you hope.

D. ADJUSTMENT OF DIFFERENTIAL CARRIER BEARING PRELOAD

The preloading of the differential carrier bearings will be accomplished by the squeeze that occurs when the cover casting is bolted to the main casting. After succeeding with the previous three measurements this final measurement will seem quite simple. Either purchase or cut a gasket to fit between the two rear axle castings. Your gasket will probably be somewhere between 0.003 and 0.010 inch thick. The thickness will not matter provided you know what the thickness is. Measure the compressed gasket thickness with a 0 - 1 inch micrometer, by tightening the micrometer snugly. Do not, however, apply a herculean force to the micrometer and distort the micrometer frame. After all, you only want to measure the gasket thickness when the gasket is compressed to the thickness it will attain under final assembly conditions.

Leave the spacer and shim undisturbed in the main casting as this determines the backlash. Again choose any thickness of T series carrier spacer and with this spacer in position on the carrier bearing, replace the cover casting in position on the main casting. Observe that a small gap exists between the main casting and the cover casting. To measure this gap provide yourself with two complete sets of feeler gauges. Always keep the two sets stacked to the same thicknesses and always insert the stacks in opposite positions across the flange. Vary the thickness of the pair of feeler gauges until a thickness is found that just drags when the feeler stacks are pushed into the gap. The gasket surface on the two castings must be free of all burrs and any kind of grit or gasket compound so a true feeler reading will be obtained. Be sure that the cover housing is also pressed down on the differential carrier and that the carrier spacers and bearings are free of any grit. When the cover casting is in its place resting on the top of the main casting, tap it with a heavy, soft mallet to insure that the cover casting is properly in contact with the differential carrier spacers. By this procedure you will find that the feeler gauge readings are generally both reproducible and do indeed accurately

(± 0.001 inch) measure the gap between the two castings. Generally the flanges are true and a uniform gap measurement is attained all the way around the flange. Occasionally, a warped flange is encountered that produces a variation in the gap measurement of perhaps 0.005 inch. There is little one can do about the warp, but with some thought in assembly it is possible to work around it. Make a chart of your measurements and then visualize the warp as being all in the cover casting. With this pattern in mind, make a guess as to how the cover will bend when it is pulled down and then estimate a nominal gap distance. I know this sounds like hocus pocus, but it does work, for there is the secondary check on the carrier bearing preload made by noting the slight increase in the pinion torque that occurs when the flanges are drawn together. You may also recheck the backlash in the drawn down condition if you are worried. If these two tests confirm that everything is assembled properly, then believe them, for all is well.

With the gap between the castings known and the compressed thickness of the gasket known, the final measurements are now complete. The thickness of the differential carrier spacer in the cover casting is now adjusted so *the gap between the two castings becomes 0.004 inch greater than the compressed gasket thickness.* This means that each of the differential carrier bearings will be crushed by 0.002 inch when the cover casting is finally bolted down. If the differential carrier spacer is too thick, take it to the grinding shop. If it is too thin, make the appropriate shim.

When final assembly occurs, please use a spray-on gasket compound (Permatex No. 99 MA). Place the gasket on a newspaper and spray first on one side and then on the other side. In this manner a beautiful, uniform coating is obtained. This type of gasket cement does not harden before the gasket can be worked down over the studs in the main casting. When tightening the stud nuts, be sure to tighten alternately back and forth across the casting so that you do not push the gasket ahead of you and hence tend to generate a gasket wrinkle at the final stud. If your measurements were correct you will find that the preload on the differential carrier bearings has increased the torque required to rotate the pinion by no more than 1 lb-ft. An indirect check may now be made on the ring gear runout. Insert the cross shaft fork back in the axle tube and rock the ring gear back and forth. Listen to the clicks and feel the relative movement of the fork. Rotate the pinion so that you go to a new position on the ring gear and then move the fork back and forth. If the ring gear "feels" and sounds the same all the way around then it is

probably running true. If the backlash diminishes on one side of the ring gear then you have to decide whether it is bad enough to warrant disassembly and correction.

This ends the explanation of the ring and pinion interchange methods. The locating tolerances of ± 0.001 inch are in agreement with the factory recommendations and are very time consuming. You can certainly err on the side of increased play and still have a silent rear end. One of the early rear end conversions had its MGA pinion located 0.029 inch too close to the ring gear axis. It made a slight whine at 4000 rpm but still it was considerably quieter than the original factory unit it replaced. When T series rear ends are disassembled for the first time after 25 years the pinion is generally found to have moved away from the ring gear. In some cases it is displaced by as much as 0.007 inch. (This assumes that they were all set up according to the factory jig 25 years ago). In any event, almost all of these rear ends with displaced pinions were silent before being taken apart so that perhaps the ± 0.001 locating tolerances are only desirable but not mandatory. I would still adhere to them but I am beginning to regard them with a little skepticism. If your original pinion has lost its preloading and turns freely, you might allow 0.003 inch for wear when making the initial pinion/plate measurement.

IV. RELATED REAR AXLE TOPICS

A. TOP PLATE FROM MGA

When the rear axle is removed from a TD or TF, the axle tubes inevitably display four grooves worn by the U bolts. This is normal since the rubber mounting pads around the spring allow a little movement of the rear axle. On those TDs and TFs where the U bolts were a little loose through perished rubber or slack-nuts, the four grooves will be quite deep. This wearing problem was recognized by the factory and was solved when the MGA was placed in production. A cap or top plate was fitted over the axle tube and under the U bolts. These MGA top plates are a direct fit for the TD/TF rear axle. Install a pair of these plates, they are a retrofit that should be added to all the TD/TF series.

B. BREATHER

All rear axle assemblies have some kind of breather to equalize atmospheric pressure changes between the inside and outside of the housing. If they were

sealed, the hypoid oil would tend to be forced out under the oil seals when the pressure became greater on the inside. On the TD and TF, this pressure equalization takes place through a small hole drilled through the top of the left axle tube. The hole is normally hidden under a brake pipe strap. If you want, you can drill and tap the hole for a 1/8 inch taper pipe thread and then install any one of several breather assemblies. These assemblies may do a better job of excluding road dust and perhaps also water when cleaning the chassis with a high pressure water hose. These breather assemblies can be obtained by purchasing the corresponding MGA or Jaguar part.

C. OIL SEALS AND OIL LEAKS

Oil seals will only work if the surface they ride on is completely free of all imperfections. The propeller shaft flange that the pinion oil seal runs against must be inspected and if necessary polished. The collars that the axle shaft oil seals run against must also be inspected. If these hardened and ground collars are not perfect, replace them. The collars are readily available since the collar is a part common with the MGB.

Grooves commonly develop in these collars as a result of cleaning off the brake backing plate. If, during a brake job, the rear backing plate is washed down with alcohol or solvent and the dirt laden liquid allowed to run down about the seal, particles of dirt will remain at the lip of the seal when the solvent evaporates. Some of these particles of dirt will later work themselves under the lip of the seal and then proceed to groove the collar and ruin the integrity of the oil seal. Hence, when you clean things up with a solvent, think of where the dirt goes and make sure that it goes in your drip pan and not under an oil seal. The lip of any oil seal must be well oiled during assembly so that the seal does not start out dry and perhaps ruin its sealing surface during start up.

There are some TD/TF rear axles in which the wheel bearings do not fit tightly enough in the axle housing or the axle end cap to stop oil from seeping out at the brake backing plate. The oil gently oozes out between the flanges before it reaches the oil seal. This can be stopped by applying a little gasket compound to the flanges. Don't cement everything together or you will later need a slide hammer to remove the axle shaft. Use the recently developed gels that serve as paper gasket substitutes (Loctite 2319 or Permatex 2001).

D. STUD REMOVAL

If you want to remove the studs in the main casting in an elegant manner, purchase the Snap-on collet type stud remover. With the exception of this one, all stud removers I have ever seen are cheap, nasty devices that mutilate every stud they are placed on. The Snap-on set consists of a threaded collet and housing that will remove a stud without leaving a mark. You will need a separate collet for each thread, but it is worth it. The collets are available in the fine and coarse American threads and hence will fit U.N.F. (Unified National Thread). In addition they are also available in metric threads. To obtain a B.S.F. (British Standard Fine) collet, purchase a collet for a smaller American thread, drill it out and then tap it for the appropriate B.S.F. size. If you purchase the following sizes I believe you will have a collet for every stud and straight dowel pin on a TD or TF:

<u>Collet</u>	<u>Part No.</u>	<u>Housing</u>
6 mm x 1.00	CG-500-30	CG-500-2
8 mm x 1.00	CG-500-36	
10 mm x 1.50	CG-500-43	<u>Box</u>
12 mm x 1.50	CG-500-49	KRA-150
1/4 smooth collet	CG-500-20	(Compartmented
3/8 x 20 B.S.F.	Make up from	box for above
	CG-500-3	tools.)
3/8 x 24 N.F.	CG-500-8	

Read 6 mm x 1.00 as 6 mm diameter x 1.00 mm pitch; 3/8 x 20 as 3/8 inch diameter x 20 threads per inch.

E. SPEEDOMETER GEARING

When the rear axle gear ratio is altered, the speedometer gearing must also be changed or the speedometer and odometer will both be in error. The three original rear end gear ratios available for the TD and TF each had their associated speedometer gears. Table II lists these gears. The early versions of the S speedometer gears (TD gearbox TW395 and earlier) relied on the clamping action of the mainshaft nut to prevent them from turning on the mainshaft. This was not always successful as a circlip further down on the main shaft would generally bend and relieve the compressive load on the speedometer gear. When this occurred, the speedometer gear slipped. It not only slipped, but under high speed operation, the

gear could suddenly emit a piercing shriek that was accompanied by a wildly fluctuating speedometer needle. At two o'clock in the morning in the middle of Texas it was a sound to test one's faith in all things mechanical. The factory later keyed the speedometer gear to the mainshaft to prevent this slippage. Hence, fit the keyed gear if you have the opportunity to do so. You will need the appropriate Woodruff key and a later mainshaft. If you can't locate a keyed mainshaft, a keyway can be cut in your old shaft. (A tungsten carbide cutter will be required as the shaft is hardened).

TABLE II

Rear Axle Ratio	Speedometer Gear Ratio (Gear Teeth/Pinion Teeth)	Gear Part No.	Pinion Part No.	Letter Stamped on Both Gears
5.125/1	5/13	AEG3110	X22797	S
4.875/1	6/15	168210	X17260	T
4.555/1	6/14	AEG3109	X19101	U

All Jaeger speedometers have a four digit number printed on their faces somewhere after the model number. This four digit number gives the number of revolutions the speedometer shaft must make to register one mile on the odometer. The constants for the TC, TD and TF are: TC -- 1675 rev/mile, TD -- 1600 rev/mile, TF (1250 and 1500) -- 1525 rev/mile, and for metric calibration, TC -- 1050 rev/km, TD -- 1000 rev/km. If your speedometer had been worked on or has a doubtful ancestry, open it up and determine the number of revolutions per mile by counting the gear teeth. As an example, the chronometric (flat face) speedometer has a single worm gear on the end of its input shaft. This worm engages a 25 tooth pinion which in turn advances a 64 tooth ratchet. One turn of the 25 tooth pinion advances the 64 tooth ratchet wheel one tooth. One full turn of the ratchet wheel will then advance the first number on the odometer by one digit. Accordingly, it takes 25 x 64 or 1600 revolutions of the input shaft to advance the odometer reading one mile.

The speedometer calibration problem is one of providing the speedometer with its required number of revolutions per mile from a gear box that now gives an incorrect number of revolutions per mile. Fortunately, Stewart-Warner makes a very

nice little speedometer correction gear box (drive joint) that can be adapted to a T series MG. You can either cut your speedometer cable and swage on the appropriate American cable fittings or you can machine the necessary collars and shafts so the speedometer gear box will screw directly on the MG gear box. The second approach is more satisfactory since it doesn't require the speedometer cable to drive a grease filled speedometer gear box. This would be a problem for the speedometer cable in the winter. With the TD/TF installation the grease could probably be omitted when the speedometer gear box is attached directly to the MG gear box. In this case I would expect the normal oil seepage along the pinion to transfer sufficient oil into the speedometer gear box.

Stewart-Warner manufactures five basic kinds of speedometer gear boxes, each of which is available with 164 gear combinations. The Model 666 is probably the best suited for T series installation. The Model 666 pamphlet has been reprinted here with the kind permission of Stewart-Warner. Let us now calculate a T series installation for the 813 rev/mile tires and the 4.300/1 rear axle. The propeller shaft will make 813×4.300 or 3495.9 rev/mile. If the original speedometer gears (S set) are still fitted in the gear box, then the output pinion in the gear box will make $3495.9 \times 5/13$ or 1344.57 rev/mile. The speedometer requires 1600 rev/mile so we will need a S-W gear box with a ratio of $\frac{1600}{1344.6}$ or 1.190. Examination of the S-W chart shows the closest ratio to be 1.2020. Hence, this is in error by less than 1%. These speedometer gear boxes (drive joints) are available at speedometer repair shops all over the country, and you should have no trouble if you know the ratio you need. Drawings of the parts necessary to attach the little S-W gear box to the MG gear box are included. As a closing comment, if your speedometer and tachometer faces show signs of age, new ones may be obtained from

René Rüfenacht,
Gibelstrasse 39
CH-2540 Grenchen
SWITZERLAND

René has made up new dial faces for the flat face chronometric instruments that were fitted to the TC and early TD. I have seen his dial faces (\$13.50 each, 1975) and they are excellent.

In closing, I hope that this booklet will allow T series owners outside of Orange County to benefit from the MG activities of the members of our club.

Best wishes, and may your MGs run until the roads wear out.

MG forever,

944 E. Redbud St., Brea, CA, 92621

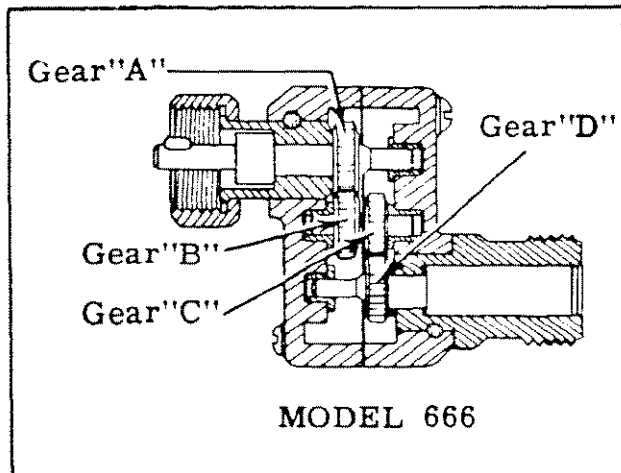
Carl Cederstrand

NOTE FOR "LOCATION OF THE PINION"

In fitting the MGA pinion there is one additional bearing problem which you might encounter. This problem is rare and has occurred only once in the past year here in the VMG. It arises when the cone of the outer bearing cannot be pushed far enough on the pinion shaft to contact its cup. This occurred with one particular pinion because the surface of the shaft was not ground far enough along its length. The cone would push to the end of the ground surface on the pinion shaft and then stop when it reached the unground portion. The cone needed an additional 0.060 inch of ground shaft. The solution was to go to the grinding shop and have the ground area extended.



DRIVEN SHAFT, (output) Purchase an S-W shaft number 15203 and bore it out to 0.302 inch. This must be bored out on a lathe since a drill will "walk" off axis because of the existing slot. The slot must also be opened up to 0.085 inch to accommodate the English key. The slot can be opened with a thin Swiss file since only a few thousandths need be removed.

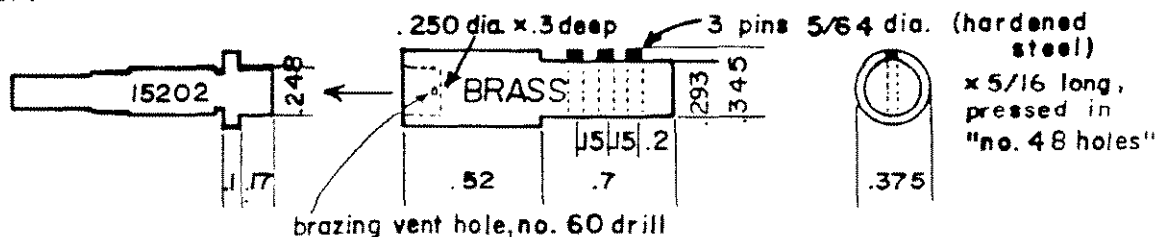


COUPLING NUT Remove a nut from either an old Smith's speedometer cable or a tachometer reduction gear box. Both of these Smith's nuts are the same and will fit properly if the I. D. of its flange is opened up slightly.

SLEEVE, CABLE END Purchase an S-W sleeve number 15204. This sleeve will have a 7/8 x 18 thread. Take the sleeve and the Smith's coupling nut to a machine shop and have a 3/4 x 26 thread cut on the S-W sleeve. The 3/4 x 26 thread is a British Brass Thread that is used in England for general brass work and on early gas burner fit-

tings. The major diameter is, of course, 0.7500 inch, the minor diameter 0.7007 inch, the included angle 55° and the profile is that of a Whitworth thread. See page 19 of *Guide to world Screw Threads*, Edited by P. A. Sidders, Industrial Press, Inc., 200 Madison Ave., New York, N. Y., 10016. Library of Congress Catalog Card Number: 71-185990.

DRIVE SHAFT, (input) Purchase an S-W shaft number 15202 and machine it as shown in the drawing. Make the additional part as illustrated and silver solder the two together.

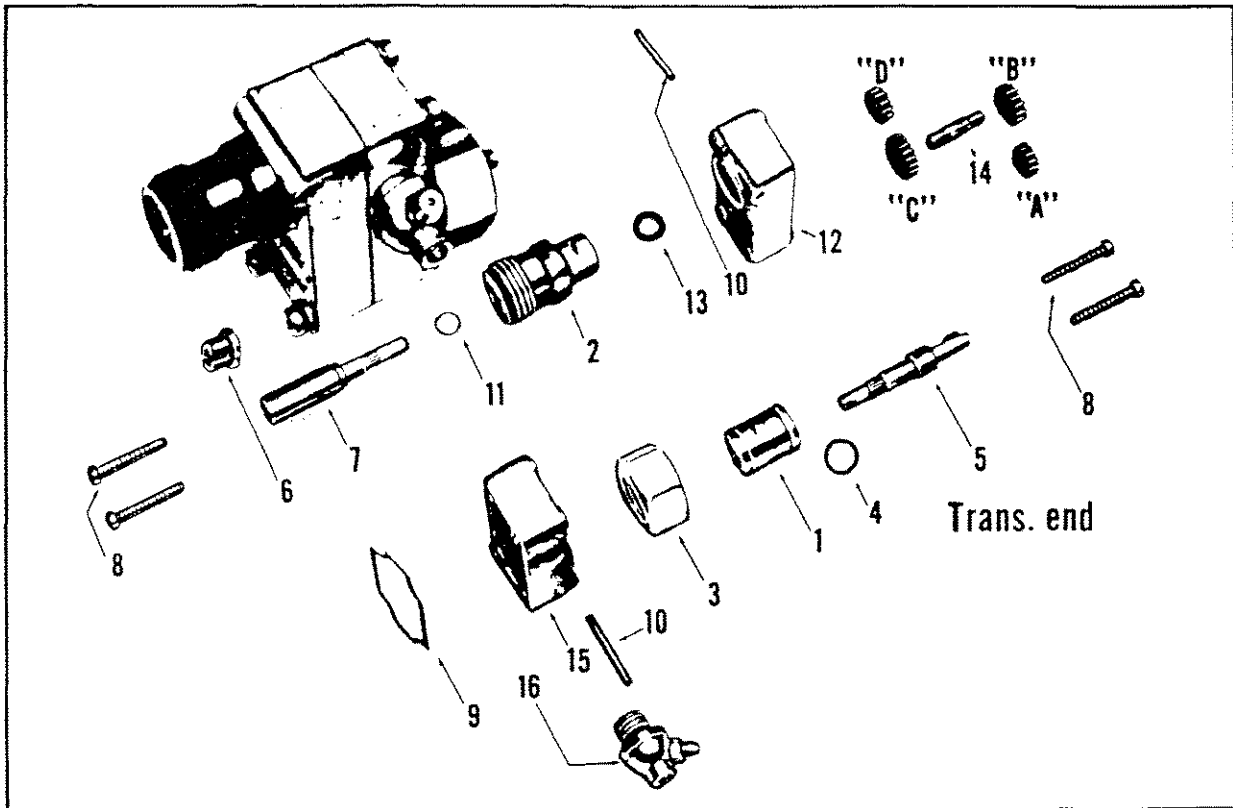




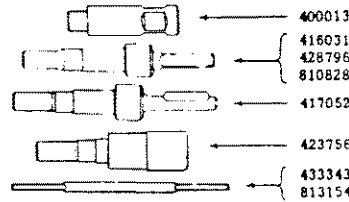
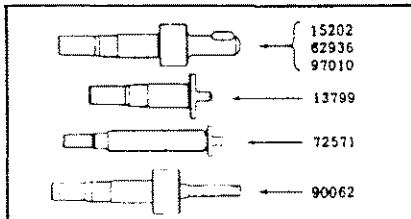
SERVICE PARTS LIST



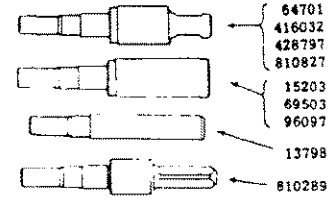
666 SERIES DRIVE JOINTS



DRIVE SHAFTS



DRIVEN SHAFTS



Model 666 Drive Joints are for correcting an instrument that registers fast or slow because tire size, axle ratio or transmission gears have been changed on a passenger car or truck. To determine correct ratio required, refer to SER-SW 4410.

HOW TO ORDER

Illustrated above is a typical Model 666 Drive Joint. Numbers in the illustration correspond to numbered descriptions in parts list. Do not order by these numbers.

Locate part required in the illustration above. Turn page to parts list column headed by model being repaired. Locate description of part by using its illustration number. Read across from description to column headed by drive joint model number to find part number. Order by this PART NUMBER and DESCRIPTION.

When ordering entire drive joint, be sure to specify MODEL NUMBER, RATIO and DESCRIPTION.

STEWART-WARNER CORPORATION

1826 W. Diversey Parkway
Chicago, Illinois 60614

Printed in U.S.A.

SER 666 (1-71)

No.	Description	666-A	666-B	666-D	666-E	666-F	666-G	666-H	666-N
1.	Sleeve (Trans. End)	13112	13112	13112	72570	13112	13112	13112	13112
2.	Sleeve (Shaft End)	13111	15204	15204	15204	15204	15204	15204	15204
3.	Coupling Nut	6071	33637	33637	-0-	33637	33637	33637	33637
4.	Washer	-0-	-0-	-0-	35029	-0-	-0-	-0-	-0-
5.	Drive Shaft	13799	15202	62936	72571	90062	15202	15202	90062
6.	Clutch or Tip	841	-0-	-0-	-0-	-0-	-0-	-0-	-0-
7.	Driven Shaft	13798	15203	69503	15203	64701	64701	15203	15203
8.	Screw	171885	171885	171885	171885	171885	171885	171885	171885
9.	Gasket	13144	13144	13144	13144	13144	13144	13144	13144
10.	Pin	13116	13116	13116	13116	13116	13116	13116	13116
11.	Washer	843	844	844	844	844	844	844	844
12.	Case (Shaft Side)	14477	14477	14477	14477	14477	14477	14477	14477
13.	Washer	433326	433326	433326	433326	433326	433326	433326	433326
Gears A, B, C & D		See Gear Chart on Page 4.							
14.	Intermediate Shaft*	13787	13787	13787	13787	13787	13787	13787	13787
15.	Case (Trans. Side)	14477	14477	14477	101487	14477	14477	101487	14477
16.	Alemite Fitting	-0-	-0-	-0-	1940-B	-0-	-0-	1940-B	-0-

No.	Description	666-P	666-R	666-S	666-T	666-U	666-X	666-Y	666-Z
1.	Sleeve (Trans. End)	13112	13112	13112	13112	423635	13112	13112	13112
2.	Sleeve (Shaft End)	15204	15204	15204	15204	15204	15204	15204	15204
3.	Coupling Nut	33637	33637	33637	33637	33637	33637	33637	33637
4.	Washer	-0-	-0-	-0-	-0-	-0-	-0-	-0-	-0-
5.	Drive Shaft	90062	15202	416031	417052	423756	97010	428796	15202
6.	Clutch or Tip	-0-	-0-	-0-	-0-	-0-	-0-	-0-	-0-
7.	Driven Shaft	64701	15203	416032	69503	69503	96097	428797	69503
8.	Screw	171885	171885	171885	171885	171885	171885	171885	171885
9.	Gasket	13144	13144	13144	13144	13144	13144	13144	13144
10.	Pin	13116	13116	13116	13116	13116	13116	13116	13116
11.	Washer	844	844	844	844	844	844	844	844
12.	Case (Shaft Side)	14477	14477	14477	101487	101487	14477	14477	14477
13.	Washer	433326	433326	433326	433326	433326	433326	433326	433326
Gears, A, B, C & D		See Gear Chart on Page 4.							
14.	Intermediate Shaft*	13787	13787	13787	13787	13787	13787	13787	13787
15.	Case (Trans. Side)	101487	414286	101487	414286	414286	14477	14477	14477
16.	Alemite Fitting	1940-B	1940-B	1940-B	1612-B	1612-B	-0-	-0-	-0-
	Relief Valve	-0-	-0-	-0-	47200	47200	-0-	-0-	-0-

-0- Indicates part not used on this model.

* Intermediate Shaft is part of Gear for Ratios .3746, .3882, 2.5757 and 2.6694 - See Gear Chart. Used on 666-T only

See Gear Chart on Page 4 to obtain correct gears and desired Ratio. Always be certain that Gear "A" drives Gear "B", and Gear "C" drives Gear "D".

Place Gears "A" thru "D" as in illustration. For Ratios 1.000 or less use top headings of Gear Chart and read from left to right ("A" to "D").

For Ratios 1.000 or more use bottom headings of Gear Chart and read from right to left. ("D" to "A", example "D" becomes "A", "C" becomes "B", etc).

No.	Description	666-AA	666-AB	666-AD	666-AF	666-AH	666-AJ	666-AK
1.	Sleeve (Trans. End)	433321	433321	13112	415245	813152	13112	813152
2.	Sleeve (Shaft End)	15204	15204	15204	15204	15204	15204	15204
3.	Cpl. Nut or Clamp	429982	429982	33637	421498	429982	36589	429982
4.	Washer	-0-	-0-	-0-	-0-	-0-	-0-	-0-
5.	Drive Shaft	400013	400013	97010	810828	400013	90062	400013
	Drive Tip	433343	433343	-0-	-0-	813154	-0-	813154
7.	Driven Shaft	64701	64701	810289	810827	64701	64701	64701
8.	Screw	171885	171885	171885	171885	171885	171885	171885
9.	Gasket	13144	13144	13144	13144	13144	13144	13144
10.	Pin	13116	13116	13116	13116	13116	13116	13116
11.	Washer	844	844	844	844	844	844	844
12.	Case (Shaft Side)	433323	433323	14477	14477	433323	14477	433323
13.	Washer	433326	433326	433326	433326	433326	433326	433326
	Gears A, B, C & D	See Gear Chart on Page 4.						
14.	Intermediate Shaft	13787	13787	13787	13787	13787	13787	13787
15.	Case (Trans. Side)	14477	14477	14477	14477	14477	14477	14477
	Washer	416789	416789	-0-	-0-	416789	-0-	416789
	Cpl. Nut	820346-22	820346-22	-0-	-0-	820346-22	-0-	820346-22
	Adapter Sleeve	433325	433325	-0-	-0-	433325	-0-	433325
	Set Screw	105179	105179	-0-	-0-	105179	-0-	105179

-0- Indicates Part Not Used on this Model.

END CONNECTIONS

Model	Drive End		Driven End	
	Drive Shaft	Sleeve or Nut	Driven Shaft	Sleeve or Nut
666-A	.093 tongue	3/4-20 Int. Thd.	.130 slot clutch	3/4-20 Ext. Thd.
666-B	.193 dia. bar	7/8-18 Int. Thd.	.213 dia. hole	7/8-18 Ext. Thd.
666-D	.152 dia. bar	7/8-18 Int. Thd.	.161 dia. hole	7/8-18 Ext. Thd.
666-E	.125 tongue	3/4 dia. sleeve	.213 dia. hole	7/8-18 Ext. Thd.
666-F	.104 square	7/8-18 Int. Thd.	.105 sq. hole	7/8-18 Ext. Thd.
666-G	.193 dia. bar	7/8-18 Int. Thd.	.105 sq. hole	7/8-18 Ext. Thd.
666-H	.193 dia. bar	7/8-18 Int. Thd.	.213 dia. hole	7/8-18 Ext. Thd.
666-N	.104 square	7/8-18 Int. Thd.	.213 dia. hole	7/8-18 Ext. Thd.
666-P	.104 square	7/8-18 Int. Thd.	.105 sq. hole	7/8-18 Ext. Thd.
666-R	.193 dia. bar	7/8-18 Int. Thd.	.213 dia. hole	7/8-18 Ext. Thd.
666-S	.183 square	7/8-18 Int. Thd.	.195 sq. hole	7/8-18 Ext. Thd.
666-T	.152 dia. bar	7/8-18 Int. Thd.	.161 dia. hole	7/8-18 Ext. Thd.
666-U	.161 dia. hole	7/8-18 Int. Thd.	.161 dia. hole	7/8-18 Ext. Thd.
666-X	.187 dia. bar	7/8-18 Int. Thd.	.191 dia. hole	7/8-18 Ext. Thd.
666-Y	.150 square	7/8-18 Int. Thd.	.154 sq. hole	7/8-18 Ext. Thd.
666-Z	.193 dia. bar	7/8-18 Int. Thd.	.161 dia. hole	7/8-18 Ext. Thd.
666-AA	.104 square	.967 dia. sleeve	.105 sq. hole	.441 I.D. Sleeve
666-AB	.104 square	.967 dia. sleeve	.105 sq. hole	.441 I.D. Sleeve
666-AD	.183 square	7/8-18 Ext. Thd.	.187 dia. bar	7/8-18 Int. Thd.
666-AF	.104 square	.874 dia. sleeve	.105 sq. hole	7/8-18 Ext. Thd.
666-AH	.104 square	.967 dia. sleeve	.105 sq. hole	.441 I.D. Sleeve
666-AJ	.104 square	7/8-18 Int. Thd.	.105 sq. hole	7/8-18 Ext. Thd.
666-AK	.104 square	.967 dia. sleeve	.105 sq. hole	.441 I.D. Sleeve

666 SERIES GEAR CHART

RATIO	GEAR "A"	GEAR "B"	GEAR "C"	GEAR "D"	
.3746	816477-11T	816478-17T	816483-11T	816484-19T	2.6694
.3882	816477-11T	816478-17T	816487-12T	816488-20T	2.5757
4186	816477-11T	816478-17T	816477-11T	816478-17T	2.3884
4313	816477-11T	816478-17T	816485-12T	816486-18T	2.3182
4427	816477-11T	816478-17T	816489-13T	816490-19T	2.2587
4444	816485-12T	816486-18T	816485-12T	816486-18T	2.2500
4561	816485-12T	816486-18T	816489-13T	816490-19T	2.1923
4621	816475-10T	816476-14T	816477-11T	816478-17T	2.1636
4681	816489-13T	816490-19T	816489-13T	816490-19T	2.1360
4761	816475-10T	816476-14T	816485-12T	816486-18T	2.1000
4853	816477-11T	816478-17T	816479-12T	816480-16T	2.0606
4887	816475-10T	816476-14T	816489-13T	816490-19T	2.0461
5000	816479-12T	816480-16T	816485-12T	816486-18T	2.0000
5032	816477-11T	816478-17T	816491-14T	816492-18T	1.9870
5102	816475-10T	816476-14T	816475-10T	816476-14T	1.9600
5131	816479-12T	816480-16T	816489-13T	816490-19T	1.9487
5185	816491-14T	816492-18T	816485-12T	816486-18T	1.9285
5321	816489-13T	816490-19T	816491-14T	816492-18T	1.8792
5357	816479-12T	816480-16T	816475-10T	816476-14T	1.8666
5392	816477-11T	816478-17T	816472-10T	816473-12T	1.8545
5555	816472-10T	816473-12T	816485-12T	816486-18T	1.8000
5607	816477-11T	816478-17T	816481-13T	816482-15T	1.7831
5625	816479-12T	816480-16T	816479-12T	816480-16T	1.7777
5701	816472-10T	816473-12T	816489-13T	816490-19T	1.7523
5709	816477-11T	816478-17T	816493-15T	816494-17T	1.7515
5777	816481-13T	816482-15T	816485-12T	816486-18T	1.7307
5833	816479-12T	816480-16T	816491-14T	816492-18T	1.7142
5882	816485-12T	816486-18T	816493-15T	816494-17T	1.7000
5929	816481-13T	816482-15T	816489-13T	816490-19T	1.6863
5952	816475-10T	816476-14T	816472-10T	816473-12T	1.6800
6037	816489-13T	816490-19T	816493-15T	816494-17T	1.6564
6049	816491-14T	816492-18T	816491-14T	816492-18T	1.6530
6190	816475-10T	816476-14T	816481-13T	816482-15T	1.6152
6250	816472-10T	816473-12T	816479-12T	816480-16T	1.6000
6302	816475-10T	816476-14T	816493-15T	816494-17T	1.5866
6481	816472-10T	816473-12T	816491-14T	816492-18T	1.5428
6500	816479-12T	816480-16T	816481-13T	816482-15T	1.5384
6617	816479-12T	816480-16T	816493-15T	816494-17T	1.5111
6666	816474-12T	816474-12T	816485-12T	816486-18T	1.5000
6740	816481-13T	816482-15T	816491-14T	816492-18T	1.4835
6862	816491-14T	816492-18T	816493-15T	816494-17T	1.4571
	GEAR "D"	GEAR "C"	GEAR "B"	GEAR "A"	RATIO

RATIO	GEAR "A"	GEAR "B"	GEAR "C"	GEAR "D"	
.6944	816472-10T	816473-12T	816472-10T	816473-12T	1.4400
7222	816481-13T	816482-15T	816472-10T	816473-12T	1.3846
7333	816477-11T	816478-17T	816494-17T	816493-15T	1.3636
7352	816472-10T	816473-12T	816493-15T	816494-17T	1.3600
7466	816477-11T	816478-17T	816482-15T	816481-13T	1.3393
7511	816481-13T	816482-15T	816481-13T	816482-15T	1.3313
7555	816485-12T	816486-18T	816494-17T	816493-15T	1.3235
7647	816481-13T	816482-15T	816493-15T	816494-17T	1.3077
7692	816485-12T	816486-18T	816482-15T	816481-13T	1.3000
7754	816489-13T	816490-19T	816494-17T	816493-15T	1.2895
7764	816477-11T	816478-17T	816473-12T	816472-10T	1.2878
7785	816493-15T	816494-17T	816493-15T	816494-17T	1.2844
7894	816489-13T	816490-19T	816482-15T	816481-13T	1.2666
8000	816473-12T	816472-10T	816485-12T	816486-18T	1.2500
8095	816475-10T	816476-14T	816494-17T	816493-15T	1.2352
8215	816473-12T	816472-10T	816489-13T	816490-19T	1.2179
8241	816475-10T	816476-14T	816482-15T	816481-13T	1.2133
8319	816477-11T	816478-17T	816492-18T	816491-14T	1.2020
8500	816479-12T	816480-16T	816494-17T	816493-15T	1.1763
8571	816485-12T	816486-18T	816492-18T	816491-14T	1.1666
8626	816477-11T	816478-17T	816480-16T	816479-12T	1.1590
8653	816479-12T	816480-16T	816482-15T	816481-13T	1.1555
8797	816489-13T	816490-19T	816492-18T	816491-14T	1.1367
8814	816491-14T	816492-18T	816494-17T	816493-15T	1.1344
8888	816485-12T	816486-18T	816480-16T	816479-12T	1.1250
8974	816491-14T	816492-18T	816482-15T	816481-13T	1.1143
9000	816479-12T	816480-16T	816473-12T	816472-10T	1.1111
9058	816477-11T	816478-17T	816476-14T	816475-10T	1.1038
9122	816489-13T	816490-19T	816480-16T	816479-12T	1.0961
9183	816475-10T	816476-14T	816492-18T	816491-14T	1.0888
9333	816491-14T	816492-18T	816473-12T	816472-10T	1.0714
9444	816472-10T	816473-12T	816494-17T	816493-15T	1.0588
9457	816477-11T	816478-17T	816490-19T	816489-13T	1.0574
9523	816475-10T	816476-14T	816480-16T	816479-12T	1.0500
9577	816489-13T	816490-19T	816476-14T	816475-10T	1.0439
9615	816472-10T	816473-12T	816482-15T	816481-13T	1.0400
9642	816479-12T	816480-16T	816492-18T	816491-14T	1.0370
9705	816477-11T	816478-17T	816486-18T	816485-12T	1.0303
9743	816485-12T	816486-18T	816490-19T	816489-13T	1.0263
9822	816481-13T	816482-15T	816494-17T	816493-15T	1.0181
1.0000	816472-10T	816473-12T	816473-12T	816472-10T	1.0000
	GEAR "D"	GEAR "C"	GEAR "B"	GEAR "A"	RATIO

*Intermediate Shaft is part of Gear for these Ratios. Used on 666-T only.

Twelve pairs of gears are required to make up all of the ratios above. Order and use only in pairs as follows:

816472-10T	816474-12T	816475-10T	816477-11T	816479-12T	816481-13T	816483-11T	816485-12T	816487-12T	816489-13T	816491-14T	816493-15T
816473-12T	816474-12T	816476-14T	816478-17T	816480-16T	816482-15T	816484-19T	816486-18T	816488-20T	816490-19T	816492-18T	816494-17T

ADDENDUM

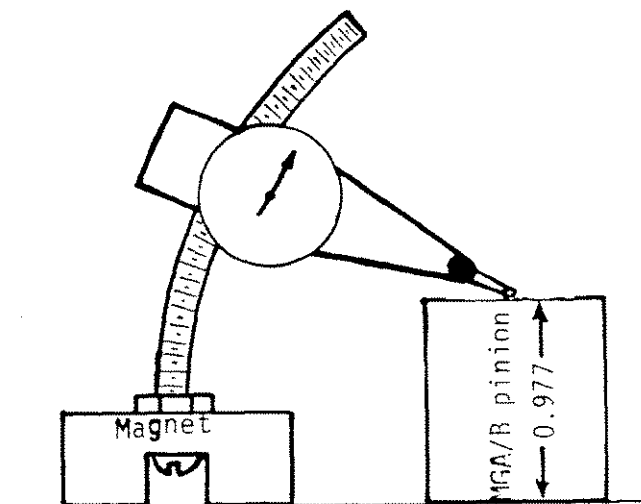
A. STEWART-WARNER SPEEDOMETER CORRECTION GEAR BOX

Unless you are skilled in silver brazing, I would suggest that you initially make the brass extension shown on p. 34 about 0.015 inch oversize. After brazing it would then be turned down to size so that any warping that might have occurred would be machined out.

B. MASTELL'S PINION LOCATION PROCEDURE (5TH METHOD)

Dale Mastell has developed the following pinion location method for the TD/TF rear end. This method is a direct measurement method that parallels the method in the MGA workshop manual. It requires a minimum of tooling and has been used successfully by Dale for several years. A 1/4 inch diameter machine screw, several inches long, is secured in the central hole of a button magnet. (Button, or pot, magnets made by Eclipse are available in hardware stores all over the country.) The screw is then bent and a dial indicator is attached to this inclined screw (see illustration). To set the dial indicator, a 0.977 inch gauge block must be made up at a local grinding shop or borrowed from a machinist friend. The exact dimension is not at all critical as long as the dimension, whatever it is, is accurately known. The magnet with its attached dial indicator is then placed on a flat plate where the reading is noted when the finger touches the top of the 0.977 inch gauge block. In this manner the dial indicator is calibrated with reference to the base of the magnet.

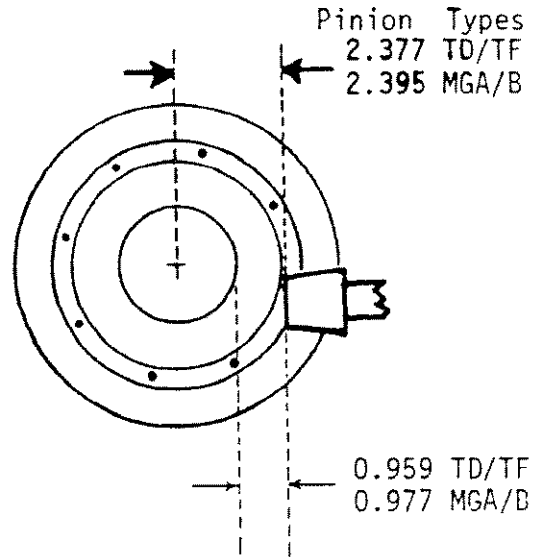
The dial indicator and magnet are then carefully moved over to the head of the pinion in the rear axle housing. With the magnet then adhering to the head of the pinion, the finger of the dial indicator is brought to bear on the bore that



Drawing Illustrates Concept Only

You must bend the machine screw so that your particular dial indicator will reach into the bore that receives the carrier bearing.

receives the differential carrier bearing. This is accomplished by rotating the pinion -- the dial indicator must not, of course, be disturbed from its gauge block setting. The pinion is slowly rotated back and forth until the finger of the dial indicator picks up the "lowest" point in the bore. By reading the dial indicator, the perpendicular distance from the head of the pinion to the bore can be calculated. For the MGA pinion, the nominal distance to be achieved (by pinion spacing washers) will be 2.395 inch minus 1/2 bore diameter. Since the bore diameter is 2.835 inch, the desired distance is 0.977 inch ($2.395 - 2.835/2 = 0.977$ inch). This is the exact locating distance as measured from the bore to the head of the MGA/B pinion when fitting the MGA/B pinion into a TD/TF rear end. The corresponding bore to pinion dimension, and gauge block dimension to be used when fitting a TD/TF pinion into the TD/TF rear end is 0.959 inch ($2.377 - 2.835/2 = 0.959$ inch).



Note that both of these bore to pinion dimensions are *before* accomodation of the production variation etched on the heads of the pinions.