TUNING for SPEED

How to Increase the Performance of A Standard Motorcycle Engine For Racing and Competition Work

by

P. E. IRVING

M.I.Mech.E., M.S.A.E.

Published in association with

MOTOR CYCLING

WITH SCOOTER WEEKLY

ALSO PUBLISHED IN ASSOCIATION WITH
"MOTOR CYCLING WITH SCOOTER WEEKLY"

"MOTOR CYCLING" MANUAL

"MOTOR CYCLING" ELECTRICAL MANUAL

MODERN MOTORCYCLE MAINTENANCE

"MOTOR CYCLING" SPORTS MODEL ROAD TESTS

TRIALS RIDING

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MOTORCYCLE ENGINEERING

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TEMPLE PRESS BOOKS
LONDON

FIRST PUBLISHED 1948
SECOND IMPRESSION 1949
SECOND EDITION 1951
SECOND IMPRESSION 1952
THIRD EDITION 1956
FOURTH EDITION 1960
SECOND IMPRESSION (REVISED) 1963
THIRD IMPRESSION 1965

*Printed by Purnell and Sons, Ltd.
Paulton (Somerset) and London*

MADE IN GREAT BRITAIN

INTRODUCTION

SINCE time immemorial man has sought to travel faster than his fellow man. This inborn human urge gained tremendous impetus some 60 years ago with the introduction of the petrol engine as a practical proposition for vehicle propulsion. Inevitably, users of the new form of transport felt their competitive instincts aroused. Their demands for more power and more miles per hour were met initially by the simple expedient of fitting larger and still larger engines. In this "brute force" era, in which sheer volume counted for more than volumetric efficiency, the results of hit-or-miss experiments were all too obvious, the scientific reasons for those results all too seldom understood.

Clearly a halt had to be called to this "Frankenstein's Monster" method of progression. With a growing understanding of basic principles in internal-combustion power production, allied with practical experiments on the road, designers turned their attention to obtaining increased performance from engines of more moderate cubic capacity. So far as motorcycles were concerned, this more scientific form of development was encouraged by the introduction of the Isle of Man Tourist Trophy Races with their strict limitation of engine sizes. This second era was one in which the observant mechanically-minded amateur rider sometimes discovered a "secret of tune" which enabled him for a time to score over his professional rivals.

With advances in metallurgy and a complete appreciation of basic principles, the point has now been reached in this, the third era, where there are no longer any "secrets". Science has replaced brute force and guesswork. Success in international events depends upon the combined efforts of the designer and development engineer, plus a host of specialist technicians responsible for ancillary equipment.

But if the day has passed when the lone amateur could hope to beat the factory representative in a major event, tuning skill, combined with riding ability, continues to bring success to the private owner in a wide variety of competitions. Nor need he possess elaborate workshop facilities to improve the performance of his standard sports model or production-type racer. What he must have, however, is the necessary "know how" and that is supplied in this Manual. Written by a well-known technician responsible for the design, development and racing preparation of highly successful "factory" models, it embodies a wealth of hard-earned knowledge yet reflects throughout the author's ready understanding of the private owner's problems.

Although intended primarily for the racing motorcyclist, much of the text will prove equally useful to those drivers of Formula III racing cars who may lack practical experience in the tuning of high-performance air-cooled single- and twin-cylinder power units. Moreover the ordinary everyday motorcyclist, perhaps not in the least degree interested in racing, can benefit from a study of the special assembly methods employed by racing mechanics; a touring engine rebuilt to racing standards of accuracy could well be more efficient in overall performance, smoother running, cleaner and quieter in operation than hitherto.

The author emphasizes that in this textbook it has not been possible to delve deeply into details of specific makes of machines and power units. Thus, should the reader find that his advice runs counter to that given in some manufacturers' instruction manual, the latter should always be heeded or the maker consulted before modifications are undertaken. Similarly, where reference is made to machine tools, heat treatments and so forth, the information is given so that the reader may explain what is required to the engineer deputed to carry out the work; specialist technicians are not always well versed in the idiosyncrasies of motorcycles and will appreciate expert advice.

GRAHAM WALKER

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AUTHOR'S FOREWORD TO THE FOURTH EDITION

As in every other branch of engineering, motorcycle design does not stand still. Steady progress is made from year to year and a book which was first written in 1948 cannot hope to remain of much value unless revised from time to time. This has already been done on four occasions, and it has now been considered advisable to prepare another edition. To this end, much dead wood in the form of references to obsolete models has been deleted, although some has been retained to assist owners of "vintage" machines or to trace the lines of development which have led up to modern designs. Much new material has been added, and several passages rearranged or altered to avoid repetition and for the sake of greater clarity, without increasing the size of the book.

In fact, I have attempted to emulate, in literary form, what the tuner has to accomplish with an engine—pack a lot more punch into the same volume!

CHAPTER I

A PRELIMINARY SURVEY

SUCCESS in tuning an engine for speed depends upon three main factors. They are: (1) getting the maximum combustible charge into the cylinder the maximum number of times per minute; (2) turning as much as possible of the heat liberated into useful work, instead of absorbing it into the combustion-chamber walls or losing it down the exhaust pipe; and (3) eliminating all unnecessary sources of internal friction.

With these three points fixed firmly in mind, some time can be spent very profitably in planning a course of action which will ultimately give the maximum benefit without wasting too much time, or money, on non-essentials.

It must be assumed that certain indispensable equipment is available, such as a good workshop bench, well lighted, and in surroundings of such a nature that scrupulous cleanliness can be observed, particularly during the final stages of assembly. It is also a good plan to prepare suitable washing and draining trays, large enough to accommodate a complete crankcase, *before* commencing work.

External and internal micrometers of assorted sizes are useful, but expensive. As one will be concerned primarily with clearances and not actual dimensions, they are not absolutely vital, but a good set of narrow-bladed feeler gauges most definitely will be. Another very useful gadget is a cast-iron surface plate at least 12 in. square. Really accurate scraped plates, too, are expensive, but a slab of cast iron or mild steel plate, surface ground on both sides, makes an excellent substitute, and there are plenty of machine shops which will carry out the grinding at quite reasonable rates. In conjunction with the surface plate will be needed a scribing block and dial gauge reading to .001 in., for checking squareness and parallelism of various components and assemblies, and a burette, or graduated glass jar, for measuring compression ratios.

A lot can be done without any equipment additional to the usual workshop hand tools, but a little capital invested at the outset is almost certain to pay a dividend later on. When lack of cash is a deciding factor, it is a good plan for several aspirants to speed honours to pool their resources and thus obtain a well-equipped workshop without an excessive drain on the pocket of each individual.

Then one must decide for what sort of event the machine is to be used—or, rather, for what it is *suitable*—grass, scrambles, hill-climb or road events. However well prepared, very few engines other than modern types can compete with any hope of even moderate success in long-distance racing when limited to non-alcoholic fuel, but they may do quite well in short-distance work—dirt or grass-tracking, hill-climbs or sprint stuff—or even long-distance track or road-racing where alcohol is permitted. Some of the engines built several years before the 1939–45 war are excellent for such work; they are usually lighter than their modern brethren, and although sometimes rather deficient in fin area judged by modern standards, this is not to their detriment if the engine is to run exclusively on "dope" fuel; in fact, it may even be an asset in short-distance events when a rapid warm-up to operating temperature is necessary.

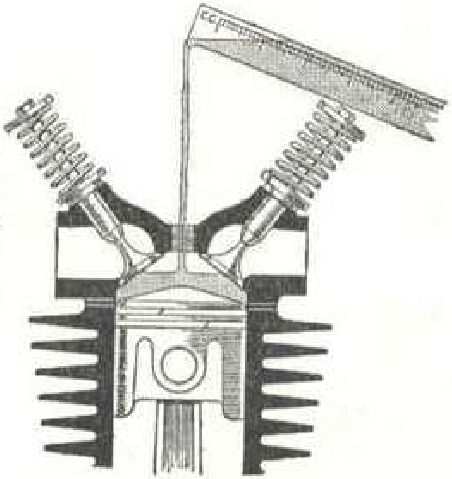
Right from the start it is best to keep a record, rather than to trust to memory, of work done and the results which have been obtained, and, in order to know where you have started, commence operations by noting carefully the ignition timing, valve timing and compression ratio.

The latter is determined by setting the piston at t.d.c. with the valves closed, then filling the cylinder head from the burette until the liquid just reaches the lowest threads of the plug hole, the engine being positioned so that the plug hole is vertical. Then rock the mainshaft very slightly until the liquid reaches its maximum height and top-up,

if necessary, until it reaches half-way up the hole. Now read off on the burette the amount of liquid poured into the head. Assuming this to be 80 c.c. and the correct volume of the engine to be 498 c.c., the C.R. would be

$$\frac{498+80}{80}=\frac{578}{80}=7.22 \text{ to } 1$$

and so on for any other capacity (*see Chapter IV*). The level of liquid in the hole should be watched closely for a while, to verify that no leakage is taking place past the rings or valves. If there is a leakage, then the real volume



The method of measuring combustion-chamber volume with a burette and liquid—either paraffin and thin oil or water is used.

of the head will be somewhat less than the apparent volume but, unless the engine is in a very bad state indeed, the inaccuracy will rarely be bad enough to matter at this stage of the proceedings. If you really want to do the job accurately, then lift off the head and carefully apply grease around the valve heads and between the piston crown and barrel to form a temporary seal.

As to the liquid used, water is the best when there is any doubt about the condition of the valves or rings, because it is less likely to leak past these points than paraffin or light oil, particularly if a drop or two of engine oil is first squirted up the inlet and exhaust ports. For checking the ratio of a newly assembled engine, a mixture of paraffin and thin oil is preferable, as it does not leave a legacy of rust behind it.

Although burettes (which can be obtained from most wholesale chemists or scientific instrument suppliers) are mostly graduated in c.c.s, there is of course no absolute need to use either that instrument or that scale. The head volume could just as easily be measured in fluid ounces or cubic inches, provided that the same measure is used for the cylinder capacity. In working out the ratio remember that 35.2 fluid ounces equals one litre (1,000 c.c.) and, using this conversion factor, the volume of any cylinder can easily be determined.

If a timing disc is fitted to the mainshaft before measuring the clearance volume and a wire pointer is fixed at some convenient location, the pointer can be bent to fix accurately zero on the disc when the piston is absolutely at t.d.c., as indicated by the liquid in the plug hole having reached maximum height.

The ignition and valve timings can now be checked, and to avoid inconsistencies it is best to adopt a standard system of measurement, preferably that recommended by the makers. For measuring the ignition, first set the points to the correct gap (.012 in. in the case of most racing magnetos), insert a 1½-thou. feeler or a piece of tissue paper between the points, and, while applying slight resistance to the armature spindle (by holding the sprocket nut between the fingers or some such method), in order to take up any back-lash in the drive, turn the motor until the feeler is *just* released as the points commence to open, the control being, of course, set to full advance.

On machines fitted with automatic centrifugal advance mechanisms, it is best to lock the mechanism into full advance with a piece of metal or wood jammed between one pair of the fixed and moving stops; checking at full retard and adding the theoretical amount of advance given by the A.T.D. is not really accurate, although it is a good

plan to determine for subsequent reference what the actual advance range is.

For the valves, methods vary: some cams are ground with "quietening curves" at the start and finish of lift, which give a very slow take-up, making the actual "lift" and "close" points difficult to determine. The recommendation in such cases is to measure the timing with the tappets set to a clearance much greater than the normal running clearance. This has the effect of making the opening and closing points more sharply defined, but, of course, it does not give the true figures, a point which must be borne in mind when comparing the timings for engines of different makes.

A table of representative valve timings and clearances is given in Chapter XII from which it will be seen that, on o.h.c. Nortons, the timing is quoted with *less* than the running clearance. As the clearance is adjusted by varying thicknesses of shims inside the valve caps on these models, the recommended checking clearance is obtained simply by using feeler gauges of the appropriate thickness between valve stem and cam follower.

On Vincent engines, it is not possible to measure the clearances with feelers; the method adopted is to set up a dial gauge in contact with the valve stem, and to adjust the tappets so that the valve is lifted one to two thous., making sure, of course, that the follower is on the base circle of the cam and not just at the start of lift. Then turn the engine until the valve lifts a further .005 in., and read off the timing at that point. This method is the one which should be used on any engine to get a really accurate reading of the valve timing.

The crank angle at which each valve reaches full lift, too, should be measured, and also the full height of lift of both valves and the amount each is open at both bottom and top dead centres *with correct running clearances*. You will then have a fairly complete picture of the valve-timing diagram, though to do the job properly, readings should be taken at, say, 2° intervals: the lift curve can then be plotted on squared paper and the effect of any subsequent cam modifications will be shown very clearly by plotting the new lift curves on the same sheet (*see Chapter XIII*).

All this measuring and note-taking may seem very upstage and time-wasting, but if you are to do any good at the game you must know what you are about, and you may as well start off correctly. You may, for instance, do something very silly later on, but will have all the information necessary to check back on earlier work and thus be able to put the motor back to a previously successful state, and then start again from there but along different lines.

The way is now clear to get down to business, the first step, obviously, being to strip the engine completely, keeping watch all the time for places where undue wear has occurred, for wear implies friction, which is one of the main things to be eliminated.

Mechanical Condition

Regarding the general mechanical condition, any parts which show signs of bad wear should be replaced, the main thing being to decide what constitutes "severe" wear. Because a bit of mechanical noise does not matter, a certain amount of slackness here and there can be disregarded; but it is a mistake to think, as some do, that everything should be sloppy. Plain bearings in the cam gear and small-end should have a clearance of between .001 in. and .002 in. per inch of diameter, with a maximum of .003 in. Ball or roller bearings should have practically no slack in them at all, and the race tracks must be examined to see that no pitting has taken place.

When dismantling any engine containing more than one set of rollers, take particular care to avoid mixing up the sets. Take each bearing apart carefully, place each set of rollers in a matchbox, or tin, and make a note on the container stating the location of the contents. Injecting fresh oil into each bearing prior to dismantling will usually

serve to retain the rollers in place long enough to stop them cascading all over the bench.

Sometimes a bearing will be slightly pitted in one small area but is otherwise in good condition. In this case, as a temporary expedient if no spare is available, the bearing can be refitted on its shaft or in its housing so that the damaged area comes into a position where it is not subjected to heavy load. The line of least load is at 9 o'clock on the drive-side main, at 3 o'clock on the timing-side main, and 9 o'clock on the crankpin—in each case looking at the crankcase from the drive side.

One point which needs to be watched closely is big-end lubrication, and another is whether the rod or cage shows any tendency to work over to one side and bear heavily against the flywheel. The latter state is a sign that roller-cage slots are not absolutely parallel with the axis. If all the slots have been incorrectly machined at the same small angle the rollers will clearly tend to screw their way to one side, though if they are out of parallel in random directions the errors may all cancel out. Feeding the oil into the bearing at one side has been known to cause trouble, but the scheme is used successfully on Velocettes from 1957 onwards.

If the compression is already fairly high, say, up to 7 to 1 or so, it is probable that enough metal can be turned off the cylinder to raise the C.R. sufficiently for petrol-benzole to be used while retaining the original piston, if this is in good condition and shows no signs of cracking in the gudgeon bosses. If "dope" is to be used, the old piston can be put quietly on the shelf, for a new high-domed pattern will be required in order to obtain the desirable 14 to 1, or so, ratio.

The condition of barrel and rings is especially important, for it is no use sucking in a lot of gas just to let it blow past the piston. If an area of wear exists at the top end of the barrel, the rings will lose contact with the walls at very high speeds, allowing pressure to escape.

Unfortunately, few cylinders will stand being rebored to the full standard oversize without bringing the engine outside the class cubic capacity limit, and you may have to look into the question of obtaining a new barrel or having the old one re-lined. The latter technique is quite a good idea for the arduous conditions of dirt-track or sand racing, because the liner can first be bored out .005 in. under-size, and, later, honed out to standard diameter: after that a new liner can be fitted and thus the expense of new barrels is avoided. If the wear is only to the extent of two or three thous. the barrel can be lapped out by hand.

The condition of the valves, guides and seats is important. Inlet valve stems should have between .002 in. and .004 in. clearance, but exhaust valves, if made of K.E. 965 or Jessop's G2 steel, need more, .002 per $\frac{1}{8}$ in. of stem diameter being a good figure to work to. If the clearance is less than this, the valve may bind at high temperatures; if more, it cannot get rid of heat properly, owing to insufficient contact with the guide.

This matter of heat dissipation is vital, and if the existing guide is of cast iron (which is a comparatively poor conductor of heat) it can, with advantage, be replaced by one of aluminium-bronze or phosphor-bronze, though the latter has been known to give trouble under extreme heat conditions through breaking away at the hot end near the port.

An alloy composed of copper with 1 per cent. chromium added, and known as chromium-copper, is even better for heat transference and its rate of wear is low; it is, however, difficult to machine and particularly hard to ream. It must be remembered that none of these non-ferrous materials will function satisfactorily without lubrication; cast-iron will work dry, but even it will benefit from the presence of a little oil. Cast iron is, however, quite satisfactory for the inlet valve guide, as heat conditions here are much less severe, and iron material can with advantage be used for exhaust guides in place of aluminium bronze in engines with aluminium heads using methanol, Shell A or 811, or similar fuels and a castor-base oil. This type of oil is soluble

in methanol and, as the mixture in such engines is often excessively rich, the guide lubrication suffers and wear is rapid.

Valve and Seat Modifications

If the exhaust valve has worn to a very thin edge or shows signs of being badly burnt under the head, it should be discarded in favour of a new one.

The silchrome or cobalt-chrome valves frequently fitted to sports engines are very satisfactory in ordinary road use for both inlet and exhaust, but where racing is concerned it is better to play safe and go in for an austenitic steel, i.e., K.E. 965 or Jessop's G2, for the exhaust valve, although the standard steel should be satisfactory for the inlet.

Apart from the faint chance that a crack may have occurred between the exhaust valve seat and plug hole (a defect which is sometimes difficult to detect), an old cylinder head which has seen much service usually shows a good deal of valve-seat wear—particularly on the inlet side—which causes the valves to be slightly masked and reduces their effective lift. If, as is frequently the case in sports engines, the valves are of equal size, the best way to cure this trouble will be to obtain a valve about $\frac{1}{8}$ in. larger in diameter, open out the inlet port to suit and recut the seat.

This procedure may not be possible in some engines where the valve is already of the maximum permissible diameter, as dictated by the position of the plug hole or the liability of fouling the edge of the exhaust valve when both valves are partially lifted. In such a case the masking must be removed by cutting it away with a tool similar to a valve seat cutter but with a much flatter angle; if the valve seat angle is 45°, it is often possible to utilize a cutter intended for a larger valve with a 30° seat. Some welding firms make a speciality of building-up worn seats by welding, and in some cases also will re-machine the head, though not of course to racing standards; both iron and aluminium-bronze heads can be reclaimed by this method, but there is a grave danger of melting the aluminium jacket or at least destroying the close adherence between the two metals in the case of bi-metal heads which have aluminium fins cast round a bronze "skull".

Boring out the seats and fitting inserts is an attractive method of head reclamation, since there is no risk of cracking or spoiling the head through excessive heat, and also the inserts can be made of material which is less prone to seat-sinkage than the original; cast aluminium-bronze is bad in this respect whereas wrought aluminium-bronze in the form of tube or extruded bar is good. Seat-rings made in plain cast-iron or austenitic cast-iron which is high-expansion, high-wear-resisting material, can be obtained from various firms—" Brico " and Wellworthy to mention only two—in standard sizes which may, however, need to be reduced in outsize diameter to be usable in some heads. Seat rings can also be purchased as spares from firms who supply aluminium heads with inserted seats as standard equipment. The nickel-iron alloy known as "Ni-resist" is of the austenitic type and is excellent raw material from which to make seat-rings.

When fitting seat-rings, the limiting factor is usually the thickness of metal which will be left between the recesses and the plug-hole and at the top of the sphere between the two recesses; if the thickness at the surface of the sphere is less than $\frac{1}{8}$ in. there is a risk that a crack will occur in service, particularly if there is much difference between the thermal coefficients of expansion of the metals of the seats and the head (p. 282). The most likely place for such a crack to start is between the plug-hole and exhaust seat, and it may be possible to improve the situation by using a seat-ring to suit an exhaust valve $\frac{1}{16}$ in. smaller; admittedly this will leave a small ridge in the port but this can be subsequently blended out and in any case the exact shape of the exhaust port is not of great importance.

Seat-rings of the same coefficient of expansion as the head do not need to be fitted very tightly : $1\frac{1}{2}$ to 2 thous. per inch of diameter will retain them indefinitely, provided that

the mating surfaces are very accurate and smooth and the bottom of each recess and ring are square to the axis. For aluminium-bronze heads, use either Monel-metal or K-Monel (obtainable from Henry Wiggin and Company, Ltd., Birmingham) or wrought aluminium bronze, preferably in the form of tube.

Rings should always be shrunk in, never pressed in cold, as with the latter method there is always a risk of scoring the surfaces. With .002 in. per inch of diameter interference a ring should just drop into a cast-iron head at 200° C. temperature difference, but, to make sure, 250° C. is advisable. This can be obtained either by just heating the head to 250° C. or by heating it to 200° C. and freezing the ring in dry ice (frozen carbon dioxide) 200° C. (400° F.) head temperature without freezing is sufficient for a bronze head; this temperature can be obtained in a domestic gas or electric stove. To ensure quick and accurate fitting a mandrel (or mandrels) should first be made up from steel bar, with a pilot which fits the guide bore and a register on which the ring fits with a little more slack than the interference of the ring in the recess (otherwise the mandrel may be gripped by the ring). In use, a light coating of grease on the register will retain the ring whilst the mandrel is pushed quickly home and pressed down for some seconds until the ring is gripped by the head.

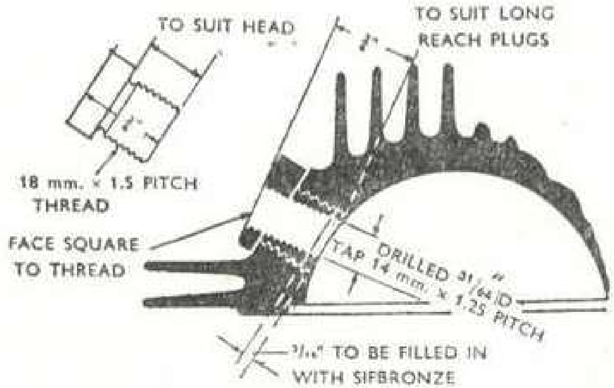
Aluminium-alloy heads with inserted seats have been used on racing engines for many years and are a feature of all post-war production racers. Various methods of fitting the rings have been tried, but the plain parallel form just described is much the simplest to make and to replace and never comes loose if properly fitted in the first place. Aluminium, besides having a high coefficient of expansion loses a large proportion of its strength at high temperatures, though some alloys such as Y-alloy and R.R. 53b are much better than others in this respect and are in consequence much used. Two conflicting factors accentuate the problem; if the initial fit is not tight enough the head will expand away from the ring and it will loosen; but if the fit is too tight, under certain circumstances the head metal may stretch or else the ring may be compressed so tightly that it collapses and in either event the ring will loosen. The most satisfactory seat materials seem to be austenitic cast iron for the inlet and wrought aluminium-bronze for the exhaust, and with these materials a cold interference of .002 to .003 in. per inch of diameter gives perfectly satisfactory operation in Y-alloy or R.R. 53b heads. Austenitic cast iron work-hardens very rapidly and any machining operations must be done with a perceptible depth of cut at a low speed; if the tool is allowed to skid on the work it will be very difficult to re-start the cut. For that reason, ordinary valve-seating cutters operated by hand are useless; the only satisfactory way to form the seats true to the valve axis in the first place is by means of a seat-grinder of the Black and Decker type.

Inserted seats which are worn but still tight cannot easily be removed by heating the head, as the temperature required to get the seat loose enough to drop out is so high that the heat-treatment of the head metal may be affected. The best method is to turn out the old seat very carefully to avoid damaging the recess, or else to run a flat-ended drill down through the ring at two or three places and break the ring into pieces. This operation must, however, be done very carefully to avoid making any cuts in the wall of the recess, or in the bottom face. Local damage of this sort is almost bound to lead to gas leakage and severe damage to the head.

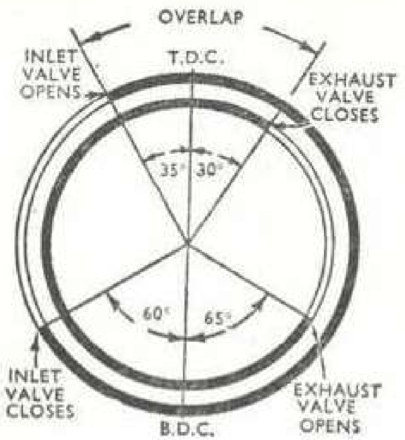
Sometimes a seat ring will be found to be loose, or there will be some evidence that it has been loose when the head has been at running temperature. If the recess and ring are still otherwise in good condition, a satisfactory repair can be made by copper-plating the ring and finishing the deposited metal by turning or filing in a lathe to a size which will give the correct interference fit. All but the outside diameter of the ring should be "stopped off" with wax before plating to eliminate any possibility of plated metal flaking off under the action of heat inside the head.

Should the plug hole be tapped to take the old-fashioned 18 mm. size, which is now almost unobtainable in "hard" grades, it can be fitted with a 14 mm. adaptor sleeve, usually obtainable from one of the accessory houses. Perhaps a better plan is to screw in a solid steel plug, lying $\frac{3}{16}$ in. or so below the inner surface of the head, and to get this recess filled in by an acetylene welder with Sifbronze. This can be done at a dull-red heat, and will not harm the head if it is made of iron. A new plug hole can then be drilled slightly off-centre and at a different angle to the original, so as to get the plug points as near to the centre of the head as possible. The diagram gives the machining dimensions.

The remaining components still to be considered are the cams and rockers. Supposing that the valve-timing figures have worked out like this: Inlet opens 35° before t.d.c., closes 60° after b.d.c. Exhaust opens 65° before b.d.c., closes 30° after t.d.c.; the cams are of such a type as to be worth retaining—at least for the time being—unless they are badly worn or exhibit wavy grooves on the flanks. But if the timing is slower (i.e., of shorter duration) than that, and particularly if the overlap at t.d.c. is less, then a new set of cams should be obtained.



An old-type cylinder head fitted with an 18 mm. sparking plug can be fitted with this special adaptor to carry a 14 mm. plug. The adaptor is drilled and tapped at an angle to bring the plug points into a more favourable position.



If your cams give a timing approximate to this diagram and with not less than the overlap shown, they can be retained temporarily and the performance improved by other means.

The lift must also be considered and should be slightly greater than one-quarter of the port diameter.*

Many makers will supply racing cams which can be fitted directly in place of touring or sports cams, and it is often possible to obtain higher lifts by utilizing cams made for an engine of similar make but of larger capacity.

The valve-timing diagram is determined not only by the cam contours but also by that of the cam-follower feet, and wear of their surfaces will affect the timing quite considerably. The grooves which eventually appear in cam followers with non-rotating solid feet have the effect of opening the valve earlier and closing it later; therefore, if considerable

*The quoted factor of one-quarter the diameter depends on the fact that the area of valve-opening measured round the circumference of the seat is equal to the area of the port at that lift, and hence, in theory, the gas-velocity is the same. However, as the valve is only at full lift momentarily, a ratio larger than one-quarter will give better total flow if the valve is on the small side, as it may be in engines of small bore-stroke ratio. On the other hand, if all other things are equal, the inertia loads and thus the spring-strength required varies in direct proportion to the lift and therefore for reasons of mechanical strength it may be necessary to keep the lift below one-quarter the diameter if the heads are very large.

wear of this nature is discovered, it is best to recheck the timing with new rockers, or with the old ones reground to the correct contour.

If the wear is more than 10 or 12 thous. in depth the rockers will, in any event, need to be reground and case-hardened anew unless new ones can be obtained, and this is obviously a job best done by the makers. In this connection it is sometimes found that an engine which has been going very well despite severely worn rockers, falls off in performance after new rockers have been installed. This is a clear indication that cams giving longer valve openings are required for maximum power at high revs.

The foregoing more or less covers the preliminary survey of the new parts which may be required and the work to be done. The methods of doing that work form the subject of subsequent chapters.

CHAPTER II

CARBURATION AND INDUCTION FACTORS

AT this juncture it is worth noting that a lot of fun can be obtained by "hotting-up" a side-valve engine; there have been quite a number of 90 m.p.h. s.v. "500 s" in existence at one time or another, and readers with long memories will recollect the astonishing little 250 c.c. J.A.P. engines which were reputed to be able to rev. at 8,000 r.p.m. The s.v. unit is usually much lighter than its o.h.v. counterpart, and can be screamed up to maximum revs. without danger of the valves hitting the piston or fouling each other.

Irrespective of the location of the valves or their method of operation, the prime necessity is to have the inlet passage as unobstructed as possible. An easy exit for the exhaust is also highly desirable, although this is as much for keeping this valve cool as for aiding power production.

There are three main factors to be considered in the inlet tract: (a) the size of the valve; (b) the shape and length of the port; and (c) the size of the carburetter. To some extent these are allied to each other. For example, it is no use using a very large-bore carburetter if the rest of the induction system is inadequate; such an instrument will give little or no extra power and will certainly spoil the acceleration. Improving the breathing capacity of the valve and port without increasing the choke size will, on the other hand, improve the power a little at high revs. without affecting the low speed end much; but, of course, the ideal arrangement is to make the whole induction system match, thus carrying out a principle once aptly described as "proportional tuning".

A 500 c.c. engine with a conventional cast-iron o.h.v. head will usually have a carburetter of $1\frac{1}{8}$-in. or $1\frac{1}{16}$-in. bore;

the latter is about the smallest size which will give anything like good results, and as progress is made, larger sizes can be tried. At the commencement it is unwise to use too big a choke, and the following ranges are recommended: 500 c.c. single o.h.v., $1\frac{1}{8}$ in.–$1\frac{3}{16}$ in.; 500 c.c. twin o.h.v., $\frac{15}{16}$ in.–1 in.; 350 c.c. o.h.v., 500 c.c. s.v., 1 in.–$1\frac{1}{16}$ in.; 250 c.c. o.h.v., 350 c.c. s.v., $\frac{13}{16}$ in.–$\frac{7}{8}$ in.; 250 c.c. s.v., $\frac{25}{32}$ in.

As to valve size, it is scarcely possible to lay down any hard and fast rule as so many factors are involved, but the following table gives an idea of the throat sizes which are advisable: 500 c.c., $1\frac{5}{8}$ in.; 350 c.c., $1\frac{1}{2}$ in.; 250 c.c., $1\frac{3}{16}$ in.

The capacities refer to the size of individual cylinders and *not* to the engine as a whole. At best, however, these figures can only be a rough guide, because other factors besides the cylinder capacity enter into the matter; the type of performance (i.e. the shape of the power curve) required, the valve timing, compression ratio, the design of carburetter and the nature of the exhaust system all exercise an effect. Generally speaking, the faster the engine is capable of revolving, judged from a purely mechanical sense, the larger the choke which can eventually be used, though a good deal of modification may be required before the ultimate size is attained; this line of development almost inevitably leads to a type of engine which, whilst it develops very high power at high r.p.m., is virtually useless below a certain minimum speed and in consequence has to be used in conjunction with very carefully chosen gear ratios and driven with great skill in order to keep it always pulling above its minimum speed. Such a power unit will show up at its best on a fast circuit, but may be equalled or even outclassed by an engine with less maximum power but with greater torque or pulling power at low or medium speeds, on a short course with many corners or in a vehicle such as a racing car where a very close set of gear ratios cannot be used. The Italians are notable exponents of large-bore induction systems; the

250 Guzzi has a choke of 32 mm. diameter, or just over $1\frac{1}{4}$ in. and, therefore, $\frac{1}{16}$ in. larger than that normally fitted to post-war 500 c.c. " Manx " Nortons, and this bore is continued right through the long induction pipe and inlet port up to the throat of the valve. The 500 c.c. twin of the same make, which is virtually two 250 singles, has even been used with 35 mm. chokes and the same size is used on the four-cylinder Gilera, in which one carburetter feeds a pair of cylinders of which each is of course only 125 c.c. At that time the fastest English "works" machines had not gone to those extremes; the twin A.J.S. had one $1\frac{1}{8}$ in. carburetter feeding each of its 250 c.c. cylinders, whilst the "works" 500 c.c. Nortons and Velocettes used $1\frac{5}{16}$ in. chokes. The foregoing data refer to engines built *circa* 1950–52. Since then engine speeds have gone higher particularly in single-cylinder types, and chokes of $1\frac{1}{2}$ in. diam. have been used on quite a number of 500 c.c. engines, running in excess of 7,000 r.p.m.

Even these sizes, though smaller in relation to the cylinder capacity than the Italian examples, furnish a gas speed considerably lower than those stated by some authorities to be the theoretical optimum, this somewhat puzzling paradox being probably caused by the aforesaid optimum having been determined in manifold-type multi-cylinder engines in which the fuel has to be vaporized and maintained in that state before reaching the cylinders to obtain equal distribution, whereas in motorcycle engines the bulk of the fuel is not actually vaporized until after it enters the hot cylinder; this process takes full advantage of the latent heat of evaporation of the fuel to cool the charge within the cylinder and so results in a greater weight of charge being induced. (For later information see page 246.)

Work on the Inlet Port

If your engine conforms reasonably well to the standards so far described and is in good condition it may be well to leave things as they are for the time being and experiment

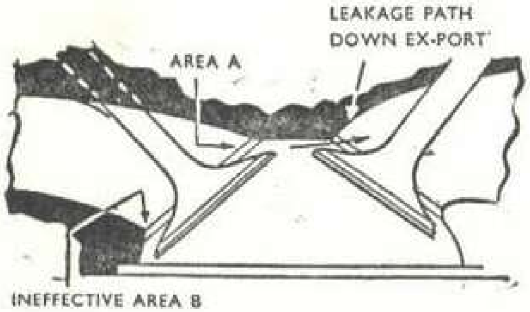
with larger valves or carburetters later. In the meantime several jobs can be done to improve the breathing ability. As the valve is operating only partially open for much of the time, anything which will ease the passage of gas past the seat at low lifts is highly beneficial, and one step towards this is to reduce the seat width from the usual $\frac{3}{32}$ in. or so down to a mere $\frac{1}{32}$ in. by carefully radiusing the corners of the valve at the seat and the top edge, and performing a similar operation on the port.

How to reduce the width of the seatings on valve face and inlet port.

If the seat is very wide, as it frequently is when interchangeable valves are used, it is also possible to open out the port by perhaps $\frac{1}{16}$ in., which provides a useful increase in total area. The correct method is to use a cutter, similar to a valve-seat cutter, but with an included angle of 45°, but the job can be done by careful work with a file and scraper.

For rounding off the valve edges, hold the stem in the chuck of a lathe or drill, remove the bulk of the metal with a smooth file and finish off with emery cloth. Should the valve-seat in the head be very badly pocketed and you decide to fit a larger valve, it is often possible to utilize one from a larger engine of the same make, as it is a frequent practice to employ the same size of stem throughout the range; alternatively, racing editions of sports engines are often fitted with larger valves, and one of these could be employed.

The shape of the inlet valve under the head is of more importance than is often realized and depends upon the shape of port. It used to be the fashion to have a comparatively flat, straight port—a shape which directs the gas flow past the valve stem into the region nearest the exhaust valve—and in this case the valve head must be fairly flat underneath and joined to the stem with a small radius, so as to allow plenty of room for the gas to flow past into the operative region. This port shape—see sketch below—is by no means the best, partly because the area (B) is relatively ineffective owing to the acute change in direction which the gas is compelled to take, and partly because much of the charge coming in at (A) is liable to rush straight across and out through the still-open exhaust valve, particularly if a highly extractive exhaust system is in use.

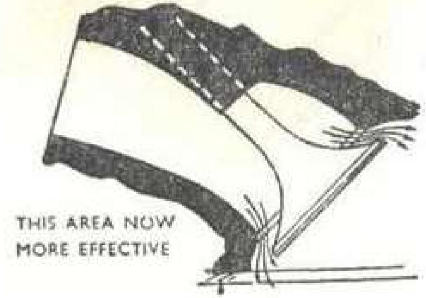


Limitations inherent in the old-type "flat" inlet port are depicted in this sketch. Note how the fresh charge tends to flow into the exhaust port.

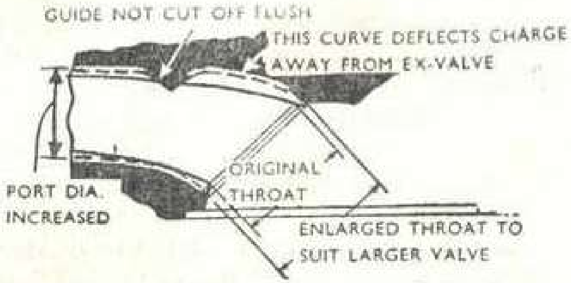
The shape shown in the second sketch—as used in modern engines—may not look so well to the casual eye, but is, in fact, better because almost the whole of the seat circumference is brought into effective use. This design of port works best in conjunction with a semi-tulip valve the shape of which has the effect of steering the gas towards the lower side of the seating. Some of the fresh charge is thus directed away from the exhaust valve, less will escape, and it will pick up less heat from that hot-headed component. Conversely, the latter is inclined to run hotter, a point which will have to be watched in petrol-benzole motors but is of less consequence with alcohol fuel.

By going up in valve size, it is possible to improve the shape of a flat port (as indicated in the third sketch by dotted

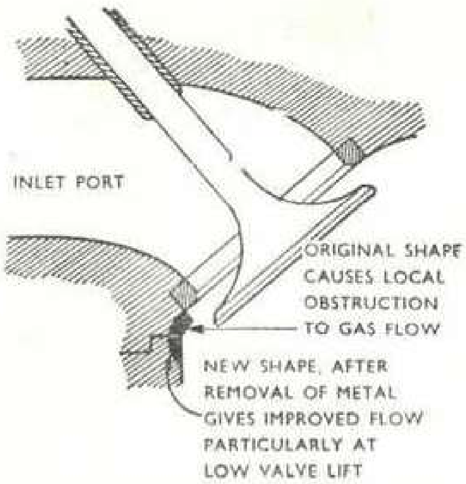
The modern down-draught port used in conjunction with a tulip-shaped inlet valve overcomes the defects of the flat-port design.



lines) in addition to increasing the valve area, but it may be better to leave this operation until a later stage, thus being able to pull something out of the bag which others may not suspect. In any case, the valve guide must be so treated as to offer the minimum obstruction, the usual method being to cut it off flush with the roof of the port. This sometimes reduces the guide length rather too drastically; equally good results can be obtained by fairing the guide off to a knife edge on the entering side and filing out the sharp corner between the guide and port, reducing the guide wall thickness down to $\frac{1}{32}$ in. or so at the sides. When a larger inlet valve is fitted, or sometimes even in standard heads, a certain amount of masking or interruption to smooth gas flow may exist in the area around the lower edge of the valve which may prevent this area being fully effective.



How gas flow can be improved by enlarging the inlet tract to suit a bigger valve. The dotted lines indicate where metal should be removed.



Gas flow may be improved by re-shaping the mating edges of the head and cylinder barrel adjacent to the inlet valve.

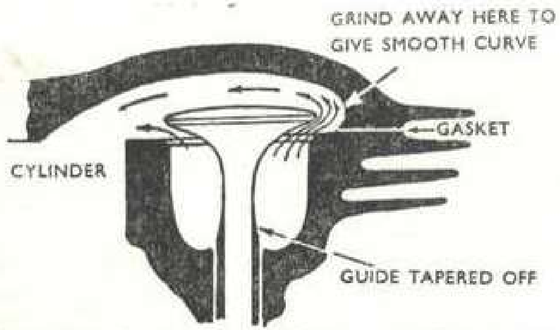
It is often possible to cut away some metal from the head and in some instances the mating edge of the cylinder barrel also, and generally shape and smooth up the surfaces so that gas coming past the seat radially to the valve is gently guided into the cylinder instead of impinging against an irregular surface, or one which is too close to the valve to permit easy entry. For these port and head modifications, rotary files or emery wheels driven by a flexible shaft from an electric motor are invaluable, but the same effect can be produced with "rifflers" or bent files. Ordinary round files can be bent to shape after heating to redness, and re-hardened by plunging into water; after this treatment the steel is very hard and brittle, but if not knocked about, the teeth remain keen for a long time.

From the carburetter flange face or stub up to the valve guide boss there should be a gradual taper in the port, entirely free from ridges or hollows, and after the whole port has been brought to a satisfactory shape it must be polished with rough and fine emery cloth to a dead smooth finish. For rough port-polishing, one of the handiest tools is a piece of steel rod slotted across a diameter with a hacksaw for about $\frac{5}{8}$ in. in from the end. This rod is held in the

flexible shaft chuck, and a strip of emery cloth or, better still, woven abrasive tape is slipped into the slot and wrapped round the rod up to about $\frac{7}{8}$ in. diameter. The wound-up tape must be inserted into the port before the power is turned on after which the flying end of the tape rough-polishes the port very rapidly. For final polishing, use felt bobs moistened with ordinary metal polish, or cotton material wound round the slotted bar. There is some controversy regarding the value of an absolutely mirror-like surface, but there is no doubt that a finish of this nature cannot do any harm, and is at least a source of envy to chance beholders; however, *shape* is of greater importance than mere polish.

Side-valve Inlet Ports

The treatment described for o.h.v. inlet ports applies equally well to side-valve types. Except in those cases where the port has been purposely elevated in order to obtain clearance over a rear-mounted electrical installation, most s.v. ports have reasonably easy curves, and, so far as the approach of the gas to the underside of the valve is concerned, may be as good or better than the average o.h.v. type. As to valve shape, the same rules apply: a flattish



In a side-valve unit the gas flow can usually be improved considerably by grinding away metal in the cylinder head at the point indicated. Great care is required in this operation. Do not overdo it.

port requires a flattish underhead profile, but a combination of a swept port with a semi-tulip shape of valve gives the best results.

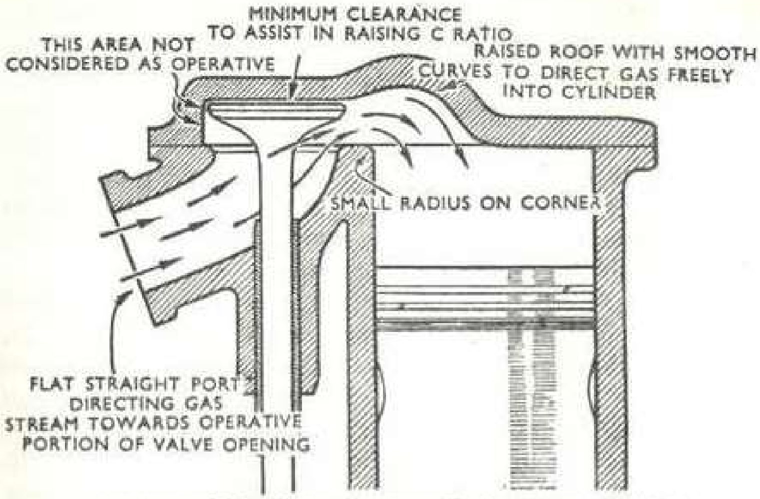
The valve guide can be shortened a little or even cut off flush with the port without detriment, but the amount to which this is done depends largely on the port shape. It may be better to leave the guide fairly long and streamline it by tapering off to a very thin section at the upper end. Certain V-twin engines were built with the inlet ports lying horizontally in order to employ a plain T-shaped induction manifold, but an incidental advantage gained is the reduction in the angle of bend in the port itself. The front cylinder of such a motor mounted on a single-cylinder crankcase and equipped with a carburetter arranged to work at 25° upward inclination offers interesting possibilities.

It is obvious that after passing the valve the charge has to undergo another change in direction before it can reach the cylinder, and this, plus the fact that a third—possibly one-half—of the valve opening is partially masked by the valve chest wall is bound to impair the breathing. The change in direction cannot be avoided, but the effect of the masking can be reduced by clearing away the chest wall with a port-grinder and eliminating as far as possible all sharp angles.

Further Aids to "Breathing"

In detachable-head engines, care must be taken to see that the inner edge of the gasket merges properly with the interior of the head. It may be advisable to cut a new gasket, from soft sheet aluminium or copper, which fits the head studs accurately, and to trim it up to the correct internal contour after bolting to the head.

As a further aid to "breathing", reduce the valve head diameter until the seat width is only $\frac{1}{16}$ in. and round off the edges of seat and valve as previously described. On account of the proximity of the chest wall, it is rarely of much use to attempt to use a larger valve, although it may be possible to do so if there is ample room available around



A section of the inlet tract of a side-valve engine modified for special high-performance valve timing.

the head, or alternatively if there is sufficient metal in the casting to allow for the necessary grinding-away.

Most side-valve engines have "slow" timings and low valve lifts, with the intention of giving good low-speed performance and, in keeping with this idea, are fitted with small-bore carburetters. After due attention to the induction tract and timing (a point still to be dealt with), a larger carburetter can be used, although it is not possible to go up to quite the same size as an o.h.v. unit will take. As a commencement, use the next size lower to that recommended for an o.h.v. engine in the capacity choke scale on page 17. Naturally, the inlet port must be increased in diameter to match up with the larger-bore carburetter. One of the difficulties associated with the obtaining of good breathing in a side-valve engine on the lines just described is that, if ample gas-flow areas are provided all round the valve head, the compression ratio may become much too low. A possible line of development worth pursuing only if valve-timing modifications can also be carried out would be to regard that portion of the valve opening furthest from the cylinder as being relatively useless anyway, and to re-design the combustion chamber with the space immediately above the inlet valve machined parallel to a radius only slightly greater than the largest inlet valve that can be worked in. The depth of this pocket should be about $\frac{1}{32}$ in. greater than the maximum valve lift, and the cam contour modified to slam the valve up to full lift as rapidly as possible and hold it at full lift for a considerable period.

The port likely to be most effective in this instance would be of the flattish type, as shown in the diagram on page 25, directing the gas stream mainly in the direction of the effective portion of the valve opening. To assist the gas in changing direction to enter the cylinder, the head may be hollowed out slightly and the top edge of the cylinder bore radiused off, though not so far down the bore that the top ring is deprived of the protection of the upper ring land.

The Exhaust Port

The exhaust valve and port still remain to be dealt with, and it should be appreciated that the gas-flow conditions are quite different from those existing in the inlet. In the latter, the gas approaches from the back of the valve, and the pressure-difference between port and cylinder is (or should be) of the order of 5 or 6 lb. per sq. in. only as a *maximum* value, and must be much less than this at the finish of the induction stroke.

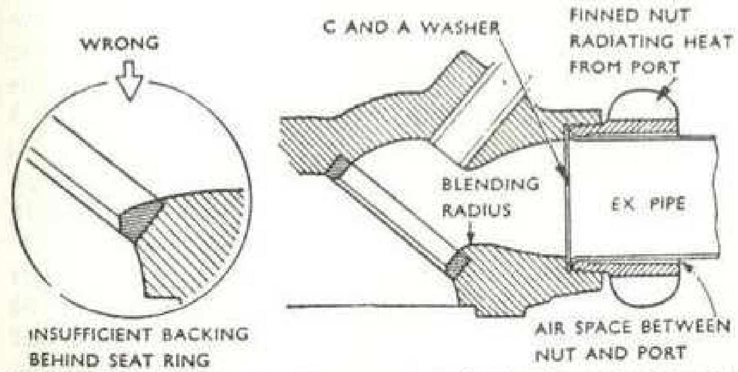
With the exhaust valve, however, the gas at the moment of opening is at high pressure—70 or 80 lb. per sq. in.—and is waiting all round the periphery of the head, ready to escape the moment the valve begins to lift. Thus, whatever the port shape, the whole of the circumference of the valve is operative, and 50% of the gas can escape straight down the port; only half of the gas is obstructed by the valve stem and guide, consequently these can be permitted to occupy quite a lot of space without detriment.

It is a mistake to carve away the guide and boss com-

pletely in order to get the maximum port area; by so doing more of the valve will be exposed to the outgoing gas and the stem will be robbed of much valuable contact area through which most of the heat picked up has to be conducted.

On the other hand, if the port is so shaped that a large proportion of the extremely hot gas is caused to impinge on the stem without a free path of escape, there is a grave risk of valve-failure, through overheating at the junction of the stem and head. Cutting away the side walls so that the port is considerably wider than the diameter at the throat reduces this effect very considerably, and is in fact the only course open in some engines where for the sake of compactness the port has purposely been kept flat.

Before doing much modification to port shape, the thickness of metal between the port and neighbouring pockets such as rocker-boxes, spring-boxes or the recess sometimes provided to accommodate coil valve springs, should be explored, preferably with indicating callipers, to ensure that there is ample metal present; nothing is more annoying (or expensive) than to find the port breaking through into a place where it should not do so, particularly just as the job is nearing completion. Also, on inserted-seat heads,



(Right) The ideal exhaust-port arrangement showing, in comparison with the inset sketch, the seat ring well supported and the valve guide protected from feed-back of heat from the exhaust pipe area.

never remove metal in any but the smallest amount from the seat ring itself, and also leave plenty of metal under the ring where it is closest to the port, as there is a possibility that explosion pressure may actually force the seat-ring back due to lack of support if there is insufficient port-metal backing up the ring at this point; the valve will then cease to be gas-tight and may shortly fail through overheating.

It is not always appreciated that one of the most prolific causes of trouble in I.-C. engines is the large amount of waste heat absorbed by the exhaust-port walls and fed back into the head. Once the exhaust valve has opened, any heat left in the gas ceases to be of any value and becomes a waste product, and the less it is re-absorbed into the engine the better. Consequently, exhaust ports should be as short as possible, and there should be a path for heat conduction between the port and the exhaust pipe nut, but not between nut and pipe. These conditions are all assisted by using a male-threaded nut, with a perceptible clearance between the bore of the nut and the pipe; at one time long ports furnished with many fins to cool the port down were fashionable, but it is far better to prevent the heat being absorbed rather than to dissipate it afterwards: an examination of almost every engine which has been a successful performer on petrol or petrol-benzole will show that they all possess this feature of short, direct ports. The point is of less importance on engines designed specifically for alcohol fuel, but that does not impair the essential correctness of the principle.

Double-port heads which were popular some years ago are, in general, not nearly as efficient as those with single ports, and most engines so designed will perform better if one of the ports is blanked off completely and as close up to the valve as possible. The term "double port" is used here as applying to a single valve; if there are two exhaust valves, it is usual for each to have its own port, and in this event the criticism, of course, does not apply.

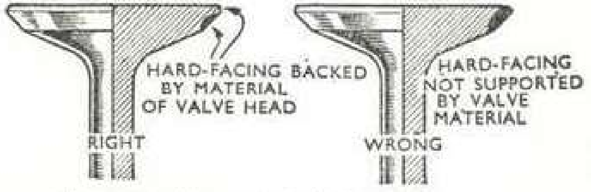
Once an exhaust port has been correctly shaped and

polished, it is best not to clean all carbon out of it, but to allow a thin skin to remain. Carbon is a poor heat conductor and acts as quite an effective insulator, which helps to limit the absorption of waste heat by the port walls.

Python engines (the name under which Rudge power units were supplied to other makers) were sometimes equipped with adaptors forced into the ports to suit $1\frac{3}{4}$-in. pipes (far too big, even for the 500 c.c. models), and these should be removed. This is done by removing the hexagon-headed setscrews which lock the adaptors in place; the latter can then be twisted out with the aid of a large Footprint pipe-wrench, an operation best conducted before detaching the head from the cylinder. The right pipe size is that which just fits in the counterbore in the port from which the adaptors have been removed.

Much of the heat absorbed by the exhaust valve head gets away by conduction through the seat; the latter should, therefore, be reasonably wide—between .07 in. and .09 in., according to valve size. For sprint work, or on engines which will be regularly stripped down after every event, narrower seats can be used with safety, but more frequent valve-clearance adjustment may be necessary. It may pay on heads originally designed with both valves of equal size to reduce the diameter of the exhaust valve, and, therefore, the seat width, in order to fit a larger inlet valve and still maintain enough clearance between the two to prevent mutual interference in the overlap period when both are partially lifted. A slightly convex head is preferable to one which is dead flat or even concave, although it means a slight increase in weight. The edge must not be too thin, otherwise it will get very hot and may warp out of shape at running temperature; as with the inlet valve, the top edge should be slightly rounded off to improve gas-flow and to avoid a local hot area.

If there is any doubt as to the quality or age of the valve, play safe—discard it and get a new one of K.E. 965 or similar austenitic steel, conforming to the British speci-



Correct and incorrect techniques in the process of hard-facing valve seat areas.

fication D.T.D. 49b. Better still, if possible get hold of a sodium-filled valve of the type commonly fitted to aero-engines and used by our leading racing firms during the last two or three years of racing before the war. These valves are larger in the stem than the " solid " ones they replace, and in consequence the guides will need to be renewed or bored out. Valves which have worn thin near the seats but are otherwise sound can be reclaimed by building up the seat with " Stellite " No. 6 grade or " Brightray ". Both these alloys are non-ferrous and have the advantage that they are very resistant to attack by the by-products of Ethyl fluid; in countries where heavily-leaded fuel is obtainable and permissible for racing it is a wise precaution to have the valves " Stellited " before they become damaged by lead attack. There are other materials such as "Cobalite" which can be used for hard-facing valves, but it must be remembered that the deposited material should always be backed up by the parent metal of the valve, as shown in the diagram; it is inadvisable to use hard-facing to increase the diameter of the head or to build up a very badly worn head to its original size. "Nimonic 80a", a non-ferrous nickel alloy, is another very good material for exhaust valves.

Side Exhaust Valves

Side exhaust valves are worse off than the o.h.v. type as regards heat, because they get little cooling from the incoming charge, and the part of the seat next the barrel is bound to run very hot. Also they are usually rather small

in stem diameter, and have cast-iron guides; an improvement in heat dissipation can be effected by increasing the stem diameter by $\frac{1}{16}$ in. and using aluminium-bronze or Barronia-metal guides, but it will be necessary with these metals to ensure a small though positive supply of lubricant to prevent seizure. A flash-coating of hard chrome plate .0005 in–.001 in. thick on the valve stem will also assist in this direction.

Absorption of heat by the head is lessened by contouring the valve chest so that the outgoing gas is allowed to flow away as smoothly as possible; this is achieved by eliminating sharp corners in the valve chest and allowing plenty of room over the head of the valve when it is at full lift.

The scheme previously outlined for deliberately masking part of the inlet valve is definitely not applicable to the exhaust valve, for it is imperative that the whole circumference of this component is brought into action at the commencement of lift in order to liberate the high-pressure gas immediately.

Valve Seat Angles

The majority of valve seats are formed at 45° to the axis, but some makers favour the flatter 30° seat, which is more effective at low lifts because the actual opening between valve and seat is greater by 22% or in the ratio of 86 to 70. This advantage ceases after the valve has lifted by an amount approximately equal to $1\frac{1}{2}$ times the seat width, and is, therefore, of less importance with narrow seats than with wide ones, but with the widths commonly employed the gain is worthwhile because much of the breathing must perforce be accomplished when either valve is only partially lifted. Conversely, the 45° seat has a greater self-centring effect on the valve, and may give better results mechanically in heads with short guides, or those in which distortion may occur at high temperatures. Some engines are made with 30° inlets and 45° exhausts for that reason.

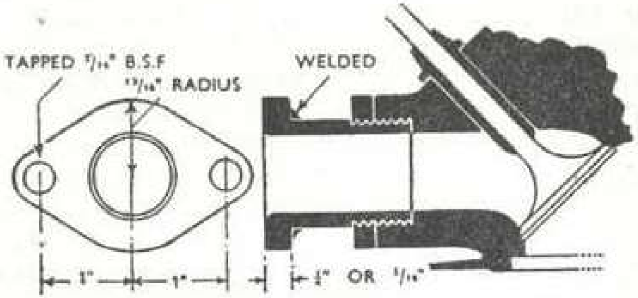
CHAPTER III

ATTENTION TO O.H.V. CYLINDER HEADS

So far we have progressed to the stage where the valve seats and ports have been attended to—at least, so far as the written word is concerned: carrying out the work in a really thorough manner takes a much longer time and far more patience than simply describing it.

Before proceeding further, the carburetter should be fitted temporarily to make sure that it lines up accurately with the port without a "step" anywhere around the circumference. If the mixing chamber pushes on to a stub, or has a spigoted flange, the two bores are almost bound to be concentric, but a "step" will exist if the diameters are unequal or if the port has been filed a trifle out of round during the cleaning-up process.

Once lined up, either of these types will always reassemble correctly, but the standard flange, which has no spigot, is sometimes a rather slack fit on its two studs. If the stud-holes are worn or have been filed out at some time, it pays to ream them out to ⅜ in. and fit bushes on to the studs;



Showing how an old-type induction stub can be modified to accept a modern flange-fitting carburetter. Note how the induction pipe is blended smoothly into the inlet port.

the carburetter and port bores can then be blended into each other and you will be sure that, even if the carburetter has to be refitted hastily at some future date, it will go back correctly and cannot possibly work out of position at any time through vibration.

Sometimes these flanges become bent (usually through the use of a packing washer which is too soft and thick), and this fault must be rectified by truing up the face. For use in later experiments, it is worth while making up some extension blocks to go between the flange faces, in order to vary the length of the induction system ; the easiest way is to machine them from aluminium castings made from a wooden pattern the same shape as the flange and ½-in. finished thickness. Four of these, plus the necessary longer studs, will enable the induction pipe length to be varied in ½-in. steps up to 2 in.

The cylinder heads of early models were fitted with screwed-in stubs to take clip-fitting carburetters, and the engine makers concerned can often supply longer patterns; alternatively, in order to use a flange-fitting carburetter, a steel flange can be welded or brazed to the stub.

It may be advisable to make up a new stub altogether rather than to modify an old one; many induction systems are neither long enough nor large enough for maximum power at very high r.p.m. (*see Chapter XVIII*). New pipes can be made up quite easily from steel tube with flanges brazed or welded on. Another simple method is to use a piece of canvas-reinforced hose attached to the existing induction pipe and to an extension piece by Jubilee clips, though some care has to be taken to avoid a nasty step or change in diameter at each change from pipe to hose, particularly if an inch or so of hose is exposed; in that case, the best thing is to line the hose with a third piece of metallic tube to preserve a constant diameter. This scheme insulates the carburetter from the high-speed vibrations which sometimes cause a mysterious loss of r.p.m. through unsuspected flooding at a certain critical speed and so is referred to again in Chapter XVIII.

Amal carburetters are made in both flange- and clip-fitting forms. The largest bore which can safely be used with the standard 2-in. stud centre-distance is $1\frac{3}{16}$ in., and even then there is only just clearance for the nuts. Clip-fitting carburetters, i.e. those which fit on stubs, do not have this limitation, and when for reasons of space or to eliminate bends in the induction system, the mixing chamber has to be mounted out of the vertical, the clip-fitting type is easier to install because it can simply be swivelled round the stub into the correct position. The angle of the mixing chamber is immaterial, within limits, provided the float level is correct in relation to the jet, though if the inclination is greater than 15° flooding from the pilot jet into the engine will occur when standing if the fuel supply is not turned off.

Standard stub diameters are $1\frac{1}{8}$ in., $1\frac{1}{4}$ in., and $1\frac{3}{8}$ in., according to the bore of the carburetter. Adaptor stubs with any of these diameters can easily be made up to fit on standard flange studs, but it is best to use socket head screws of the Allen or "Unbrako" type with the largest size to avoid having to cut too far into the wall to obtain head clearance. By this method it is also possible to make up a stub of 1.422 in. diameter to utilize the 32 mm. carburetter.

Around 1950, a new standard Amal flange with 65 mm. bolt centres was introduced for very large carburetters and this point must be watched when contemplating a change-over. It is possible to make a double flanged adaptor with 2 in. hole centres in one flange and 65 mm. in the other, but the largest bore you can get without having two local flats adjacent to the Allen screw heads is $1\frac{7}{16}$ in.

Coil and Hair-pin Springs

Hair-pin springs came into use many years ago for racing, because the coil springs in use at that period were not always reliable and became less so as engine speeds rose above 6,000 r.p.m. and valve lifts increased above $\frac{3}{8}$ in.; the combination of these two factors creates a difficult situation for any spring and even hair-pin springs have

undergone considerable design modifications before reaching their present state of reliability. Most sports models and genuine "racers" of 1934–39 vintage were fitted with hair-pin springs with the notable exception of the speedway J.A.P. which has always retained coils, these being, in any case, sufficiently reliable for an engine intended primarily for short-distance events. Since 1945 there has been a considerable change back to coils, largely because of the difficulty of enclosing hair-pins without serious interruption to air-flow through the head fins, particularly on a parallel twin. Coil-spring fracture is mainly caused through "surging" or a state of rapid vibration of the centre coils, and the way to overcome this is by using only a limited number of coils; all modern coil springs are duplex or even triplex with not more than six free coils in the outer. The inner spring often has one or two more coils of lighter gauge wire and it is an advantage for it to be a push fit within the outer one as this provides a certain amount of friction damping and so helps to prevent surging.

Side-valve engines are usually equipped with single springs of fairly light poundage, sometimes of the "floppy" type, with a large number of coils. These should be replaced by others of similar proportions to those just described, which will probably entail replacing the top and bottom spring caps and the split collets. It is also probable that pieces will be required to make up for the reduced length of spring and these should be made with the smallest possible area in contact with the head to minimise transfer of heat to the uppermost coils.

With regard to spring pressure, it is a mistake to think that enormously strong springs are a *sine qua non*; any pressure in excess of that required to prevent valve chatter up to the usable maximum revs.—say, 7,000 for a 500 c.c. unit, up to 9,500 for a 250 c.c. plot—simply wastes a little more power and loads up the valve gear unnecessarily.

The actual strength required at full lift depends upon the cam contour and lift, as well as the revs. and weight of valve gear, so it just is not possible to give any hard-and-fast rule, but, apart from these considerations, it is usual in two-valve heads to provide 100–120 lb. pressure when the valve is seated, and, unless the cam design is very peculiar, 140–180 lb. should suffice at full lift (the greater figures obviously applying to the larger-size engines). Four-valve designs, with their much lighter individual components, require only about 80 lb. seat pressure per valve.

It is possible to make up a rig to test this pressure but makers' genuine replacements can be accepted as giving the correct amount provided that nothing has taken place to alter the installed length of the spring appreciably. Excessive seat wear, or possibly a stretched exhaust valve may account for this length being $\frac{1}{16}$ in. or so longer than it should be and this is sufficient to reduce the pressure by several pounds. For workshops which do a considerable amount of tuning work, it is a wise plan to acquire one of the special spring-testing machines that give an accurate measurement of the poundage given at any particular length. The actual installed poundage can then be determined by first measuring the distance between the top and bottom spring collars and then determining the pressure of the springs at that length on the machine.

Apart from being obviously broken, coil springs often shorten or settle and so lose their effective strength; attempting to restore the situation by packing up the lower seating is sometimes effective, but it is then essential to verify that the springs are not "coil-bound" at full lift; to do this see that the valve can be raised a further $\frac{1}{16}$ in. beyond the maximum point to which the rocker lifts it.

Another expedient is to place the weaker set of springs on the inlet valve, as insufficient strength on this component will at worst only reduce the top r.p.m. a little, but severe mechanical damage may result from lack of spring strength on the exhaust valve.

Hair-pin Springs

Hair-pin springs were originally made with fairly small diameter coils and long legs, but the modern variety has

much larger diameter coils and shorter legs. Further, the design is such that over the working range of the spring, the loop in contact with the valve moves almost in a straight line parallel to the valve stem and there is little or no tendency, as there was with earlier designs, for the stationary legs to move in and out on their abutments. When for reasons of space it has been necessary to reduce the overall width of the pair of springs, as, for instance, on parallel twins, like the twin-cylinder A.J.S. racing machine, the ends of each pair of springs are overlapped, and careful design is necessary to ensure that no interference occurs between the springs at any part of the valve movement. When fitting these overlapping springs, particularly to an engine for which they were not originally designed, care must be taken to see that the end loops do clear each other at all points from closed to full lift.

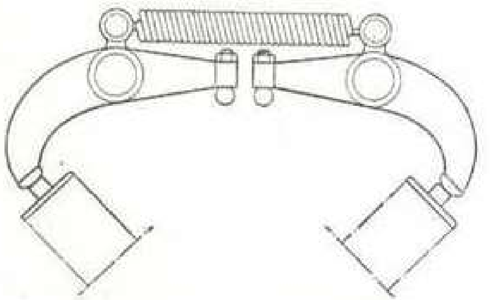
After a while, wear occurs on the straight piece of the loops of hair-pin springs, caused by rubbing against the spring retainer, and replacement is called for if this wear exceeds one quarter of the wire diameter. The force exerted by the springs in each pair should be equal to avoid tilting the valve, therefore it is essential to replace in pairs, not just one at a time. When they do break, hair-pins usually fail just where the top leg commences to bend round into the coil and every care must be taken to see that no cuts or deep scratches are made in the wire at this point when handling such springs.

It is often considered advisable to convert a coil-spring engine to hair-pins and this can be done with a little wangling in some o.h.v. designs. On MOV and MAC Velocettes of the original design with bolted-on rocker boxes, standard KTT springs, and spring collars can be used, but the rocker-box extensions which normally surround the spring cups have to be cut away, leaving the outer ends of the rockers exposed, and the rocker-box standards on the push-rod side must be thinned down to clear the springs which are installed at an angle.

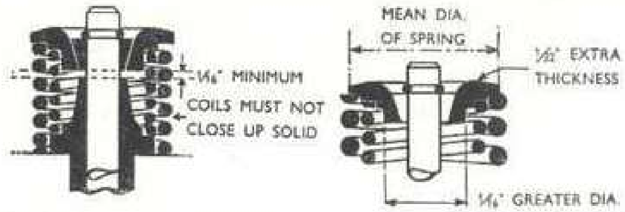
Removal of the coil valve spring enclosure has the effect of depriving the valve guides of lubrication which they normally receive by leakage from the rocker bearings and it is, therefore, desirable, particularly for long-distance work, to drill $\frac{3}{16}$ in. holes through the head into each valve guide to receive oil pipes which can be supplied with lubricant either from a small external tank or from the normal lubrication system with metering jets to limit the supply, which need only be small in quantity.

Auxiliary Return Springs

Some designers prefer to utilize return springs on the rockers or push-rods, thereby either reducing the strength necessary in the valve springs or, alternatively, permitting higher revs for the same spring strength; examples are the hair-pin springs surrounding the rocker bosses on KTT Velocettes and the springs fitted on the lower ends of the push-rods of certain J.A.P. motors. The effect of such springs is to maintain the cam follower in contact with the cam continuously, and excessive pressure leads to local heating and wear of the cam-follower foot unless a roller is employed. A neat way of adding return springs to engines with hollow rockers is to house a coil spring loosely in the bore of the shank with one end fixed to the rocker arm and the other attached to the rocker enclosure in such a way that the spring can be wound up to any desired amount; this scheme has the merit that it is easy to experiment with varying



A method of obtaining increased return-spring action without creating excessive cam-follower loading.



(Left) (see page 42) At maximum valve lift there must be adequate clearance between valve guide and spring cap and between the spring coils. (Right) (see pages 40–1) Note how the outer diameter of the alloy cap is that of the mean diameter of the outer spring.

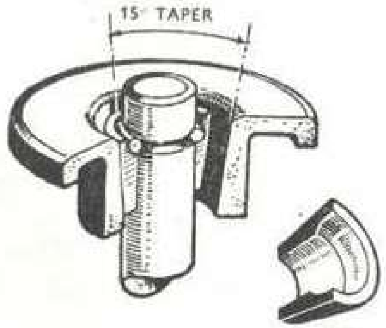
amounts of pressure and thereby to discover whether stronger valve springs are required.

Another method is to attach a tension spring to a short vertical arm on each rocker; this idea is good because when the exhaust valve is closing, the inlet is lifting and, therefore, the tension in the spring is increased just when it is most wanted, and the required effect is gained without excessive cam-follower loading during other parts of the cycle.

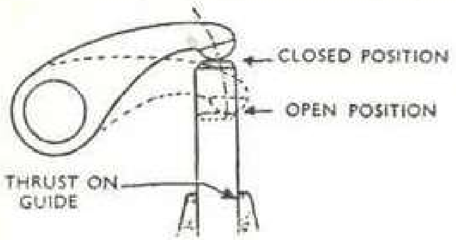
In some designs the exhaust spring settles very quickly because the coils nearest the exhaust port become overheated; a remedy can be effected by insulating the lower spring coils by mica washers, or packing the spring seat up on narrow washers so that there is the minimum of metallic contact and the maximum of air space between the spring seat and the hot port metal. Generally speaking, enclosed coil springs can be kept as cool or even cooler than open springs by the expedient of pouring enough oil over them through jets to carry away the heat.

Before leaving this subject of springs it may be well to dispel a popular misconception as to their function: some think that the cam pushes the valve open against the spring right up to full lift and after that the spring takes charge and performs the closing action. In fact, the cam operates against the spring plus the inertia of the valve for the first portion only of the lift; after a certain point and up to full lift, the spring is doing all the work of slowing the valve down until it is actually stopped by the time full lift is reached. After that, the spring accelerates the valve back again until a certain point is reached on the closing flank of the cam; from that point onwards the valve is being slowed down again by pressure between the cam-follower and cam so that it is eventually let down on to the seat at a very low velocity. In other words, the valve is under cam-control for the first and last portions of the motion but for the remainder of the action, i.e. over the nose of the cam, the valve is under spring control. The pressure between the cam and follower during periods of cam-control is equal to the spring pressure plus the inertia force or momentum of the valve gear, whilst during the periods of spring control, the pressure is the spring pressure minus the valve inertia, and if at any point the inertia becomes greater than the spring pressure the cam follower will leave the cam contour giving rise to the condition known as valve float. This may not be very noticeable, in fact if the cams are inadequate in lift or duration, may even be beneficial from the power production point of view, but if the speed rises still further, the separation will become greater and greater until the valve hits the piston or the other valve and suffers in the process, or else very heavy impact loads (as indicated by the onset of excessive noise) will be set up somewhere in the valve gear with the certainty of failure if the conditions are allowed to continue.

Touching on the matter of spring caps, many can be lightened judiciously by reducing their diameter to the



Construction of the split collet and circlip assembly used on J.A.P. and various valves.



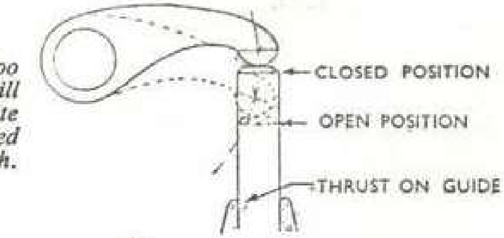
An over-long valve stem causes thrust to the right.

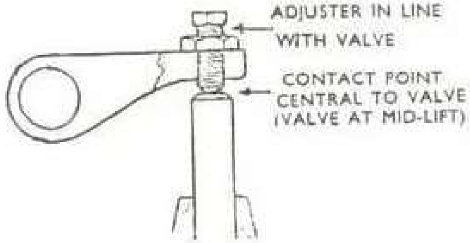
mean diameter of the outer spring and filing scallops in the edge, so that the spring bears only on six or eight projections. Alternatively, new caps can be turned from a *strong* light alloy (preferably R.R. 77, but, failing that, R.R. 56, or Duralumin B) to the same dimensions as the original steel ones, except for allowing a little extra thickness where the flange merges into the taper bore, and around the collet boss. Titanium is also suitable.

If you are rejuvenating an old engine and are using non-standard valves, the easiest way to retain the spring is by using a spring cap, collets and wire circlip of J.A.P. design, since all the machining called for is to turn in each stem a narrow semi-circular groove to accommodate the circlip, and to verify that the stems accurately fit the collets, which are obtainable with bore sizes of $\frac{5}{16}$ in., $\frac{11}{32}$ in., and $\frac{3}{8}$ in.

The position of the upper end of an o.h. valve-stem is important, but is difficult to check dimensionally without special measuring fixtures, so it will need to be done by eye. Assemble the rocker gear on the head and observe the behaviour of the rocker-tip relative to the valve stem as the mechanism is operated by hand from the closed to the full-lift position. If the stem is too long, the contact point will

If the stem is too short the thrust will occur on the opposite side to that indicated by the above sketch.





Just right! A valve stem of the correct length has the rocker at right angles to it, with the adjuster dead central, when in the mid-lift position.

move outwards, as shown in the diagram; if too short, the movement will be in the opposite direction. If the outer end of the rocker is fitted with a screwed adjuster, this should be exactly in line with the valve stem in the *half-lift* position with the contact point dead central to the stem at the same time.

There is little likelihood that a fairly new engine using standard components will not be correct, but numerous causes—excessive cutting-back of the valve seats, stretched or incorrect valves, or damaged rocker boxes—may all contribute their quota to incorrect alignment in old power units. Sometimes a cure can be effected by shortening the stem or fitting hardened end-caps of the appropriate thickness; in other cases packing up the rocker box will do the trick. The point may seem to be of little importance, but it must be remembered that, however strong the springs may be, as maximum revs. are approached they have less and less margin of strength over that required to return the valves to their seats; thus, an amount of friction anywhere in the whole of the valve gear, which is barely perceptible at hand-cranking speeds, becomes a serious item at top revs.

The clearance at full lift between the valve guide and spring collar must also be checked. This should be a full $\frac{1}{16}$ in., otherwise, if the motor is over-revved, the parts may hit, and no valve will stand up to impact of that nature for very long. Similarly, if coil springs are used, they must not be "coil-bound," i.e., compressed solid at full lift, which might occur if the bottom washers have

been packed up to increase the spring pressure, or if incorrect springs are installed; this point is *most* important.

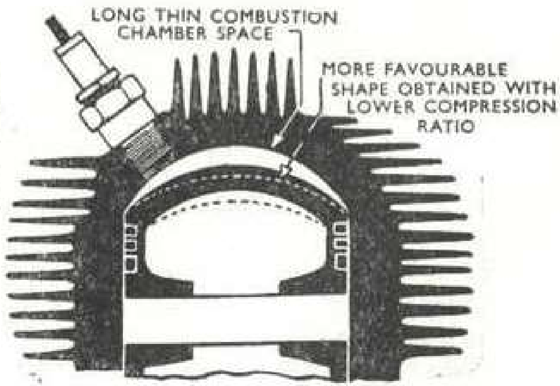
As a general rule, it is best to defer the assembly of the head until any necessary alterations to the barrel or piston have been effected; the subject is dealt with in Chapter X. Such modifications may be required to obtain the correct compression ratio, or to ensure that the valve-clearance "pockets" in the piston crown are deep enough and of the correct radius.

The correct compression ratio depends upon the cylinder capacity, type of fuel, material of the head and many other items, such as stroke-bore ratio, effectiveness of the induction system, and so forth, but the following table gives an approximate guide to the ratios which can be employed with safety in o.h.v. engines under racing conditions when using 50/50 petrol-benzole fuel, although they by no means represent the ultimate, or necessarily the most suitable, figures:

	Cylinder-head Material		
Cylinder capacity	Iron	Alum. Bronze	Alum. or Bi-metal
250 c.c.	8.5	9.5	10
350 c.c.	8.2	9	$9\frac{1}{2}$
500 c.c.	7.8	8.5	9

Elderly engines with finning a bit on the scanty side may not be able to use ratios quite as high as those quoted, save

The benefits of a higher compression piston can be lost if the resultant combustion space becomes unduly distorted. Better results will be attained with the lower piston dome, illustrated here by a dotted line.



for very short-distance events or when using alcohol fuel. Ratios as high as 16 to 1 can be used with fuels such as Shell "A", J.A.P. fuel, or pure methanol, irrespective of the cylinder size, but in "square" engines—i.e., those in which the bore and stroke are equal, or nearly so—the piston may need to have such a pronounced dome on it that the shape of the combustion chamber becomes unfavourable to good combustion.

In such cases better results may be obtained by keeping the ratio down to, say, 12 to 1. Of course, "dope" fuels *can* be used with ratios suitable for petrol-benzole, but the engine will not give much more power—it will simply run cooler and be less liable to give trouble with valves or plugs. This subject is a wide one and has a section devoted to it in the next chapter.

Assuming that the compression ratio and valve clearances are correct, final-stage work on the head can be commenced by polishing the valve stems with dead-smooth emery cloth (rubbed *along* the surface, not round it) and then grinding-in in the normal manner, but using a fine abrasive, finishing off with powdered Turkey-stone or metal polish to obtain really smooth seatings entirely free from ridges or grooves. During this process the inlet-valve seat, which has been reduced to $\frac{1}{32}$ in. wide through rounding-off the edges of valve and port (*see page* 19), will increase slightly in width to 45 or 50 thous., at which it is wide enough to run for some time without undue hammering back. It is best to use some flexible device rather than a rigid tool for oscillating the valve so that it can seat itself freely; a rubber suction-cup on a wooden handle fills the bill, though it may be necessary to apply a drop of Bostik to the polished valve head to obtain sufficient grip to turn it. Another method is to push a piece of tight-fitting canvas-lined hose on the end of the stem and oscillate it, either by twirling between the hands or with a hand drill. When grinding in Vincent valves the upper guide *must* be in place, for the lower one is too short to locate the valve accurately by itself and is not intended to do so.

After grinding in, wash away all traces of abrasive with petrol or paraffin, smear a drop of *clean* mineral oil on the seats, and assemble each valve in turn—first applying some graphited running-in compound to the stems to minimize the risk of scoring or "picking-up" when the engine is first started.

Make sure that the split collets are paired up correctly, and, after fitting the springs with the aid of the invaluable Terry spring compressor—an essential part of the shop equipment—verify that both collets are hard against the abutment on the stem and that the spring cap is fitting snugly on the taper; if not, one or two light blows with a hide mallet or the end of a hammer-handle on the spring cap (*not* the valve stem) will usually jar the assembly into position. If you are in any doubts as to the fit of the collets in the spring cap, it is advisable before assembling the valves to lap these components together in position on the stems, using ordinary valve-grinding paste as the abrasive.

Wire circlips sometimes become stretched when being removed and it is best to close them up before reassembly and then to make absolutely sure, even by using a magnifying glass, that each is actually seated in the groove and nestling snugly in the recess provided in the top of the split collets when the full pressure of the valve spring is on them. Occasional cases of valves dropping in with this form of fixing are usually attributable to failure to observe this point during assembly.

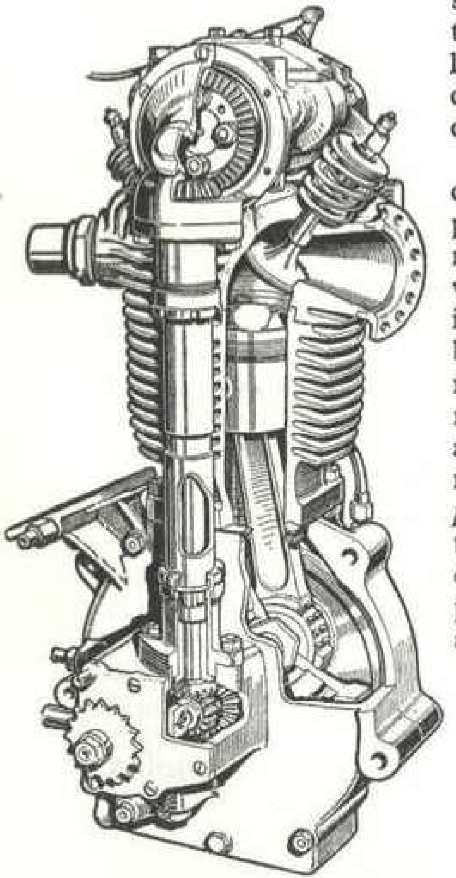
Split collets are often supplied in pairs, partially cut through. They should always be left in this condition until required, then separated and the small burr left in the cut cleaned up prior to fitting. Mixing up split collets indiscriminately is a fruitful source of valve failure, though it may take some time to develop.

Finally, don't leave the head on the back of the bench to get filled up with dust and swarf. Smear oil lightly inside the ports and block them with pieces of clean rag, or, better still, large corks, which are less likely to be left in place when they should have been removed.

CHAPTER IV

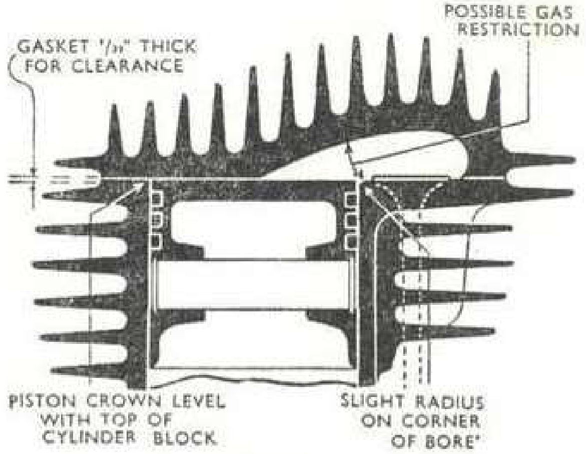
COMPRESSION RATIOS AND FUELS

The previous note on compression ratios suitable for various engines needs amplification. In the first place, the ratios quoted are for racing only, being far too high for ordinary road work with silencer and "pump" fuel, one to one and a quarter ratios less being about as high as could be used in the latter circumstance.



Scanty finning, characteristic of design of the 1920's, is still satisfactory where cool-running methanol and similar fuels are used.

Owing to the shape of the combustion chamber, it is impossible to obtain such high ratios in a side-valve cylinder without restricting the breathing ability—about 8 to 1 being as high as one can reach—and it must always be remembered that it is not the actual *measured* compression ratio which counts, but the *pressure of the gas* at the end of the compression stroke. This depends as much upon the perfection of cylinder filling as on the ratio; therefore a



Points which must be checked when a side-valve cylinder head has been machined to raise the compression ratio.

balance must be struck between the interior shape which gives the best gas-flow and that which gives the highest ratio.

Most s.v. engines are now of the detachable head pattern, with the combustion chamber concentrated over the valves, and in this type the throat between chamber and barrel sometimes constitutes a restriction, particularly if much metal has been machined off the head to raise the compression. A small radius on the top edge of the barrel next the valve chest will improve the flow lines, but must not be carried so far down that the top ring is in danger of being exposed. The thickness of the gasket is sufficient clearance between the flat portion of the head and the piston crown, which should, therefore, be just level with the top face of the barrel at t.d.c. An aluminium head is particularly desirable, except, perhaps, for alcohol fuel, when the cooling problem is not so acute.

Reverting to o.h.v. engines, the aim is to obtain the correct ratio with the least possible area of metal exposed to flame. However impressive it looks, an excessively domed piston crown is not desirable, partly because its exposed area is great, also because the "hump" hinders flame propagation, and anything in the nature of the converging pocket which is formed by the angle between dome and head produces a likely locality in which detonation can take place.

It is quite a good scheme to make a cast of the combustion chamber by pouring melted wax into the plug hole with the piston at t.d.c. This shows up any pockets or restrictions, and the wax can be softened and remoulded with the fingers after detaching the head, to give an improved shape with the same volume of wax and, therefore, the same compression. As quite a number of different makers have used the same bore diameters—74 mm., for instance, is a popular size for 350 c.c. power units—it is often possible to utilize a "foreign" piston of different proportions to replace a standard article and thereby obtain a better shape of combustion chamber.

Unless shims are already fitted under the barrel flange, the usual method of finally adjusting the ratio is by shortening the barrel. A great deal of laborious setting-up and remeasuring can be saved by making up a two-column table showing the C.R. and equivalent compression space and determining, as follows, the differences in barrel length which are equal to 1 c.c.

Bore	1 c.c.=
60 mm.	.014 in.
70 mm.	.010 in.
80 mm.	.008 in.
90 mm.	.006 in.

Knowing the existing ratio, a very simple piece of arithmetic will show how much length alteration is required for any other ratio. Skimming the barrel flange is an operation which has to be done very accurately to ensure that the new surface is dead square to the bore, and some prefer to turn the top joint face where squareness is not so vital. On the other hand, a little more than is necessary at the moment can be turned off the flange and thereafter adjustments to increase or decrease the ratio can readily be made by shims. These should be made up from good material of even

thickness and free from wrinkles, and should be provided in assorted thicknesses, as a large number of thin shims do not provide a firm foundation for the cylinder.

Short-stroke Manx Norton engines are fitted with squish-type heads, and in these the gap between the flat lands on the piston crown and the steps machined in the combustion chamber *must* be maintained within the limits of 45 and 55 thou. This means that no significant alteration in compression ratio can be obtained by shimming the barrel, but only by changing the piston. On the other hand, shimming to obtain this optimum clearance may be necessary after any change of components or dimensional alteration has been made. The same remarks apply to late A.J.S. 7R and Matchless G50 engines.

Determining Compression Ratios

For maximum power, the rule is to employ the highest ratio *at which the engine will run without distress*, on the fuel which is specified for the race in question. There are, of course, many factors, both internal and external to the engine, which have an effect on the ratio which can be used, but the prime factor is the quality of the fuel.

J.A.P. fuel, methanol and similar fuels composed mainly of alcohol can be run at anything up to 16 to 1, even in engines with iron heads and barrels and scanty finning, this being partly because alcohol is almost knock-free and partly because the large amount of heat required to vaporize these fuels exercises a great internal cooling effect.

Engines for use on official dirt tracks are, however, limited by the regulations to 14 to 1 ratio. Even in events where there is no such limitation, there is not much to be gained by going any higher as very often the obstruction in the combustion chamber caused by the high piston-crown required to obtain these ultra-high ratios more than offsets the slight increase in power which might otherwise be gained.

With petrol or petrol-benzole detonation will occur at much lower ratios than are usable with alcohol, the highest ratio that can be satisfactorily employed depending upon the "octane rating" of the fuel and the design of the engine. The octane rating varies from country to country, and may be as low as 70 in some and as high as 100 in others. If fast road work is the main objective, it is better to choose a ratio which will suit the octane value of the best fuel normally obtainable in the area where the machine will be used; as an instance of the effect of octane rating, engines which could be run at 10.5 to 1 in the 1939 T.T. races using 50/50 petrol-benzole mixture of around 87 octane rating, had to be dropped down to 7.5 to 1 when using plain petrol of 72 octane, which was the only fuel permitted in 1947 by the F.I.M. regulations.

Since then the regulations have been progressively relaxed and the requirement in 1959 is that the fuel must be only of the best quality commercially obtainable in the country where the race is being conducted, but in no case must it exceed 100 octane, Research method. This rules out the 100/130 aviation gasoline which is permitted for Formula 1 car racing, but even so allows ratios of 10.5 to be used if in fact 100 octane fuel is available. This may not be so, of course, in all countries and where "pump fuel" is specified in the local regulations, care should be taken to discover, well in advance, the octane rating which the available fuel possesses, and, if possible, obtain a sample.

The octane rating is improved either by the chemical analysis of the fuel or by the addition of tetra-ethyl-lead or "Ethyl fluid". This additive has the disadvantage that as a by-product of combustion it forms lead-bromide which attacks poor-quality exhaust valves with ferocity, but will have little effect on a well-cooled valve material corresponding to DTD49b or Nimonic 80. Consequently TEL additions of up to 3½ c.c. per gallon can be tolerated by any good racing engine fitted with such valves.

Besides its inherent freedom from detonation, alcohol requires a great deal of heat to vapourize and if no external heat is applied, the result is a great reduction in charge-temperature, which can be seen by the rapidity with which

ice forms on the induction-pipes. Methyl alcohol (methanol) has a greater cooling effect than the lesser-used ethyl alcohol (ethanol), but in either case the result is to allow a greater weight of air to be induced, since cold air is heavier than hot air. This improves the volumetric efficiency of the engine, and as the mixture will tolerate very high compression ratios and thus can be burnt at a higher efficiency, in theory a great increase in power should be obtained. In practice the increase is not as great as might be expected, partly because the engine must be run hot enough to give good burning and this offsets to some extent the cooling effect of the alcohol. Alcohol-engines are prone to be sluggish until fully warmed-up and usually perform best with limited cylinder cooling, so that this fuel is particularly suitable for getting a high performance from engines with ordinary cast-iron barrels and heads especially if the stroke is longer than the bore and a high compression ratio can be obtained without spoiling the combustion chamber shape.

Apart from this aspect, the amount of energy liberated in the cylinder per cubic inch of mixture of correct strength is very much the same whatever combustible substance is employed, the theoretical quantity being around 48 foot-pounds. No amount of juggling with any of the regular fuels will alter this fact, and the use of alcohol blends only increases the power for the reasons outlined above. On the other hand, the calorific value of alcohol is very much lower than that of petrol and for the same amount of power developed, the consumption is very much higher, up to two or even three times as great. This is naturally a serious matter in long-distance racing, as apart from the matter of expense, it entails either a vast tank capacity or too many pit stops, but a compromise may be affected by using alcohol blends such as 80 per cent methanol, 10 per cent benzole and 10 per cent petrol, or equal parts of methanol, benzole and petrol which will run very satisfactorily with a reasonable consumption at ratios of 10 or 11 to 1. The various oil companies used to supply such blends ready-mixed; a table of these with their approximate analyses and compression ratios appears on page 286, but this practice has now ceased in England. Instead, each company supplies a standard blend of methanol, and acetone, also quoted in the table of fuels, which can be blended with petrol and/or benzole to make any required mixture.

Blending methanol with petrol is not quite a straightforward matter; the methanol must be of the quality known as "anhydrous" or "dry blending", and even then it will only mix with some types of petrol if a proportion of benzole equal to that of the petrol is added as well. Acetone, besides being a fuel in itself, is a help in preventing the constituents from separating out after mixing and some of the present-day petrols, with this help, will mix with methanol without the addition of benzole. The point can always be checked by making a trial mixture in a bottle and observing whether the liquids settle into two layers or not after standing for a while.

For sprint work where a high consumption can be tolerated, it became fashionable to employ nitro-methane as an addition to alcohol fuel. This compound contains an excess of oxygen which is liberated during combustion and thus enables much larger amounts of alcohol to be burnt; in effect, it provides a sort of chemical supercharging, and as such its use is frowned upon by some race organizers. It is, however, permissible at the time of writing in sprint events, but it requires careful handling at all times, as unlike petrol or alcohol, it is chemically unstable.

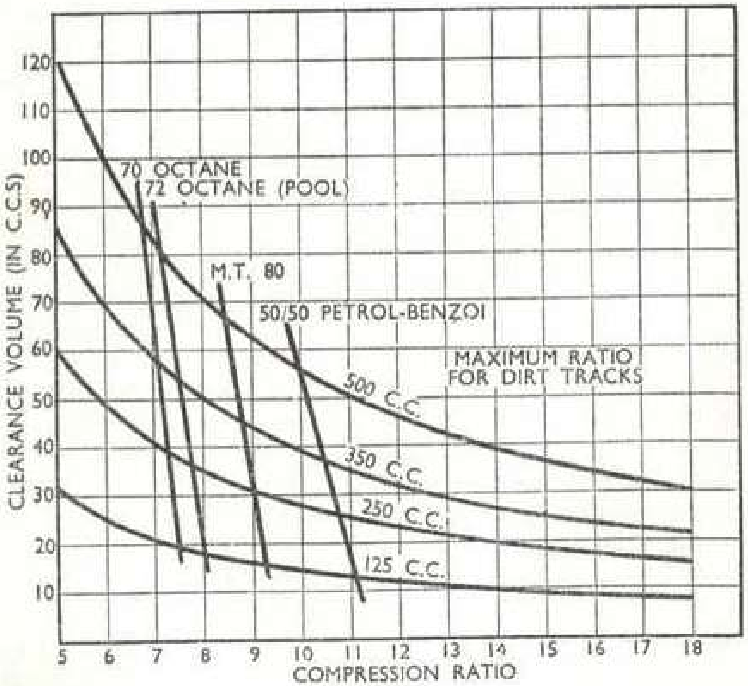
For that reason, it is not supplied neat, but diluted with an equal proportion of methanol; even then, any containers must be de-pressurized occasionally by loosening the filler-caps. Drums, fuel tanks and so forth must be kept scrupulously free from any contaminants, and the entire fuel system should be drained completely after each event. Some riders are in the habit of flushing the system through with petrol but if this is done all petrol *must* be removed before starting-up on the nitro blend.

Fifty per cent nitro-methane can be used, with a theoretical power increase (according to W. B. Rowntree, M.Inst. Pet.) of 40 per cent, provided the engine is strong enough to stand it, but it is more usual to dilute the basic mixture with Shell A.M.1 (94 per cent methanol, 6 per cent acetone) to bring the nitro content down to 25 or 20 per cent which should give a theoretical power increase of 15 per cent. To utilize the liberated oxygen, much more fuel is required; if, say, an 1800 main jet is necessary for straight A.M.1 fuel, it may have to be increased to 2500 for 25 per cent nitro. In any case, it is wisest to err on the large side, partly because alcohol fuels give their best power at rich mixture strengths, and partly because the free oxygen present when there is a deficiency of fuel attacks and burns a hole in the piston crown with astonishing rapidity. Great care must be taken to see that the fuel can flow through to the carburetter absolutely freely; it may not do so under severe acceleration because the fuel then gravitates to the rear of the tank and there may be practically no "head" of liquid existing on, say, the front carburetter of a V-twin and this instrument will be starved in consequence. An adequate breather must also be used on the tank, particularly if this is of the tiny sprint variety. These may seem to be obvious points, but all too often they are overlooked.

It must never be forgotten that the rate at which fuel can enter the float-bowl must always be much greater than the rate at which it is being used. Otherwise the level in the bowl may at some time be much lower than it should be: the position of a carburetter in relation to the tank and the conditions existing during violent acceleration or braking may seriously affect the rate of flow. If, for instance, the down-draught angle is appreciably steepened to obtain a straighter inlet pipe the bowl may be brought so close to the tank that the head of liquid present when the tank is nearly empty is so small that the flow will be inadequate unless two very large taps and pipes and possibly twin float-bowls are fitted. Also under braking conditions, the outlet may be uncovered as the fuel surges forward thereby admitting air into the pipes and this must be got rid of in some way before fuel can again commence to enter the bowl. Lack of attention to these matters has more than once caused trouble which has been wrongly attributed to the carburetter itself.

Finding Compression Ratios

The graph on this page has been drawn to provide an easy method of finding out the C.R. given by any particular combustion space volume (or vice versa) in conjunction



Relationship between combustion-space volume and compression ratio in the most commonly used cylinder capacities is readily seen from this graph.

with cylinders of various capacities; these curves represent the most commonly used cylinder (*not* total engine) capacities, the 125 c.c. curve, for example, applies also to 250 c.c. twins or 500 c.c. four-cylinder engines. To find the ratio given by any known combustion chamber volume, read across from the volume figure to the curve of the cylinder capacity concerned and then downwards to the ratio line.

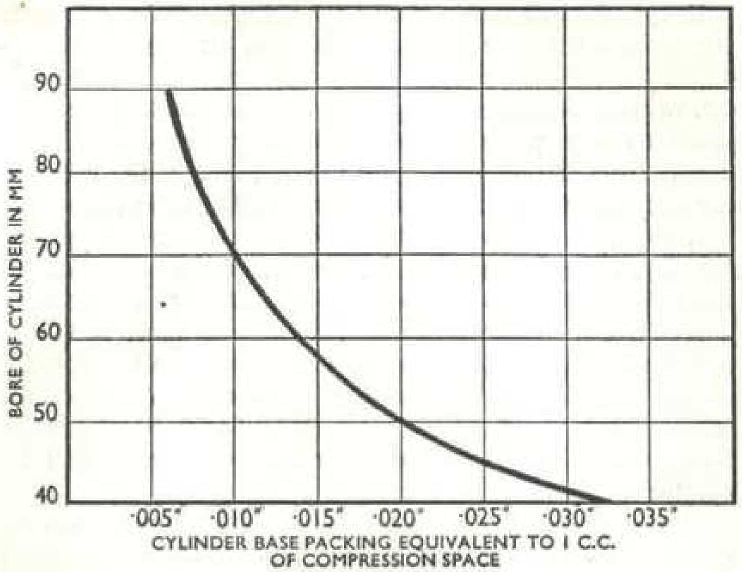
Conversely, to find what compression space is required for, say, 7½ to 1 in a 250 c.c. cylinder, read up from 7.5 on the ratio line to the 250 curve, then across to the volume line, where it will be seen that the required figure is 38 c.c. If it is desired to raise the ratio to 8, for which the volume should be 35½ c.c., the amount of packing (or material which has to be removed from the cylinder base) can be found by reference to the second graph, which will be mentioned in a later paragraph.

The diagonal lines drawn across the capacity curves indicate roughly the maximum ratios which can be used with various grades of fuel, though (except in the case of alcohol) the indication is necessarily only approximate owing to the aforementioned varying factors. An engine with an iron head and barrel and poor finning cannot employ nearly as high a ratio as one with an aluminium head and extensive finning, unless the latter's breathing is so poor that the cylinder does not fill properly at maximum r.p.m., and then its *actual* compression pressure is very much less than the measured value.

For the same reason, reduced cylinder filling, engines which are to be raced at high altitudes, and consequently low atmospheric pressures, will stand a higher ratio than when run at sea-level, which is worth remembering in some countries where competitions are held in mountainous regions. A damp atmosphere also helps to suppress detonation.

Detonation the Limiting Factor

A peculiar thing about almost all engines is that with any particular fuel there is a limiting ratio above which one



Here is shown the thickness of packing equal to 1 c.c. for given bore dimensions.

dare not go without an almost certain risk of piston trouble, yet the ratio at which the engine will run happily is frequently only very slightly lower; dropping from, say, 9.8 to 9.5 may make all the difference between success and failure, with but little or no diminution in performance.

Apart from audible " pinking," the signs that detonation has been occurring are unmistakable when the piston is examined: on the side of the crown remote from the plug some of the carbon film will be burnt away and the surface of the aluminium will be locally pitted and rough, for which condition the colloquial description is that " the mice have been at it."

To some extent detonation due to an excessively high ratio can be suppressed by retarding the ignition and using a jet giving a very rich mixture, but the result will only be diminution of power; in other words, there is nothing to be gained and everything to be risked in attempting to use

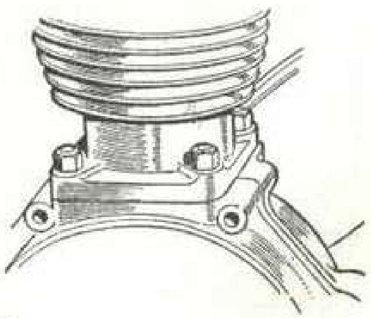
a ratio higher than the engine is willing to accept. This is particularly true in long-distance races in which the throttle is wide open most of the time; for small tracks and sand racing, it is usually possible to go up a little in the ratio, because as the engine is not running for very long, and in any case is being shut off fairly frequently, the internal temperatures do not have time to become dangerously high.

The graph on page 56 enables the amount of packing which must be removed or added when adjusting the ratio to be determined very easily. Referring to the previous example, it was found that to raise a 250 c.c. cylinder from 7.5 to 8 to 1 the compression space must be reduced from 38 to $35\frac{1}{2}$ c.c. With this second graph, supposing the bore of the cylinder to be 68 mm., if we read across from 68 mm. on the vertical line to the curve, and then down to the horizontal line, we find that .0115 in. is equivalent to 1 c.c.; therefore the thickness of packing must be reduced by 2.5.× .0115=.029 in., or just under $\frac{1}{32}$ in.

By reading in the reverse direction, i.e., upwards from the horizontal line, then across, the difference in volume effected by removing or adding a known thickness of packing can be read off directly, thus saving a lot of calculation. With the aid of this graph, a series of plates can easily be made up so that the ratio can be altered rapidly to any desired figure.

If the barrel is held down by studs which go through the

If the barrel is attached to the crankcase by short studs it is inadvisable to remove metal from the base. When long bolts such as those illustrated on page 128 are used, a small amount of metal can be removed from the base with safety.



flange only, care must be taken to see that the latter is not made too thin, as it is in a highly stressed locality, and in overhead camshaft engines with the usual Oldham couplings in the vertical shaft there is a likelihood that the small amount of end-float normally allowed may all be swallowed up—a state of affairs which will put excessive end-thrust on the vertical bevel shafts. This point must, therefore, be verified.

On the other hand, if at some future date the barrel is stacked up again with shims by an appreciable amount, it may be necessary to obtain a longer shaft or a pair of coupling-discs of greater thickness. On no account must these couplings be run with the tongues only half-engaged in the slots, as the metal is then subjected to a form of stress which may cause failure.

It goes without saying that the cylinder bore should be in perfect condition, particularly at the upper end, where the heaviest wear invariably occurs. The wear practically ceases about an inch or so below the ridge which forms just level with the top ring at t.d.c., but, in effect, over this short length the barrel is tapered. In their attempts to conform to the varying diameter, the rings have alternately to expand and contract, a feat which may be beyond their capacity above a certain speed.

When this occurs, gas will be able to blow past and also oil can work its way up into the head. Worse still, if the rate at which the rings are being forced in and out corresponds to their natural frequency of vibration, they will get into a state known as "flutter," and after a while will break, usually at a point fairly close to the gap. If the ring design is incorrect, flutter may occur even in a perfect bore, or may be aggravated by barrel distortion of an unsuspected nature, which may be caused either by thermal distortion or through the base flange or crankcase face not being dead flat.

Another snag about altering compression ratios by omitting compression plates or reducing the base flange is that the piston ring will inevitably strike any ring ridge which has formed; occasionally this has even been known

to cause the piston ring land below the top ring to break away. Even a change from one type of piston to another may introduce the same effect due to varying ring heights, so it is always wise to remove every vestige of ridge by very careful work with a sharp hand scraper or a ring-ridge remover if any possibility of fouling exists. This point has already been noted in Chapter II but is sufficiently important to warrant repetition.

Re-conditioning Worn Barrels

Badly worn barrels may be rejuvenated by reboring or by linering, though the first method is rarely applicable for racing, as mentioned in Chapter I; Norton engines of 490 c.c. are one exception as they can be bored out from 79 to 79.6 mm. before exceeding the 500 c.c. limit. Not all barrels are thick enough to stand being linered without danger of fracture at the flange, although the type in which long bolts extend from the crankcase up to the head are much less likely to fail than those with a bolted flange. Also it must be remembered that however well the liner fits, its junction with the barrel constitutes an additional heat-break, though this is of less moment with alcohol than with petrol fuel.

Barrels can be brought back to original size by chrome-plating, though this is a specialized process, as it is necessary for the plating to have a porous surface to provide a multitude of oil reservoirs; the processes are known by various trade names such as "Listard," the Van der Horst patent administered in England by R. A. Lister and Co., Ltd., Dursley, Glos., and "Honeychrome," a speciality of Monochrome, Ltd., Alcester. Laystall Engineering Co., Ltd., London, S.E.1, market a range of liners under the name of "Cromard," these being very thin mild-steel tubes with chrome-plated bores supplied finished to size; all that has to be done is to bore the barrel to the correct size and press the liner into place, no subsequent boring being required. Included in the range is a liner, specially designed for fitting to the speedway J.A.P. engine.

This liner is slightly reduced in diameter near the lower end and the cylinder must be machined with a corresponding step in the bore, so that there is no possibility of the liner, which has no flange at the top, being pulled down into the crankcase should it come loose or if the piston seizes badly. Such an occurrence is nothing short of a calamity and, where possible, liners should always be provided with a flange at the upper end which can be gripped between the head and barrel and thus prevent the liner from moving in any direction.

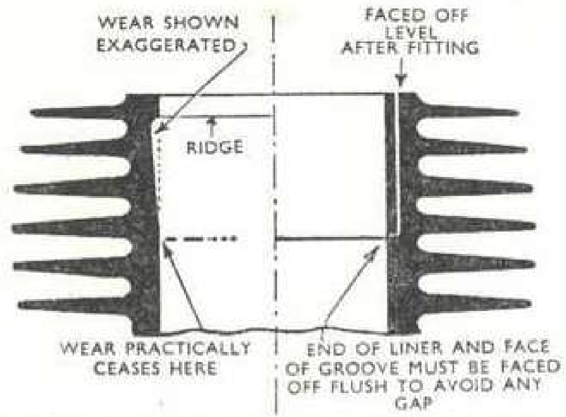
Other points which must not be overlooked are the drilling of any oil-holes and cutting out any con-rod clearance slots which may be provided in the barrel originally.

If the skirt of the barrel, which projects inwards into the crankcase, is long and of fairly thin section, as it is for instance, in MOV and MAC Velocettes, it is a wise plan to reduce the liner diameter at the lower end so that it is just the same size over the length of the skirt as the bore into which it is pressed. This eliminates the danger of the skirt developing a crack when the liner is pressed into position.

Incidentally, liners in aluminium cylinders should be fairly thick—not less than $\frac{1}{8}$ in., and preferably more—to resist distortion, but when used in an iron barrel they may be made as thin as $\frac{1}{16}$ in. Provided a locating flange is present, an interference fit when cold of .001-.002 in. is sufficient, irrespective of diameter, as the liner is bound to be hotter than the jacket when running, and it will never loosen. In aluminium jackets a greater interference is required, and allowance must be made for the high expansion, on the following basis, according to the alloy used:

High silicon (L.33, "Alpax," "Lo.-Ex"), .0009 per in. of diameter.
Y Alloy, R.R. 50 or 53, "Birmabright," .0011 per in. of diameter
Magnesium Alloys (Elektron, Magnuminium), .0016 per in. of diameter.

After a short period of running, liners in aluminium shells



Short liners may be used very effectively for barrel renovations without reducing strength near the base flange.

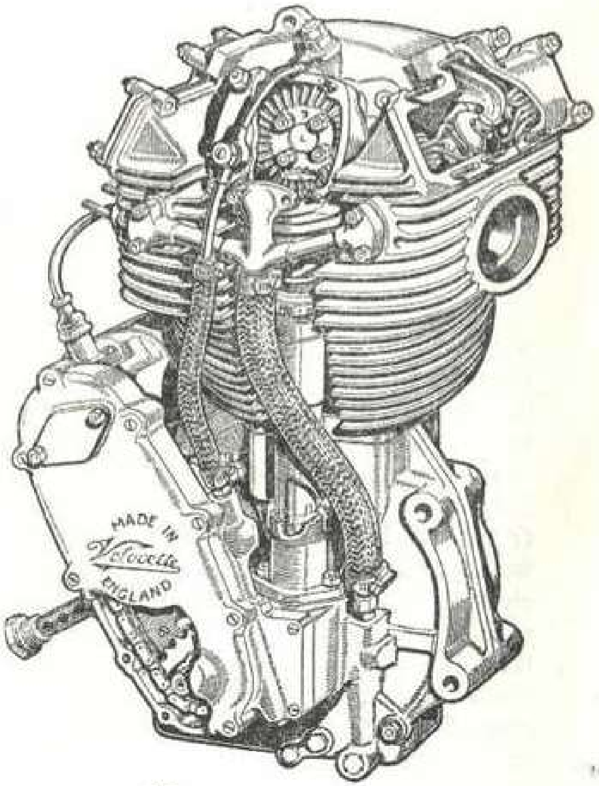
sometimes bed down a little and exhibit either tight spots or hollows in the bore; it is therefore a wise precaution to hone them out initially 2 or 3 thou. small and re-hone or lap out to the correct size after a short period of running-in with an old or slightly undersized piston.

Shown above is a simple method of rejuvenating a cast-iron barrel by boring it out and fitting a short liner, which can be made from any good grade of cast-iron but preferably an alloyed iron containing $1\frac{1}{2}$ to 3% of nickel. The important point to watch is squareness of the shoulder and liner to avoid any semblance of gap; if properly done the joint will be almost invisible and will remain so in service.

The quickest method of truing up worn or tapered bores is by means of a cylinder hone which can be rotated and reciprocated in the bore either under a drilling machine with a long spindle travel or by means of a slow-speed electric drill. A copious and continuous supply of kerosene or paraffin oil is vital to prevent scratching the surface, but there is no need to attain a mirror-like finish. The ideal surface consists of a large number of very fine criss-cross markings which retain oil and give a quick bed-in of the rings.

Hand-lapping to obtain a perfect finish or to clean up a slightly damaged bore can be done with a discarded piston, split in halves and expanded by two springs slipped over the gudgeon bosses. Mounted on an old rod or a piece of wood, this device is pushed up and down the barrel by hand and given a partial rotation at each stroke, the abrasive being flour emery mixed with oil if much metal has to be removed, and metal polish for obtaining the final finish. All traces of abrasive must, of course, be finally eliminated by several washings in clean spirit.

If this treatment is applied to side-valve barrels a hollow will sometimes be noticed in the surface adjacent to the



Higher power output calls for greater heat dissipation; it is interesting to compare the finning on this 1939 Velocette T.T. engine with the 1926 version shown on page 46.

exhaust-valve port; this is a sign that, when hot, the barrel distorts inwards, and the " bump " so formed has in time been rubbed away by the piston. When cold, of course, the former " bump " now appears as a " hollow," and it is better not to continue lapping in an attempt to eliminate it. If that is done, the bump will reappear when the engine warms up and may cause trouble if full-throttle is applied without a lengthy preliminary running-in period.

Enclosure of side valves in large boxes which eliminate a lot of fin area is bad from the cooling aspect, and in such cases the covers should be liberally drilled and fitted with scoops in order to stimulate a flow of air through the valve chest; the fins sometimes cast on the covers are not really effective, as the stagnant pocket of air in the valve chest is the real barrier to heat dissipation.

For short-distance events with alcohol fuel—grass- and dirt-tracking and sprint work—deep barrel fins are not necessary (readers will remember the early dirt-track J.A.P. power units, which might almost be called " clean-shaven "), but when road racing, complete with mudguards, barrel cooling becomes worse as the speeds become higher, due to the " dead air " region behind the front-wheel assembly becoming more and more pronounced. Hence the use of fins up to 8 in. in diameter, which would be of prohibitive weight in iron and are, therefore, made of aluminium or, in rare instances, magnesium alloy.

For anyone with a little skill in lathe work there is no great difficulty in making up a bi-metal barrel out of a solid casting, or, better still, a forged bar of light alloy, with the fins turned from the solid. The tapered form of fin has been found to be the most effective, although a little more difficult to machine than the parallel variety. The bore should be finished at as high a speed as the lathe will permit, using for preference a tungsten carbide tool and a feed of about .003 in. per revolution, as successful heat transference between liner and barrel depends very largely upon the perfection of finish of the contacting surfaces.

The actual composition of the alloy is of little moment, except in so far as the coefficient of thermal expansion varies. High expansion rates are not advantageous, and in this respect the magnesium alloys and the L5 aluminium alloy commonly used for general castings are the worst of the lot at .000026 in. per degree C. High-silicon alloys to specification L33 are the lowest at .000019 in., but do not machine well. "Lo-ex," the alloy from which most motor-car pistons are made, has the same coefficient as L33 but machines quite well, especially if given a low-temperature heat-treatment of 10 hours at 170° C.

The best compromise is afforded by Y-alloy or R.R.50, with a coefficient of .000022 in., while the aluminium-magnesium group—such as "Birmabright" or Hiduminium 40—are only very slightly higher and have the additional advantage of being very resistant to corrosion, even by sea-water.

A process which has lately come into prominence is the "Al-Fin" system, in which a true metallic bond is formed between the iron and aluminium, so eliminating a heat-break at this locality. The process is handled by Wellworthy Ltd., Lymington, Hants, who are prepared to cast solid jackets on to barrels, the fins, of course, having to be subsequently turned by the customer. Racing Norton and J.A.P. engines have been equipped for some time with "Al-Fin" barrels, with fins cast to size. The main patents covering the process are held by the Fairchild Aviation Corporation, Farmingdale, N.Y., U.S.A., and there are several licencees operating in that and other countries. A recent development has been the bonding-in of austenitic cast-iron ("Ni-resist") seat inserts in cylinder heads, thereby obtaining better conduction of heat from the insert to the head than with shrunk-in inserts.

After considerable usage, the bores in aluminium jackets sometimes increase in size to such an extent that the grip upon the liner is insufficient; this may show up by the liner turning in the jacket either in service or when being honed. When the engine is at running temperature, the heat conductivity between liner and jacket will be very

poor, and, moreover, movement of the liner will be resisted mainly by the top flange instead of by the grip of the jacket. This is likely to cause the flange to break away and the liner to drop down, so if looseness is suspected, then either a larger liner should be obtained or the old one increased in size by copper-plating, after making quite sure there is no sign of a crack around the flange.

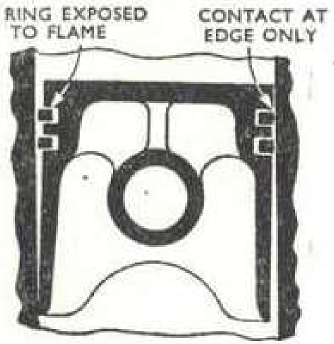
Vincent liners can be supplied .010 in. oversize on diameter. Before fitting the jacket should be re-bored to .006 in. smaller than the liner.

CHAPTER V

CHECKING PISTON AND RING WEAR

JOINTLY or singly, the piston, rings and gudgeon-pin have a number of jobs to perform. They must prevent leakage of gas from, and the passing of oil into, the working part of the cylinder, and also act as a cross-head to resist the side-thrust of the con.-rod and absorb the minimum of power in so doing. As each is dependent upon the others to a greater or less extent, they can be considered as a little family group in which the behaviour of one affects the well-being of the remainder.

All these components work in highly adverse conditions of high and fluctuating temperatures, high pressures and rubbing speeds and scanty lubrication; it is not surprising, therefore, that nearly half the power lost in internal friction is absorbed by them. Luckily, as no medals are awarded for mechanical silence in racing, the problem of eliminating piston-slap does not arise; thus the clearances which give the best results can be used irrespective of any noise which may occur. Nevertheless, it is a mistake to

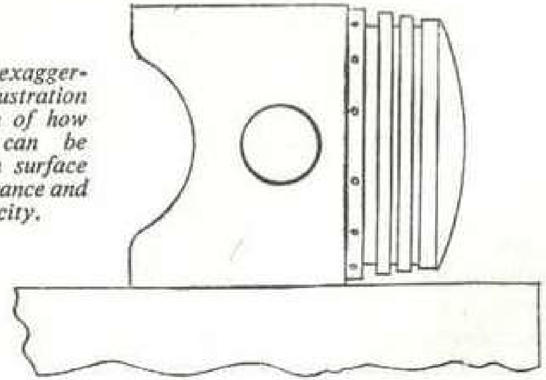


A piston with excessive clearance in the barrel tilts at t.d.c., thus exposing the top ring to flame contact and rounding the edges of all rings.

think that ultra-wide clearances are essential or even beneficial; the reverse is the case in fact, for piston rings cannot function properly in a piston with an excessive amount of "slop." Whenever the piston is tilted, the rings, instead of being in contact over their full face-width, will make only edge contact over a large part of their circumference, and so cannot seal properly; in time the rings will wear to a barrel shape, and this will diminish their ability to prevent the upward passage of oil. Also, as the piston alternates in position from side to side of the barrel the rings have a tendency to follow suit (due to the friction between them and the ring grooves), and even if their contact with the wall is not momentarily broken, the action is very prone to initiate ring-flutter. KILLER

These effects are governed by skirt clearances, i.e., the diameters measured *at right angles to the gudgeon* just below the lowest ring and at the bottom edge of the skirt. Since there is, or should be, very little oil present above the rings, the lands between them cannot be expected to carry any load at all, and should run just clear of the barrel. As there is a sharp rise in temperature as the crown is approached, the lands must be progressively smaller, but again it is undesirable to make the top land too small, since this would expose the top ring to the full fury of the combustion, whereas a close-fitting land will shield it to some extent.

Deliberately exaggerated, this illustration gives an idea of how ring lands can be checked on a surface plate for clearance and concentricity.



A correctly fitted piston, after running at racing speeds for some time, should show an even bearing over the thrust faces of the skirt and just the faintest witness of rubbing contact on the ring lands. To obtain this result it is imperative that the relationship between skirt and lands is correct—that is to say, in addition to being the right relative sizes, the lands must be concentric with the skirt, which cannot be determined simply by direct measurement with a micrometer.

It is, of course, possible for a piston which is too small all over to exhibit the appearance just described, but this can be detected by the presence of thick carbon deposits in the plane of the gudgeon pin, or by the skirt showing areas of heavy bearing at the upper and lower extremities and light bearing in the centre, which appearance is caused by the piston running tip-tilted, in turn permitting the ring lands to make contact. Easing down these areas of heavy bearing under the erroneous impression that they are " high spots " will only make matters worse.

Perhaps at this juncture it might be as well to deal with this matter of easing down the high spots, which is frequently misunderstood. Obviously the piston must touch the barrel somewhere, but if every time the motor is down all the rub marks are carefully eased off with a file, the piston will simply become smaller and smaller until it is useless. The thing is to distinguish between the appearance of *normal* and *abnormally heavy* contact areas. The latter are potential sources of seizure, and the surface may be either torn and rough (in which event momentarily seizure has actually occurred) or possess a smooth polished appearance easily visible by viewing tangentially across the curved surface.

Areas of normal pressure have a dull-grey matt surface, possibly intersected by machine marks of trifling depth, and should be left undisturbed, but the polished areas, or true " high-spots," should be eased down with a dead-smooth single-cut, or Swiss file used very sparingly, for it is a simple matter to convert a high spot into a hollow,

and, once done, the fault cannot be rectified—it can only be disguised by still further filing. The file must, of course, be used with a circular motion, so that the cut washes out smoothly to zero at the perimeter of the area being worked on.

The appearance of the wear-marks on the skirt is also an indication of whether or not the piston is running true in the bore. If the markings veer to one side just below the oil ring and to the other side at the bottom of the skirt, there is a remote possibility that the skirt was not finish-machined true to the gudgeon-hole, but it is more likely that the con-rod is bent. Filing the skirts to remove the uneven marking is useless in this case, because the original error still remains and there is a likelihood that the ring grooves which should of course be absolutely square to the axis of the piston and of the bore will in fact be lying at a slight angle. This will prevent the rings from seating properly and after a while they will wear barrel-shaped, a condition which is bad both for gas-sealing and oil-control. The only solution is to track down the cause of the error and eliminate it, not to disguise it by relieving the skirts.

As regards the actual clearances, conditions vary so greatly that it is virtually impossible to give any hard-and-fast rules and in case of doubt the manufacturer's advice must be sought. Generally speaking, the temperatures in racing engines, even with 50/50 fuel, are not a great deal higher than those sometimes encountered in touring engines, thanks to the use of open exhaust systems and the fact that the racer is almost always making a pretty fair breeze of its own. As a rule, therefore, an increase of .001 in. to .003 in., according to diameter, will suffice, in most cases when the piston being fitted is the same one or is of the same type and material as the original.

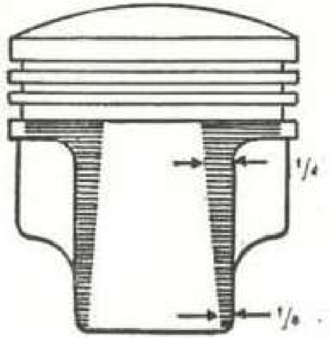
However, it is often advisable to change over to a different design or to use a more suitable material when maximum power is being sought. Alloys such as RR53b or Y-alloy in the fully heat-treated condition are preferable to the medium-silicon, low-expansion varieties known under various trade names such as Lo-ex, Heplex, etc., or the U.S. specification A143, because of their greater strength at high temperatures and slightly less coefficient of friction, but allowance must be made for their greater thermal expansion if a change-over is made. As an approximate guide, the following table gives the clearances *per inch of diameter* which will be found suitable for Y-alloy or R.R. 53 B pistons running in iron barrels:

Top land	.0065 in.
Second land	.0055 in.
Third land	.005 in.
Top of skirt	.0036 in.
Bottom skirt	.0027 in.

Bi-metal barrels with shrunk-in liners have a slightly higher expansion rate than plain iron, and the clearances in such cases can be about .001 in. *less* per inch of diameter. These clearances apply to pistons without T-slots (which are not commonly employed in racing), and are measured at right angles to the gudgeon.

All pistons distort to some extent under the influence of heat and load, and pistons of the plain cylindrical pattern (sometimes referred to as the "pot" type) are oval-turned to the extent of .006 in. or .010 in. to give greater clearances on the sides, although the ring lands are frequently circular; when cold, therefore, the skirt clearances progressively increase from the central plane outwards.

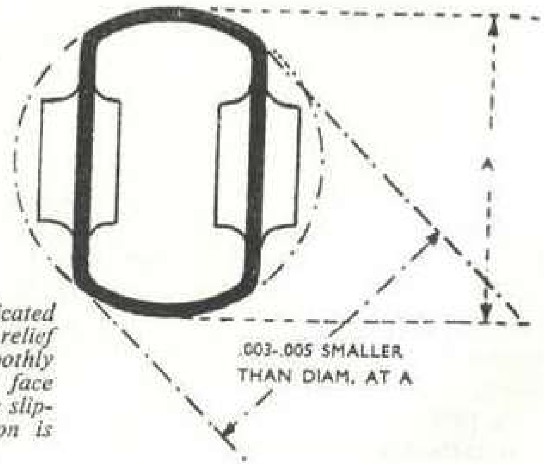
Actually, quite a small area is sufficient to carry the con.-rod side-thrust, and this fact led to the development of the slipper piston, which is both stronger and causes less friction loss through oil shear than the "pot" variety, but to some extent the latter can be improved by heavily relieving the sides so that they run well clear of the walls. Owing to the heat-flow down the side ribs and their consequent expansion, the thrust faces of slipper pistons depart from their cold shape quite a lot when hot, and the clearances should be increased perceptibly towards the lateral



The shaded portion suggests how the contact faces of a slipper piston can be relieved.

edges, particularly just below the top ring, as indicated in the diagram on the left.

There are two ways of measuring clearances: (a) by measuring the bore with an internal micrometer or a dial gauge such as the Mercer, and subtracting the various piston diameters as measured by "mike" or vernier; and (b) by direct measurement, using various feeler gauges. The latter is the means which most private owners will have to adopt, and is quite accurate if carefully performed. For the wide clearances round the top lands it is best to use a number of thin feelers rather than one or two thick



And here is indicated how the depth of relief should merge smoothly into the contact face when work on the slipper type of piston is completed.

ones, as the latter give a false reading through being too stiff to conform with the barrel curvature.

As mentioned, the ring lands must be concentric with the skirt, and this is easily proved by placing the piston on a surface plate and measuring by feelers the gap between plate and lands, which should be equal for corresponding localities on each side of the piston (*see page* 67).

It is sometimes necessary to reduce an oversize piston to fit a standard bore, or to finish to size a semi-finished piston, i.e. one with all machining completed except the final sizing of the ring-lands and skirt. This job can be done in a lathe using a large 60° cone centre in the open end with a loose pin or blade bearing on the gudgeon bosses to provide the drive, and a regular centre in the tail-stock. Alternatively a soft spigot can be turned in the chuck to fit the register bored in the open end of the piston which is then pulled back on the spigot by a loose pin and bolt passing through the hollow head-stock. In either case the ring-grooves must be checked with a dial-gauge to see that they run free from wobble; with any reputable make of piston, this is an indication that the piston as a whole is running true and finish-turning can commence.

Camming, or relieving, the sides as just described, is best done in one of the piston-finishing machines used by engine re-conditioners. Of the range of cams developed and widely adopted for these machines, type "C" is usually most suitable. It gives .009 in. total ovality and .006 in. reduction in diameter measured at the 45° points. Failing access to one of these devices, camming can be done either by hand or in the lathe by offsetting the piston .010 in. in the direction of the gudgeon and relieving the high side to a depth of .005 in. and then repeating the process on the other side. This will leave about half the skirt on each thrust face still circular and the sides of these areas must finally be merged into the relieved portions by hand filing.

A precaution which must always be taken, even with new factory-made pistons, is to see that the corners of the

ring-grooves are slightly chamfered. If not, a fine square file run round the grooves will do the trick. The purpose of these chamfers is partly to provide a small channel for oil distribution, but mainly to prevent the rings being locked into the grooves by metal being pulled over at the corners should the lands happen to run in contact with the cylinder. Locked rings permit local blow-by to occur and destruction of the piston in that area may then occur with startling rapidity.

Another worthwhile idea is to chamfer the open end at 45° almost to a knife-edge. This has the effect of skimming excess oil off the bore and directing it up towards the piston crown, whereas a square shoulder just rolls the oil up ahead of itself, creating oil-drag and increasing the work to be done by the oil-control ring.

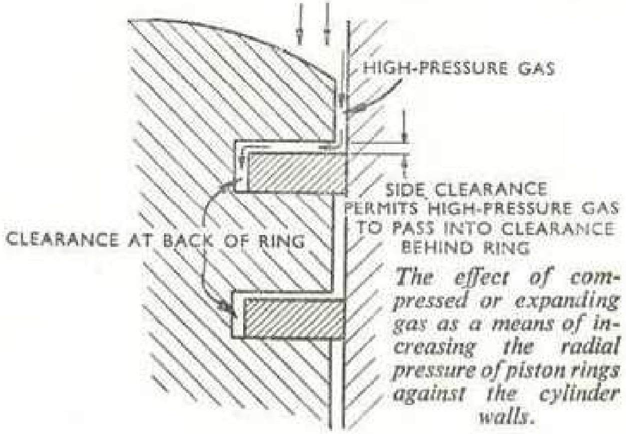
The skirt clearances of two-stroke pistons are of great importance because the piston has to act as the pumping element for the induction system. Pressures of up to 8 lb. per sq. in. are involved in the crankcase breathing cycle, and as there is very little oil present and no rings at all on the skirt, leakage can only be prevented by the latter being a close fit all round its circumference. Extreme ovality leads to loss through the transfer ports, round the piston sides and out through the exhaust. Consequently many such pistons are ground circular or with not more than .003 in. camming, and that concentrated in the region of the gudgeon bosses, leaving the last ½ in., or so, of the skirt parallel and with a clearance of .0015 to .002 in. per inch of diameter.

The bores of air-cooled split-singles distort quite considerably on account of the difficulty of cooling the metal between the bores. It pays to make these pistons a fairly close fit, then, after a warm-up period, to give the motor a short burst at full-throttle, shutting off immediately at the first indication of tightening-up. Then dismantle the engine, ease down the areas of high pressure and repeat. This process will give much better results than any amount of running-in at light load and low temperature.

The most satisfactory ring equipment is two narrow pressure rings, and one scraper, usually of the slotted type, which is so effective that a very copious supply of oil can be fed to the barrel. The pressure rings should have .0015 in. to .002 in. side clearance in the grooves, but scrapers perform their work better with about twice that amount. Oil wiped off the bore is thereby enabled to pass through to the back of the ring and out through the drain-holes. Sometimes a row of drain holes is provided below the oil-ring; these should be horizontal and not drilled downwards at an angle, as the rapid acceleration of the piston away from t.d.c. then has the effect of driving any oil in, or near, the holes back through the piston.

On the other hand, the primary job of the upper rings is to maintain gas-tightness, but this is done not so much by the natural pressure of the rings against the walls as by the action of the high-pressure gas passing through the clearance between ring and groove into the clearance space behind, and thus forcing the ring outwards against the cylinder. If the side clearance is insufficient the gas cannot get through sufficiently fast to build up this vital pressure in the very short time available, but if it is greater than .003–.004 in. there is a likelihood of the grooves being quickly hammered out wider still as the rings change rapidly from one side to the other at each end of the stroke with not much oil present to provide a cushion.

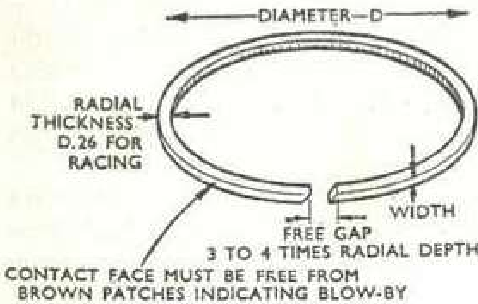
To assist in building up gas-pressure almost instantaneously behind the rings, the grooves should only be .005–.010 in. deeper than the radial thickness of the pressure rings, but much more clearance than that should be allowed behind the oil rings to assist oil to leave via the drain holes. A small clearance helps to damp out flutter which sometimes causes top-ring breakage, and in all cases the sides of rings and grooves must be free from waves or ridges. These defects do not often occur in the grooves unless the piston has seen a great deal of service, but are sometimes noticeable on rings; the remedy is to lap the sides on a flat iron plate with flour emery and finally metal



The effect of compressed or expanding gas as a means of increasing the radial pressure of piston rings against the cylinder walls.

polish, using only light finger pressure. Incidentally, it is better to have dead-flat rings running at slightly excessive clearances than wavy ones with the correct amount of side-clearance.

The grooves in pistons which have been run for some time in severe conditions often wear unevenly. When new rings are fitted in such grooves they may feel as if the side clearance is correct because, being of correct radial depth, they are located in the unworn portion of the grooves. However, under working conditions, the top ring in particular is subjected to a severe twisting action, due to lack of support at the outer edge, and it will almost certainly



Important ring dimensions which must be taken into account when in search of the last ounce of performance.

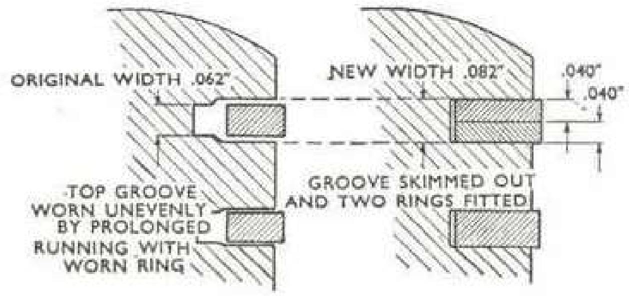
fail in short order. The only cure is to skim the grooves out parallel and, if wider rings are not available, to fit two rings per groove by grinding down standard rings by the requisite amount. There is, of course, a limit to how much can be taken from the grooves because the lands eventually become too thin and may break under load. Definite figures are hard to give but, as a general rule, $\frac{1}{16}$ in. would be about the safe minimum land width and then only if the piston is of strong material such as Y-alloy or R.R. 53.

Effective gas sealing is achieved mostly by the top ring, the second acting as a standby should the top one fail, and also as an oil controller. For sprint work, or in very small engines when there is little room for rings, it is possible to omit the second ring and use a single groove, with two narrow side-by-side rings therein as just described. By placing the gaps on opposite sides minute leakages through this point will be almost entirely eliminated.

A recent development is the Dykes ring, which is of "L" section, the horizontal leg providing location in the piston, while a comparatively large clearance at the top of the ring allows a clear path by which high-pressure gas can flow rapidly into the space behind the vertical portion, and thus expand it against the bore. Only one such ring per piston was employed in the 1954 A.J.S. racing twin, and this practice is continued on the 7R and G50 models.

There is little to choose between the various well-known brands of ring, but each maker has perfected types specially suitable for racing, notably the Wellworthy "Thermochrom" and "Limalloy" or Hepworth and Grandage "Phormicast." Whatever the make, it is advisable to obtain rings with lapped sides, since they maintain correct side clearances for long periods, and there is much less friction between the ring and groove when subjected to heavy gas pressure.

The usual radial depth of English or Continental piston rings is $D/30$ (D is the barrel diameter), but lately high-pressure rings of greater radial depth, $D/26$, have come into use. The latter type are less prone to flutter at high speeds, but if



A piston ring reclaim method which may be used in cases where spares are not available.

they are used care must be taken to see that the ring-grooves are deep enough to accommodate them. If they are fitted in grooves of normal depth there is a likelihood that they will stand proud of the ring lands even when forced down to the bottom of the grooves, and, in service, the ring-faces will then be forced to carry the side thrust which should be borne by the piston skirt. This condition, although easy to overlook, is very bad indeed, and must be rectified by deepening the grooves. Another point about these rings is that, being stiffer, they will not conform to a worn, or oval, barrel quite as freely as will the lighter variety, and it is advisable to lap them in very carefully to ensure gas-tightness if the barrel is not so perfect as it might be.

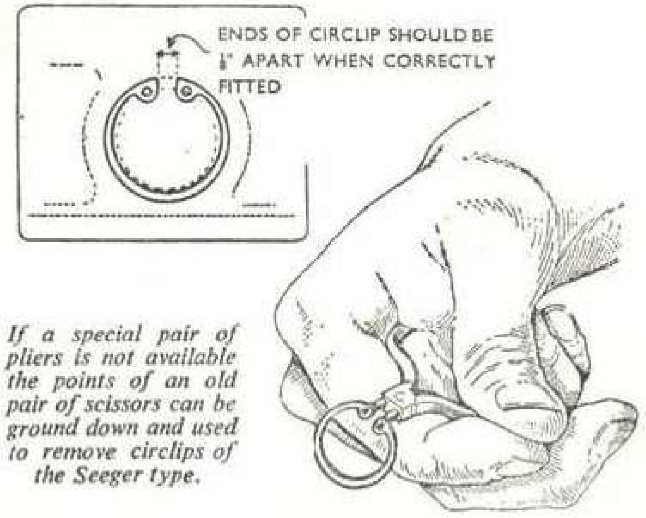
In side-valve engines and most two-strokes some barrel distortion is bound to occur due to unequal distribution of metal and varying temperatures around the cylinder, and in these motors it is a good plan to peg the rings so that they cannot rotate in the grooves. They then bed-in closely to the shape which the cylinder attains when running and will maintain good gas-tightness and oil control even when the cylinder has worn appreciably.

Two-stroke rings are also pegged for another reason—to prevent the ends springing into the ports and becoming broken, and they are also generally made wider in proportion to their depth than four-stroke rings of equivalent diameter to enable them to traverse the ports with less shock. When fitting new rings care must be taken to clean out the corners adjacent to the pegs very thoroughly, otherwise the rings will be propped up above the lands and either they or the barrel will suffer as soon as the engine is started up. The tiny lips on the ends of the Puch rings are very frail and can easily be broken off during assembly if held proud of the piston by the pegs or by carbon.

Wide ring gaps are advisable, particularly for engines subjected to rapid cold starts; .010 in. per inch of diameter is a good figure to work to. To make a thorough job, after rough gapping, lap the rings in to the barrel with metal polish until they are bedding uniformly all the way round, and finally finish the gaps to the correct width.

Regarding gudgeon pins, it is a great aid to assembly to make these an easy hand-push fit in the piston when cold. Some makers prefer to fit them a little tighter, but this entails driving the pin out with a drift, or warming up the piston with hot water or a bit of meta-fuel in a cocoa-tin lid; as in any case a $\frac{3}{4}$-in. pin will have about .002-in. clearance at running temperature, another .0005-in. will make very little difference. The best way to take a thou. or so out of gudgeon pin holes is with a hone of the Sunnen or Delapena type, using kerosene or paraffin liberally. It is not easy to take out small amounts with an ordinary reamer without leaving small chatter-marks in the bore: indeed, sometimes a better job can be made with a triangular hand scraper.

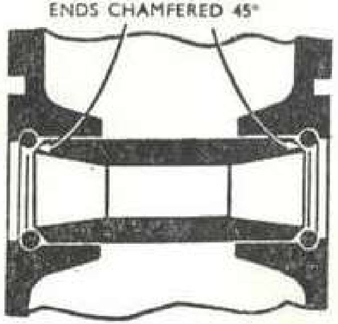
The main attributes of a gudgeon pin are stiffness and high surface hardness; pins which have become "blued" are softened and should be replaced. Insufficient pin rigidity is a fruitful source of cracked piston bosses, and the most satisfactory designs have a central bore not greater than 0.6 of their diameter. Although this can be tapered out for one-third the length of the pin at both ends, the bore must be left soft and is preferably polished to remove surface blemishes from which cracks may start. A nickel-



If a special pair of pliers is not available the points of an old pair of scissors can be ground down and used to remove circlips of the Seeger type.

chrome steel such as K.E. 287, case-hardened, .025 in. -.035 in. deep, is an excellent material for the purpose.

The best form of end location is by circlips, either of the spring-wire type (without bent-in ears, which sometimes break off) or the Seeger pattern, although for fitting the latter variety you must have the right sort of pliers, available quite cheaply from the makers of the circlips. The wire type is perfectly satisfactory provided the ends of the gudgeon pin are chamfered, and the overall pin length is



Chamfered outer gudgeon pin edges act as circlip retainers, whilst the tapered bore, left soft, reduces weight and is polished so that flaws may be detected.

such that the final circlip can just be sprung into place with the pin thrust hard against the opposite one. The chamfers then act as retainers, and there is no likelihood of the circlips ever managing to work out of the grooves. Seeger circlips require flat-bottomed grooves turned to accurate depths in accordance with the table which covers the range of sizes most likely to be met. This type of circlip must never be fitted in the half-round grooves used with wire circlips because, although it may seem to fit correctly, in fact it would only be seating on the corners and in a short period of running the corners dig in, the circlip becomes loose and eventually comes right out of the groove, with disastrous results.

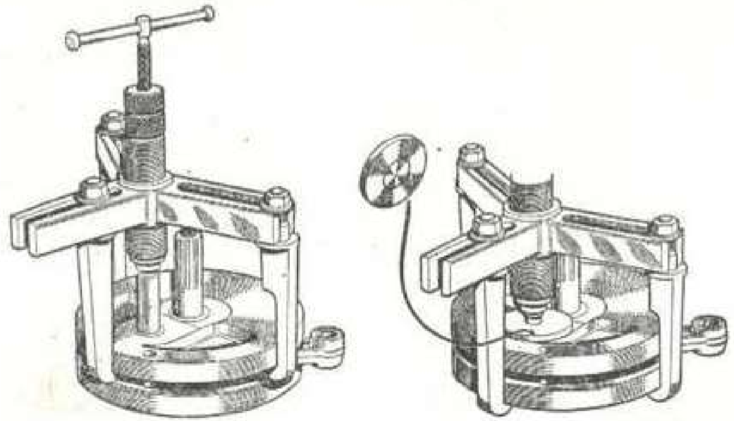
Wire circlips should always be stretched out before refitting to make sure that they will fit tightly in place. Occasionally an engine will exhibit a tendency to hammer the circlips loose, more so on one side than the other; in such a case examine the alignment of the small end bush for both squareness and absence of "twist"; the latter is difficult to detect but may be the unsuspected cause of the trouble.

If a spare is not available at all, or the standard article is known to be unreliable, a new piston can be made from a solid bar with only a lathe, a drill, and the usual hand tools. The process is rather too long for inclusion here but as a guide it will help to note that the best material is aluminium alloy R.R. 59, which is specially made for pistons and retains strength very well at high temperatures. The American material to SAE260 specification, and known commercially as 14S also is very suitable. Alloys such as R.R. 77, which are even stronger at atmospheric temperatures, contain zinc and, therefore, are not so good at higher temperatures.

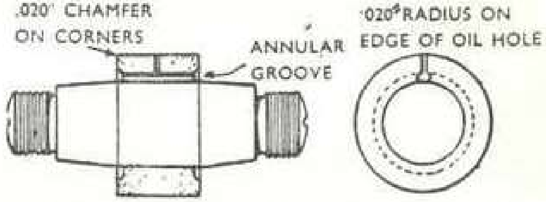
CHAPTER VI

THE CONNECTING-ROD AND BIG-END

The function of the flywheel assembly is to convert the straight-line motion of the piston into rotary motion at the engine sprocket. The crankcase also forms a vital link in this mechanical chain by providing a rigid location for the cylinder and mainshaft bearings, whilst, in addition, housing timing gears, oil pumps and so forth. Referring back to Chapter I, it was pointed out that any areas where undue wear has occurred are open to suspicion and that the cause of the wear (which may be malalignment, lack of lubrication or faulty material) must be investigated with an eye to its cure. This advice should come in useful at this stage, for the main job of work to be tackled in the "basement" will consist of the elimination of friction usually due to malalignment.



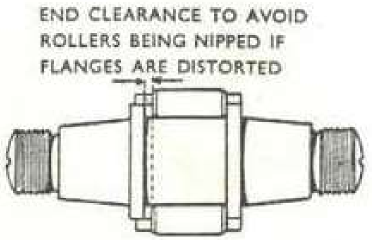
The Velocette slow-taper crankpin-flywheel assembly calls for the use of a specially modified "Pickavant" hydraulic tool when dismantling (left) or reassembling (right), unless a power press is available.



A worn big-end assembly can be renovated by grinding down and fitting a press-fit hardened sleeve to the crank-pin. Where the pin is not drilled centrally for lubrication, the sleeve must be provided with annular grooves and a transverse passage.

The big-end is the most highly-stressed bearing of all, and if there is any sign, however small, of flaking of the roller track or the crank-pin, that component should be discarded; wear on the areas rubbed by the cage is not of such great detriment as long as it does not encroach on to the track.

Some crankpins (e.g. in certain Velocette, A.J.S. and Matchless engines) are made in two pieces with a hardened sleeve pressed on a tough steel centre, and if new complete pins are unobtainable, new sleeves can be made up, and any pin in which the central portion is considerably larger than the ends can be repaired by grinding down this portion and fitting a similar sleeve. If the outer race in the con.-rod is also worn, this should first be ground or lapped out true—preferably without removing it from the rod—after which operation the pin-sleeves can be ground to the particular oversize necessary to suit the finished outer race diameter and whatever rollers are available.



With the shouldered type of crankpin there must be end-float between rollers and shoulders to avoid danger of the former becoming nipped when the pin is tightened in the flywheels.

The pin-sleeve should preferably be made from ball-race steel (1% carbon, 1% chromium), which hardens right through by quenching in oil after soaking at 820° C. for half an hour. After quenching, temper for half an hour at 200° C. and cool in air; this procedure relieves internal stresses without materially affecting the hardness, which should be Rockwell C. 60-64 or diamond hardness number 750-800.

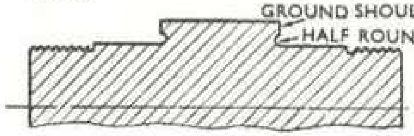
If the crankpin is already drilled centrally for oil, a mating hole must also be drilled in the sleeve before hardening. Where the pin is not drilled, an annular groove should be formed at one or both ends of the sleeve to collect lubricant from the flywheel oil-hole, a shallow transverse groove with central hole finally delivering it to the rollers as shown in an accompanying sketch. After hardening, the sleeve must be ground internally to a diameter .001 in. smaller than the pin, then forced into position under a press. Finally, it must be ground externally and on the two end-faces; squareness of the latter is *particularly* important, as the slightest error will be greatly magnified at the mainshafts.

If carbon-chrome steel is not available, a case-hardening variety can be used, in which event the bore can be left soft if desired, thus avoiding the necessity for internal grinding. The scheme is first to rough-turn the blank to grinding sizes and then to carburize for six hours to obtain a case-depth at least .045 in. After cooling, rough-bore then recess the end faces to $\frac{1}{8}$ in. larger diameter than the bore to a depth of $\frac{1}{16}$ in., drill the oil hole and finally heat-treat. After hardening, the sleeve can be finished internally either by boring or grinding, fitted to the pin and then finish-ground exactly as previously described.

Mild steel is not up to the arduous nature of this work, the case being liable to flake away from the soft core; 3% nickel to specification S15 or EN 33, or single-quench 3% or $3\frac{1}{2}$% nickel will be satisfactory for moderately heavy duty, but better varieties are S82, S90, EN 34, 36 and 39, the last three being wartime specifications. The heat-treatment for all these grades is practically the same, viz.: carburize at 900-920° C. Refine by quenching in oil from

850-860° C. Harden by quenching in oil from 760° C. Temper by cooling in air from 150-170° C., except for the 3% and 3½% nickel single-quench grades, in which the refining and hardening can be effected simultaneously by quenching in oil or water from 760-780° C.

Equivalent American specifications are S.A.E. 51100 for carbon-chrome ball-race steel, S.A.E. 2317 for 3½% nickel case-hardening steel and S.A.E. 3316 for 3½% nickel-chrome C.H. steel. If it is necessary to make up a completely new pin, either of the two case-hardening steels mentioned will do. The plain nickel steel, though having less core strength, gives a slightly harder case than the nickel-chrome steel. Steels which harden right through, such as the ball-race steel mentioned or medium-carbon direct-hardening steels must not on any account be used for complete pins, because, when hardened sufficiently on the roller track, such steels are far too brittle in the core and are certain to break in service. In any case the threaded ends must always be left soft and this can be achieved by copper-plating prior to carburizing and hardening. Another method is to rough-turn the threaded portions .100 in. oversize, carburize the whole pin, turn off the carburized skin locally and screw-cut the threads and finally harden. Incidentally, great care must be taken to ensure that the threads are square to the axis, otherwise it may be difficult to maintain tightness of the nuts.

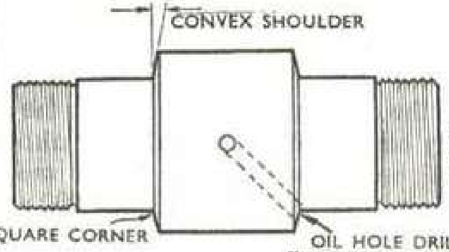


This method of machining a groove and providing a ground shoulder obviates a possible weakness without reducing the effective length of spigot.

Breakage of crank-pins of the shouldered type almost invariably occurs in the corner of the shoulder, and it is desirable to have a definite fillet in this corner, even at the expense of reducing the length of bearing in the flywheels. The faces of the shoulders should be slightly concave; on no account may they be convex even to the slightest

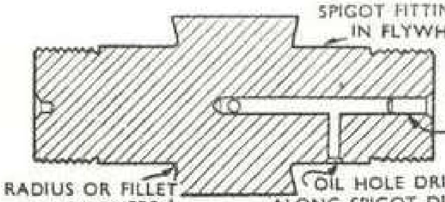
extent. Unfortunately, with normal grinding equipment, it is difficult to attain this finish, but somewhat the same effect can be gained by turning a shallow undercut in the faces which also provides a little extra room for the vital fillet.

Details of crankpin design which are likely to cause weakness under high-duty conditions.



Sometimes the oil-hole in the pin is drilled inwards from the corner of the shoulder, causing a local weakness from which a fatigue crack will start. It is best to modify the drilling to avoid this danger; it may also be necessary to drill another hole in the flywheel to coincide with the new position of the hole in the pin (*see* illustration below).

When reassembling the big-end new rollers should be fitted for preference, but old ones can be used, provided they are absolutely free from flats or other surface blemishes (which can be detected by examining them through a magnifying glass under a fairly strong, oblique light) and are all of the same diameter to within one or two tenths of



a thou. One or two oversize rollers will rapidly ruin any roller bearing, this being the reason why new and old rollers should *never* be run together.

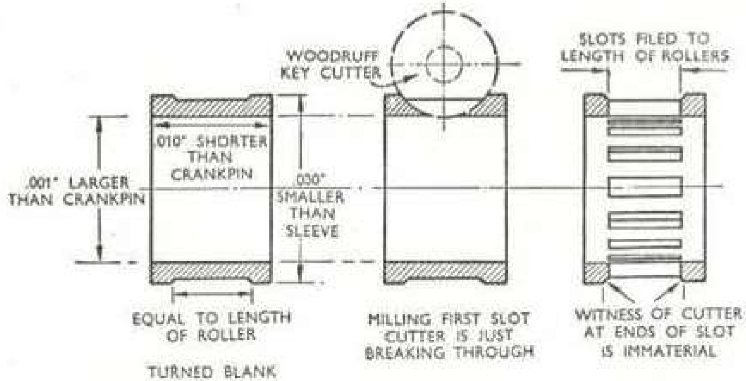
Caged rollers are maintained in parallel by the spacing bars, and, if the pin is of the shouldered type, the rollers can have several thou. end clearance—in fact, this is advisable, otherwise if any deflection takes place the shoulders may close in a trifle and nip the rollers endways. If the pin is without shoulders, the bars of the cage should be relieved on the internal diameter so that the cage bears on the pin only outside the roller path, for although the cage is soft it will wear away the pin surface much faster than will the rollers. In engines running at 7,000 r.p.m. or more, this form of wear can become very serious, and it may be advisable to hard-chrome-plate the rubbing areas; .001 to .002 in. thickness will be ample, but the deposit must be finally ground to a dead-smooth finish and must on no account be allowed to encroach on the roller tracks. Useful as it is in other ways, hard-chrome cannot stand up to the action of heavily loaded rollers and will inevitably flake off if applied to the tracks.

What makes the life of a big-end so strenuous is not so much the speed or the loading but the fact that the roller assembly does not rotate round the pin at a steady speed, as it does in a main bearing. Instead, it has alternately to speed up and slow down because at or near t.d.c. the swing of the connecting-rod is in opposition to the motion of the pin, whereas at bottom dead centre the two motions are in the same direction though of course not at the same speed. Like all objects possessing weight, the roller assembly objects to this process and relatively heavy loads are created between the rollers and the bars of the cage and a certain amount of sliding motion takes place in addition to normal rolling, when the engine speed exceeds a figure at which the friction between rollers and tracks is insufficient to overcome the inertia of the roller assembly.

The lighter the assembly, the less its inertia, and for that reason racing cages are made of one of the strong wrought

aluminium alloys such as R.R. 56 or 24 S. Some cages are made from cast aluminium which, though light, is relatively weak and for serious work should be replaced by parts made of bar or tube. Factory-made cages have the slots formed by broaching, a method which is not available to everyone; an alternative process is to end-mill the slots and then machine the ends out square. A cage with relatively long slots to accommodate long rollers such as $\frac{3}{16}$ in. by $\frac{9}{16}$ in. (a very useful size) or three short rollers placed in each slot can easily be made by milling the slots with a standard Woodruff key-seat cutter, and finishing the ends out by hand using a square file ground to width so that it does not damage the machined sides. The blank should be turned with a shoulder or groove at each end to mark the length of the slots, which is not vitally important. What is important, however, is that the slots must be dead parallel to the axis, otherwise the whole roller assembly will run over to one side and, therefore, accurate setting-up of the milling machine and dividing head is essential.

Large uncaged rollers are not good for high-speed work because of the heavy pressure between them due to centrifugal force. This force can be reduced to an acceptable figure by using rollers which are small in diameter and



Three stages in the fabrication of a big-end roller bearing cage, using bar or tube material.

consequently light. The Vincent big-end uses three rows of rollers 3 mm. diameter by 5 mm. long per rod, separated by parallel rings which prevent the rollers skewing; the total end clearance must be held to limits between .004 in. and .008 in. if skewing or binding endways is to be avoided. The 250 Guzzi also used small rollers of 3 mm. diameter by approximately ⅝ in. long with no cage, but in this instance the big-end is split, just as if the rod had a plain bearing. Plain big-ends have been successfully used for some time now on high-speed engines though they are necessarily smaller and more heavily loaded than their counterparts in the automobile world. Rigidity of the metal surrounding the big-end is of more importance than lightness, therefore no attempts at lightening or altering the design of plain rods should be attempted without consultation with the makers, who in some instances supply rods of different design for racing. The correct clearance for plain big-ends is .001 in. to .0015 in. per inch when the rods are cold though a much greater clearance is permissible provided the oil pressure is maintained above the allowable minimum figure as quoted by the makers.

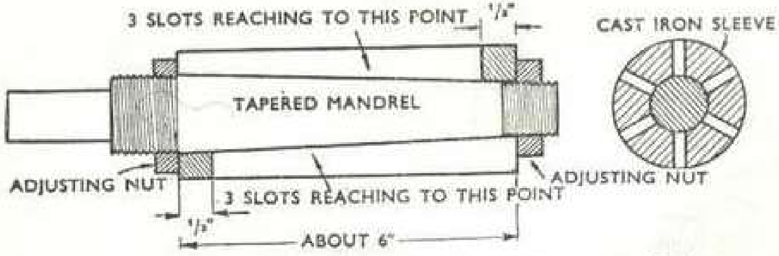
In any design of roller bearing there is bound to be a certain amount of rubbing going on, which, at ultra-high speeds, will generate sufficient heat to cause seizure unless it is carried away by the lubricant. For reasons of strength, the oil-holes are never very large, and care must be taken to check that they are clear by feeding oil through them with a force-feed oil can. Incidentally, when doing up a "vintage" model, it is often possible to obtain a more up-to-date big-end which will fit the old pattern wheels with little or no modification.

After assembling the rollers with oil (*not* grease, which is likely to block the oil-holes) the con.-rod should just slide into place. If it has to be pushed or screwed on, the fit is too tight; if there is more than .002-in. diametral clearance it is too loose. The best way to get an exact fit is by means of several sets of rollers graduated in steps of .0001 in. but if these are unobtainable the outer race can be lapped out

with flour emery and a mild-steel lap, but on *no* account must any attempt be made to ease a tight big-end by rotating it with abrasive applied to the rollers; such a procedure will almost certainly lead to rapid breakdown in service. A piece of hard wood turned to size makes quite an effective lap for emergency use, but when many rods of the same diameter have to be handled, it pays to make up an expanding lap. This tool can either be held in a vice and the rod turned by hand or it can be used in a drill; in either case the rod should be allowed to "float" on the lap and be frequently removed and reversed to avoid tapering the bore.

When spun round with the pin held horizontally, the rod should not show the slightest sign of working over to one side; if it does, either the pin or the outer race is tapered or the slots in the cage are at a minute angle. Reversing the rod may effect a cure, in which event mark the correct position of pin, rod and cage; if not, the cause should be located and rectified. Sometimes deep grooves are formed in the flywheels through rods running over, and if these are present they should be skimmed out to accommodate phosphor-bronze or hard-steel washers of suitable thickness. The correct amount of side-clearance varies according to the design, but is rarely of great importance, .010 in. to .020 in. being about right in most cases.

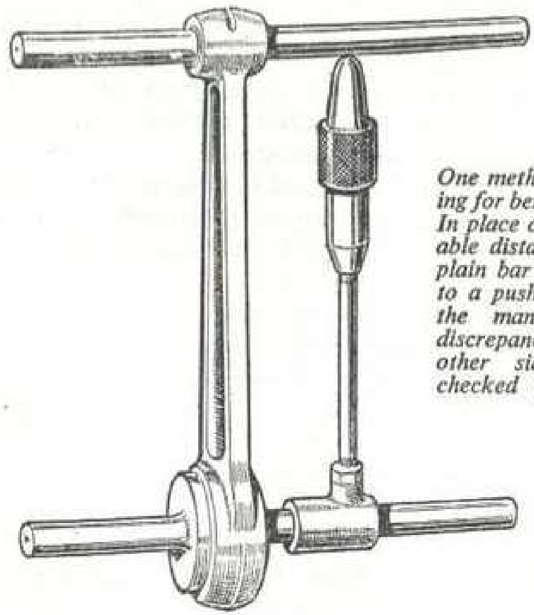
If not already done, it is a good idea to polish the con.-rod all over, since the removal of the rough outer skin greatly diminishes the chance of fatigue-cracks developing.



Details of a lapping tool of the split-adjustable type suitable for the cleaning up of the big-end eye.

Although rare, cracks of this nature usually occur at the junction of the shank and small-end. They can be detected by immersing the rod in hot paraffin for some time, and, after drying, dusting the surface with french chalk; if any cracks are present they will be indicated by the chalk adhering to the paraffin retained in them. Cracks may also be detected by the Magnaflux, or similar methods, which are commonly employed in aircraft manufacturing or maintenance establishments and it is a good idea to avail oneself of such facilities if they are easily accessible.

Shot-peening the surface is of great value in preventing fatigue-cracks—the system is to bombard the surface with steel shot, projected at high velocity by compressed air through a nozzle. This process must not be confused with shot-blasting, which uses sharp-edged grains and removes a certain amount of metal. The *smooth* shot used in shot-peening removes no metal at all but, through its compacting action, places the surface skin of the metal in compression.

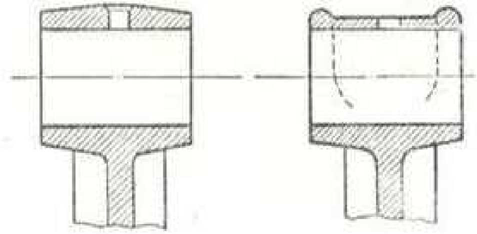


One method of checking for bend in the rod. In place of the adjustable distance piece, a plain bar can be filed to a push fit between the mandrels. Any discrepancy on the other side can be checked with feelers.

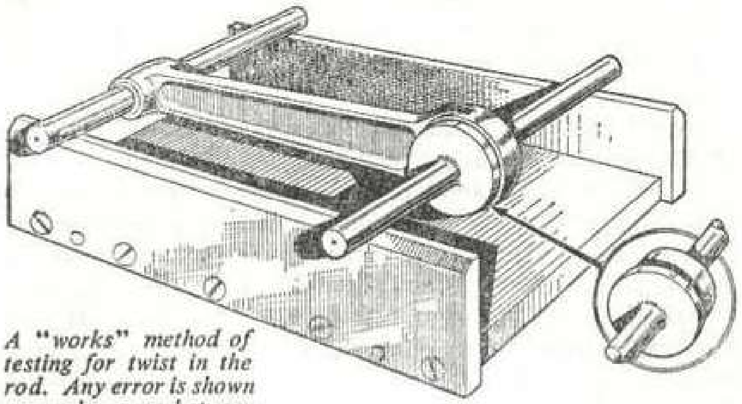
Obviously, shot-peening must be done *after* polishing, as any work done on the surface after peening will remove the compressed layer and destroy the effectiveness of the process.

Experiments in lightening rods extensively are *not* advisable, unless the maker's advice has been obtained. There is sometimes a fair amount of excess metal round the small-end due to taper in the forging dies and manufacturing tolerances and much of this can safely be removed by filing. As the stress concentration falls mainly on the extremities of the small-end, these should be left untouched. It may be possible to obtain a lighter rod made specially for racing from higher-grade steel; as one example, the early KTT Velocette rod can be used in the push-rod MOV engine. "Manx" Norton rods can also be used in "International" angines, and Vincent rods, with the big-ends narrowed cen be used in speedway J.A.P.s.

A guide to the amount of material which may be removed from the small-end eye for lightening purposes.



Rods must be tested carefully for alignment in all planes. Checking off the sides of the big-end, the small-end should be central within .005 in. to .008 in. The big-end and small-end bores must be dead parallel; although it is possible to carry out an approximate check for this by using the crank pin and gudgeon pin as mandrels, a better way is to make up a pair of bars, at least 6 in. long, which are a tight push fit in the respective bores. For preference, they should be hardened and ground, but soft ones will suffice if carefully handled. Commercial bright-drawn bar is usually .003 in. less than nominal size and quite effective mandrels for use



A "works" method of testing for twist in the rod. Any error is shown as a clearance between one mandrel and the knife-edge jig.

in small-ends can be made from such material, hard-chrome plated up to finished size.

With these mandrels in place, the distance between them measured at the extremities should not differ by more than .002 in. In the absence of the necessary measuring equipment, a steel bar can be filed so that it will just fit between the mandrels at the tightest end; the gap at the larger end can then be measured by feeler gauges. The bearings must also be free from " wind " or twist, i.e., the mandrels should be parallel when viewed along the length of the rods. Accuracy in this respect can be checked quite easily on a surface-plate by the use of blocks and a dial gauge or feeler gauges.

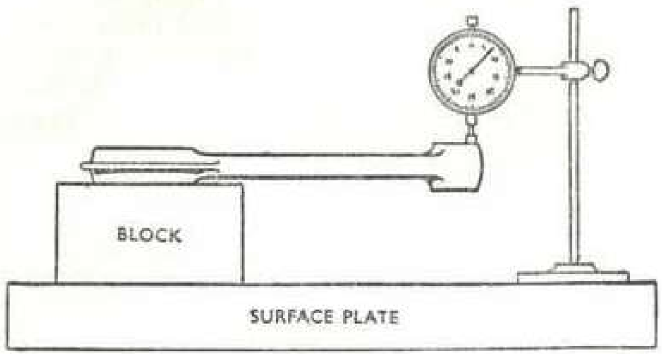
It is quite likely that at least one of these errors will be found, possibly all three. To avoid pulling the rod about unnecessarily, the three checks should first be made and the situation carefully weighed up, otherwise one may cure one error by making another worse. As a general rule, first eliminate any " twist " and then set about getting the centrality and parallelism of the small-end correct. If it is central but inclined, bend the rod close to the small-end;

if off-centre and inclined in the same direction, bend near the big-end. The rod should never be gripped by the big-end sleeve when being "set"; this action might just cant the sleeve in the bore and in service it would revert to its original position. Setting should preferably be done in a straightening press, but failing that a strong vice and three bolts can be used, two being placed on one side of the rod and one between them at the point where the bend is required; the bolts should be of large diameter and soft to avoid local damage to the rod. Alternatively, narrow strips of thick brass can be used with much less possibility of damage.

If these errors are in opposite directions, the rod will have to be set at both ends, but do *not* use the checking mandrels for the purpose and bend these as well. Since the side clearance between small-end and piston bosses can be fairly generous, it is permissible to face off one side of the small-end if all the errors cannot be corrected otherwise.

Considerable patience may be needed, but it is vitally important that the con.-rod be absolutely true; even the slightest error can have a serious effect at very high r.p.m.

Rods which have been set only slightly often show a tendency to revert to their original position, so the best



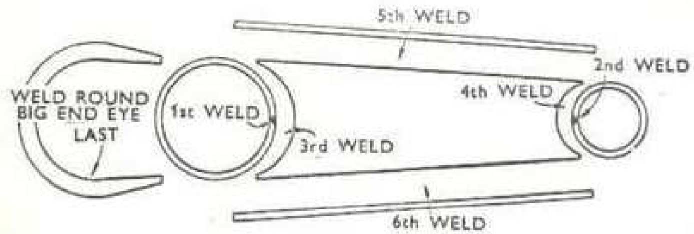
A third method of checking for bend in the rod. Dial gauge should show same reading when rod is reversed.

plan is to set the rod too far and then correct the amount of overset to bring the alignment right. Rods treated in this manner are unlikely to lose alignment again.

The small-end bush for racing must not have less than .001 in. clearance, but more can be allowed without detriment except for the noise caused. Nortons can be permitted up to .005 in. before replacing the bush. Phosphor bronze is the usual material but any of the light alloys, such as R.R. 56 or R.R. 59 or Duralumin H, which retain strength at high temperatures, make very good bearings and save perhaps an ounce of reciprocating weight.

It occasionally becomes advisable to make up a completely new connecting-rod, and at the same time to redesign for greater rigidity. In this connection, the best design is the simplest; the flanges of the I-section should be tangential to the small-end boss, and taper outwards to a width equal to the bore of the big-end eye. Circumferential rigidity of the big-end is most important, and to obtain this without excessive weight there should be either one or two deep ribs running right around the eye. For material, "Vibrac" V 30 is extremely good, and for best results the rod should be rough machined in the soft state, then heat treated to 85 tons tensile. All polishing of the exterior should then be completed and the big- and small-ends finally ground to size. Rods which are not likely to be run at ultra-high revs, do not require a steel of this nature, and carbon-manganese steel to Spec EN 16, heat-treated to 65 tons tensile, gives a very good result; it is a standard steel used in many English machines built after 1945. It can be machined in the fully heat-treated state and, therefore, does not require to be ground in the bores after hardening, which makes the manufacture somewhat simpler.

Where extreme lightness is sought, the weight of the big-end sleeve can be saved by making the rod from 5% nickel or 3½% nickel-chrome case-hardening steel, case-hardened to a minimum of .045 in. in the big-end eye only, the rest of the rod being kept soft by copper plating before carburizing. Many two-stroke rods are made on this



The author's idea of a connecting rod built of high-tensile sheet steel and fabricated by welding.

principle and it was standard practice in Rudges for many years and is to-day on many Italian engines.

Although it seems a rather revolutionary suggestion, the writer sees no reason why it should not be possible to fabricate a rod from high-tensile alloy steel sheet, united by welding. If the small and large ends are composed of strips cut along the grain of the steel and bent to shape, and the flanges of the I-section are carried round as shown in the diagram, the grain flow in every part of the rod would be ideal, which is not always the case with one-off hand forgings. The joints could be made with filler rod of the same analysis as the components and the whole assembly heat-treated before finally machining the rod to size.

Aluminium rods are not normally fitted with small-end bushes, but if badly worn, the holes can be opened out and fitted with aluminium or B.R.56 bushes; bronze bushes almost invariably come loose in time.

Some steel rods have been made without small-end bushes, the eye being either hardened and ground or simply left soft. This idea is satisfactory up to a point, but if a seizure occurs, it may be almost impossible to extract the pin. A good way to avoid this without incurring the extra weight of a bush is to adopt a system developed for the heavily-loaded articulated con.-rods of radial aircraft engines, which consists of silver-plating the small-end bores with silver to a thickness of .002 in.

CHAPTER VII

TRUING AND BALANCING FLYWHEELS

THE maximum of rigidity in the flywheel and crankpin assembly is essential to avoid power-wastage through internal vibration. This is obtained partly by providing heavy sections of metal in the regions of maximum stress, and partly by the manner in which the assembly is held together.

Some years ago it was common practice to grind fairly steep tapers on the pin, and to pull these into the wheels by fine-thread nuts. This method of construction makes a good job, provided that the tapers fit really accurately; thus, in doing-up such an engine (particularly if a new pin is being used) the fit should be checked by means of prussian-blue smeared on the pin, which is then lightly rotated in the holes. If contact is not made over the whole internal surface of each hole the parts can be lapped together, using a fine-grit abrasive, but great care must be taken to see that the holes are not lapped out-of-square, in which event the second state will be worse than the first. This process will cause the wheels to come a little closer together and the side-float of the con.-rod will be reduced by a like amount, thus the clearance must be checked carefully on final assembly. The tapered crank-pins used in J.A.P. Speedway engines are supplied in three lengths, standard, plus $\frac{1}{32}$ in., and plus $\frac{1}{16}$ in., to allow for any enlargement that may, in time, occur in the holes. By selecting the correct length of pin, the con.-rod side-clearance can be maintained within the correct limits of .015 in. to .030 in.

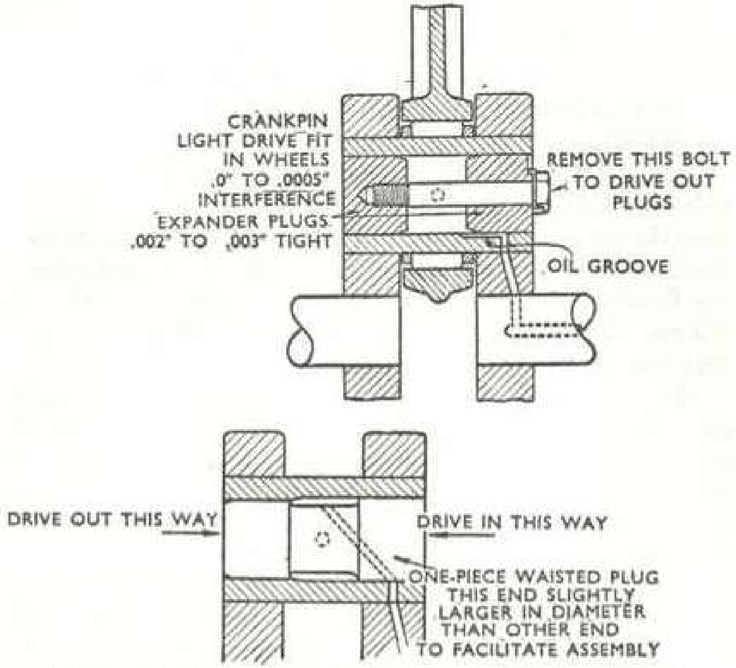
It is now more usual to pull the wheels up against substantial shoulders on the pin, rigidity being gained more by this action than by the actual fit of the reduced portions of

the pin in the wheels. If parallel, these are usually made a light drive fit (i.e., about .001 in. larger than the hole), and if for any reason the fit is much looser than this, the pin can be hard-nickel-plated and ground to correct size. Early Velocette pins are ground with an almost imperceptible taper of .001 in. per inch length, and should push in by hand for half the depth of the flywheel hole; the nuts will then do the rest, but if the pin will not go in so far as that, the shoulders may not be pulled up into hard contact with the flywheel as intended by the makers and not only will the assembly lack rigidity but there will be a grave danger that the crankpin will break in service.

Whatever the form and fit of the pin, the whole assembly eventually depends upon the nuts, which should be examined for possible damage or distortion of the threads and scrapped if there is any doubt about them. The abutting faces must be square to the threads and can be checked and, if necessary, rectified by turning up a mandrel with a thread tightly fitting the nuts, which are then screwed on and skimmed up dead flat. If it is necessary to make new nuts because spares are unobtainable, mild steel is not good enough: nothing less than 45-ton tensile steel should be used, and alloy steel of 55–65 tons such as K.E. 805 is better still for racing. Most English crankpin nuts are tapped either 20 or 26 threads per inch, Whitworth form, the commonest sizes being $\frac{3}{4}$ in., $\frac{7}{8}$ in. and 1 in. and, therefore, it is often possible to utilize nuts of another make if genuine spares are not available.

Recently there has been a growing tendency to do away with nuts altogether and rely simply upon a press fit. This system was introduced on Velocette and Villiers two-strokes, and has been adopted for the 86 mm. four-stroke Velocettes. In this case, the pins are tapered .008 in. per inch and should push a little less than half-way by hand; 4 tons pressure should then be required to force the wheels hard against the shoulders.

An excellent method of eliminating nuts which is worth consideration by experimenters is the S.K.F. system of using



Two types of expander-plug fitting to supersede a conventional crankpin.

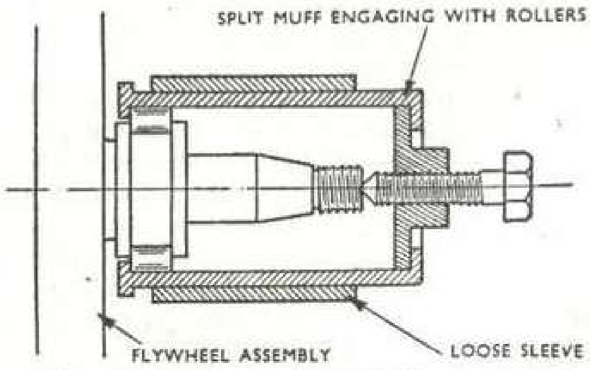
expander plugs in a hollow pin or shaft. The scheme is to make the shaft a tight push fit in the hole so that initial assembly is easy with no danger of scoring. Then a hard steel expander plug is forced into the bore, which is about one third to one half the diameter of the shaft, and the latter is thereby expanded and locked firmly in the flywheels. Diagrams show two methods by which this scheme could be applied. A somewhat similar idea was used in the 1954 A.J.S. racing twin though, in this instance, the expander plugs had a fairly steep taper.

Regarding the flywheels, there is very little point in going in for a big course of weight reduction; in fact, it is possible to reduce rather than increase the performance by such a proceeding, except perhaps for speedway work. Some

riders find they get better results with light flywheels on short tracks where acceleration out of the corners is of paramount importance, but for road work, where full use can be made of the gearbox and clutch, there is not much in it. However, a general cleaning-up of all surfaces will not be a waste of time and is of decided benefit on the rim surfaces, as oil-drag is perceptibly reduced by a high polish.

Another method of reducing oil-drag is to chamfer the outside diameter of the rims, leaving only a narrow land next the inner faces running in close proximity to the edge of the crankcase scraper. As the majority of the oil to be skimmed off travels down the inner faces of the flywheels, the scraping action is not unduly impaired but remember that, during assembly, the wheels cannot be roughly aligned by a straight-edge across the rims if this modification is performed.

The correct fit of the main-bearing inner races on the shafts varies according to type; in no circumstances should there be any actual slack present, but if the races are locked up endways in some manner, they can be quite an easy fit. If not so retained, there must be sufficient interference between bore and shaft to prevent " creep " which, once it commences, will cause the shafts to wear, particularly if



Split puller for removing inner main-bearing races.

they are not case-hardened; a light drive fit is the ideal at which to aim.

One snag about these tight-fitting races is the difficulty of removal without damage; applying two levers to the roller-cage will almost inevitably distort it, thus nipping one or two rollers endwise, and this defect should always be looked for in old engines. The best scheme is to make up a split puller which cannot harm the cage, but, failing that, two sharp-edged chisels ground to a narrow taper may be driven in from opposite sides between the race and flywheel until there is a gap wide enough to insert a pair of levers. Of course, before reassembling, any burrs raised on the faces must be filed off flush.

On all models, Velocette mainshafts have an almost imperceptible taper of .001 in. per inch on the bearing seats; the inner bearing races are also ground to the same taper and to a diameter such that they have to be lightly driven on for the last quarter-inch, or so. If looseness develops, the inner race may appear to fit correctly if it is reversed on the shaft but this must *not* be done on any account, as the race is then only in contact at one end and will rock about under load.

If slackness is found, do not adopt the barbarous process of centre-punching or chiselling the shaft, which, however good it may seem at the time, ceases to be effective after a very short while. Building-up the shaft by nickel-plating or metal-spraying with zinc are both effective and lasting repair methods and are necessary for these and similar shafts which are forced into the flywheels and permanently locked by grub-screws. Though the course is not advisable, Vincent shafts can be pressed out after driving out the Mills pin fitted through at an angle. The correct interference fit is .003 in. and this must be verified before pressing-in the new shafts. On the drive side, press in the new shaft hard up to the shoulder, with the angle-hole about 60° away from the original position, then drill back through the shaft at 45° with a $\frac{5}{16}$ in. drill, and fit a new Mills pin.

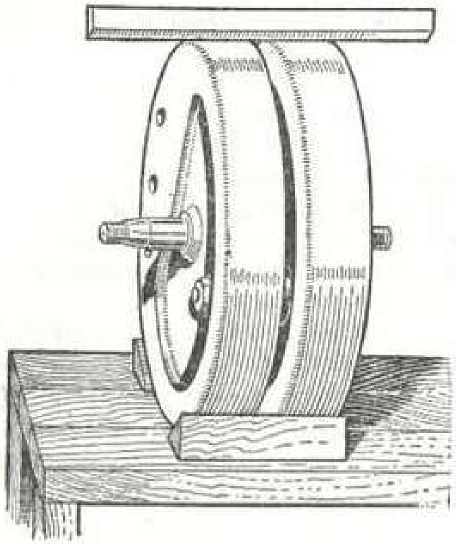
Where the main bearing rollers run direct on the shafts, as

in J.A.P. engines, renewal is the only course open if wear shown by pitting of the roller tracks is present. As previously noted, chrome-plating cannot be used as a reclaiming process for roller races. Take particular care to line up the oil-holes in shaft and wheel accurately on the timing side, and, as a precaution, check that the axial distances of the hole from the face of the wheel and from the shoulder of the shaft are the same, otherwise the oil flow may be restricted. If any error is found, full flow-area can be obtained by grooving in the required direction with a half-round chisel.

It is essential after new shafts have been fitted, and advisable even if not, to check, the truth of each wheel before final assembly commences. One method is to apply a dial indicator to the outer side of the wheel near the rim with the shaft supported between lathe centres; another is to rotate the flywheel with the shaft running in an accurately-bored bush, or even in its own bearings. If there is more than .003 in. run-out, the shaft will have to be set in the requisite direction; it will be impossible to line both shafts up accurately, if one, or both, are not dead square to their own wheels.

Strictly speaking, any balancing which is thought to be necessary should be done at this stage so that each wheel can be dealt with individually; it is, however, easier to work on the wheels after they are assembled, so unless two odd wheels are being made into a pair the job can safely be left until later, on the assumption that they were individually balanced by the makers in the first place.

After all work on the components has been completed, assembly is a straight-forward job, although it is easy to overlook the obvious precaution of ensuring that the oil-holes in the crankpin and timing side wheel are accurately in line. This precaution is not required when fitting pins with annular oil grooves, which should be placed with the hole leading to the rollers at the 3 o'clock position, looking towards the drive-side with the pin at the top, this being the position of least load. Cleanliness is essential, not only for the sake of the bearing but also to ensure that no particles of foreign matter are trapped between the

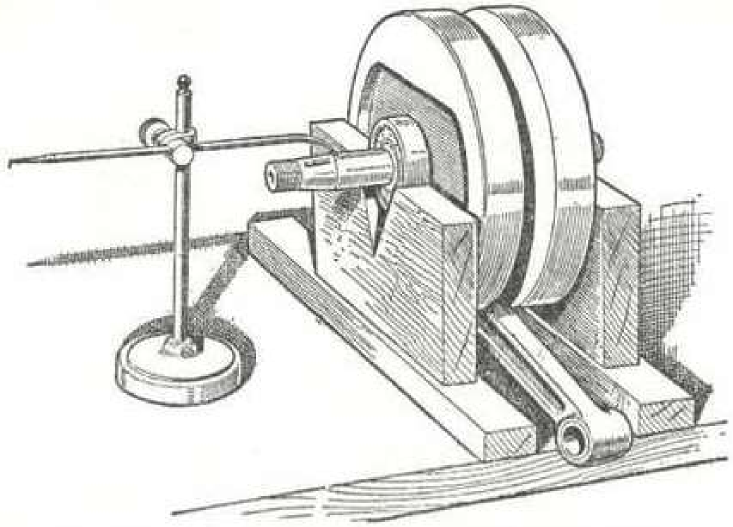


A simple method of rough-aligning the flywheels with the aid of two wooden chocks and a straight-edge before final checking and tightening.

abutting faces to the detriment of mainshaft alignment.

If the pin is of the solid type, it can be inserted and the nut fully tightened on one wheel only before assembling the rest, but if it is of the two-piece variety with a pressed-on sleeve, it is not advisable to tighten either nut fully unless the other wheel and nut are in place, otherwise the pin may be pulled over too far to one side, leaving insufficient thread for the nut on the other end. Great care must be taken with tight parallel-fitting pins to avoid damaging the hole when fitting the second wheel, which is liable to cant over due to its overhung weight. The temptation to pull the wheel on with the nut as soon as a couple of threads project through must be resisted, as so doing will overload, and very probably damage, the thread inside the nut.

The Velocette pressed-in pin is best fitted first to the drive-side wheel and forced home under a press with a ring supporting the flywheel as sometimes the pin may project a little. After fitting the con-rod, the other wheel is then partially pressed on and the wheels lined up as accurately as possible with the end of a straight-edge laid across

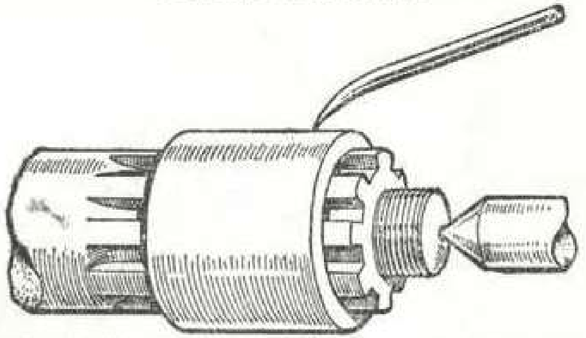


Where a pair of centres is not available, the flywheel assembly may be supported on a pair of bearings in a simple wooden jig. A steel base plate makes for added accuracy.

the rims. Similarly, with other designs, initial lining-up is facilitated if the second nut is not quite fully tightened. Final tightening or pressing home is done after correct alignment has been achieved.

The best way to move the wheels relative to each other is to hold the assembly in both hands and bump the rim of one wheel on a heavy lead block or the end grain of a hardwood post; the inertia of the wheels does the trick and no damage is done to the rims, as there might be if they were struck with a hammer.

For final tightening a properly fitting box-spanner with a strong tommy-bar at least 2 ft. long is essential; the normal type of tubular spanner is not really up to the job (unless reinforced by a brazed-on ring turned to clear the crankpin nut counterbore), but if made from chrome-molybdenum steel tube, this type will do. The best spanners, of course, are those made from solid steel with integral tommy-bars, which can be purchased from any good small-tool factor.



When checking a splined mainshaft for truth with a scriber or dial gauge, an externally ground sleeve, which is a push fit on the shaft, will prove of great assistance in making accurate measurements.

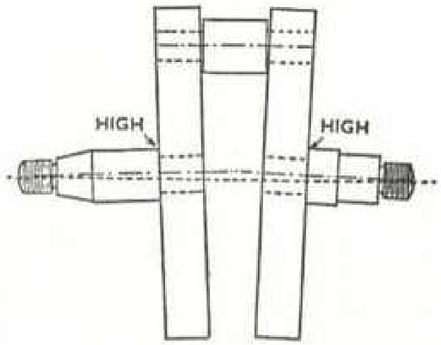
Real strength must be put into the final tightening, something of the order of 400 pounds-feet being necessary.

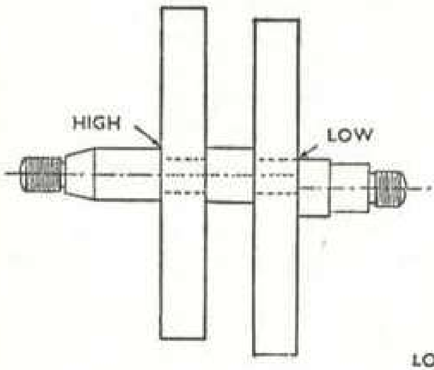
The final check must be made on the main shafts, using an indicator reading to thousandths of an inch. If a lathe or a pair of centres is available, the assembly should be mounted thereon, taking care that the wheels are not deflected inwards by excessive pressure applied to the centres. Next apply the indicator to each shaft as near as possible to the ends, as all too frequently the centres in these shafts get knocked about; any defect will immediately be shown by the indicator, and must be rectified by scraping, lapping or re-turning the centres until concentricity is obtained. A number of engines have one or both shafts splined, and in such cases a tightly fitting, accurately ground concentric sleeve fitted over each spline will greatly assist matters.

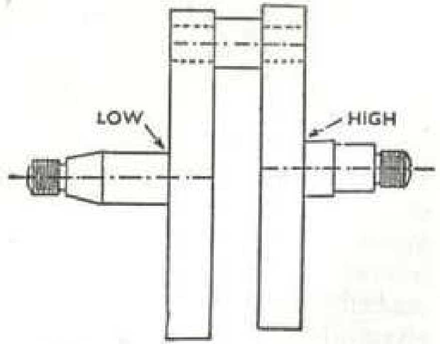
The next move is to check the shafts close up to the wheels; this will probably show one shaft to be "high" and the other "low" in a direction at right angles to the crankpin. If this is the only error present the situation is good, because bumping the wheels in the appropriate direction will (albeit after quite a number of shots) eliminate it entirely, but if in spite of all your efforts both shafts are "high" in the *same* direction, the whole assembly is in effect "bent" in the centre.

This may be due to swarf between wheels and pin, lack of parallelism between the outer faces of the pin shoulders, or similar causes. If not due to dirt, the trouble can sometimes be cured by refitting the pin in a different position should the oil-hole arrangements permit. Failing that, some very careful work can be done with a scraper applied to the flywheel faces, but most people adopt the easy way of nipping the rims in a vice, in the plane in which both shafts are "high." This process is all right if the wheels are of steel and the pin is robust, but is not to be recommended if the pin is of light section or the wheels are cast iron.

These deliberately exaggerated diagrams depict three faults which may be discovered upon assembling the wheels. Their causes and the appropriate remedial action are outlined in the text.







(Top right) Wheels out of parallel. (Above) Wheels out of alignment, an easily corrected fault. (Right) Odd wheels, necessitating a difficult machining operation.

There is a third error which is rarely met with unless the wheels were not originally a pair, and that is when one shaft is high and the other low in the plane of the crankpin. This is almost certainly due to the radius of the pin-hole in one wheel differing slightly from that in the other and can only be remedied by a machine shop operation.

If a pair of centres is not available, or the shaft-centres are too badly damaged to be usable, another method of checking is to support the assembly by its own or a similar pair of main-bearings resting in V-blocks. Alternatively, if there are only two bearings the crankcase itself can be utilized, although it is rather laborious having to separate the whole issue in order to bump the wheels whilst carrying out the truing process—needless to say, it is a bad plan to attempt to shift them while supported in their own bearings. If there are two fairly widely spaced bearings on either side of the case the job is a little easier, as then one shaft can be run in these bearings, leaving the other wheel and shaft completely exposed for checking. These last two methods are, however, only suggested as makeshifts where facilities for employing one of the others are unavailable. Another workshop method is to bore out a block to a close running fit on the shafts and rotate the assembly with one shaft fitting in the hole and the other in the air, a dial gauge fitted to an arm attached to the block indicating the truth or otherwise of the free shaft. One shaft is first tried, and then the other, and this method is just as good as doing the job between centres except that one cannot check both shafts at the same setting.

The question now arises—how true should the shafts be? Well, the correct amount of error is zero, but this is rarely attainable, and if the sum of the errors indicated on both shafts comes to less than .002 in. there is nothing much to grumble about. Two-bearing assemblies can run with a greater error than those with three or four, particularly where the latter are housed in a very stiff crankcase. Having made the shafts as true as they can be made, the nuts must be given a last nip-up, and the shafts finally rechecked. If all

is well, a couple of squirts of oil up through the holes drilled in flywheel and pin to clear out any possible grit will finish the job off, unless for any reason, such as a big change in piston weight, you wish to rebalance the engine.

The Balance Factor

The lower half of the rod can be considered as rotating weight, and it, together with the crankpin, can be completely balanced by an equal and opposite counter-weight, or by removing metal from the crankpin side. But it is not possible to do this with the reciprocating weight in a single-cylinder or parallel-twin engine for, if an equal and opposite weight were added, the balance would be good in the direction of the cylinder axis, but just as bad as before along a line at right angles to the cylinder. Hence a compromise must be effected by adding only a certain percentage of the reciprocating weight, this figure being termed the balance factor.

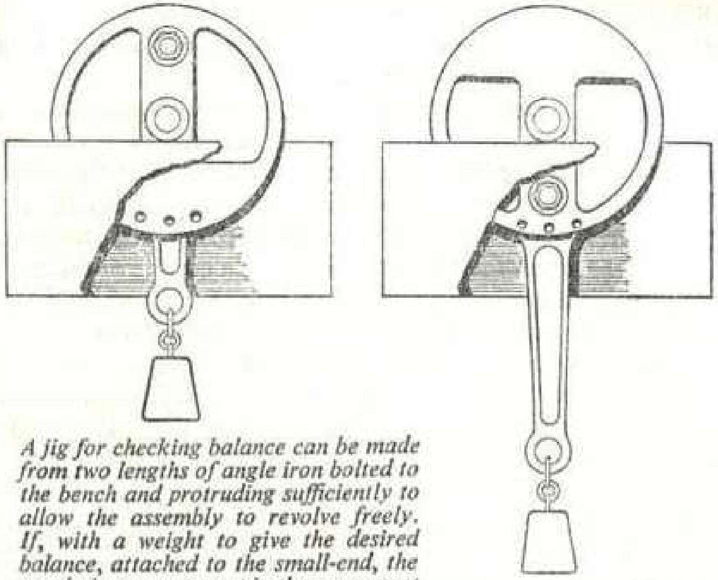
It is quite useless to postulate any particular balance factor as being the ideal; so many considerations enter into the matter that it varies with almost every design of engine, or even the type of frame in which engines of the same kind are mounted. That being so, do not be misled into rebalancing your engine just because one of your pals with an entirely different machine thinks he has some magic formula of his own.

The only source of reliable information is the parent factory but, failing advice from that direction, a factor of .66 of the reciprocating weight usually gives reasonable results. Strangely enough, some makers use a smaller factor for racing than for touring, while others do the reverse; the idea in all cases, however, is to get an engine which runs most smoothly in the speed-range at which it is intended to operate for the majority of its life. It does not matter much if the engine feels "rough" at 4,000 r.p.m. if it is to be raced and feels smooth at 6,000, whereas such an engine would be very undesirable for fast touring, where much of the running is done at the lower r.p.m. Altering the balance factor will usually succeed in moving the rough period, if any, well away from the most-used speed.

The aforesaid balance-factor applies, of course, to the reciprocating parts only—i.e., piston, rings, gudgeon pin and *the top half* of the connecting rod. A lot of weighing and measuring can be eliminated by dealing with the complete flywheel assembly only, thus-wise. Arrange it on a level surface so that the con.-rod (minus piston) is lying *horizontally*, with the small end resting on one side of a pair of scales, or supported by an accurate light-spring balance. If the balance pan is other than flat the rod must be supported in the plane of the gudgeon pin. This is best done by passing a short bar through the bush and resting the former on the edges of the pan. First, a few nuts and washers can be placed in the other pan to counterbalance the short bar and then there is no likelihood of getting confused with the other weights used to find the weight of the small end, which we will suppose is found to be 6 oz. Next weigh the complete piston assembly, which comes to, say, 15 oz. Thus the total reciprocating weight is 15 + 6 = 21 oz. and employing a factor of .66 the amount to be balanced is, therefore, 14 oz.

The wheels are already bound to be balanced to *some* percentage, and if they are placed with the shafts resting on a pair of accurately horizontal metal straight-edges they will eventually come to rest with the crankpin vertically upwards, although they may show a tendency for the pin always to be slightly to one side of the dead centre-line. This indicates that the bob-weights are off-centre, a fault which must first be rectified by drilling equal-sized holes into the side of each rim, in a position at right angles to the pin and, of course, on the "heavy" side of the vertical centre line.

When symmetrical balance has finally been obtained, attach to the small end a weight equal to the amount to be balanced *minus* that of the small end; using the figures quoted above, this would be 14 — 6 = 8 oz. The manner of



A jig for checking balance can be made from two lengths of angle iron bolted to the bench and protruding sufficiently to allow the assembly to revolve freely. If, with a weight to give the desired balance, attached to the small-end, the crankpin comes to rest in the uppermost position, the counterweights must be drilled as shown on the left. If the crankpin stops at the bottom the wheels must be drilled adjacent to the pin.

making up this weight or of attaching it to the small end is purely a matter of choice—it can be, for instance, a bag full of oddments, or a bolt with the requisite weight of washers. If the balance-factor does actually correspond to the figure desired, the wheels will roll freely along the straight-edges and show no tendency to settle in any one position; if not, the pin will go to the top or bottom according to whether the bobweights are too heavy or too light.

Correction is usually made by drilling the rims in the appropriate positions, being careful to take equal amounts out of each wheel and on each side of the centre plane, but it can equally well be done by tapping and plugging existing holes. If a bit of experimenting to find the best balance is part of the tuning programme, it is a good idea to drill or tap a few holes of, say, $\frac{3}{8}$-in. or $\frac{1}{2}$-in. diameter, into which plugs of the required weight can subsequently be fitted or removed. In some engines it is possible to drill a hole somewhere in each crankcase wall below the main-bearings, and tap it ½ in. gas (or more correctly, ¼ in. B.S.P.), thread which is the size commonly employed for drain plugs, and drill several holes in the flywheels at the same radius, tapping these out ⅛ in. B.S.P. Plugs of various lengths in steel or bronze can then be inserted or removed from the flywheels through the crankcase holes in a couple of minutes without disturbing the engine and experiments to find the most satisfactory balance can be very rapidly conducted. This is a very good scheme to employ when adapting engines for use in 500 c.c. racing cars, because the different method of mounting as compared to a motorcycle very often leads to trouble in obtaining smooth running.

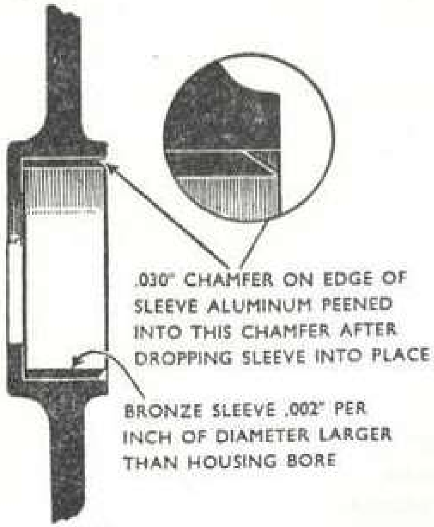
In closing, a word or two about those "horizontal straight-edges." For permanent workshop use it is nice to make up a proper jig, but there is no necessity to go to such lengths. There are plenty of other ways: for instance, two lengths of angle iron can be bolted to the bench-top overhanging the latter by about a foot. The only vital qualification is that the top edges must be flat, smooth and absolutely horizontal and level with each other when the wheels are resting on them. A good spirit level is the best aid to checking this point, but failing that a dead-round bar such as a piece of silver steel will prove the point, since it will obviously tend to run towards whichever is the lower end. Most machine-shops equipped with cylindrical grinders possess a pair of straight edges on which the grinding wheels are balanced and, with a little blandishment exercised in the right quarters, it is often possible to obtain the use of such equipment.

On A.J.S. 7R and G50 models, special circular nuts are used, which must be split (and are thus destroyed) to remove them. Replacement nuts must always be on hand if the big-end is to be inspected; these are made with a reduced hexagonal portion which must be sawn off after the wheels are fully tightened.

CHAPTER VIII

WORK ON THE CRANKCASE

THE primary duty of the crankcase is to provide a rigid mounting for the main bearings and a foundation for the cylinder sufficiently solid as to maintain that component square to the mainshaft axis at all times, irrespective of temperature changes or cyclic load variations. The rigidity of the bearings is settled by the original design; thus in the search for speed little can be done other than to see that the outer races are a good fit in the case when the latter is hot. If they are not, the races will show signs of "creep" indicated by a polished appearance of the outer surface. If there is any doubt about the matter, each side of the case, complete with bearings, can be immersed in boiling water, after which the bearing rings should not be free enough to turn by hand.



How worn main-bearing housings may be renovated by the insertion of a shrunk-in bronze liner. (Inset) How the edge of the liner is chamfered to permit of peening the aluminium surround.

Several methods are available for curing loose races, of which the easiest—tinning the surface—is not to be recommended, as it is almost impossible to get an even coating, and in any case it inevitably gives away after a while. Chromium or nickel-plating the races is satisfactory, provided that the finished outer surface is absolutely circular and true to the bore; if only about .001 in. thickness of deposit is required to restore the fit it will not need to be finished to size if the job is done by a competent plating shop, but deposits of greater thickness will need to be finally sized by grinding owing to the tendency of these metals to build up more thickly at the edges than at the centre. Copper-plating on the other hand does not show this tendency and can easily be cleaned up to size with a file whilst the race is rotated in a lathe chuck. Owing to its softness, copper plating does not always last very long but is such a convenient method that it is often utilized.

Probably the best method of effecting a permanent cure is to bring the housing back to its original size by boring it out and fitting a bronze sleeve about $\frac{1}{16}$ in. thick and .002 in. per inch of diameter larger than the housing. Having heated up the crankcase to 200° C. (400° F.)—which can be done with the aid of a domestic oven—the sleeve can be dropped into place, and after peening the aluminium over the edge as shown in the diagram, the sleeve is then finish-bored to .001 in. per inch smaller than the race. A job such as this requires to be done extremely accurately; the crankcase must be set up in the lathe with the joint face and spigot diameter both running dead true. In some instances, one or more of the outer races are clamped endways, either by plates or ring nuts, in which event they cannot turn or work out sideways, and, provided there is no actual looseness perceptible, the diametral fit need not be so tight. "Manx" Norton bearing sleeves, besides being retained in this manner are machined with grooves on the sides, into which the metal of the retaining plate is punched to form keys to prevent rotation. When refitting these sleeves they must be

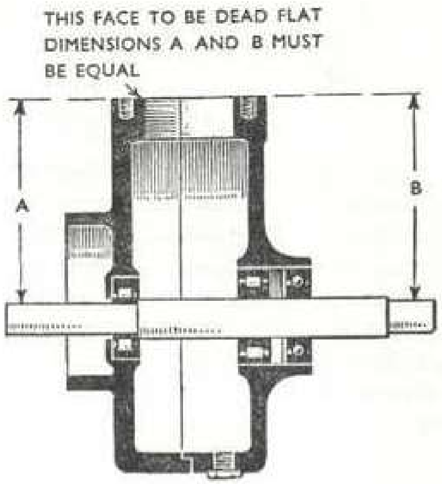
placed with the grooves visible and the retaining-plate punched in to them in the prescribed manner.

Main Bearing Clearances

If any bearings need replacement through excessive wear or pitting of the race tracks, remember that the outer races close in by approximately half the amount of interference between them and the case. Consequently, the bearings should have a slight amount of slackness in them before fitting, otherwise the rollers or balls may be subjected to a damaging overload, even though the inner races may still *seem* to turn quite easily. Some makers supply bearings with various amounts of diametral clearance to allow for the tightness of shaft or housing, and mark the races accordingly. Two small polished circles on the edge of a Hoffmann outer race, for example, indicate that it has the right clearance if the outer race is tight, but the inner a push fit; three circles are visible if both races are intended to be tight. Apart from this indication, the inner assembly of a roller bearing should just slide into place by hand if the fit is correct, and ball-races, when free of oil, should spin for several turns. Most ball-bearings, rotated dry, emit slight noise but, if the noise sounds irregular or rough, the condition of the bearing is open to suspicion and its replacement is the wisest course. The inner tracks of roller bearings can be examined by prising a roller out of the cage.

Another important matter is the alignment of the bearings, particularly if there are two in either or both halves of the case. The only check of any real value is to turn or grind a mandrel to a size (or sizes) which will just push through the inner bearings without shake, and then insert this through the whole set of bearings with all the crankcase bolts tightened up. The mandrel should push through without effort and turn freely when in place; if not, the error must be tracked down and eliminated.

Sometimes in old engines the spigot on one half-case is a poor fit in the register in the other half, and relative move-



The studs should be removed, the case placed mouth downwards on a surface plate and the distances A and B then checked for equality with a mandrel. The end diameters of the mandrel must be of equal size.

ment which would alter the alignment can take place. This can be prevented by reaming out two existing bolt-holes to accommodate special oversize bolts acting as dowels; before reaming, the cases must be tapped lightly this way and that until the best position is attained, as indicated by the freeness of the test-mandrel.

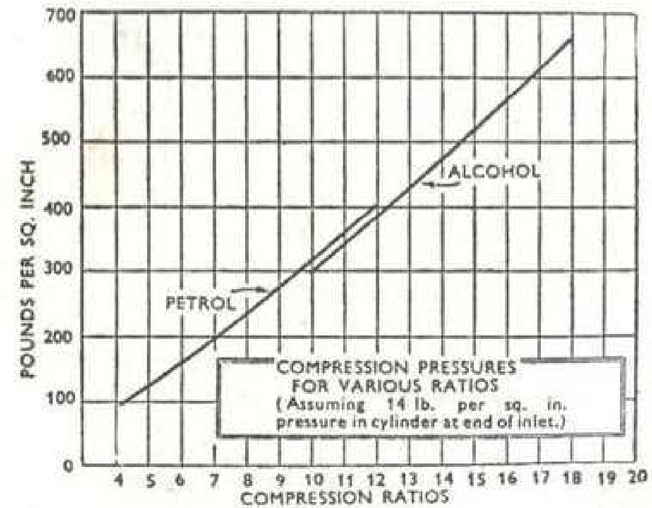
Fortunately, since it is a difficult fault to eradicate, incorrect bearing alignment is a rare disease, but it should be looked for just the same.

If plain bearings are included in the make-up and have been renewed, it is best to leave the bores a few thous. undersize and finally line-ream with the reamer piloted in the bearing on the opposite side. When a bearing of this type is used to transfer the main oil supply to the crankshaft the fit is particularly important and the maker's instructions regarding clearances must be rigorously followed. Naturally any communicating oil-holes must be accurately lined-up and checked by injecting oil through them from a pressure can.

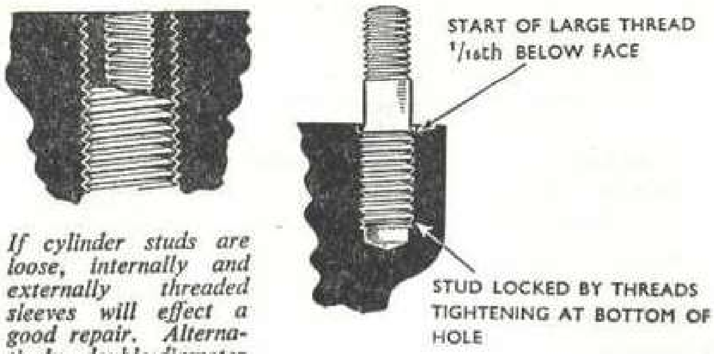
The test-mandrel also comes in useful for checking the cylinder-base, which must be parallel to the main-shaft centre line in addition to being flat and free from any step

at the joint. Presupposing that it is accurate as to flatness, parallelism can be checked by inverting the case on a surface-plate and measuring the distance on each side between the plate and mandrel. Should the bearings be of differing bores on each side, a short length of the mandrel at each end can be turned to exactly equal diameters—which avoids having to make allowance for varying sizes. If the face is not flat it must be re-machined or trued up with file and scraper until it is correct in all respects. Lack of flatness is bad; it permits oil leaks, allows the barrel to rock about under load, and can be an unsuspected cause of barrel distortion or cracked base-flanges.

At this stage it is as well to cast a critical eye over the cylinder studs—particularly if the compression ratio is being raised considerably, because the gas pressures will be greatly increased thereby; the accompanying graph shows the actual rise of compression pressure with increase of ratio, and as explosion pressures are some $3\frac{1}{2}$ to 4 times



A study of this graph will reveal the reason why cylinder studs should be examined when the compression ratio is raised.



If cylinder studs are loose, internally and externally threaded sleeves will effect a good repair. Alternatively, double-diameter studs can be made up.

the compression pressures, it will be appreciated that a good deal depends upon the cylinder fixing in an ultra-high compression motor.

Almost invariably the studs are tapped direct into the aluminium. As the latter metal is relatively weak, the length of thread in engagement should preferably be twice—and certainly at least one and a half times—the stud diameter. In addition, the threads should tighten at the *bottom* and not at the top of the hole; in the latter case the run-out of the stud thread exerts a powerful wedging action on the surrounding boss, which may be the cause of a crack making an unwelcome appearance either at the first onset or later on when the engine is put to work.

Should the crankcase threads appear to be damaged or partially stripped, there is usually enough metal present to permit drilling and tapping to a greater depth. If not, two courses are available. One is to re-tap to a size 1/16 in. larger and fit a double-diameter, or "bull-headed", stud; for example, a stripped hole previously 5/16-in. Whitworth can be re-tapped to 3/8-in. Whitworth, having first opened it out with a letter N drill. In other cases it may be preferable to retain the existing size of stud by tapping out the hole to take a bronze or steel bush, screwed inside and out to the appropriate sizes, this being particularly advisable if there

TAPPING SIZES FOR VARIOUS THREADS

SIZE	PITCH THREADS PER INCH	CORRECT DRILL SIZE	NEAREST FRACTIONAL DRILL SIZE
2 B.A.	31.4	No. 23	5/32 in.
1/4-in. B.S.F.	26	No. 5	13/64 in.
1/4-in. Whit.	20	No. 8	13/64 in.
5/16-in. B.S.F.	22	Letter G	17/64 in.
5/16-in. Whit.	18	Letter F	1/4 in.
3/8-in. B.S.F.	20	Letter O or P	21/64 in.
3/8-in. Whit.	16	Letter N	5/16 in.
1/8-in. B.S.P.	28	11/32 in.	—
1/4-in. B.S.P.	19	29/64 in.	—
3/8-in. B.S.P.	19	37/64 in.	—
7/16-in. B.S.F.	18	3/8 in.	—
7/16-in. Whit.	14	23/64 in.	—
1/2-in. B.S.F.	16	27/64 in.	—
1/2-in. Whit.	12	13/32 in.	—
9/16-in. C.E.I.	20	33/64 in.	—

is any future likelihood of the studs being removed and replaced at intervals.

For the external threads on these sleeves, either 1/4-in. or 3/8-in. gas threads—both having 19 threads per inch—are very suitable. The table on this page, which may come in useful for other jobs, gives the tapping sizes and drill designations for a number of pitches.

Still another scheme is to instal Armstrong or Cross wire inserts, which somewhat resemble springs made from steel wire of diamond section, and are screwed into an oversized tapped hole and finally locked in by punching or "staking" the end coil.

Timing-side Mechanism

It is now time to give a little attention to the mechanism contained in the timing-side case, to wit, the timing-gear and oil-pump. Despite their relatively small size and heavy loading, the camshaft bearings in push-rod engines give very little trouble, and unless they are in a very bad state it is best not to disturb them. Provided the surfaces are not torn or worn into ridges, clearances up to .005 in. are permissible, and if new bushes are needed they should be reamed to .002 in. clearance, except in the rare instances where they form part of the lubrication system, when slightly less clearance is necessary in order to prevent the escape of too much oil.

The modern tendency is to provide fixed spindles upon which the cam-wheels rotate, the spindles being drilled for lubrication, and it is an easy matter to make sure that all oil-ways are clear and that the wheels spin freely. Designs in which the outer cam-bearings are located in the timing cover are not so easy to check or to rectify if incorrect, the chief difficulty being to get both pairs of bearings in line simultaneously.

If the cover is attached only by set-screws its location may not be sufficiently positive as to ensure that it remains in the correct position indefinitely when under the influence of irregular load and severe vibration. This contingency can be entirely prevented by fitting a couple of dowels, which need only be quite small—$\frac{1}{8}$-in. diameter will do if there is no room to fit a larger size—the holes being drilled and reamed after the cover has been worked into a position where the cam-wheels are at maximum freeness *with all screws tight*; with journals and bushes free from oil, either wheel should spin for several turns if given a sharp flick with the fingers.

The dowels should be a light drive fit in one component and a good push fit in the other, otherwise it is difficult to get the cover off subsequently. If there is no room to fit dowels, two of the existing holes can be modified to accept screws with close-fitting plain shanks, which will serve the same purpose.

Replacement bushes are almost invariably bored slightly undersize to allow for reaming. This operation must be done with a piloted reamer to ensure alignment, and

occasionally a special jig-plate is really required to do the job properly. Failing this device, which is only worth having if it is extremely accurately made, the bushes, after being fitted with the crankcase or cover heated, can be reamed a little on the small side, and then eased out with a scraper, using prussian blue on the shafts to show up the tight spots. Do not attempt to lap the shafts in with emery powder, as this will embed itself into the soft bearing metal and subsequently cause rapid wear of the shafts. In the matter of end-play, it is better to err on the generous side rather than have too little, except when the gears have helical teeth, in which event only .001 in. to .003 in. should be allowed. Excess play in the timing gears of Velocette engines is the frequent cause of harsh, noisy running.

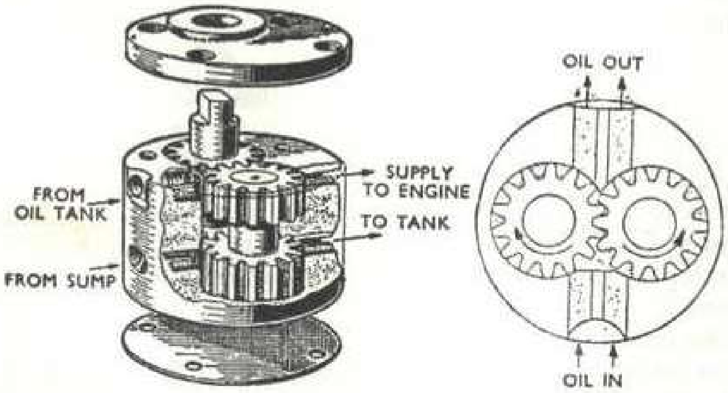
Several modern engines, the A.J.S., Matchless and Vincent for example, employ porous-bronze bushes for camshafts and also at other localities. These bushes should never be reamed if at all possible; the correct method is to size them, after fitting, with a planishing broach—i.e. one with no cutting edges—or else bore them with a sharp single-point tool. The ordinary shop reamer almost invariably smears the surface instead of cutting and thus blocks up the pores in the metal, though a very keen reamer will not do much harm if the amount of metal to be removed is very slight.

Increased Lubrication

Lubrication of the timing-gear is not always well carried out, and can frequently be improved by chiselling extra V-grooves leading to oil-holes in the bushes, or by arranging an extra oil-feed to the timing-chest. In dry-sump engines an easy way to accomplish this is to take a lead off the pipe which returns oil to the tank and couple this up to a union and internal jet in the timing-case placed where it will do the most good—preferably so that the oil drops directly on to the cams. Naturally there is a limit to how much oil can be re-circulated in this way, and the flow needs to be restricted, otherwise the scavenge pump will be overloaded.

A fixed jet with an orifice about .030 in. in diameter should suffice to start with, or alternatively an Enots adjustable drip-feed can be used. Small jet orifices are rather prone to blockage when return-oil is being fed through them, but this can be largely overcome by using a larger hole into which a loose wire or split pin is fitted. This scheme is used by Vincents in the rocker-feed jets, but, for racing, the restrictor wires should be removed from the exhaust valve feeds and also from the inlet-valve feeds if the valve guides are in good condition.

If the lubrication is of the total-loss type, it may be worth while fitting an additional sight-feed pump, or if the existing pump is of the single-delivery type, to replace it by a duplex pattern, the aim being to let one pump feed the big-end while the second takes care of the timing gear. Each supply can then be adjusted to suit its own department without risk of swamping or starving other parts of the motor. This of course is the system used on J.A.P. Speedway and 8-80 engines which are intended for short-distance work, but full dry-sump systems are used on these engines for long-distance racing and Formula III cars.

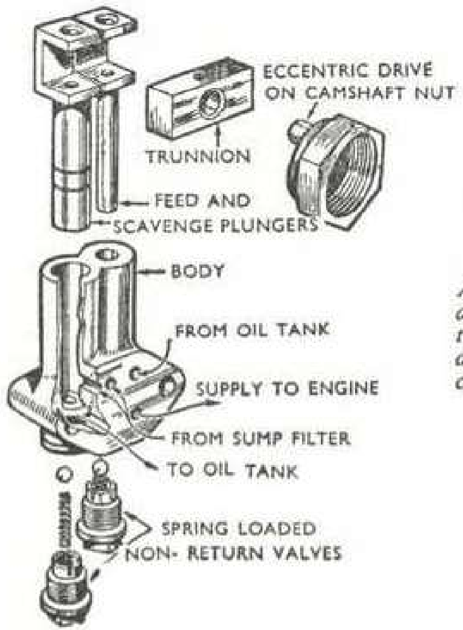


The efficiency of a gear-type oil-pump depends upon the minimum possible side clearance between wheels and case. The case faces can be rubbed down when clearance becomes excessive.

The oil-pump in dry-sump engines is usually housed directly in the crankcase, and in most o.h-camshaft designs is tightly fitted into a recess bored in the back of the timing-chest. To remove, it is necessary to heat up the case and tap it, timing-chest down, on the bench-top. It may sometimes be necessary to run a tap into two of the retaining screw holes, so that a couple of screws can be inserted by which the pump can be pulled out. The internal bore of the housing must never be enlarged to make refitting of the pump easier, as this is almost certain to lead to leakage between the pressure and scavenge side of the pump. As a rule, gear-pumps give very little trouble, unless a piece of metal has found its way in and damaged the teeth or the gear-pockets, but in time side wear will develop and reduce the delivery rate.

It is sometimes thought that backlash between the teeth affects the pumping, but it makes no difference at all—in fact, for smooth functioning at least .005 in. backlash should be present. The side clearance, however, should not exceed .001 in. (particularly if the gears are thin in relation to their diameter), and if more is present the faces of the pump body should be very carefully dressed to the required amount. Should a paper packing be a standard fitting, the simplest course is to lap the pump-face on a surface-plate with the gears in place until the whole surface, including that of gears, is flat and level; the paper packing will subsequently provide the requisite clearance. All push-rod Velocettes have double-gear pumps, somewhat similar to those in the o.h. camshaft models, but driven at a much slower speed by a single-start bronze worm-wheel engaging with a straight-cut pinion on the pump spindle, the pump housing being bored at a small angle to give correct engagement between worm and pinion teeth. On later models the pump speed was doubled by using a two-start worm and helically-cut pinion; and it is possible to utilize the later design of gearing to increase the oil-flow of earlier engines.

In those pumps which have a plunger which rotates and



As in the case of the double-gear wheel pump, the plunger type employs a scavenge outlet with a capacity larger than that of the feed pump.

reciprocates, the points to note are the condition of the gear-teeth, the fit of the plunger portions in their respective housings at each end, and the state of the small driving peg which is screwed into the crankcase and engages with an inclined groove machined in the plunger. As these pumps turn at a relatively low speed, wear is very slow as a general rule; it is a wise plan, nevertheless, to verify that all is as it should be, for even a momentary shortage of oil at maximum revs. is fraught with the most expensive, and sometimes physically painful, consequences. The driving worm, if renewable, should be replaced without hesitation if the teeth have worn thin, and, when the design has been changed from bronze to hardened steel, as in Vincent engines, it is desirable to instal the new design in any event.

Rotary plunger pumps are not designed for working against pressures of more than a couple of pounds or so to the square inch, and if any blockage should occur

in the outlet passage or pipes from either the pressure or scavenge ends, either the worm drive or the driving peg which engages with the cam-slot in the plunger will fail in a fairly short time. Therefore this point must be carefully watched both inside the engine and in all the external connections.

Apart from that eventuality, this form of pump lasts for a long time and has the merit that it does not allow oil to syphon back through it whilst standing, which is an annoying defect of the gear-pump.

When plain big-end bearings are used, it is essential to maintain a high oil pressure, consequently they are lubricated either by a gear-pump or one with purely reciprocating plungers as used on Triumph and some Ariel engines. These also give little trouble but the trunnion block should be examined for wear and care taken to see that the plungers, ball-valves and springs are all operating correctly.

Should an engine ever suffer the misfortune of having a piston break up badly, or have a hole burnt in the crown, the resulting mass of aluminium particles penetrates into every nook and crevice with astonishing rapidity, and is likely to congregate in odd corners or pockets such as exist where two drilled oil passages meet, or where unions are screwed into bosses. Flushing-out cannot really be guaranteed to get rid of them all and the only safe way is to remove all plugs and unions and make a thorough job of the cleansing process. When replacing these parts, remember that the slightest trace of an air-leak on the suction side of the scavenge pump is certain to lead to poor scavenging and, even if the engine does not oil plugs as a result, it will be down in speed due to oil-drag. Consequently, too much care cannot be taken to see that every joint, or plug, is completely air-tight and locked up so that it will remain so indefinitely, this remark applying also to the driving-pin of a rotary pump which, after being fully tightened, should have metal punched into the screw slot as an insurance against loosening.

CHAPTER IX

REASSEMBLING CRANKCASE COMPONENTS

WITH the exception of the cam-followers and overhead gear, practically all the components within the engine which demand attention or possible modification have by now been dealt with sufficiently thoroughly to indicate the lines upon which to proceed.

If everything has been done to your own satisfaction—and the more self-critical you are the more likelihood there is of your beating other and more easily satisfied rivals—a spot of assembly work can now be put in hand prior to getting down to the vitally important matter of valve and ignition timing.

It is probably unnecessary to mention the point again, but, to be on the safe side, let me remind you that before commencing this work the bench and its surroundings must be swept clean of all foreign matter—dust, filings, cigarette butts—and the crankcase halves, flywheels, bearings, and so forth laid out, preferably on sheets of paper, after having been thoroughly cleaned.

The first step is a trial assembly of the flywheels in the crankcase in order to verify the centralization of the con.-rod and the amount of end-float, if any, on the mainshafts. Lateral location of the flywheel assembly is normally effected by one of two main methods: either one of the mainshafts is clamped endways in its bearings (usually on the driving side), or, alternatively, the main bearings on both sides of the wheels share the duty between them, each preventing movement towards its own side of the engine.

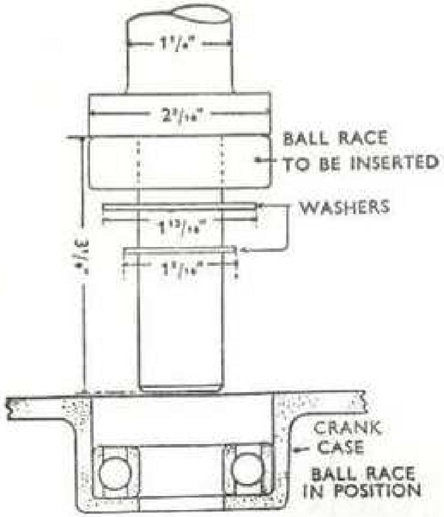
In the first-mentioned category the question of getting the amount of end-float correct does not arise, since it is fixed and limited to a very small amount by the design of the location bearing. Care must be taken, however, to see

that any distance-pieces or washers, either next to the wheels or between the various bearings, are in place in their correct order; otherwise, there is a risk that some bearing not intended to take end-thrust may be subjected to loading of this nature, which, in addition to causing stiffness, will rapidly destroy the bearing involved.

In the G3L Matchless, for instance, two ball bearings, with a pair of spacing washers separating their inner races, are fitted on the drive side, and should these spacers be inadvertently omitted, the pressure of the shock-absorber spring will put very heavy thrust-loads on the bearings, and may ultimately cause the whole flywheel assembly to move out of position in the direction of the chain. In this design, as in quite a number of others, the end-thrust of the shock-absorber spring is ultilized to clamp the drive-side mainshaft in its bearing or bearings; thus, when verifying the flywheel location, the spring should be lightly tightened with the sprocket and face cam (or an equivalent distance piece) in place.

In this condition there should be no measurable side play

It is an easy matter to mislay or misplace essential washers or packing pieces which vary considerably with individual makes. This drawing shows the correct method of assembling drive-side bearings used for many years for the Matchless G3L series.

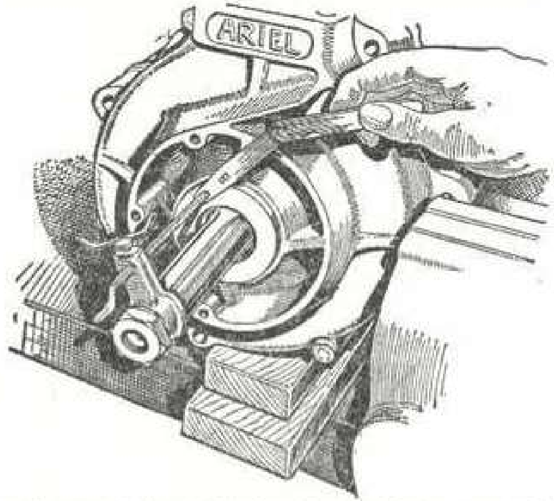


present, but if the spring is then removed, there should be a perceptible amount; if not, the bearing on the timing side is too close in and appropriate measures must be taken to obtain, say, .005-in. float and thus avoid the possibility of undesirable end-thrust being placed on the bearing.

In other designs such as the post-war Vincent singles and twins, a distance-piece is used between the inner and outer main bearings. If this is inadvertently omitted, the shaft will be locked up solid when the main shaft nut is tightened up, an operation which may not take place until the motor is almost completely assembled. It is therefore a golden rule *always* to assemble all relevant components on the mainshaft and tighten the nut fully early in the piece just to make sure that all is in order. There is never very much clearance to spare in this locality and it may be that a new crankpin is a fraction longer than the original and fouls the crankcase, though this might otherwise have escaped notice. In some engines, the crankpin nut runs so close to the bearing housing that a portion has to be cleared away with a chisel and this naturally must be done before proceeding further.

In those designs in which the bearings on both sides of the wheels assist in their location, it is not uncommon to allow up to .010-in. end-float, except in the case of o.h.c. Velocettes, in which the mesh of the bevel timing gears is partly controlled by the position of the mainshaft. These engines, and also those push-rod models equipped with taper-roller main bearings, *must* be adjusted so that there is precisely 4 thou. "nip" or negative clearance when the crankcase is cold. This is checked by noting with feeler gauges the gap between the two halves when tightness in turning the wheels becomes apparent, and shims to the requisite thickness must be placed behind either or both outer rings to ensure the correct "nip".

In o.h.c. Nortons, the assembly is located by the timing-side ball-race in order to limit the amount by which thermal expansion can affect the bevel mesh. Centrality adjustment is made by fitting shims between bearing and flywheel and

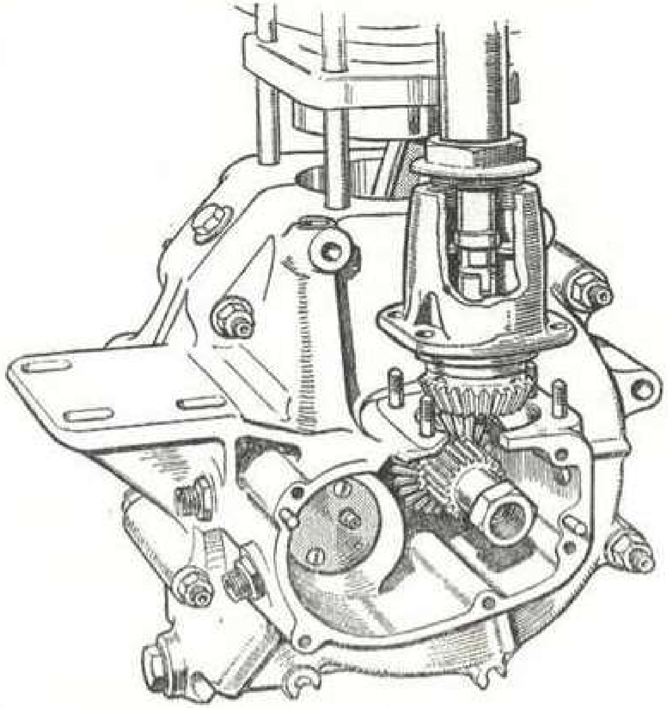


Another method, employed in the Ariel factory, is to fix a pointer on the drive-side mainshaft just touching the crankcase when the wheels are pushed fully over to the timing side. The wheels are then pulled in the opposite direction and clearance between case-face and pointer checked with a feeler gauge.

should be verified with all timing side components fitted and the retaining nut pulled up tight.

With the wheels placed correctly in the case and the latter tightly bolted up, the small end of the connecting-rod ought to be exactly central in the cylinder-barrel register, but as in almost every design $\frac{1}{32}$ in. or more side clearance is allowed between the small-end bush and the piston bosses, a little latitude in centrality is permissible. There is always a certain amount of side play of the rod itself if the big-end has been correctly fitted up, and so the simplest method of checking centrality is by callipering the distance between the register and each side of the small end, with the rod lightly held over in the opposite direction. The two measurements should not differ by more than .010 in. when checked by this means.

If the difference is perceptibly greater, it is *not* a good idea to attempt to correct it by bending the rod over; such a



The overhead-camshaft engine differs from the o.h.v. and side-valve types in that end-float in the flywheel assembly must be avoided, otherwise the bevel gears will tend to mesh incorrectly, thereby permitting backlash. Full details of the correct treatment are given in the accompanying text.

procedure would simply undo the value of the work previously carried out in truing up the rod. Instead, correction must be effected by altering the lateral location of the flywheels—hence the reason for the trial assembly before finally adjusting the end-float. Should the latter be excessive, obviously the side on which it is taken up should be the one which will tend to centralize the small end.

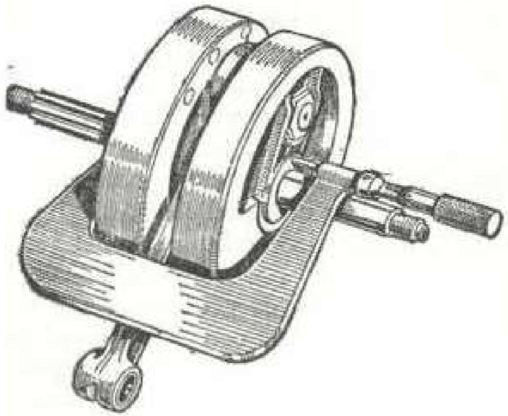
There are various methods of adjusting end-float, but most of them entail the use of shims, which, if not available from the makers, can be cut from shim-steel stock or, alternatively, use can be made of the shim-washers commonly

employed for adjusting the end-float in the steering knuckles of cars. The location of the shims varies, but the golden rule is never to put them between two parts which rotate relatively to one another, but only between faces which are relatively stationary. Thus, they can be placed on the mainshafts between the inner races and flywheel bosses, or between the bearing outer rings and the inside of their housings.

The latter entails rather more work in fitting, but in some designs is the only suitable place. For example, Velocette mainshafts are ground with a very narrow taper, and the races are designed to fit tightly thereon when hard up against the flywheels; obviously, if shims were interposed the fit of the inner races would be slack, so the shims must be placed behind the outer races.

Again, in many J.A.P. models the rollers run direct on the hardened shafts and end play is fixed by the clearance between the faces of the outer rings and the flywheel bosses. If this is too great, these rings must be shimmed up from behind. At the same time, it is wise to verify that the rollers have sufficient end clearance between the flywheels and the hardened washer placed at the other side of the outer race, otherwise they cannot rotate freely and will rapidly destroy the shaft surface. In the racing twins of this make, phosphor-bronze thrust washers are interposed between the flywheels and bearing sleeves and side-float can be regulated by using shims of varying thickness.

When building up an old engine with new parts, it will occasionally be found that there is no end play; there is, so to speak, negative clearance. In that event, first heat the case, drop out the main bearings, remove any shims there may be, and then try again. If this treatment is of no avail, the only thing to do is skim the requisite amount of metal off the flywheel bosses—the amount being determined by measuring the distance between the main bearings with the case bolted up and subracting this from the width over the flywheel bosses. For this job a large pair of external and internal callipers (or an internal micrometer) are required.



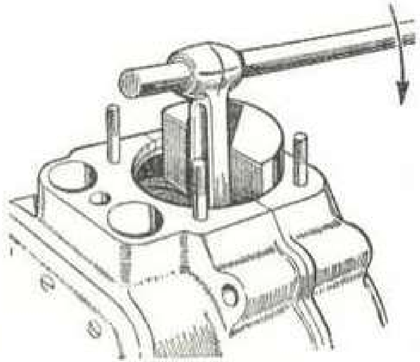
The use of inside and outside micrometers will determine the amount of end-float; the minor reading is subtracted from the major one. This is a good method for camshaft engines which employ "nil" clearance, but the amateur who has no such elaborate equipment will find a pair of inside and outside callipers a reasonably satisfactory substitute.

Having finally obtained a state of perfection in which the rod is central and the end-float correct, the flywheels should spin with the greatest ease as the rod is pumped up and down with the fingers, and there should be no perceptible difference in " feel " with or without the sprocket and shock absorber in place. If there is, the cause must be tracked down and eliminated forthwith and not glossed over in the hope that it will run off shortly in service. If all is well, the crankcase joint can be coated with the preferred form of jointing compound and reassembled. The author's preference is for one of the non-hardening cements such as "Gasket-goo", which does not dry quickly so that there need be no undue haste to slap the case together before the compound hardens. Make absolutely sure that nothing has been omitted which should be fitted at this stage, particularly in unit-construction engines in which certain gearbox components may have to be in place before the case goes together. Even though some may have to be refitted subsequently, it is preferable to fit and tighten all the crankcase bolts at this stage, but at all events it is very bad practice to put in a few in the region of the cylinder, leaving the bottom ones out because they are not easy to get at just at the moment. Subsequent joint leakage may be traced to this cause.

Some crankcase bolts are drilled for cylinder-base oil-feeds and care must be taken not to overstress them. As part of the checking routine, inject oil with the pressure can through every oil passage there may happen to be to make absolutely certain all are clean and to wash out any foreign matter. Sometimes an oil hole may be blocked with excess gasket cement and it is better to clear it at this stage than after some mechanical damage has been wrought later on.

The next step is to fit the piston, minus rings, as a preliminary to checking its " lie " in the barrel. If the piston is new or undamaged, the bottom edge of the skirt should

An error in workmanship may necessitate setting of the connecting rod; this can safely be attempted if a block of metal is first placed between crankcase and connecting rod, with the latter at bottom dead centre.



be parallel to the crankcase face in a direction parallel to the mainshafts, but, owing to the various amounts of side play present and the shape of the bottom edge, which is frequently by no means flat, this is not an easy matter to check with real accuracy. In any case, it does not matter a great deal; what *does* matter is whether the piston lies centrally in, and parallel to, the barrel.

This is checked by lightly bolting the cylinder in place and viewing the piston from the open end of the barrel under a good light. It is then quite easy to see just where the piston does lie, and if it shows any undesirable tendency to work over to one side when the flywheels are rotated.

If it does so, it is a sign that an error—or, more likely, a multiplicity of small errors all tending the same way—is causing trouble. Sometimes reversing the piston will effect a cure, although this method, of course, cannot be adopted (except as a temporary method of checking) with a piston which has unequal-sized valve clearance pockets in the crown, or with the split-skirt pistons fitted to a few sports models which must not be run with the split towards the rear, or thrust, side.

Even if the top lands appear to be central, this provides no guarantee that the piston as a whole is lying parallel. This can best be verified by turning the engine over several times after coating the cylinder bore with a very light (almost imperceptible) smear of prussian blue. Both thrust faces of the piston should then be symmetrically marked, but if differences of only a minor nature exist, a little careful work with a dead-smooth file will usually rectify matters. Severe cases call for a little more drastic treatment, and it may be necessary to set the small end slightly, although this will be a reflection on the quality of the previous work. To perform this operation, position the rod at bottom dead centre and select a block of metal which will just jam into the space between the crankcase mouth and the shank of the rod about half an inch below the little end. Insert a mandrel (*not* your checking mandrel!) into the small-end bush and bend in the requisite direction, remembering that only a very small amount is required to make a big difference to the position of the piston, owing to the latter's relatively greater length.

Once the piston is really true, it ought to be possible to make it reciprocate (without rings) at least half a dozen times by giving it a smart push with the thumb downwards from t.d.c. and it should always come to rest at b.d.c., because the flywheel counterbalance weight is less than that of the piston and rod.

Before finally completing the assembly, temporarily fit the barrel and head and pull the nuts down to correct tension to verify that no barrel distortion is thereby brought about. If distortion is present, it will be shown up by the engine not being quite so free to turn as it was before. Lack

of flatness or squareness at the cylinder-base flange or crankcase face is the most probable cause, and this can sometimes be detected by observing the effect of loosening the bolts one or two at a time. Final fitting is dealt with at the end of Chapter X.

The golden rules during assembly are *always to check everything you do as you go along, and not to assume anything is correct without verifying it.* Many hours of hurried mid-night toil would have been avoided time and again had these simple precautions been observed.