

## CHAPTER X

### "TOP-HALF" REASSEMBLY

We have now proceeded to the point where the barrel is ready for final bolting-down, provided that there is no further work to be done either upon it or the piston.

If, however, larger valves or a high-domed piston have been fitted, or the barrel shortened to any extent, it is essential to verify that the valve pockets in the piston crown are sufficiently deep, and if not, to deepen them, which will mean that the barrel must come off again in order to carry out the re-machining or to get rid of swarf if the pockets are deepened with the piston in place. Another matter which can be attended to at this stage is the head joint, a vital factor which will be touched upon later.

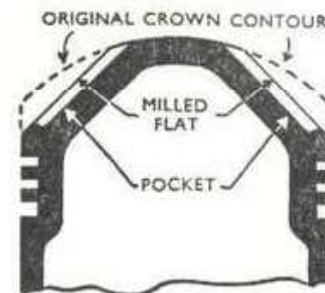
Reverting to the question of valve clearances, in themselves piston recesses are no earthly use—in fact, they are detrimental, inasmuch as they break up the crown contour and create pockets of gas in which detonation may take place. In extreme cases they may even render a considerable portion of the valve-opening area inoperative for a few degrees before and after t.d.c., so reducing the effectiveness of high-overlap valve timing at an exceedingly important period of the cycle.

Although this is only likely to be serious if ultra-high compression ratios are being sought after, the tendency to masking of the valve heads can be reduced by milling flats on the crown—to, say, half the maximum depth of the pocket—and counterboring into the flats to make up the rest of the clearance required.

Another point to be watched is the thickness of metal between the corner of the pocket and the top ring groove; this can be quite thin on the inlet side, down perhaps to  $\frac{1}{16}$  in., but on the exhaust side as much metal as possible

### "TOP-HALF" REASSEMBLY

*The masking effect of deep recesses may be offset by milling down the original crown contour and forming flats.*



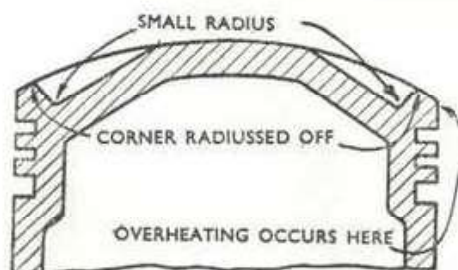
must be left. Insufficient thickness at this point allows an excessive amount of heat to pass through to the top ring; moreover, the upstanding rim of metal will become very hot and consequently weak and liable to crumble away if for any reason the carburetter is delivering a weak mixture.

This brings up the question of how deep the pockets really need to be, to which the answer depends to a large extent on the type of engine and the conditions of racing. In o.h. camshaft designs (provided the springs are of the correct strength) the valves follow the cams very closely up to a certain designed speed. Provided, therefore, the revs. are studiously kept below the danger mark by the use of a rev. counter, the clearance need be only a little greater—say,  $\frac{1}{16}$  in.—than that necessary just to clear the valve-heads at the point at which the valves and piston are in the closest proximity. *This is not necessarily at t.d.c. but may be a few degrees to either side*, depending entirely upon the cam contour and timing in relation to the piston movement. The danger points can be determined by temporarily setting up the head with light springs on the valves and checking the amount of clearance under each head at  $1^\circ$  intervals of crank rotation, measured by a degree-plate on the mainshaft, and with both valves correctly timed.

Another method is to fill the pockets of an old piston with deep recesses with plasticine and rotate the engine so that the valves form impressions in the soft material. These will be the minimum depths, but thermal expansion of the piston, any rocking of this component in the barrel



## TUNING FOR SPEED



*Recesses in the piston crown should be radiused off. If they are too deep the top ring land will tend to over-heat.*

and take-up of bearing clearances, all tend to reduce the measured clearances, and the actual pockets should be cut 30 to 40 thou. deeper.

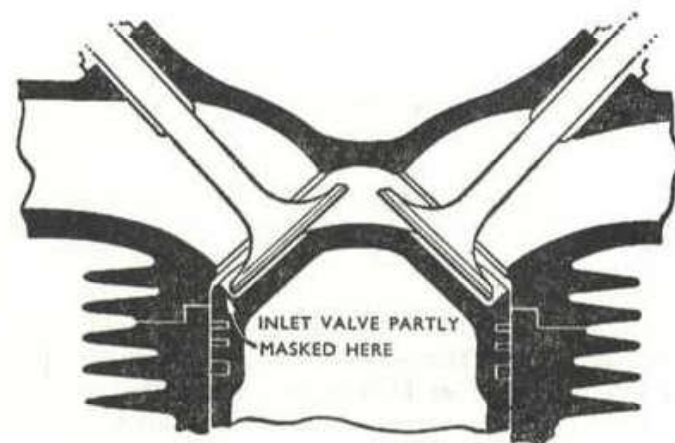
Owing to the greater possibility of whip or spring in push-rod mechanisms, particularly if the rockers are not too well supported, a little more clearance is desirable in o.h.v. units, even for road racing work where the rev. counter can be kept under observation. In other varieties of racing, particularly scrambles or sprint events where, in the heat of battle, engines are liable to be grossly over-revved, it is best to play safe and to cut the pockets sufficiently deep to clear both valves at their full lift.

Failing the use of a milling machine, the common method of deepening the pockets is to file cutting-teeth on the heads of an old pair of valves and, after bolting the cylinder head on with the piston locked at t.d.c., to rotate those valves with the aid of a tap-wrench or similar tool clamped on the stems. This scheme has the merit that the actual clearance can be measured as the work proceeds; but, of course, the radius of the pocket thus formed is exactly equal to the valve head. It is obvious that in running conditions the rim of the valve head will almost certainly scrape the pocket sides, therefore it will be necessary either to use valves (or, better still, a proper cutter, which can easily be made from a piece of hardened tool-steel) about  $\frac{1}{8}$  in. larger in diameter than the real valves, or else to remove the necessary amount of metal by scraping, a laborious and not very easy job to do nicely.

## "TOP-HALF" REASSEMBLY

If you are intending to experiment with compression ratios and have provided for one or two packing plates beneath the cylinder, it is advisable to arrange the valve clearances to suit the highest obtainable ratio.

As a final touch the edges of the pockets should be rounded off and the piston crown polished, preferably on a power-driven cloth mop with Tripoli compound; small mops or "bobs" which can be fitted in a drill-chuck, or to a



*Unduly deep recesses tend to decrease the effective area of the inlet valve.*

flexible shaft as used for port-grinding, can be purchased for a shilling or two, and this form of polishing gives a surface superior to anything which can be obtained by hand with the aid of metal-polish.

The value of crown polishing, like that of port polishing, is a debatable point, as some engines do not give of their best until a perceptible film of carbon has formed. The sequence of events is actually this: a bright, highly-polished surface absorbs less heat than a dull black one, and as less heat is extracted by it from the burning mixture, more is available for conversion into mechanical work. This is the



state of affairs when the engine is first run, but shortly after, if using fuel other than alcohol, a film of carbon forms which greatly increases heat absorption, thereby reducing power and increasing the piston temperature. Carbon, however, is a very bad conductor of heat; therefore, as the thickness increases the heat able to pass through the deposit to the metallic surface becomes less, and so the power returns almost to its former figure.

If using alcohol a polished crown will remain bright for a long time, and, whatever the fuel used, is much easier to decarbonize. Generally speaking, a high degree of polish is preferable.

When an extremely high dome is used in an endeavour to obtain a very high compression ratio, there is a danger that the plug will be badly masked. The close proximity of the crown to the plug points blocks the rapid spreading of the flame-front which is vital to good power production at high speed, and this masking may commence quite a long way before t.d.c. measured in crankshaft degrees, particularly if the shape of the head approximates a hemisphere rather than being flat. The result will be that engine power will fall off badly at high r.p.m. unless the ignition is advanced an excessive amount, and even this expedient is no real cure as it is merely trying to eliminate one error by putting in another.

Sectioning a wax cast of the combustion space as recommended in Chapter IV will clearly show the true state of affairs which can be improved by forming a local depression in the crown near the plug, or even making the whole crown unsymmetrical. In any event, it is better to use a lower compression and obtain good combustion than to aim only at maximum compression ratio.

### Cylinder-head Joints

Next, attention can be given to the head joint, which can be made gas-tight in a variety of ways. The "double-ground" variety in which simultaneous contact is made

between two areas (but at different pressures) is widely used, and is particularly good if the head is not very robust and therefore liable to be distorted by the tension of the head-bolts.

This design takes the form of a recess in the head which fits over a spigot on the barrel, recess and spigot being of substantially equal depths. To form a gas-tight joint the two components are ground together, with *fine* grinding paste between the spigot faces and *coarse* paste between the broad outer faces through which the bolts pass. The result of this method is to leave a minute gap between the broad faces when the spigot faces are in actual contact; thus when the bolts are tightened heavy pressure is applied to the latter to form the gas-seal, but distortion is prevented by the broad faces coming into contact.

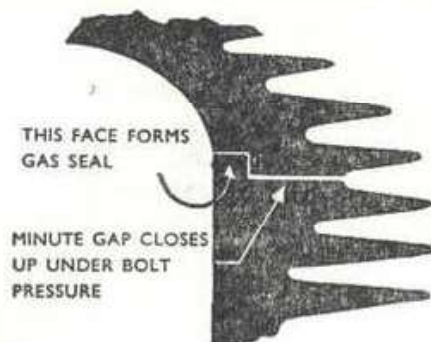
During grinding the coarse paste should be continuously renewed, but as the spigot faces approach perfection the fine paste can be gradually reduced in quantity by wiping the barrel spigot clean and adding a drop or two of oil, after which attention the process is continued until the spigot and its recess acquire a bright, smooth finish over their entire area. The broad faces will have (when the coarse paste is cleaned off) a matt surface which ought to be continuous, although one or two small patches of indifferent contact will not be seriously detrimental.

In the final assembly no jointing compound is necessary—in fact, it is undesirable. Besides being gas-tight, this type of joint permits a good flow of heat from the underside of the exhaust valve seat to the barrel, and so assists in keeping this region of the cylinder head cool. On the other hand, feeding heat back locally into the barrel is not good practice, since it is liable to cause distortion therein.

This snag is avoided in some of the later designs (employing aluminium heads with thick basic sections and particularly effective finning) by permitting the two components to be in contact only over a small area, preferably at the top of the barrel register, since this method eliminates the small annular gap which would be present if contact took place on the outer faces.



## TUNING FOR SPEED



*The principle of the "double-ground" head joint, to be found in several successful racing designs.*

For this type of joint laminated copper gaskets are excellent, but they should always be annealed every time they are fitted. Annealing of copper is done in precisely the reverse way to steel—i.e., the gasket is supported on a flat metal surface, heated to redness, and then dropped, *edge-on*, into cold water, from which it emerges in a dead-soft condition.

In cases where long bolts extending from head to crankcase are used, and particularly if the head and barrel are of light alloy, the bolt tension with the engine hot is considerably greater than when cold, possibly to such an extent that unsuspected distortion may take place, or the bolts may actually yield and become permanently elongated. These conditions should be suspected (a) if the engine does not hold its compression as well when hot as when cold; (b) if the head-bolts frequently have to be re-tightened.

If the engine on which you are working is known to possess these faults, it may be as well to make up a new set of bolts in high-expansion austenitic steel such as K.E. 965 or Jessop's G2—which conform to Specification D.T.D. 49B—or the "18/8" grade of stainless steel to Specification D.T.D. 176A.

Each of these steels has a tensile strength of 37 tons or more, but all are a little difficult to machine due to their work-hardening propensity. A tip to remember when working on them is to use keen-edged tools with 8° to

## "TOP-HALF" REASSEMBLY

10° top rake, a slow cutting speed and a comparatively heavy feed. On no account must the tool be allowed to rub or skid on the work; if this happens the surface will immediately become hard and impossible to cut unless the tool is run back and then dug in under the skin.

The coefficient of expansion of the steels mentioned is .000017 in. per in. per deg. C., but there is another type—the Firth-Brown product called N.M.C.—which has a coefficient of .000022 in. per in. per deg. C., nearly equal to that of many aluminium alloys.

Owing to space limitations the head studs on KTT Velocettes are only  $\frac{5}{16}$  in. in diameter and, being tucked away under the rocker boxes, necessitate a special short-headed T-spanner, which must be handled with care to avoid overstressing the studs, which if replaced must be of high-tensile steel. Early-pattern Vincent engines are fitted with large hollow bolts, through which pass the smaller bolts that carry the frame stresses. The large bolts have large nuts but must not be tightened to more than 30 pounds-feet torque, otherwise they may break when the cylinder expands; the later-pattern solid bolts are not subjected to this restriction.

In some engines it is necessary to fit the head before assembling the rocker gear, but in others it is more convenient to reverse the procedure. In view of work described in Chapter XI, dealing with valve mechanism as a separate subject, let us suppose your engine to be in the first category and that it is, therefore, now ready to have the barrel and head bolted on.

The procedure is quite straightforward. First, just make sure that the gudgeon circlips are all in place, then, with clean fingers, smear the rings and skirt of the piston with oil of the same grade you intend to use for racing, making sure that the ring grooves are full, for this will provide a store of oil to cover the starting-up period. Smear the whole surface of the barrel likewise, then slip it over the piston, easing the rings into place with their gaps staggered round



the circumference; do not omit the base gasket or packing washers if any are to be used.

On some engines it is not easy to handle the rings as they enter the barrel, due perhaps to the presence of a forest of studs, and none of the conventional ring compressors are much use either. The simplest and handiest tool is a strip of thin sheet metal, about  $1\frac{1}{2}$  in. wide, bent to the circumference of the piston and pulled up with a small bolt through the ends. The top-edge is lightly crimped with pliers to prevent jamming in the chamfer of the barrel, and after the piston has entered half-way, the strip can be pulled off simply by removing the bolt. All piston rings, and slotted oil rings in particular, are fragile, but as one cannot tell whether they are broken or not once the barrel has gone on, the operation cannot be carried out too carefully. Afterwards rotate the engine a few times to spread the oil evenly and wipe excess from the piston crown.

Next, after making sure the contacting faces of head and barrel are clean, bolt the head down by carefully tightening each bolt a fraction of a turn in rotation in order to obtain an equal amount of tension on each.

Finally, check your compression ratio exactly and enter it up, together with any relevant details such as type of piston, number of packing plates and so forth, in your notebook, and one more stage will have been completed.

## LIGHTENING THE VALVE MECHANISM

IRRESPECTIVE of any air of finality about the reassembly work comprising the last part of Chapter X, if cams giving a modified timing are to be used it may pay to make a trial assembly with only a single spring on each valve, or to put in only one rocker at a time when checking the opening and closing points. Otherwise it may be difficult to measure the top overlap with exactitude, because near t.d.c. the exhaust valve, as it closes, tries to turn the camshaft forwards, thus taking up all the backlash in the drive in the wrong direction.

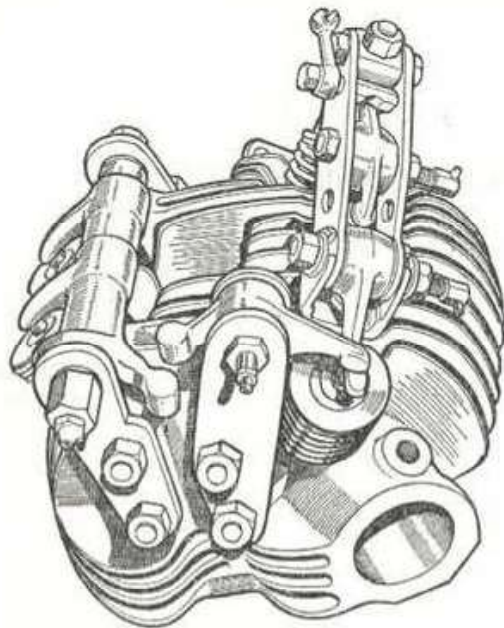
The best sequence of operations for any particular engine depends entirely upon the design, but in any case the points to watch closely in connection with the rocker gear are similar. In addition to operating correctly in relation to the valve stems, all the moving components must be as friction free as possible and all sources of lost motion, such as flexure in the rockers or rocker supports, must be eliminated.

Friction in valve gear comes into a different category from that of friction in the rest of the engine. The latter, while being very undesirable, simply subtracts from the amount of power the engine is potentially capable of developing at any speed. The former, even though seemingly insignificant in amount, can have the much more serious effect of limiting the maximum revs. by lowering the speed at which the valve gear can be returned from full lift by the springs, which have to overcome any friction present *before* they commence to close the valves.

At low speeds the inertia of the valve gear is small, and there is plenty of excess spring-force available to overcome friction, but as the speed rises towards the peak the excess of spring-force over inertia becomes progressively less. The



## TUNING FOR SPEED



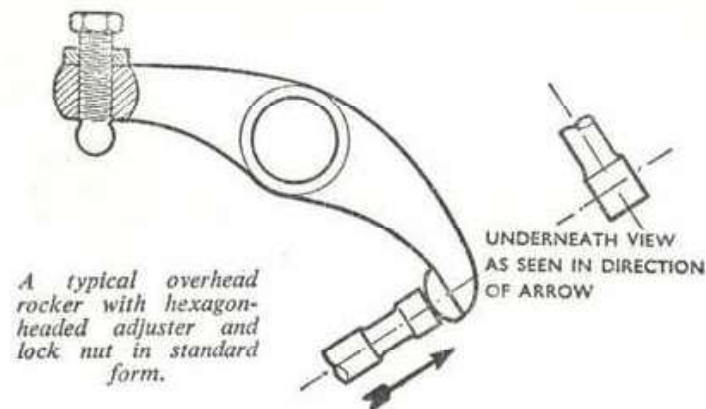
*Wear may be occasioned by cross-loading on the rocker mechanism of semi-radial-valved Rudge and Python engines.*

ill-effect of friction thus becomes proportionately greater and either or both valves cease to follow the cams on the closing side at a much lower speed than one would think without giving the matter some consideration.

Rockers and cam-followers must, therefore, be absolutely free on their pins and have enough end-float to ensure that they cannot possibly bind at running temperature. Occasionally spring washers are fitted to take up end-play and eliminate a possible source of rattle, but for racing it is better to replace these by distance pieces of the appropriate thickness. Felt washers are sometimes used to retain oil, and these can be an unsuspected cause of binding if they are too thick or made of insufficiently resilient material.

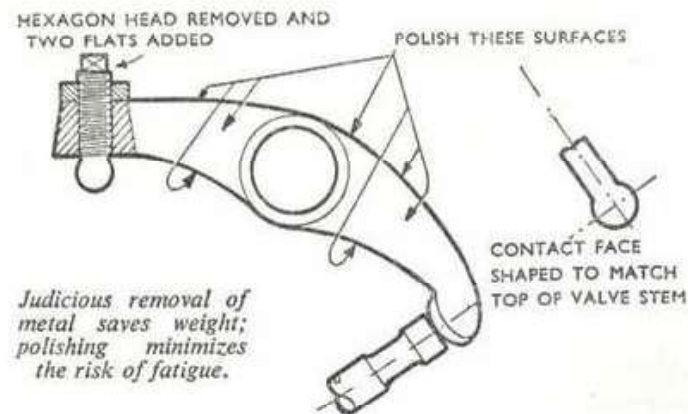
Radial-valve Rudge and Python engines have an unusual form of rocker layout which puts a certain amount of cross-loading on the bearings of those rockers which act at right-angles to each other. After a while wear occurs on the sides of the central bosses and the hardened washers against

## LIGHTENING THE VALVE MECHANISM



which they bear. In addition to causing noise, this wear diminishes the valve-lift and should be taken up by renewing the washers and shortening the hardened central roller-race until the side clearance is reduced to .001-.003 in. The length of the needle rollers must also be reduced if necessary so that there is no chance of their being nipped endways when the rocker pins are fully tightened.

Lightness is a very desirable quality in valve gear, but it is fatal to achieve it at the expense of rigidity. Where the latter is lacking at high speeds, when the loads are heaviest,



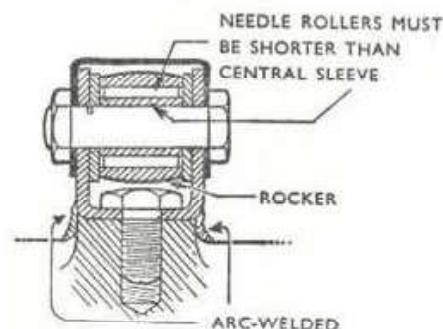


the motion of the valves will be quite different from the theoretical figures; in fact, the rockers in some racing engines are actually made heavier than those in their touring counterparts to ensure maintenance of power at high r.p.m. In this connection one of the main advantages of the double o.h. camshaft layout is the elimination of places between cams and valves where flexure or lost motion might occur.

Broadly speaking, the arms of a rocker are subjected to bending loads which are heaviest close to the point of support and fade out to zero at the points of contact at each extremity. The central part of the Z-shaped rocker usually fitted to push-rod engines is subjected mainly to torsion or twisting of constant value from end to end.

The only safe places to remove metal are those at which stress is clearly low or non-existent, paying most attention to the ends of the arms, since these portions move at the highest speeds. It is often possible to grind a sizeable amount off the faces which contact the valves, or to reduce the weight of the screwed adjusters sometimes fitted. Although reducing the section of the arms is *not* to be recommended, it is a good plan to polish the whole rocker in order to minimize the chance of fatigue cracking.

Flexure may occur in the rocker mounting unless this is bolted very firmly to the head. Anyone tuning one of the older engines with the rockers held between sideplates should verify that the attachment bolts are a really good fit



*The U-shaped rocker-brackets of Rudge and Python radial-valve gear can seldom be kept tight with the standard bolt. An effective cure is to weld them to the head.*

in the mating holes. Some editions of the Python engine were poor in this respect, the cross-rockers being held in a steel pressing attached to the head by rather inaccessible set-screws which were both difficult to tighten and inclined to work loose. The sovereign remedy is to have the pressings electrically arc-welded to the head, this process being preferable to acetylene welding because of the absence of distortion risks.

Push rods are normally made either from Duralumin or high-tensile steel tube. It is important to check that they are absolutely straight (by rolling them along a surface plate) and also that they have not been badly bent and re-straightened, because a rod which has been so treated is always liable to go again. If spares are not obtainable, light-alloy tube to specification 4 T 4 is the correct grade, the usual size being  $\frac{3}{8}$ -in. diameter by 16 S.W.G. This is a handy gauge because the bore is slightly under  $\frac{1}{4}$  in. and can just be reamed out to that size to take the end-fittings. Whether these are fitted internally or externally, they must be tight, otherwise the ends of the tube will hammer away quite rapidly. If the tube is steel instead of light alloy the end-fittings should be securely sweated in place with tinman's solder, completely penetrating the whole joint.

In some designs the push rods are of considerable length, and it may pay to experiment with tubes of larger diameter or heavier gauge; unsuspected buckling may take place at high speeds, and this will have the same bad effect as flexure in the rockers.

Tappets or push-rods are occasionally on the heavy side, due to the presence of adjusters. If you are well acquainted with the engine's habits when cold and hot with regard to correct valve clearances, something is to be gained by making up a set of non-adjustable components, finally setting the clearances either by adjustments to the push-rod length or by using hardened caps of the appropriate thickness fitted to the ends of the valve stems. These components are very often fitted as standard to minimize stem wear, but for making them up individually K.E. 805 steel hardened in oil



at 830° C. and tempered by quenching in oil from 550° C. cannot be bettered.

The top cups on Velocette Venom and Viper pushrods are a sliding fit in steel sleeves, clearance being measured by feeler-gauging the gap between the components. A slight saving in weight can be gained by keeping the original pair for checking and making up a new pair with fixed cups, without the sleeve, for running. The scheme is very easy to apply to side-valve engines with their flat-topped tappets; the method is to use the existing adjustable pair, set to the correct length, as gauges from which to obtain exactly the length of the non-adjustable pair, which can be made up out of silver steel, hardened on the ends by quenching in water from a bright red heat.

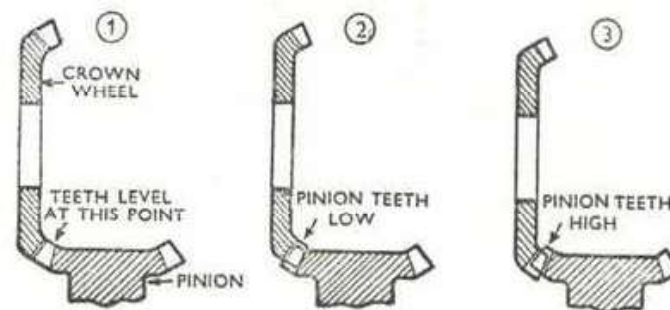
Sometimes shims of the correct thickness for placing under the valve stem cap are not available when making adjustments to o.h. camshaft engines which employ this system. Satisfactory substitutes can, however, be made by cutting discs from a set of feeler gauges, this material being sufficiently hard to withstand the rigorous duty entailed.

### Camshaft Drives

Even in o.h. camshaft engines, motion can be lost in the valve gear unless all is as it should be. If bevels are used they must be in correct mesh, which means the minimum of backlash consistent with absence of whine. Fortunately, neither these gears nor their bearings wear very rapidly, but if the adjustment is faulty it may not be possible to rectify it by moving only one of a mating pair of gears. As an approximate guide, the inner ends of the teeth on each pair of gears should be just level with each other as they pass through the point of intersection.

If this is not the case, one or both gears must be moved in the appropriate direction, either by adding shims or skimming a small amount off the face of the vertical shaft bush housing, or the bush itself. Since the camshaft, and hence its crown wheel, is usually located by a ballrace some distance

### LIGHTENING THE VALVE MECHANISM



- (1) Pinion and crown wheel in perfect mesh, with slight backlash. (2) If excessive backlash, raise pinion. If no backlash, move crown wheel to left. (3) If too much backlash, move crown wheel to right. If no backlash, lower pinion.

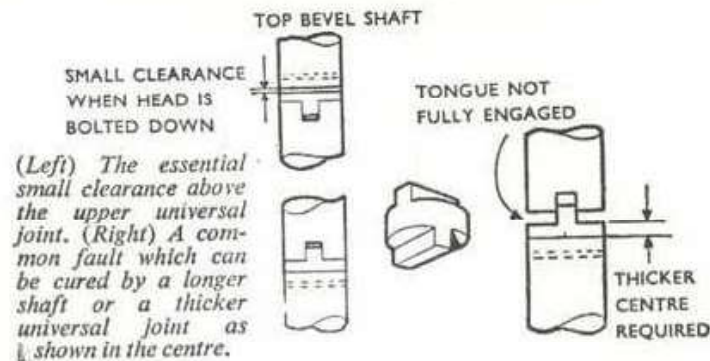
from the gears, thermal expansion has the effect of increasing the backlash by a small amount, and it will often be found that a slight whine, noticeable when the engine is cold, will cease as running temperature is attained.

Another point to notice is the fit of the tongues of the Oldham couplings in the bevel-shaft slots; they should be just free enough to slide, but have no trace of clearance sideways. If any shake has developed, new couplings are required, or, alternatively, the originals can be built up with hard nickel plating or chromium plating.

Conversely, a small amount of end-clearance must be present in the vertical shaft assembly. If this is not provided, severe thrust load will be applied to the bevel shafts when the head is bolted down, and this point must be closely watched if the barrel has been shortened appreciably. If no clearance exists it is possible to gain a little by grinding a small amount off the ends of the bevel shafts and vertical shaft. This process, however, must not be overdone or the area of contact of the coupling tongues will be seriously reduced. If at a future date the barrel is packed up again on compression plates, it will be necessary to obtain either a longer shaft or some couplings with thicker centres, otherwise the uppermost tongue will not run fully in engagement



## TUNING FOR SPEED



with the top bevel-slot. This condition is likely to lead to failure as the metal in the tongue will be subjected to a destructive combination of shear and bending loads instead of pure shear as intended by the designer.

There is a small opportunity for saving weight in the vertical tube and its retaining ring-nuts by making these in light alloy, using Duralumin or R.R. 56 for the tube and any reasonably strong alloy, such as L 1 or D.T.D. 423, for the nuts and possibly the bevel-shaft housings.

## Attention to Cam Followers

Coming back to push-rod and s.v. motors, freeness of the cam followers is important. The procedure with lever followers is similar to that for the overhead rockers; in addition, if any of the rocker bearings or pins show patches which are unduly rubbed or worn, indicating lack of lubrication, an additional oil hole or a nick ground into the end face of the bush may improve matters.

A little weight may be saved by judicious reduction here and there, but it is a mistake to remove too much from the region of the follower-foot, as this may lead to local overheating with consequent softening and rapid wear. Depending on the shape of the cam and the foot, there is a certain minimum area over which rubbing contact takes place, and it is an advantage for the foot to be appreciably longer than

## LIGHTENING THE VALVE MECHANISM

this, as the excess portion acts as a lead-in for oil and thus improves lubrication at a vital spot.

Followers which are only slightly worn can be reshaped to their original curvature by hand with an oil-stone, but if wear to a depth greater than .030 in. is present, there is very little of the original hard case left and new rockers are required—unless the originals can be re-casehardened and ground, a process which some service departments are willing to carry out.

If it is desired to embark on the project of making a new set of followers, 3½% case-hardening nickel steel is an excellent material; the hardening should be .045 in. to .055 in. deep on the contact faces, but the remainder of the surface should be kept soft by copper-plating before hardening. An even better job can be made by using 5% air-hardening steel to Specification S 28 with a coating of "Stellite" welded to the contact faces.

Following the application of the "Stellite" the component is heat-treated by cooling in air from 830° C., after which the faces are finally ground to shape. Welded "Stellite" is not very hard but will withstand very arduous treatment without appreciable wear as, unlike case-hardened steel, it does not soften under the influence of heat. "Stellited" faces must be finished by polishing to a dead-smooth surface, otherwise the cams will suffer undue wear.

## Points to Bear in Mind

When reconditioning or making new followers (or o.h. camshaft rockers) with curved feet two things must be borne in mind. One is that the radius of curvature does *not* alter the timing at all, but that it *does* affect the rate of acceleration; in other words, if you substitute a radius of ½ in. for one of ¾ in. the opening and closing points measured in degrees will remain the same, but the initial acceleration will be greater with the larger radius. The second point is that the distance between the rocker pin centre and the line of contact with the cam *does* affect the timing. A difference



## TUNING FOR SPEED

in length of .004 in. makes one degree difference with a cam base circle of 1 in. diameter; obviously, therefore, the presence of a flat due to wear at this point will lengthen the period of opening in proportion to its width. This matter will be gone into in greater detail later, when valve timing comes up for discussion.

### Varying Types of Follower

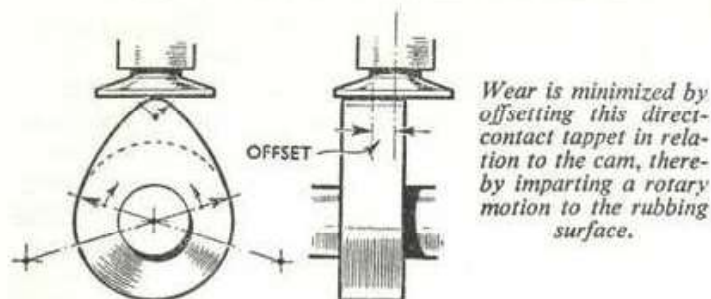
A few designs employ lever followers with flat feet and with these the rate of wear is usually lower owing to the greater area over which the rubbing action is spread. Eventually grooves will appear near the middle of the faces but except in extreme cases the parts can be reconditioned simply by smoothing on a flat oil-stone, taking care that the faces are maintained parallel with the rocker-pin holes.

The only other common variety of lever follower is fitted with a roller in place of a solid foot, and as the bearing in the roller is necessarily very small, heavily loaded and not too well lubricated, wear develops after a period. Replacing the pins usually effects a cure, but while the parts are dismantled it is advisable to check the roller holes for ovality. If they are out of round they can be lapped out with a mild steel bar and emery paste, after which oversize pins will be required of a size giving .002-in. clearance in the lapped-out holes.

The holes in the follower itself must be enlarged to a light drive fit on the new pins, which can be made from silver steel hardened and tempered to a straw colour in the centre but blued at the ends to enable them to be riveted over after assembly. Some racing J.A.P. engines employed rollers running on needle-roller bearings which were quite an improvement over the plain variety.

If some serious defect, such as extensive flats or pitting of the circumference of the rollers, is in evidence it is best to replace them. If spares are unobtainable the 1% carbon-1% chrome steel known as ballrace steel (Spec. EN 31) is about

## LIGHTENING THE VALVE MECHANISM



the best material from which to make new ones; heat-treat by quenching in oil after heating to 820° C. for one hour, then temper at 200° C. for half an hour and allow to cool in air.

“Direct attack” tappets bearing on the cams without the intervention of followers are used in many s.v. and a few o.h.v. designs. They are usually of the mushroom type offset from the cams a little so that a slow rotation is continually taking place, this distributing wear over the entire surface of the head, which in consequence remains flat almost indefinitely. Non-rotatable patterns wear, of course, in just the same way as lever followers, and require the same reconditioning treatment.

Working loads on the stems and tappet guides are heaviest at the ends and become zero in the central portion; thus a small reduction of weight can be effected by heavily relieving the tappet in that region. Another scheme, applicable only to non-rotating tappets, is to grind two flats on the central portion leaving a rectangular section to take the load. It is worth noting that on most valve-gear one ounce of weight creates an inertia force of thirty pounds or more at top speed, so there is much to be gained by reductions which seem trifling in themselves.

The exhaust lifter is not required for racing. In the case of standard A.J.S. and Matchless the control lever in the timing case can be disposed of, and after extracting the guide and tappet from the crankcase, the valve-lifter collar can be removed through the slot cut in the side of the guide; a small



point, but one which illustrates further how the odd fraction of an ounce of unwanted weight can be saved.

In any case, it is wise to verify that the exhaust lifter is not holding the follower off the base-circle or fouling the cam nose if any change on cam form or type is made. As an instance "Lightning" cams when fitted to a Vincent "Rapide" or "Shadow", foul the exhaust lifter levers: the latter must therefore be filed away to clear the cam-nose, though it is probably better for most racing to remove the entire lifter mechanism and block up the hole through which the lifter rod passes with a  $\frac{1}{4}$  in. B.S.P. plug, the same as that used in the crankcase drain hole.

Frequently, there is not overmuch clearance between the pushrod end of the rocker and the inside of the rocker housing or cover. If cams giving a higher lift are installed, it is essential to check that no fouling occurs here at full lift, otherwise the pushrods will be bent or the cams and followers seriously damaged.

## ADJUSTMENTS TO VALVE GEAR

THIS and the next chapter are closely related and may well be studied in a single reading. The aim is first to deal with standard valve settings and, in Chapter XIII, to indicate adjustments which may result in improved performance.

Newcomers to speed work would be well advised to stick to the maker's timing for a start, and it is also a wise plan for more experienced riders to follow the same course if major modifications have been made elsewhere to the engine. As a general rule, it is a bad policy in experimental work to alter more than one thing at a time, otherwise it is difficult to assess the value of any single modification.

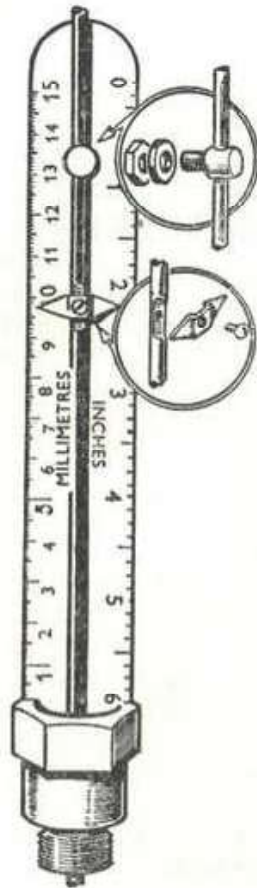
Engines usually have their timing gears clearly marked by dots on the mating teeth or lines inscribed on the rims of the gears. When taking notes on the engine before commencing the initial dismantling, a diagram of these marks should have been included, together with any information relating to the position of the small timing pinion on the mainshaft, since it is a common practice to use a number of keyways, any one of which may be selected, in order to provide a fine variation of timing at this point. If this is the case, it is usual to mark the "standard" keyway in some manner, though this may not be visible when the pinion nut is in place.

By no means do all engines employ a keyed-on pinion. In Rudge and Python power-units, for example, the pinion fits on a plain taper, and thus can be set readily in any position desired.

Marks or no marks, before finally fitting the timing cover it is best to play safe and to check the setting with a timing disc and pointer set to zero with the piston at top centre. It is not too easy to detect exactly where this occurs to within a degree or two just by feeling the movement of the piston



## TUNING FOR SPEED



*An excellent timing stick for engines fitted with a central plug can be made up from an old plug body, a steel rule marked in inches or millimetres, a piece of rod, a bicycle brake draw bolt and a simple pointer.*

crown through the plug hole, but it can be determined quite accurately by means of a depth gauge of such a length that it contacts the crown a few degrees—the precise number is immaterial—before top centre. If the pointer is correctly set to zero, the readings at which the crown touches the gauge on the upstroke or downstroke on each side of top centre will be equal, except in the unlikely event of the cylinder being “offset” or “desaxé”. Another method is to apply a film of thick soap solution over the plug-hole, or better still over a small hole in a dummy plug body; the point at which the film balloons out to its maximum extent as the piston is rocked over centre will clearly be t.d.c.

With this important matter settled, the timing can be determined and compared with the maker's figures, if this knowledge is available, not forgetting that many makers specify valve clearances for checking which are wider than those at which the tappets are eventually set for running. This is done because with some cam contours the rate of lift over the first few degrees is so gradual that it is very difficult to determine the precise instant at which the valves open or close. As was pointed out in Chapter I, this must be borne in mind when making comparisons between the timing diagrams of various makes. Nortons reverse this

## ADJUSTMENTS TO VALVE GEAR

practice and specify closer clearances for setting than for running, giving the appearance of greatly increased overlap compared to some other makes. On the short-stroke editions from 1956 onwards, the timing is not quoted in degrees, but in the amounts by which each valve is lifted at t.d.c. and b.d.c.

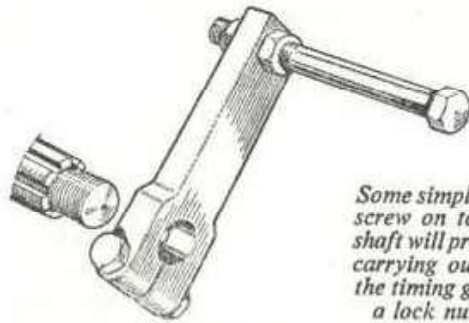
When the inlet and exhaust cams are separate so that the overlap can be varied either on purpose or accidentally, care must be taken to see that the valve-heads clear each other at part-lift. If the timing is altered from standard, or oversized valves fitted, one valve may quite possibly foul the other, and at least 40 thou. clearance should be present at the closest point to avoid this happening at speed. Except on overhead camshaft models this point is difficult to check and is often overlooked in consequence.

Whatever clearances are used, it is very rarely that the figures obtained will agree *absolutely* with those quoted by the maker, so that a little common sense has to be used to obtain the closest approximation. Generally speaking, the least important point is the exhaust valve opening, and the most important is the relationship between inlet opening and exhaust closing with reference to t.d.c. Sometimes it may be found that the timing is several degrees out on one or more points at the specified clearance but the error is reduced if the clearances are set a trifle wider; whilst not being a hard-and-fast rule, this at least gives an indication that the error is not so serious as it appeared and may even be disregarded. It is often difficult to get an equal timing diagram for both cylinders of a twin unless some allowances of this nature are made, particularly if some wear has taken place in the cams or cam followers. It is as well, while the timing disc is fitted, to obtain some more figures for future reference, and as an aid to getting a clearer picture of the valve-lift diagram, without which any subsequent experimenting will be simply like taking leaps in the dark.

The more obvious additional things to measure are the amount each valve is open at top and bottom dead centres; the actual amount of lift (which is not necessarily the same



## TUNING FOR SPEED



*Some simple form of handle to screw on to or grip the main-shaft will prove invaluable when carrying out experiments with the timing gear. If screwed on, a lock nut will be required.*

as the height of the cam on account of leverage-ratios which may exist in the rocker gear); the crankshaft angles at which the valves reach full lift, and the dwell at full lift, if any is present—"dwell" representing the number of degrees of movement during which the valve is on full lift.

The best way to measure lifts is by means of a dial-gauge rigged up in contact with the valve-spring collar. With this simple apparatus a really painstaking individual can draw a complete lift-curve for both valves by taking readings at intervals of, say, two or five degrees. These results are plotted on squared paper as vertical measurements, crankshaft degrees being measured horizontally. The graph so obtained can be made to give quite a lot of information, as will be shown later on.

There has been such an enormous number of engines manufactured in various types and editions that it is not possible to quote all their timing figures, even if they were available, but on pages 163-165 is a list of quite a few, not all of which are modern. Included are a number of figures for both sports and racing engines of the same make, the object being to show how the timings vary according to the type of performance required.

The checking and running clearances are also given—it being understood that "running" means "as set for running" with the engine *cold*, although at operating temperatures the clearances will probably be very different. As a general rule when hot, the clearances of both valves in

## ADJUSTMENTS TO VALVE GEAR

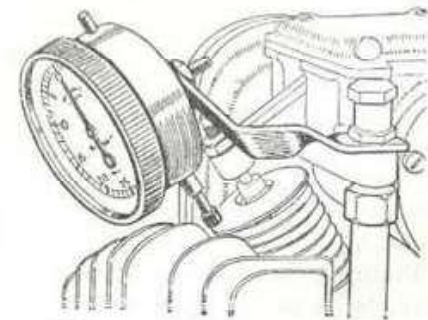
push-rod engines tend to widen, though more on the inlet than the exhaust. On camshaft engines both clearances close up, though more on the exhaust than the inlet, and in side-valve engines the inlet widens, whereas the exhaust closes up—the latter point in particular must be watched carefully if a K.E.965 valve has been fitted in place of one of silchrome or some other steel with a much lower expansion coefficient.

It is best when measuring valve lifts, to set the clearances to zero; the readings will then indicate whether the cams are ground with quietening ramps or not. The actual timing can then be shown by drawing lines on the graph parallel to the base-line at heights equivalent to the running clearances.

Each valve should be measured separately, with the other one rendered inoperative by leaving out its pushrod or rocker so that the backlash in the drive is not taken up in the reverse direction by the cam not under consideration.

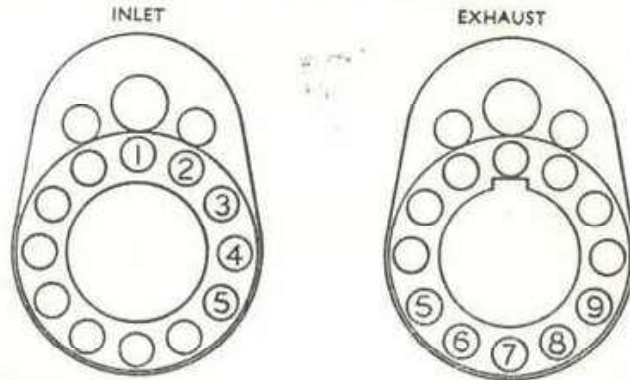
The rockers on Matchless/A.J.S. twins and KSS/KTT Velocette units with fully enclosed valves are mounted on eccentric spindles, as a means of adjusting the clearances, and either rocker can be removed, even if the valves are fully assembled, simply by unscrewing its spindle, the rocker in the process partially lifting the valve at each revolution. When finally setting the clearances, care must be taken to see that the plane of eccentricity lies in the right direction, otherwise the timing will be badly out and the rockers will not contact the valves properly. Arrows are stamped on the

*A dial gauge, contacting with the valve-spring and cap, read in conjunction with the timing disc, will enable the tuner to prepare an accurate valve-opening diagram.*





## TUNING FOR SPEED



By a series of hole-and-peg locations, Norton "Manx" cams can have vernier setting in five different positions.

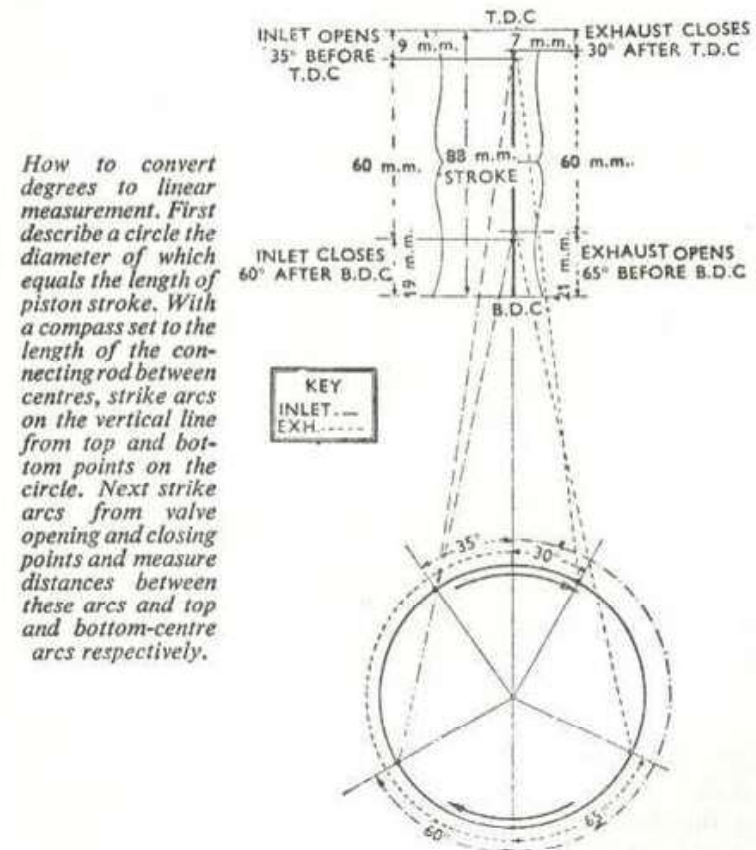
rocker spindle ends to indicate their correct positions, but any doubts can be settled by checking that the rocker-ends contact the valve stems centrally; if not, the error will be quite noticeable and can be cured by moving the pins half a turn in either direction.

A glance at the timing data shows that there is fairly close agreement between the timings used by various makers for fast main-road work or for racing, and an engine which does not figure in the list is bound to perform reasonably well if timed in a similar fashion. A closer study shows that, on the whole, not only is the overlap on engines intended for use with a silencer much less than on those made mainly or wholly for racing, but it is also split up differently.

In the latter case, the crank angle at which the inlet valve opens before t.d.c. is usually several degrees greater than the angle at which the exhaust valve closes, whereas in sports engines the angles are more nearly equal; indeed, in some cases the inlet angle is a few degrees less than the exhaust. The reason for this is that better low-speed performance and fuel economy are thereby obtained, and, of course, in standard production machines these aspects, although perhaps not so important as flat-out performance, have to be given a fair degree of attention.

## ADJUSTMENTS TO VALVE GEAR

Engines with "short" timings will respond to some extent to the fitting of an open exhaust system and an inlet pipe of the correct length, but they cannot be expected to perform as well as those with racing timings in conjunction with the correct pipe lengths. Conversely, if maximum performance with a road silencer fitted be the aim, there is nothing to be gained—in fact, a good deal may be lost in speed, economy and acceleration from low speeds—by attempting to utilize cams of the T.T. variety. If it were





# TUNING FOR SPEED

otherwise, manufacturers would not go to the trouble of developing different cams for various types of engine.

In road-racing, engine speeds can almost always be kept high by proper use of a multi-speed gear-box; good torque at low speeds is not essential though it is very desirable on short circuits where time lost in gear-changing may become too high a proportion of the actual running time. In scrambles and trials in which a speed test is incorporated, conditions are very different because one needs an engine which is fast but can put up with large variations of speed because of the fairly wide ratio gear-boxes which have to be used. On speedways it is also essential to have an engine with good pulling power from 3,000 r.p.m. upwards, and for all this sort of work overlaps are not generally greater than 80°, and may be much less for competition models where flexibility is as important as sheer power.

Engines with separate inlet and exhaust cams, as for instance the Triumph twin and the double o.h.c. Norton, lend themselves well to variations in the respective cam settings. In fact the Norton is specifically made so that the inlet cam can be set to the best position for the exhaust system in use by means of a vernier device; as shown in the table, the inlet cam is set to open 10° earlier (and close earlier by the same amount) when a megaphone is fitted than it is with a straight-through pipe.

Actually, the timing figures given do not convey any real information as to the efficacy of the opening diagram—which varies, of course, with the valve lift and the steepness of the cam contours—but assuming a reasonable degree of similarity in the height of lift in relation to the capacity of the cylinder between various designs, they *do* indicate the lines along which to proceed in the search for speed.

It is frequently possible to utilize cams from one type of engine in another type of power unit from the same stable and thus obtain either a longer timing or higher lifts, or both. As an example, the “45°, 65°, 70°, 35°” cams used in the 350 c.c. and upwards racing J.A.P.s can be fitted to the 250 c.c. model. Again, some of the push-rod,

# ADJUSTMENTS TO VALVE GEAR

SPECIMEN VALVE SETTINGS WITH COMPLEMENTARY DATA

MAKE	MODEL	INLET		EXHAUST		CLEARANCES				IGNITION ADVANCE
		OPENS	CLOSES	OPENS	CLOSES	CHECKING		RUNNING		
						IN.	EX.	IN.	EX.	
A.J.S.	1930 Camshaft .. ..	40	63	70	35	—	.016	.014	.022	40°
	o.h.v. 250, 350 and 500 ..	20	67	78	28	.016	.016	nil	nil	37°
	7R 48-52 .. ..	62	71	74	44	.005	.014	.005	.014	40°
	7R 1953-54 .. ..	49	70	70	47	.007	.012	.008	.012	38°
	7R 58-62 .. ..	55	78	78	45	.008	.012	.008	.012	34°
Ariel	250, 350 and 500 Red Hunter ..	22	70	70	25	nil	nil	nil	nil	1 in. max.
	250 Model LG .. ..	22	70	70	25	nil	nil	nil	nil	1 in. max.
	350 Model NG, 500 Model VG ..	15	55	60	20	nil	nil	nil	nil	1 in.
	1949 NG, NH, VG, VH ..	18	68	63	23	.010	.010	nil	nil	1 in.
	Competition .. ..	50	65	65	50	.010	.010	nil	nil	1 in.
	650 Twin (Home) .. ..	30	70	65	25	.010	.010	.010	.010	1 in.
	650 Twin (Export) .. ..	42	62	67	37	.010	.010	.010	.010	1 in.
	500 Empire Star .. ..	35	75	70	40	.004	.006	.004	.006	7/16 in.
	500 Gold Star .. ..	25	65	65	25	.003	.003	.003	.003	7/16 in.
	B31, B32, B33, B34 .. ..	25	65	65	25	.003	.003	.003	.003	7/16 in.
B.S.A.	B32 Gold Star Touring .. ..	43	73	64	34	.018	.018	.008	.010	37 1/2°
	International Trials .. ..	50	80	70	45	.018	.018	.008	.010	37 1/2°
	Scrambles .. ..	60	85	80	55	.018	.018	.008	.010	38 1/2°
	Racing, 50-50 fuel .. ..	43	73	70	45	.018	.018	.008	.010	34 1/2°
	Racing, alcohol .. ..	43	73	70	45	.018	.018	.008	.010	34 1/2°
	B34 Gold Star .. ..	As for B32, Ignition approx 3° earlier								
	A7ST, A10SF .. ..	42	62	67	37	.010	.010	.010	.010	1 in.
	175, 250 and 350 s.v. .. ..	15	50	50	20	.004	.006	.004	.006	40°
	500 and 600 Sports s.v. .. ..	16	65	65	25	.004	.006	.004	.006	40°
	1,100 s.v. A/c 60° twin and ..									
J.A.P.	1,100 s.v. W/c 60° twin .. ..	16	65	65	25	.004	.006	.004	.006	38°
	1,100 o.v.h. W/c 60° twin .. ..	16	65	65	25	.002	.002	.002	.002	38°
	175, 250 o.h.v. Std. and Racing ..	27	67	67	27	.002	.002	.002	.002	45°
	1,000 o.h.v. Std. and Racing ..	15	60	63	23	.002	.002	.002	.002	45°
	1,000 o.h.v. Racing, 1932-34 ..									



## TUNING FOR SPEED

SPECIMEN VALVE SETTINGS WITH COMPLEMENTARY DATA—continued

MAKE	MODEL	INLET		EXHAUST		CLEARANCES				IGNITION ADVANCE
		OPENS	CLOSES	OPENS	CLOSES	CHECKING		RUNNING		
						IN.	EX.	IN.	EX.	
J.A.P. <i>contd.</i>	1,000 o.h.v. Racing, 1935-37 ..	25	66	65	23	.002	.002	.002	.002	45°
	350, 500, 8/75 o.h.v. Racing ..	45	65	70	35	.002	.002	.002	.002	38°
	350, 500, 8/80 Speedway ..	45	65	70	35	.002	.002	.002	.002	38°
Levis ..	350 (high-lift cams) ..	25	60	60	25	—	—	.002	.002	40°
	500 (high-lift cams) ..	30	78	66	30	—	—	.002	.002	35°
Matchless ..	350 G3L (early) ..	20	67	78	28	.016	.016	nil	nil	7/16 in.
	350 G3L (late) ..	32	63	65	30	.016	.016	nil	nil	7/16 in.
	G80 ..	32	63	65	30	.016	.016	nil	nil	3/4 in.
	G1CS, G80CS (1959) ..	59	69	69	48	nil	.005	nil	.005	39°
	G80R (1959) ..	67	85	83	62	nil	nil	nil	nil	40°
	G9CSR ..	24	65	63	25	.006	.006	.006	.006	37°
	G12CSR, 600 c.c. twin ..	34	67	67	34	.006	.006	.006	.006	37°
	G12CSR, 650 c.c. twin ..	38	78	73	42	.012	.012	.008	.006	37°
	G45, 500 Racing twin ..	35	68	70	44	.004	.008	.004	.008	38°
	G50, 500 o.h.v. ..	55	78	78	45	.008	.008	.012	.012	34°
New Imperial	250 and 350 Grand Prix ..	28	62	60	30	nil	nil	nil	nil	14 mm.
Norton ..	350 and 490 International ..	47 1/2	70	85	42 1/2	.004	.004	.010	.020	42 1/2°
	490 Mod. 18, ES2, 588 Mod. 19 ..	25/30	43/48	60/65	25/30	nil	nil	nil	nil	42-47°
	30M, 40M, o.h.v. straight pipe ..	47 1/2	70	85	42 1/2	—	—	—	—	—
	30M, 40 M, o.h.v. megaphone ..	57 1/2	60	85	42 1/2	—	—	—	—	—
	30M, 40M, Double o.h.v. long-stroke ..	60	67 1/2	85	45	—	—	.012	.024	36°
	30M Short-stroke, 1954-55 ..	74	94	72	64	.005	.005	.014	.028	35°
	40M Short-stroke, 1954-55 ..	82	95	94	74	.005	.005	.014	.028	36°
	30M Short-stroke, 1956-58 ..	70	100	82	64	.002	.002	.014	.028	36°
	40M Short-stroke, 1956-58 ..	74	85	89	70	.002	.002	.014	.028	36°
	30M, 1956-58 } lift at t.d.c. ..	.278	.342	.280	.180	.002	.002	.014	.028	35°
	40M, 1956-58 } lift at b.d.c. ..	.280	.315	.280	.160	.002	.002	.014	.028	40°

## ADJUSTMENTS TO VALVE GEAR

SPECIMEN VALVE SETTINGS WITH COMPLEMENTARY DATA—continued

MAKE	MODEL	INLET		EXHAUST		CLEARANCES				IGNITION ADVANCE	
		OPENS	CLOSES	OPENS	CLOSES	CHECKING		RUNNING			
						IN.	EX.	IN.	EX.		
Norton <i>contd.</i>	30M, 1959 .. .. .	67	98	85	64	.005	.005	.010	.028	34°	
	40M, 1959 .. .. .	74	97	90	78	.005	.005	.010	.028	38°	
	30M, 1959 } lifts at t.d.c. and b.d.c. ..	.330	.342	.280	.260	.002	.002	.010	.028	—	
	40M, 1959 } .. .. .	.330	.310	.280	.260	.002	.002	.010	.028	—	
Rudge*	Ulster measured on stroke ..	10 mm.	13 mm.	16 mm.	10 mm.	.020	.020	nil	.003	15 mm.	
	350 Replica " " " ..	9 mm.	14.4 mm.	14.4 mm.	9 mm.	.008	.008	nil	.003	16 mm.	
	500 Replica " " " ..	9 mm.	14.4 mm.	14.4 mm.	9 mm.	.010	.010	nil	.003	18 mm.	
Sunbeam ..	Model 90 (Wolverhampton) ..	30	60	60	30	.002	.012	.001	.012	44°	
Triumph ..	250 and 350 o.h.v. single ..	36	70	70	36	.001	.001	.001	.001	1 in.	
	500 o.h.v. single ..	26 1/2	62 1/2	75 1/2	20 1/2	.001	.001	.001	.001	1 in.	
	Speed Twin and Tiger 100 ..	26 1/2	69 1/2	61	35	.001	.001	.001	.001	1 in.	
	Grand Prix, 1948 ..	31	42	47	32	.020	.020	.002	.004	37°	
	T100 (Racing Kit), 1950 ..	52	70	72	50	nil	nil	.002	.004	42°	
	T100, T110 Racing Kit ..	35	56	56	35	.020	.020	.002	.004	42°	
Velocette ..	Cub Racing Kit ..	59	81	85	55	nil	nil	.002	.004	39°	
	KTT, 1931 ..	43	70	68	48	.012	.012	.012	.022	42°	
	KSS (iron head) ..	39	69	60	40	.020	.020	.012	.020	42°	
	KTT, Mark IV and V ..	51	57	71	43	.020	.020	.015	.025	35°	
	KTT, Mark VI ..	55	65	75	45	.020	.020	.015	.025	32°	
	KSS (aluminium head) ..	34	47	64	29	.025	.025	.012	.012	38°	
	MOV, MAC ..	50	60	70	40	.010	.015	.003	.006	40°	
	MSS, early iron head ..	50	60	70	40	.015	.020	.003	.006	40°	
	MSS, iron head ..	30	60	60	30	.025	.025	.005	.010	40°	
	MSS, Viper, M17/7 cams ..	19	49	49	19	.030	.030	.005	.005	38°	
	Viper, Venom, M17/8 cams ..	55	65	75	45	.030	.030	.006	.006	38°	
	Vincent ..	Rapide, Shadow, Comet ..	42	68	72	30	Valves lifted	Valves lifted	nil	nil	40°
		Lightning, Flash ..	56	68	72	50	.005 off seats	.005 off seats	nil	nil	38°



Velocette cams are interchangeable, although the helical teeth gears, unique to this make, have been made with two or three different angles of helix and may not, therefore, be interchangeable. However, the camwheels are a press fit on the cams and can therefore be pressed off and changed over; if each pair is marked before being separated by scribing a line radially inwards from the marked tooth, and then refitted with the marks in line, the timing should be correct. Vincent camwheels are also pressed on, but in this instance the holes are parallel and not tapered as in the Velocette. Slight variations in shaft or bore diameters may lead to insufficient tightness if shafts and wheels are interchanged; the interference fit must be *at least* .001 in. requiring 1 to 1½ tons fitting pressure. If less than this the shaft should be nickel or chrome-plated up to the requisite size, otherwise the wheel may move on the shaft when running and damage to the valve gear will result. Rudge cams are yet another example of interchangeability, in this instance for the 500 c.c. and 350 c.c. capacities.

The flexibility and low-speed pulling of side-valve engines is due in large measure to the "slow" timing usually employed, but there is no reason why quite an appreciable amount overlap should not be used. Here, again, it is often possible to substitute the original cams by a pair from a corresponding o.h.v. engine. Many years ago, I remember fitting the cams from a "big-port" A.J.S. into a 350 side-valve of this make with really remarkable results.

Owing to the comparatively poor breathing and lower compression ratio inseparable from any side-valve design, its volumetric efficiency is low at high revs, and it pays to hold the inlet valve open a few degrees longer than on an o.h.v. unit, since it is no use closing this valve until *true* compression actually begins. That does not occur, if the cylinder filling is not too good, until the piston is well up on the compression stroke. This subject of cam design is very complex and will be dwelt upon at greater length in the next chapter.

If the timing has been altered from standard, or there is any reason to doubt the markings, set the engine by means of the degree plate with the crankpin in the position corresponding to the "inlet opening" point. The tappet clearances having previously been set to the amounts specified for use when checking the timing, the cams (or cam, if two separate shafts are used) are rotated until the clearance is just taken up, and the half-time pinion is then fitted to the main shaft in the appropriate position by means which vary according to the detail design. On J.A.P. engines there are five keyways in the pinion and the most favourable position for it is found by trial and error, using each keyway in turn.

A similar scheme is used on Vincent engines, the key in this instance being parallel so that it can be tapped into place after the best position for the pinion has been found. Normally the key is fitted in the keyway marked with a punch-dot.

B.S.A. cams are listed, paired up by part numbers, with the appropriate pinion to give the correct timing, but pinions are also supplied which advance the whole timing by 10°. See Table on page 169 for details.

If the design allows of but one position for the pinion, variation in timing can be accomplished only in steps of one tooth at a time, which is usually far too great. An exception, however, is found in some Velocette push-rod models; very fine-pitch helical teeth, of which there are 48 on the half-time pinion, provide a variation of 7½° per tooth.

One method of surmounting this difficulty is to make up some stepped keys, remembering that at 1 in. radius 1° equals .017 in.; from this data the amount of "step" to be filed on the key for any particular diameter of shaft and angular variation required can be worked out by direct proportion. Keys so filed must, of course, fit the keyways in both shaft and gear closely, otherwise looseness may eventually develop.

If the engine has separate camshafts for inlet and exhaust, next turn the flywheels until the crankpin is at the "exhaust



## TUNING FOR SPEED

opening" position and insert the exhaust camshaft so that the valve is just about to lift (or at the nearest tooth to this position), and slip the timing cover into place to steady the outer ends of the camshafts—a precaution which is not necessary if the shafts run in the modern manner on fixed shafts though it is advisable to fit the steady plate temporarily to eliminate spring.

The complete timing figures can now be checked. It will probably be found that they do not come out exactly as anticipated owing to small differences here and there in the mechanism, or because an error has been made in positioning the half-time pinion. Usually a compromise has to be effected, bearing in mind that the *overlap points* are more important than the others. In this connection it is better, in most instances, to have the inlet valve opening earlier rather than later than intended.

Several attempts may have to be made to obtain the most favourable figures, particularly if the cams have been modified. The stepped-key dodge may be very helpful in such cases. For future reference, make a careful record in the notebook of the timing eventually obtained, together with any remarks, such as clearances used at the time of checking, the valve lifts, and so forth.

Then oil all bearing surfaces copiously and fit the timing cover, after making dead certain that everything is correctly in place. At this juncture it is all too easy to omit perfectly obvious things, such as distance pieces or thrust washers on the rocker spindles, any rubber washers for the transfer of lubricant, or perhaps the small spring-loaded jet which feeds oil to the big-end of certain J.A.P. models, or even to overlook the elementary precaution of fully tightening the half-time pinion nut and locking it by whatever means are provided in the design.

If you have gone to the trouble of dowelling the timing cover in correct alignment, it can now be put in place with the appropriate gasket or any good-quality jointing cement thinly applied, and the screws or bolts finally tightened. If not dowelled, the cover may have to be juggled about a

## ADJUSTMENTS TO VALVE GEAR

little to ensure that the cam spindles are not binding. Incidentally, although brown paper is commonly used for gaskets, a higher-grade material, such as "Oakenstrong," is to be preferred for high-class work, as it is less likely to require renewal if the engine has to be worked upon when far away from its rightful garage. If the joint faces are in good condition, perfectly oil-tight joints can be made by omitting the gasket and using only a thin application of "Gasket-goo."

### B.S.A. CAM PART NUMBERS AND TIMING, AT 0.018 IN. CLEARANCE

		<i>Opens</i>	<i>Closes</i>	<i>Lift</i>	<i>Total crank angle</i>
Inlet	65-2454	50°	80°	0.415 in.	310°
Inlet	65-2446	63°	72°	0.400 in.	315°
Inlet	65-2442	65°	85°	0.442 in.	330°
Exhaust	65-2450	70°	45°	0.385 in.	295°
Exhaust	65-2446	80°	55°	0.400 in.	315°
Exhaust	65-1891	85°	60°	0.428 in.	325°

Normally used with mainshaft pinion 65-692. 65-696 advances timing by 10°.



## CHAPTER XIII

### IMPROVED CYLINDER FILLING

ONE reason for providing much longer periods of valve opening on racing engines, as shown in the tables in Chapter XII, is purely a matter of mechanics, for the loads in the valve gear are diminished (or, alternatively, higher speeds can be attained with the same stresses) if the angular periods of opening and closing are increased. Provided that the lift is kept the same, an increase in speed from 6,000 to 7,000 r.p.m. puts up the loads in the ratio of 36 to 49, an increase of 36%, but if the angular periods are increased in the same proportion—for example, by increasing the duration of opening from 260° to 303°—the loads in the gear will be brought back to their original values and the higher revs. can be attained with the same valve springs with no potential loss of reliability, as might be incurred if the spring strength were to be increased from, say, 120 lb. to 150 lb.

Lightening the valve gear, of course, reduces its inertia and consequently the force required to operate it, but this can be done only to any marked extent *in the design stage*. Apart from the minor savings which have been previously described, a really desperate attempt to reduce the weight of a well-designed mechanism by one-third would, in addition to introducing a distinct risk of failure, almost certainly defeat its own object by bringing in a much greater amount of flexure under load.

Despite their seeming lightness, all gases possess quite appreciable weight (as a matter of interest, the air in a room 10 ft. square and 10 ft. high weighs 80 lb.) and consequently also possess inertia; that is to say, they resent being rapidly accelerated, and when they are on the move are reluctant to

### IMPROVED CYLINDER FILLING

stop. The effects are similar to the inertia of the valve gear, inasmuch as they become increasingly serious at high speeds. Fortunately, the expedient of lengthening the valve-opening period, used to reduce the valve gear stresses, is also of the greatest value not only in overcoming the adverse effects of gas inertia, but in turning it to good account.

When the exhaust valve opens, the pressure in the cylinder is quite high, somewhere in the region of 80 lb. per sq. in. (the exact figure depending, of course, upon the characteristics of the engine), and the bulk of the exhaust products are entering the pipe as a "slug" of gas at well over atmospheric pressure. Towards the end of the exhaust stroke, i.e. as the piston is slowing down, the column of exhaust gas tends, under the influence of its own inertia, to continue travelling along the pipe and to draw the gas still remaining inside the cylinder out through the valve. Superimposed upon this so-called extractor action is an even more valuable effect created initially by the "slug" of high-pressure exhaust gas.

This "slug" forms the starting-point of a pressure-wave which travels down the pipe at the speed of sound, until it reaches the open end. Here it is reflected back up the pipe but is reversed in the process, becoming a wave of low pressure instead of high. This low-pressure wave travels back to the valve and augments to a considerable extent the extractor action already mentioned, provided that it reaches the port at the right time, which is of course at or near top dead centre. It is possible for the wave action to create negative pressures of up to six pounds per square inch in the port and almost all of the residual gas can be evacuated from the cylinder head if the exhaust valve is still open by a considerable amount at t.d.c. and does not close until 45° or 50° after. Furthermore, the inlet valve can commence to open long before t.d.c. without fear that the spent gases will try to get out of the cylinder by flowing back through the inlet tract: in fact if everything is correctly proportioned the depression created in the exhaust port by



the combination of column inertia and wave action will even cause a fresh charge to start moving into the cylinder before the piston begins to move downwards on the induction stroke. The inlet valve can be lifted  $\frac{1}{8}$  in., or more, off its seat at t.d.c. and can attain full lift nearly as soon as the piston reaches its maximum speed, near mid-stroke, without undue stress in the cam gear such as would be caused if the valve had not commenced to lift so soon.

The speed of sound in exhaust gas surroundings varies with the temperature and pressure, but, for practical purposes, can be taken as 1,500 ft. per sec.—considerably faster than its speed in air, which is 1,100 ft. per sec. The time taken for a wave to travel down any given pipe and back again to the port is practically constant, so that the effect of the wave action just described is most beneficial only over a limited range of speeds. At other times the effect will be absent or can even be adverse, when there is a pulse of high pressure instead of the required low pressure in the exhaust port at t.d.c. Such a pulse will cause a back-flow of gas out through the inlet valve and may upset the carburation to such an extent that the engine cuts out completely.

Altering the natural frequency of vibration of the exhaust system by altering the pipe length, or diameter, or adding a plain or lipped megaphone, will alter both the maximum power range and the minimum usable speed. Shortening the pipe and, to a lesser extent, reducing its diameter, raises the frequency. Adding a megaphone accentuates the wave effect considerably but tends to narrow down the usable range of speed. High-revving, road-racing, or record-breaking engines, therefore, employ short pipes with megaphones, but, where greater flexibility and a somewhat lower peak power speed are required, a longer, straight pipe is better. For extremely high r.p.m. the correct pipe length may be too short to be practicable or it may not comply with regulations, and it is then necessary to use double the optimum length.

The whole subject of wave formation in exhaust systems

is very complex and not a great deal has been written about it. Dr. Schweitzer's book, *The Scavenging of Two-stroke Cycle Diesel Engines*, published by The Macmillan Company of New York, provides some very useful information, and in Volume XXXIV of the *Proceedings of the Institution of Automobile Engineers* there is a very informative paper on exhaust systems written by J. G. Morrison.

By mid-stroke the incoming charge is rushing through the inlet pipe at several hundred feet per second, and it will continue to do so until the piston has commenced to travel upwards again on the compression stroke. If the valve and inlet pipe areas are inadequate in size, or of poor aerodynamic shape, the cylinder will not be completely filled by the time b.d.c. is reached, and holding the inlet valve open for a further 60° gives a little more time for filling to be completed. But, of course, as the piston has by then risen some distance, the amount of new charge taken in will be considerably less than the swept volume of the cylinder—in other words, the volumetric efficiency will be low.

On the other hand, if the valve area and pipe size are large enough the cylinder should be very nearly filled at b.d.c., and after this the high-speed column of gas will continue under the influence of its own inertia to travel into the cylinder. It is possible by making full use of this "ramming" effect to obtain a small, but very useful, amount of super-charge at certain speeds.

The distance from the valve seat to the outer end of the air-intake of the carburettor has a large influence on this, and on most racing machines is of the order of 10 in. to 13 in. This is several inches longer than is usual on ordinary o.h.v. and s.v. engines—particularly the latter, in which the whole inlet tract is frequently very short—whereas readers with long memories will recollect the much greater induction length used on fast side-valvers such as the long-stroke Sunbeam. The excellent results which used to be obtained from fitting a Binks "mouse-trap" or "rat-trap" carburettor were possibly attributable as much to their extra

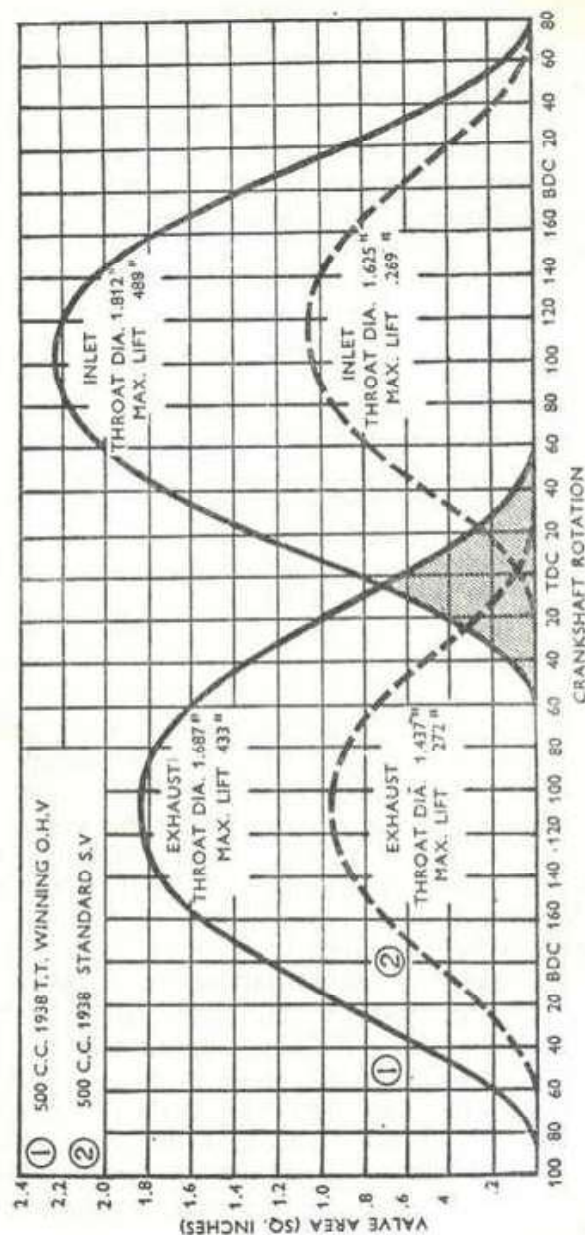


length as to the variable choke-tube which was the main feature of these instruments.

This may seem somewhat of a digression from the subject of cylinder-filling, but it has to be mentioned because filling and pipe design are to a large extent interdependent.

If all the factors involved are mutually in harmony a very large amount of overlap can be employed, and the long valve-opening periods so obtained will, in turn, permit higher lifts to be used without unduly stressing the valve gear, thus leading the way to a further increase in volumetric efficiency. The extent to which one may go is well illustrated by the 500 c.c. Norton timing diagram, originally published in an I.A.E. paper read by Mr. J. Craig during the war. The Norton diagram also shows very clearly the higher lift, longer duration of opening and larger port diameter of the inlet valve in comparison with equivalent figures for the exhaust. It is also noticeable that the exhaust opening point is a little earlier and the inlet closing point a few degrees later than usual, but the outstanding features are the 115° overlap and the amount by which each valve is away from its seat at t.d.c. Since that time, further research plus the ability to run at still higher r.p.m., gained by using larger valves in "square" engines, have permitted even longer timings to be employed. Reference to the valve-timing tabulation in the previous chapter shows that the inlet valve on the 76 mm. by 76.7 mm. "Manx" Norton model 40M, after allowing for clearance, opens at 75° before t.d.c., which is actually when the piston is just half-way up on the exhaust stroke, while the exhaust valve does not close until the piston is nearly half-way down the inlet stroke. It would be impossible to get an engine to run at all on such a timing unless full advantage were taken of correctly proportioned inlet and exhaust systems.

Diagrams of this nature are most easily obtained without loss of mechanical reliability on engines with valve gear of the double overhead camshaft type—and at the other end of the scale, in side-valve engines—where the best possible combination of lightness and rigidity is attained by reducing



This double graph, originally published in Mr. Joe Craig's paper, "Progress in Motorcycle Engines," read before members of the Institution of Automobile Engineers, illustrates in an extremely clear manner the increase in valve area and overlap timing possible with a racing power-unit in comparison with an engine of the touring type.



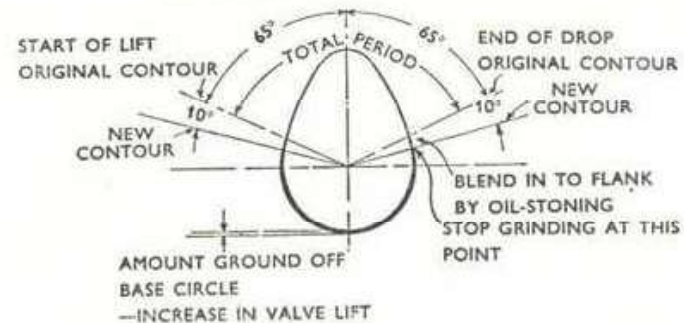
the number of components between each valve and its cam to a single tappet in direct compression. With such a layout the actual motion of the valves follows very closely the motion theoretically imparted to them by the cams, whereas in designs where flexure exists, even to a limited extent, this is not the case and the valve-motion departs more and more from the theoretical as the speed rises.

Extremely good results can be obtained from push-rod engines if this fact is recognized, and legislated for in the cam design or timing, by making allowance for the loss in motion which occurs most noticeably at the commencement of opening of each valve. This can be done by (a) advancing one or both cams by, say,  $5^\circ$ ; or (b) modifying the cam contour to give a slow initial rate of lift commencing  $10^\circ$  or  $15^\circ$  before the actual point at which the valve is desired to open. Scheme (a) has the merit of being simple to carry into effect, and comparative tests with various cam settings are easily conducted, but there are limits to what can be done in this direction, since any alteration in the opening points automatically changes the closing points by an equal amount.

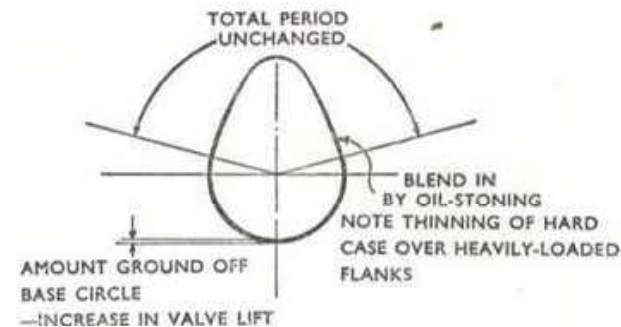
However, an engine with "30, 60; 60, 30" timing will almost certainly perform better at high speed with an open exhaust if the inlet can be advanced  $5^\circ$ , making the timing "35, 55; 60, 30," particularly if the inlet porting has been improved and the compression raised, for with the improved volumetric efficiency given by the two latter factors the inlet valve should not require to be held open for so long as it was before such improvements were effected.

Scheme (b) is standard practice with some makers whose engines are notable for their good all-round performance; it has the effect of taking up all clearances, and part of the unavoidable flexure, comparatively steadily and before the valve has lifted off its seat by any appreciable amount. At low speed the valve will leave its seat a little earlier than it does at high speed, but the actual area of opening is so small that there is scarcely any adverse effect on the low-speed performance.

## IMPROVED CYLINDER FILLING



(Above) A method of increasing simultaneously the valve lift and period of opening by grinding the cam base circle. (Below) To obtain increase in lift whilst retaining the original opening period.



Cam design is, in fact, quite a complicated business. Apart from determining the most suitable lifts and opening points, the contours of the cams and followers need to be very carefully worked out in order to give the minimum stresses in the gear consistent with the type of valve-lift diagram required; the most that the majority of private owners can do is to experiment with different settings or to obtain a set of racing cams.

There are, however, one or two simple methods of modifying the timing diagram with ordinary workshop equipment,



although experiments of this nature must, of course, be conducted solely at "owner's risk"!

One of these methods, which has as its objects increasing both the lift and the angular period of the valve opening, is to reduce the radius of the cam-base circle, a process which needs to be done very accurately to achieve good results, and requires the use either of a proper cylindrical grinder or a lathe equipped with a tool-post grinding attachment. The base circle is reduced a thou. or so at a time in successive cuts, the cam being partially rotated by hand between the centres of the grinder or lathe.

To avoid grinding too far in either direction, and thereby ploughing a hollow into the flanks, it is advisable to determine beforehand—by rotating the cam in contact with a dial-gauge—the points at which the cam-flanks join the base circle, and to mark these in some manner which will not subsequently become obliterated. Being dead-hard, it is next to impossible to mark the surface with a scriber. The best method is to make up a solution of copper sulphate (or "bluestone") in water and apply it to the cam surface. Instantly a very thin film of copper will make its appearance, and scriber lines will show up on this surface very clearly.

A carrier should be attached to the cam spindle and two rigid stops rigged up so that the wheel cannot grind beyond its allotted area—in fact, it is necessary to run out the cut some distance *before* the marks are reached, and subsequently to blend the new base circle into the existing flanks by handwork with an oil-stone—an operation requiring no little skill to carry out in such a manner that a gentle rise is obtained without forming any suspicion of a flat or hollow at the new junction of the flanks with the base circle.

Many workshops are now equipped with tap-grinders which form the lead-in portion of the tap with the correct cutting clearance. These machines can be adapted without much trouble to modify cams and can even be used to grind on new cams completely from master-cams by fixing a disc,

which is the same size as the grinding-wheel, to the bed of the machine and mounting the cam to be copied and the rough cam-blank on a mandrel held in the swinging head normally used to carry the tap when it is being sharpened. The master-cam then rotates against the fixed disc, and moves the head in and out as the spindle is rotated with the blank in contact with the grinding wheel. This process gradually forms the new cam, with a high degree of accuracy if the work is not rushed.

A simple method of generating cams with longer timing is to cut through an existing cam and rotate one half backwards in relation to the other by the desired amount, perhaps  $8^\circ$ , then sweat or dowel the halves together with a thin piece of steel between them. This filler piece is then filed to make the contour and to provide a smooth radius at the nose of the cam and the resulting article makes a very satisfactory master for use in a conventional cam-grinder or in the converted tap-grinder just described. It is possible to make up a whole family of cams in this way without much trouble, and the scheme eliminates all calculation or guesswork in determining the correct lift curves.

The majority of cams are hardened to a depth exceeding .040 in. As there is practically no load on the base circle (unless the tappets are wrongly adjusted, or heavy push-rod return springs are fitted) it is feasible to grind off as much as .025 in. and still retain sufficient hardness to give satisfactory running, if not for an indefinite period at least for long enough to prove the success, or otherwise, of the experiment.

Unless the new contour blends correctly into the base circle, valve clatter may be caused and, worse still, surprisingly heavy loads may be generated. The final shape should be checked with a dial-gauge and a degree-plate attached to the cam-spindle, and supposing .020 in. has been ground off and it is desired to increase the total angle of the cam by  $20^\circ$  ( $10^\circ$  on each flank), quiet operation will be obtained if the .020-in. rise is split up in the manner suggested on the following page.



## TUNING FOR SPEED

Measuring the lift directly off the cam with a dial-gauge does not give necessarily the actual lift which will be imparted to the valve, so this should finally be determined by assembling the valve gear and measuring the lift curve off the valve itself. In passing, a good idea of how much flexure exists in the system can be obtained by first plotting the lift curve with a very light temporary spring, and then with the correct springs; the result is sometimes rather surprising! When running at high speed the difference will be greater still, particularly at the start of lift, because of the valve inertia, which is, roughly speaking, about equal at peak revs. to the spring strength.

DEGREES FROM START OF LIFT	RISE OF CAM IN INCHES	DEGREES FROM START OF LIFT	RISE OF CAM IN INCHES
1	.001	6	.009
2	.002	7	.011
3	.0035	8	.014
4	.005	9	.017
5	.007	10	.020

*For quiet operation, the cam-lift modification described above should be graduated.*

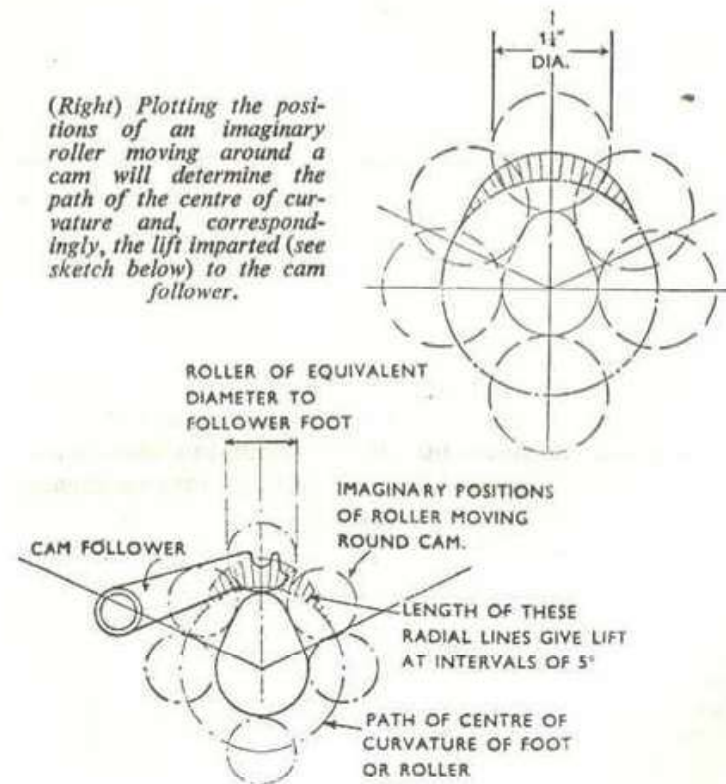
If this modification is correctly performed, a greater lift will have been achieved without the necessity for an increase in spring-strength. Should the period already be of satisfactorily long duration but the lift somewhat inadequate, the latter can be increased by itself if the base circle is ground off right up to the marks denoting the original start and finish of the flanks, after which a lot of stoning will be necessary to avoid a sudden and undesirably rapid rate of rise at the junction of flank and base circle. The final effect will be to reduce the case-depth over the heavily loaded flanks and as, in general, stronger springs will be required, subsequently the running life will probably be short. In some designs it is possible to overcome this defect by re-hardening the cams, but this may bring about difficulties

## IMPROVED CYLINDER FILLING

through destroying the accuracy of other dimensions, although the latter might be restored by grinding after being built up with hard chrome plating by the "Fescol" process or kindred methods.

It has already been mentioned that altering the cam-follower radius affects the acceleration rate but *not* the timing, and this offers a method of varying the opening diagram without altering the timing or the lift. A curved follower foot, even though it is non-rotating, is moved by the cam exactly as if it were a portion of a roller which *does* rotate. The lift imparted to the follower is determined by plotting out the path of the centre of curvature of the foot or axis of the roller, as the case may be.

(Right) Plotting the positions of an imaginary roller moving around a cam will determine the path of the centre of curvature and, correspondingly, the lift imparted (see sketch below) to the cam follower.



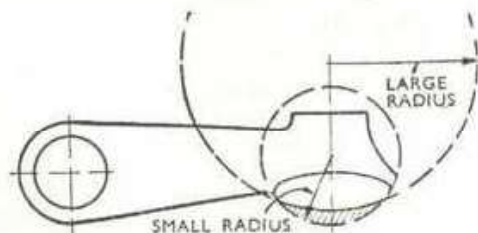


## TUNING FOR SPEED

Even if the follower is flat, the same statement applies, as a flat surface can be considered as part of a cylinder with an infinitely large radius. The lift curve can be depicted, first by plotting out the rise of the follower-centre as it traverses the cam, and then, in effect, straightening out the curved line so obtained; this has been done in the accompanying graphs (see page 183), which show the effect of different follower radii. While the rate of rise or fall is greater with the larger follower at the start of lift and finish of the drop (which increases the stresses in the gear at those times), the accelerations and decelerations near the crest of the cam are less and higher revs. can be attained with the same strength of spring. The main gain, however, is to be expected from the increased valve-lift at the vital period extending for a few degrees before and after t.d.c.

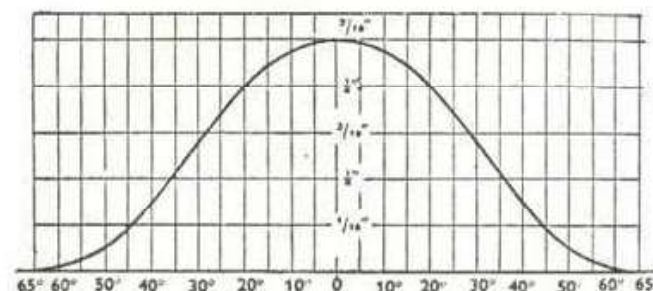
Followers can be ground to shape by hand on an emery-wheel, using a sheet-steel template previously made to the desired curvature as a gauge, but care must be taken to see that the surface is at all points parallel to the axis of the follower-pivot, otherwise highly concentrated loads and rapid wear will ensue. A good method is to make up a simple swivelling jig, which can be clamped to the machine-table and fed in to the required depth. If more than .020 in. is taken off, the follower will definitely require to be re-casehardened.

A factor which must be allowed for when lever followers are used is the effect of their angular swing; if the cam is symmetrical, the effect of the swing will be to make the lift-curve unsymmetrical and the flatter the follower the greater

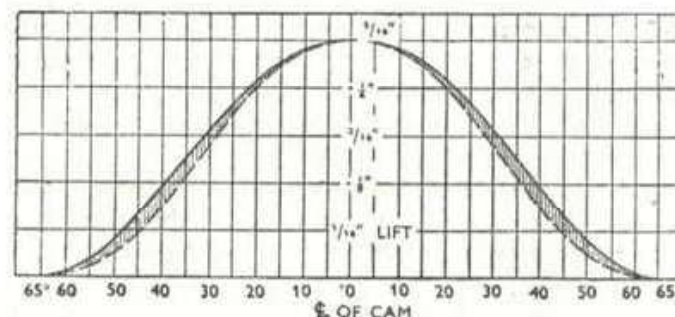


Varying the radius of a cam-follower base circle serves to alter the acceleration rate but not the overall valve timing.

## IMPROVED CYLINDER FILLING



Above is plotted the lift curve of a cam operating the valve via a follower with a  $\frac{1}{16}$ -in. radius foot. Below is the lift curve provided by the same cam operating via a follower with a  $\frac{1}{8}$ -in. radius foot, the shaded area representing the increase in valve opening.



this effect will be. The distortion of the lift curve varies according to whether the follower is leading or trailing the cam, and occasionally the effect is turned to good use to obtain a rapid lift and slow drop (or *vice-versa*) with a symmetrical cam contour.

In this volume it is possible to touch on the fringe of the subject only. It cannot be emphasized too strongly that cam modification is a thing not lightly to be undertaken, and is likely to do more harm than good unless carried out with extreme care and with the aid of good equipment.



## CHAPTER XIV

### IGNITION TOPICS

If there is any doubt about its internal condition, the best plan is to get the magneto overhauled by the makers or a reputable specialist in the art. Symptoms such as heavy sparking at the points, a noticeable grey deposit in the region of the points which this malady causes, or slackness or tightness of the armature spindle would justify such a course of action, since they are indicative of actual or impending troubles which are beyond the ability of most amateurs to rectify. When having a magneto tested, particularly if it appears to be correct but the engine misfires or just refuses to run up to speed after a period of running, get the test conducted after the instrument has been heated up to around 100°C. in a stove. This sometimes shows up a fault which was not obvious before.

Dead-certain starting, even with fully retarded ignition, is absolutely essential. The magneto must, therefore, give a good spark even when just flicked over with the fingers. If it does not do so, although otherwise in good condition as regards the points, brushes and slip ring, observe whether there is an appreciable magnetic resistance to rotation just prior to the points opening. If this "armature pull" is weak, it indicates, in conjunction with the poor spark, that re-magnetizing is necessary. This is not a usual malady with modern instruments having cast-in magnets of the "Alnico" type, but it does occur in older patterns and also in flywheel magnetos.

While not wishing in the slightest degree to cast any aspersions on them, combined magneto-dynamo instruments are intended primarily for touring purposes. For racing it is best to discard these devices and obtain one of the several types of magneto which have been built for the job. The ideal

### IGNITION TOPICS

is, of course, a modern B.T.-H. or Lucas instrument, but failing these one of the old M.L. models will give excellent results if properly reconditioned.

Given everything in order in the sparks department, the magneto can be attached to the motor and, if driven by chain, adjusted so that the chain has  $\frac{1}{4}$  in. up-and-down play in the middle of its run. Turn the engine over several times, feeling the chain meanwhile to check if its tension varies. This will be the case if the chain is worn unevenly, or the sprockets are eccentric, or one or both shafts are not dead true. Uneven tension in the magneto drive means that the ignition timing will be perpetually varied over a small range; thus the cause should be discovered and eradicated.

If the magneto holding-down bolts pass through slots, adjustment is effected quite easily, but in other cases (mainly on o.h.c. engines) where there is no such provision the base will have to be packed up on steel or brass shims of the same area as the base and suitably drilled. Owing to the high expansion of aluminium, the chain will usually tighten a little as the engine warms up, and some allowance must be made for this; besides wasting power, an overtight chain will ruin the armature bearing in a very short time.

Frequently there is little or no provision for preventing grit entering the slot provided in the chain case to allow for adjustment. This omission should be rectified by cutting out a piece of felt large enough to cover the slot and a little thicker than the gap between the case and magneto. The felt is slipped over the shaft before assembling the magneto and is of particular value if racing on sand or dirt is contemplated.

Gear-drive magnetos invariably have some form of oil-retaining device fitted behind the pinion. This should be examined and, if necessary, components such as felt washers, which have a limited life, should be renewed. There is a number of 100% efficient proprietary oil-seals on the market to-day. They cause only a negligible amount of friction, and in obstinate cases of oil leakage it may be worth while looking into the matter of fitting one, although this



will, in all probability, entail some additional machining operations. If a gear-driven magneto is found to be full of oil, probably the seal surrounding the mainshaft has become ineffective, so that cleaning out will be merely a temporary measure. The seal should be renewed; this can be done externally in B.T.-H. magnetos, but Lucas instruments must be first dismantled. In this event, make sure that the new seal is of the type which has a garter spring to contract it on the shaft, and not one of the early pattern which does not possess this feature.

There is little else to check in a gear drive unless the wheel is made of some non-metallic material such as "Tufnol" or "Fabroil." In such a case, if any teeth appear to be damaged it is advisable to obtain a new wheel, as if one tooth fails completely the rest will follow suit very shortly. Owing to the high expansion rate of these materials the teeth should have a perceptible amount of backlash, even when new; also the tips of the teeth must be well clear of the roots in the other gear, otherwise an annoying whine, which will *not* cure itself as time goes on, will be set up when running.

Racing magnetos are commonly equipped with manual advance mechanism, but on many sports machines fixed magnetos are used in conjunction with a centrifugal A.T.D. These are quite satisfactory, but for racing the scheme has two disadvantages: one is that the accuracy of the timing cannot be checked when running; the other is that it is not possible to drive a rev-counter direct from the magneto nut, which is one of the accepted methods, because the power absorbed by the drive, though slight, might prevent full advance being obtained. There are methods by which the drive can be taken off the fixed stops on the pinion, allowing the advance mechanism to work freely, but this is rather a matter for personal ingenuity. Some riders prefer to lock-up the device by welding together the fixed and moving stops after taking out the bob-weights, and either convert the magneto to hand advance, or use fixed ignition.

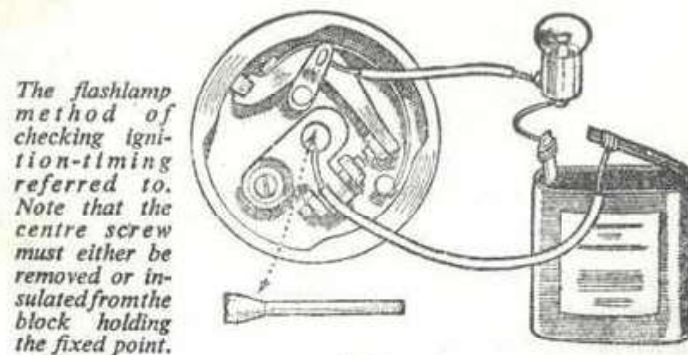
Engines running on alcohol, and not requiring more than

35° advance, will usually start quite well with no method of retarding but, if petrol is used, or much more than 35° is required, the engine may be prone to kick back, and so some form of retard control may be essential.

With the adjustment of the drive correct, and the contact-breaker points set to the correct gap (.012 in. in the case of racing instruments), the ignition can be timed in the usual manner, making sure that the cam-ring is in the fully advanced position and that the points are just breaking, with all the slack taken up in the drive. With automatic advance, it is also necessary to wedge the device into the fully-advanced position, as this is more accurate than setting at full retard and trusting that the extra advance will be as stated by the maker.

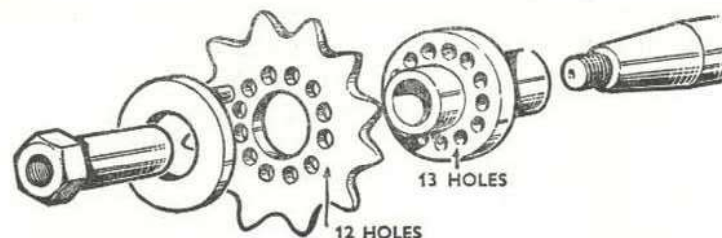
The method of using a flashlamp battery and bulb to determine the exact point of breaking is very good, but the time-honoured scheme of inserting a cigarette paper between the points and noting when this just comes free has the advantage of being available in almost any circumstances. Whichever method is used should be adhered to regularly—mixing up two schemes is bound to bring in discrepancies.

When one finally tightens the magneto drive nut, the spindle frequently creeps on a little way and thus retards the timing a degree or so; it is therefore essential to check the timing, and, if necessary, to reset it, if some particular "spot on" advance is being aimed at. The vernier device used on A.J.S.



*The flashlamp method of checking ignition-timing referred to. Note that the centre screw must either be removed or insulated from the block holding the fixed point.*





*A great aid towards obtaining optimum timing of ignition and valves is provided by a vernier adjustment. Here is the simple but excellent system applied to the magneto sprockets of certain early A.J.S. models, also to the cam-shaft drive.*

machines does not suffer from this trouble and provides a very simple method of setting or altering the timing.

On twin-cylinder engines the timing after being set on one cylinder must be checked on the other one. It will very likely be found to be a degree or two different owing to small variations or errors in the contact breaker or cam. If the error is only slight, the easiest way is to split the difference between the two cylinders; if more serious, careful grinding or oil-stoning of the cam ring will effect a cure. On single-carburettor V-twins the rear cylinder filling is rarely so good as that of the front one; consequently it will stand a little more advance and thus a difference between the two timings, provided it is not more than a degree or so, will not be a disadvantage. This does not apply to V-twins with two carburettors, which should take the same advance on both "lungs." Vincents, prepared for fast road work, run rather more sweetly low down if any difference is arranged to give the front cylinder less advance than the rear, rather than *vice-versa*.

In Chapter XII various makers' timings are quoted. These should provide a useful guide so long as it is appreciated that in many cases they are intended for use with comparatively slow engines. Broadly speaking, racing engines require less, not *more*, advance than their touring counterparts for the following reasons.

Rapid combustion is aided by high compression-pressure, a clean mixture (the result of good scavenging) and a high rate of turbulence within the cylinder. All these factors are present to a greater extent in racing power units than in touring engines. Matters of detail design, such as the shape of the combustion chamber, the location of the plug and the angle of inclination of the inlet port, exert an effect and account for the big differences in timing found in engines which, to a casual glance, appear to be fairly similar.

As a general rule, if an engine has had its compression raised from, say, 6 to 8 to 1 and has been otherwise attended to as regards its ports and exhaust system, the ignition point will need to be brought back by about 5°, and possibly more. The spark advance also varies according to the fuel or more accurately according to the flame-rate or speed of combustion of the mixture. The rate of burning of methanol is slow compared to petrol and for this reason requires nearly as much advance at 14 to 1 compression as petrol-benzole does at its maximum permissible compression ratio in the same engine. But with blended fuels such as Shell 811 or Shell M, which contain a proportion of petrol and benzole yet can be run at over 12 to 1 C.R., the ignition point may be perhaps 4° later. Some representative examples may be seen in the table overleaf. The only way to discover the best setting—and even an error of 1° in either direction will make quite a big difference in a highly tuned engine—is by test under power, preferably on a brake, a method of which few amateurs can avail themselves.

Failing that, the best method is by road test using a rev. counter, the ignition being first set a little on the early side. When the engine is at its maximum revs. in top gear, retard the ignition very slightly. If the revs. increase, retard a little more until the needle begins to fall back again. Then leave the control in its last position, check the timing, and reset the magneto to give the same timing, but with the lever at full advance. For this test it is best to select a road which is slightly uphill, so that the engine is definitely pulling hard. Another method is to leave the ignition at full advance and



# TUNING FOR SPEED

MODEL	FUEL	COMP. RATIO	IGNITION ADVANCE
B.S.A., B 32	Petrol-benzole	9	38½°
	Methanol	13	34½°
B.S.A., B 34	Petrol-benzole	9	40½°
	Methanol	11	38½°
NORTON 30 M (79.6×100 mm.)	Petrol 80 Octane	7.5	37½°
	Petrol-benzole	7.5 to 10	36°
	Methanol	12½	34°
NORTON 40 M (71×88 mm.)	Petrol 80 Octane	7.5	37½°
	Petrol-benzole	7.5 to 10	36°
	Methanol	13½	34°
NORTON 30 M (86×85.6 mm.)	Petrol 80 Octane	9.5	34°
	100 Octane		
NORTON 40 M (76×76.7 mm.)	Petrol 80 Octane	9.7	39°
	100 Octane		
VINCENT	Petrol 80 Octane	9	38°
	100 Octane		
	Methanol	12½	34°

make several runs, altering the contact-breaker gap in stages by one-sixth of a turn on the points. A good magneto will fire perfectly between the limits of .010 in. and .014 in. gap measurement. If a notable increase is gained by, say, the wider gap, check the actual timing, then reset the gap to the correct figure of .012 in. and re-time the magneto to the same figure. This is about the only satisfactory method with a fixed magneto or when using an A.T.D.

Unfortunately, though the correct advance can be determined very accurately by either of these methods, they cannot be applied until the whole machine is ready for the road. In the meantime a fairly accurate estimate can be made from the maker's figures, taking into account the effects of any modifications which have been made to the engine internally.

## CHAPTER XV

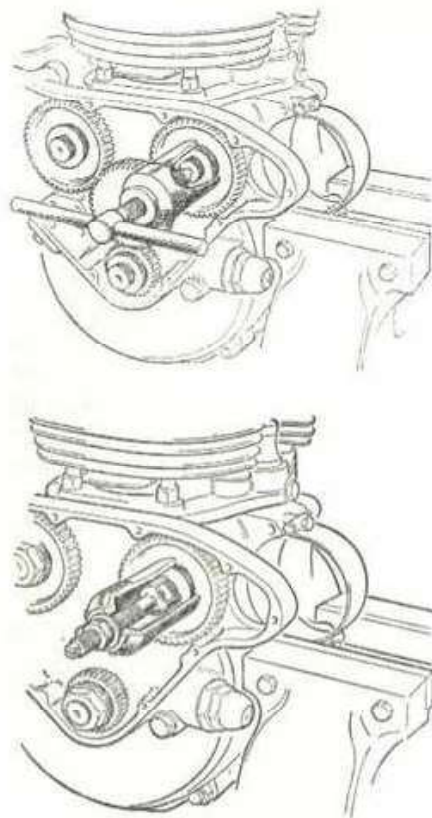
### TWIN-CYLINDER ENGINES

So far as the cylinder head and valve gear are concerned there is no fundamental difference between the work to be done on a single-, a twin- or even a four-cylinder engine, provided that they all have individual inlet and exhaust ports and pipes of equal length. Multi-cylinder engines built in that way act simply as a collection of singles and will give off power proportionate to the number of cylinders. However, when a single carburetter feeds more than one cylinder, the inlet passages are necessarily less direct and, therefore, offer more restriction to flow, whilst slight differences in the manifold shape frequently cause the mixture supplied to one cylinder to be richer than the other.

If for reasons of cost, or because the race regulations demand it, a single carburetter has to be retained, there is the choice of running one cylinder at correct mixture strength and the other too lean, or of increasing the jet size to make the weak cylinder correct and possibly dropping speed by reason of the other being over-rich. If the difference is slight, the last-mentioned expedient is the best alternative as there is less risk of damage to the engine, but it is usually possible to eliminate the trouble by inclining the carburetter towards the weak side. For initial experiments, a thick gasket of medium-hardness rubber can be fitted and the inclination can be adjusted by tightening the flange bolts more on one side than the other. This enables rapid comparisons to be made in a short period but, when the angle has been established, a solid gasket should be made up for permanent use.

Most parallel twins have a manifold with fairly sharp bends, forcing the mixture to turn two right-angled corners, this arrangement having been found to give better all-round





*Special tools are necessary to remove or replace (see sketches above and below respectively) the cam wheels of Triumph twin-cylinder models. Similar tools are available from other manufacturers.*

results than one in which the two ports blend together at a narrow angle close to the carburettor flange. For maximum power, it is of course best to fit two carburetters mounted on spacers of the same thickness as the manifold, or possibly a little greater. If the head studs and the carburettor bolt holes do not coincide, a little ingenuity is required in making up the spacers if it is not possible to buy them as spares.

Owners of speed-twin Triumphs are fortunate in that they can purchase a racing kit containing all the necessary bits and pieces. This is intended for engines supplied as standard with aluminium heads and "Grand Prix" type con-

rods, but can be for earlier engines with iron heads which are quite satisfactory on alcohol fuel. The rods in these engines were of a lighter pattern than those fitted to the genuine "Grand Prix" engine and it is best to obtain a pair of the latter type if possible. The change of rods and pistons will call for some re-balancing and, though the factory balance factor is 62%, some riders find that 70% gives better results.

Triumph camshafts are driven by wheels which are a tight fit on the shafts and have three keyways for accurate adjustment of timing. When fitting racing camshafts, the wheels are pulled off with a special remover (part number D178) and forced on again with a replacer (part number D182), and it is wise to obtain these tools before commencing the change-over. The sequence of re-assembly is to fit the crankshaft and camshafts into the drive side, not forgetting to place the rotary breather-valve disc and spring in the inlet cam-bush, then the timing-side case with the usual precautions for making an oil-tight joint and again not forgetting the two internal bolts just inside the crankcase mouth. Assemble the pistons, cylinder block and head in the normal manner, but fit only the exhaust rocker-box and pushrods with the clearance adjusted to zero so that the rockers are just free enough to slide sideways.

Replace the key and half-time pinion, fit a timing disc and pointer, accurately adjusted to t.d.c.; set the crankshaft at 70° before b.d.c. and rotate the exhaust camshaft forwards until one rocker is just tight enough to resist side movement. Without allowing anything to move, offer up the camshaft wheel so that one of the three keyways is in line with the key at the same time that the teeth line up with those on the intermediate wheel. Pull the wheel into place, and rotate the crankshaft backwards until the rocker again becomes tight. This is the closing point and should be 52° before t.d.c. is reached.

Check the figures on the other cylinder and if the discrepancy between the two is greater than 4° it should be equalized by re-positioning the wheel. It is unlikely that



the exact figures quoted will be realised at all four points on both cylinders, but aim to obtain the nearest approach, remembering that the exhaust opening point is the least critical and a degree or two early will make little difference. Before worrying too much about minor differences, check again with the clearances set to, say, .005 in. The figures will, of course, then be different, but may be more consistent between the two cylinders, in which event the difference between the figures at zero clearance is not unduly serious.

Procedure for timing the inlet valves is exactly the same as for the exhaust, after which the rocker-boxes are fitted permanently and the clearances set to .002 in. inlet and .004 in. exhaust. The ignition timing varies according to fuel and is 42° for petrol or petrol-benzole, or 38° for methanol, or alcohol-rich fuels with the appropriate high-compression pistons. If alcohol is used with low-compression pistons just for the sake of cooler running then an advance of up to 45° may be necessary (*see* page 206 for later engines).

Triumph crankshaft assemblies are built up with the flywheel sandwiched between flanges. This is a very rigid construction but, with the main bearings spaced very widely apart, flexure is bound to occur to some extent and it is prudent to avoid exceeding the maker's recommended limit of 7,500 r.p.m. When working on an engine of unknown history, get the crankshaft tested for cracks in the region of the junction of crankpin and web and, if any defect is found, replace the component. The halves are accurately spigotted and dowelled to the flywheel and alignment will be accurate provided cleanliness is observed during the change-over. This precaution should be taken with any make of two-bearing shaft after a considerable period of use.

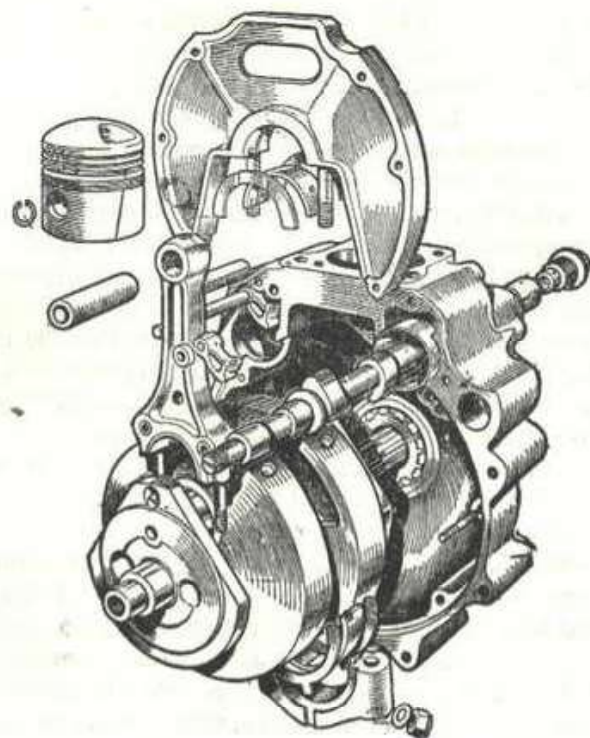
The Norton "Dominator" shaft is of similar three-piece construction and it is wise, when tuning an example which has done much road work to dismantle the shaft and clean out the large cavity which forms a sludge-trap in the centre of the flywheel assembly. Mark all three components beforehand so that they can be refitted exactly as they were so as to preserve original accuracy of alignment and balance.

As a rule parallel twins utilize forged aluminium rods, and though those in up-to-1955 Triumphs run direct on the shaft with white-metalled caps to provide a measure of safety in the event of lubrication failure, it is usual to employ precision-type thin-wall renewable steel big-end shells, such as are commonly used on cars. These components must be handled with care and, in particular, must not be assembled unless the locating nibs are snug in their recesses in rod and cap. If the clearance with the bearing dry exceeds .003 in. the shells should be renewed; the temptation to file the faces of the rod and shells must be firmly resisted, as the amount of "nip" or pinch on the shells has been very carefully determined and it is extremely difficult to maintain the same fit exactly. In addition, once a rod or cap has been filed, replacement shells cannot be fitted correctly unless they also are filed, and, thus, the valuable feature of interchangeability is lost.

Plain bearings must be given a greater quantity of oil at a higher pressure than roller-bearings; they are more easily damaged by fine abrasive which embeds itself in the white metal and acts as a lapping compound. The greatest care, therefore, must be taken to see that every vestige of dirt is removed from oilways and filters; moreover, the efficiency of the oil-feed mechanism to the crankshaft must be checked. The "Dominator," for instance, has a rubber oil-seal installed in the timing cover; it fits over an extension of the crankshaft. This cannot be removed without it being destroyed completely, but it is a simple matter to fit a new one, which is pressed into place with the metal-covered face visible.

Matchless and A.J.S. twins employ a three-bearing shaft, which imparts extra rigidity and permits oil to be supplied through the centre main bearing, and with little likelihood of over-supply to one big-end at the expense of the other. When working on an early A.M.C. model it is worth considering changing the original head with small fins to the later large-finned pattern, which affords considerably greater cooling area. Similarly several pounds of weight





*An extended impression of the A.J.S./Matchless crankshaft layout showing the camshaft and centre bearing support. A.M.C. Ltd. are unique in providing this design feature.*

can be saved by fitting the late-pattern aluminium head to an early Norton "Dominator" with a cast-iron head.

There are three examples of V-twins in current use, namely the 1,100 c.c. and 1,000 c.c. J.A.P. engines, built expressly for racing, mainly in sprint events, and the 1,000 c.c. Vincent in its various guises, "Rapide," "Black Shadow" and "Black Lightning." Additionally, there must still be in existence a number of obsolete makes, of design which could be well worth working on in the light of present-day knowledge. There are of course also the American Harley-

Davidsons and Indians from which formidable power can be wrung.

Dealing first with current J.A.P. engines, the 1,000 c.c. model, known as the "8.80," is virtually two speedway singles on a common crankcase, and has cast-iron heads and barrels with shallow finning and therefore can only be run on pure methanol, Shell "A" or J.A.P. fuel consisting of 97½% methanol, 2% acetone and ½% Castrol "R." Lubrication is either on the total-loss system with sight-feed pump, or on the dry-sump system, according to model; the former is usually used for pure sprint work and sidecar racing on speedways, the latter is superior for longer distances, as some measure of internal cooling is obtained from the oil circulation. Being designed expressly for speed, this engine does not call for much work apart from the routine assembly procedure described in previous chapters.

One major constructional feature is the con-rod arrangement, the rear being forked and the front one plain and running on needle roller bearings on the outside of the sleeve which forms the outer race of the main big-end. The two rods cannot be separated unless the sleeve is pressed out, and as it has three diameters, it can only be pressed out in one direction. This scheme is adopted mainly to simplify assembly as the sleeve can be pushed in by hand for two-thirds of the full distance.

When reassembling, locate the needle rollers in the plain rod with thick grease or with a dummy disc which can slide out ahead of the sleeve as the latter is pressed home.

The 1,100 c.c. version was designed for use in cars and has aluminium heads with inserted seats, and aluminium-jacketed barrels; though usually used with alcohol fuel at around 14 to 1 C.R. it can also be run on petrol merely by changing to 7.5 to 1 pistons and tuning the carburettors to suit. The crankpin has parallel ends and shoulders, instead of the usual J.A.P. tapers, and floating thrust washers, which must not be omitted, are used to locate the rod assembly. Similar washers are also used on each side of the flywheel assembly and their thickness can be varied if



necessary to adjust the end-float to the design figure of .010 in.

The overhead rockers and also the valve guides are lubricated by a suction system common to o.h.v. engines of this make; oil-mist, escaping past the rotary release valve which ventilates the crankcase, condenses in a box below the timing-case and is drawn thence up small-bore pipes to the rocker-boxes by virtue of the depression which exists within the engine. This system only functions if the rocker-covers are in place and are reasonably air-tight, so that it is not possible to check the flow of oil to the rockers by inspection with the covers off. The rotary release valve is driven by the rear camshaft at engine speed and must be replaced with the timing marks in line. In the unlikely event of having to renew the bronze bush, this must be fitted with the slot vertically downwards so that the valve actually closes when the crankpin is a few degrees past its lowest position in the crankcase, or about 30° after rear cylinder b.d.c. The timing can be verified by inserting a piece of wire into the outlet pipe after removing the screwed plug in the bottom of the oil-box. The two rocker-oil pipes projecting into this compartment clear the floor by  $\frac{1}{8}$  in.; this dimension is important and must not be altered.

The duplex pump controls both the pressure and scavenge oil supplies, the latter being drawn from the sump by an external pipe. The slightest air-leak in this suction line will impair the scavenging much more than one would think, and it is vital to make sure that the unions are seating properly and that the nuts are undamaged, otherwise the engine will be sluggish and prone to oil the rear plug.

As two magnetos are fitted, each cylinder can be timed individually. The recommended figure is 36° for the 1,100 c.c. engine and 34° for the smaller one both at a 14 to 1 C.R.

The three versions of the Vincent unit are basically the same; crankcase, flywheels, barrels and timing gear are identical, except for the cams, and so are the heads and valve gear, except for the port shapes and polishing of the rockers.

"Black Shadows" have the same cams as "Rapides," except that they are selected to give "long" timings within the limits of manufacturing tolerances, and con-rods are selected for 65-tons minimum tensile strength and polished. "Black Lightnings" have different cams, 85-ton "Vibrac" con-rods, and a steel idler wheel is used in some early examples; larger inlet ports, and of course racing carburettors and a special racing Lucas magneto are used, with fixed timing pinion instead of the A.T.D. used on the two touring engines. It will be seen therefore that either of these can be "Lightningized" so far as performance goes, without too much expenditure of time or money, and without fear of overtaxing the "downstairs" section.

Standard heads have  $1\frac{1}{8}$  in. inlet ports which can be opened out to any size up to  $1\frac{5}{8}$  in. according to the size of carburettors to be used— $1\frac{3}{8}$  in. T.T.10 Amals are [the usual choice, for which a pair of flanged adaptors, part Nos. ET32/6 and ET 32/7, are required. The ports can be enlarged on the lines described previously, taking care not to go through into the rocker box. As the guide is very short, it is not advisable to cut it off, and a streamlined boss leading to the guide must be left to avoid cutting through into the recess in which the guide lock-ring is screwed. When ultimate in power output is desired, obtain another front head and a pair of  $1\frac{1}{8}$  in. 5GP carburettors, mounted on adaptors 3 in. long, though very good results can be achieved with 32 mm. carburettors. The reason for using a front head on the rear cylinder is that as the port is not positioned on the same side as the rocker box, it can be opened up to  $1\frac{1}{2}$  in. diameter.

These engines are very suitable for short sprint work such as speedway sidecar racing, and quite good results can be obtained by retaining the standard port size and using ordinary  $1\frac{1}{8}$  type 29 carburettors as fitted to the "Black Shadow," but suitably modified for alcohol fuel. "Black Shadow" adaptors, which are bronze, are also required, as the standard aluminium ones are rather too thin for safety when opened up to suit the larger carburettors. Inlet valves



with the heads  $\frac{1}{16}$  in. larger than the standard 1.800 in. diameter can safely be installed and the seat-ring tapered to suit; it is, in fact, preferable to do this rather than to fit new seat-rings if the latter are badly worn.

The inner valve guides are normally of aluminium bronze and held in place with lock-rings. When alcohol fuel and castor-base oil are used, guide-wear may be excessive due to lubricant being washed out by the fuel and cast-iron guides may be a better proposition. Their reduced heat conductivity is not a detriment owing to the cool running nature of the fuel. A special tool is required to remove the lock-rings, which only need firm pressure when being replaced, but must be retained by punch-dots at two points other than opposite the slots. "Black Lightning" cams have about .040 in. more lift than standard cams and, unless the seats have sunk considerably, the guides must be shortened by .050 in. to avoid any chance of the collar contacting the guide. This job can be done *in situ* with a spot-facing tool cutting the guides back until there is  $\frac{7}{16}$  in. space between the guide and the step on the valve stem. Do not use an ordinary drill for the work, as the countersink which it will form in the bore acts as an oil-collector and plug-oiling may result.

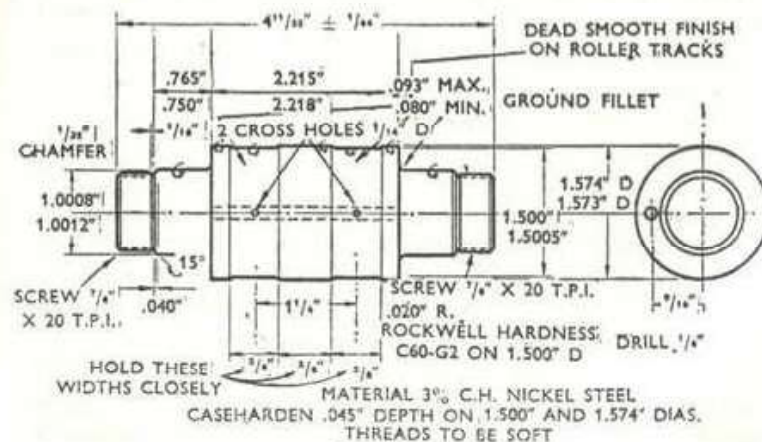
Six types of piston are available, giving ratios between 6.8 and  $12\frac{1}{2}$  to 1. The latter ratio can be used with Shell "M," but Shell 811 gives slightly more power. Although the difference in weight between the extremes in the range is over 3 ounces the change-over causes no perceptible roughness up to 6,800 r.p.m., and, therefore, the motor can be changed from petrol to alcohol without being re-balanced.

Although the ring equipment is apparently the same, the compression rings in pistons E 7/9, 10 and 11, which are the three highest, are of thicker radial depth than those in E 7/6, 7 and 8, and, while it is possible but not advisable to use standard rings in the racing pistons, it is fatal to use thick rings in standard pistons as they project above the ring lands. The oil ring, however, is identical.

For adjusting the compression ratio, base washers up to

.062 in. thick may be used, or a small amount—not more than .030 in.—may be turned off the jacket, though, generally speaking, this should be avoided. There is not much free length on the rocker adjusters, neither is there much room between the head of the adjuster and the inside of the rocker inspection cap at full lift, and damage to the valve gear will result if the adjuster cannot lift freely, which will be the case if the barrels are shortened excessively. However, the pushrods are easily shortened by grinding up to .030 in. off either, or both, ends, and, if necessary, new ones can be made from  $\frac{9}{32}$  in. silver steel, with the ball-ends hardened simply by heating them to redness and plunging into water. Apart from polishing, do not endeavour to lighten the rockers; more will be lost in flexure than will be gained by the reduction in weight.

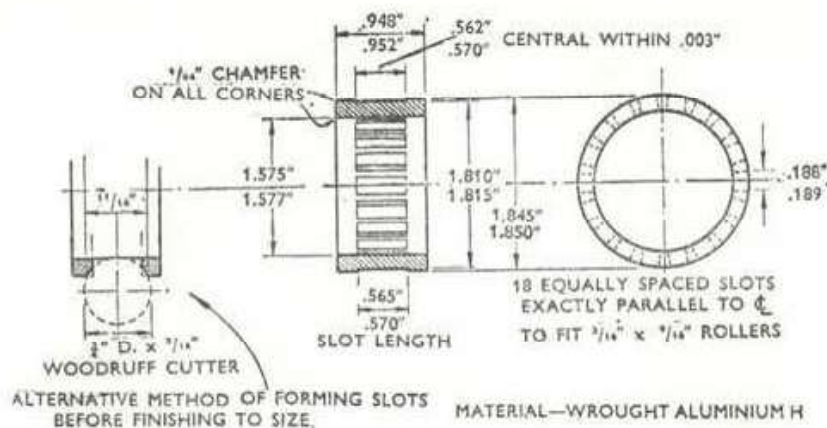
The rocker spindles are .497 in. in diameter and, if one loosens in the rocker, it is often possible to fit a .500 in. gudgeon pin shortened to suit, or else use a piece of hardened silver steel which is simply driven into place, though, naturally, the rocker-bearing bore must be reamed



Details of a modified crankpin suitable for use in a 1,000 c.c. Vincent engine.



## TUNING FOR SPEED



*Dimensions of the MSS-type bearing cage, two of which are used in conjunction with the modified crankpin illustrated earlier.*

out to suit beforehand. Looseness of the rocker-bearing in the head is of no consequence so far as speed is concerned.

In the crankcase section, the standard big-end bearing with three rows of 3 mm. by 5 mm. rollers works satisfactorily for a long period if revs are kept below 5,800, but prolonged work above that speed shortens its life, so also does racing the engine from cold before the oil has a chance to warm up and circulate freely. If genuine spares cannot be obtained, a very satisfactory big-end can be made by making a new pin as per the diagram, using two MSS Velocette cages with 18  $\frac{3}{16}$  in. by  $\frac{3}{16}$  in. rollers in each. Four of the original roller spacers must be ground down to .078 in. thickness and placed one each side of the cages to centralize them, while the original sleeves which are hard all through can be ground out without moving them from the rods to 1.875 in. The MSS cages must be bored out to slip neatly over the shoulders of the pins but, if these components are not available, cages may be made up. Their width should be .950 in. so that the steel spacers can be omitted altogether, although, of course, the central distance-piece must be retained.

## TWIN-CYLINDER ENGINES

As the big-end oil is fed into an annular space, the oil supply cannot be cut off by incorrect assembly, but the two oil holes leading to the rollers should be placed at a three o'clock position looking towards the drive side. If placed at 12 o'clock, the additional oil pressure created by centrifugal force causes almost all the oil to pass out through the first hole, thus starving the drive-side big-end.

In the timing-gear department, the front exhaust cam-spindle should be locked, after fully tightening, by angle-drilling a small hole from the end of the slot in the head back towards the camshaft and fitting therein a split-pin. Worn cam followers can be trued by grinding or oil stoning, but the face must be kept parallel with the hole. Care must be taken to note the positions of all the thin hardened washers behind the steady-plate and to replace them in exactly the same order. The lowest hole in the plate is purposely made a loose fit on the idler wheel spindle to allow for adjustment, but it is best to cut a strip of soft aluminium as wide as the plate thickness and  $\frac{1}{32}$  in. thick, wrap it into a circle round a  $\frac{5}{16}$  in. bar and pull it into position in the steady plate with the idler-spindle nut. This holds the outer end of the spindle positively in position, the soft bush being able to flow somewhat to accommodate itself to any slight eccentricity of the hole and spindle.

Better crankcase ventilation is obtained by filing straight across the breather-valve sleeve until the slot is fully  $\frac{3}{8}$  in. wide, then rounding the outer edges of the flat so that the outgoing air has a clearer passage; the edges of the slot should be left sharp to act as oil scrapers. Replacing the external banjo and pipe with a straight union and a large-bore pipe running upwards and rearwards also helps ventilation.

Cams and camwheels are press-fitted together and are regarded by the factory as inseparable units. Consequently, "Black Lightning" cams are only supplied with the wheels attached, but it is possible to change cams and wheels provided that the interference fit is between .001 in. and .002 in., otherwise either the wheel may slip or it may crack through overtightness. All cams are ground in relation to



the slot in the end-face, which must be placed in correct relationship to the marked teeth. This can be accomplished by scribing lines on both the faces of each gear, and accurately lining up the new cams with the lines before pressing them home level with the face of the gear. There are timing dots on both gear-wheel faces, so be careful to place each gear on the same way as it was originally; the timing should then be correct when all the gears are again assembled to the marks.

A rev. counter cannot be driven from the magneto-pinion nut satisfactorily, partly because it is a clockwise magneto and the nut would tend to unscrew, and partly because the automatic advance would not operate correctly. It is possible to devise a component which fits loosely over the A.T.D. and is driven from the fixed stops on the fibre pinion, but the better scheme is to obtain a fixed pinion as used on the "Black Lightning," together with the driving dog and special magneto pinion cover.

Strictly speaking, the magneto should be converted to manual advance, but with high-alcohol fuel the engine will start at the full advance of  $34^\circ$  at a  $12\frac{1}{2}$  to 1 C.R.

As there is no provision for measuring tappet clearance the method adopted for checking valve timing is to attach a dial indicator to one head stud and screw the tappet adjusters down until the valve is just lifted from its seat by .002 in. to .003 in., the exact amount being immaterial as it merely serves to take up all backlash. Re-set the dial gauge to zero and turn the engine until the valve is lifted a further .005 in.—this is important. The point at which this lift occurs is taken as the opening or closing point of the valve according to whether the valve is lifting or dropping, and all the points so obtained should agree with those quoted in the Table in Chapter XII. If there is an average discrepancy of more than  $4^\circ$  between the two cylinders and there is still a discrepancy when re-checked at .010 in. lift, it may be necessary to re-fit one of the two camwheels; the timing as a whole can of course be set by re-positioning the half-time pinion on any one of its five keyways.

Before replacing the timing cover, see that all four of the synthetic rubber seals are in place in the recesses provided, and in no circumstances omit to fill the filter chamber with the correct oil before fitting the banjo plug in the cover. Failure to observe this precaution may lead to damage occurring in the time lag before the pump fills the chamber.

In final assembly, verify that all shock-absorber springs are nestled in the recesses in the spring plate and then get the mainshaft nut really tight. The hexagon size is  $\frac{3}{4}$ -in. Whitworth (the same as the crankpin nuts), and an S.A.E. spanner  $1\frac{5}{16}$  in. across flats also fits. The tube spanner in the kit is inadequate for the job, which requires a solid steel box or ring spanner.

The best scheme is to hold the crank assembly with a  $\frac{7}{8}$ -in. bar through one small-end eye; in any case it is always advisable to tighten up the shock-absorber assembly before reassembling the cylinders, so being able to verify that all is correct down below, and to measure the centrality of the small-ends in the crankcase bores. The Series "C" Lightnings were fitted with caged big-ends; each cage has 12 slots containing three  $\frac{1}{4} \times \frac{1}{4}$  in. rollers. The  $1\frac{5}{16}$  in. diameter crankpin, instead of being shouldered down where it abuts against the hardened side-plates is deeply recessed into the flywheels, which are pulled up by the usual nuts against the ends of the pin. Because of this, this design of big-end cannot be fitted directly to standard wheels, but these can be re-machined to suit by very accurately boring and facing the required recess. New side-plates and spacer are also required, though the big-end sleeves remain as before. "Picador" engines (the version developed for small pilotless target aircraft) utilized a similar big-end, but with a parallel crank pin, without nuts, which has a .006 in. interference fit in the flywheels and needs special equipment for assembly or dismantling. The main-shafts are larger than standard, so the "Picador" assembly cannot be used as a direct substitute for the standard assembly.

Series "D" engines were fitted with a much improved shock-absorber with a greater number of springs and pro-



vision for positively locking the mainshaft rut. No additional machining is required to make the conversion, which is well worth doing.

The foregoing notes are in the nature of underlining some features of this power unit which are a little unusual to those not familiar with its construction, and must be read in the light of what has been said elsewhere in this volume.

Earlier in this chapter the Triumph GP conversion kit was described. For T100, TR5, T110 and TR6 engines a set of components is available, including a cylinder head with splayed ports which besides affecting gas-turbulence provides plenty of room for two carburetters. The new camshafts supplied must be used in conjunction with the new followers to obtain the correct valve-opening diagram, and the method of timing them is substantially as previously outlined, but instead of setting the tappets to zero clearance, they must be set to .020 in. and the camwheels positioned to obtain the nearest approach, on all four valves, to the figures shown on page 165. For running, of course, they must be re-set to the correct clearances, and as an illustration of the effect which the amount of clearance makes, the actual timing then alters from 35°, 56° to 59°, 80°, an increase in valve-open period of 48°. Ignition point is 42° for the 500 c.c. engines, but only 39° for the 650 editions.

The splay-port head will not fit directly on early cast-iron or aluminium barrels, because of an alteration in spigot height from .187 in. to .124 in. When using alcohol with a compression-ratio of 12 to 1, an iron barrel should be used, partly because of the thermal requirements of this fuel and partly because of the added strength in the base flange. However, irrespective of which type of barrel is used the spigot height must be checked and reduced if necessary to .124 in. if a splay-port head is being fitted; alternatively, iron barrels with the low spigot height are obtainable as spares.

### Twin-cylinder Carburation

Mention has been made already of the advisability of using separate carburetters on a vertical twin, and the need is even greater on a V-twin since it is virtually impossible to obtain equality of mixture with a single carburetter owing to the unequal periods between induction strokes.

The usual method of operating two carburetters is by means of a single wire from the twist-grip to a junction box, from which separate wires run to each throttle. This system sometimes causes the grip operation to be rather heavy and some riders are persuaded to shorten the throttle springs as the easiest way out. This is a bad idea because, if the slides do not close properly, erratic tick-over results. Pay attention to the run of the wires, leaving off all clips so that the wires can adopt their natural position. This usually effects a cure, but the best method is to obtain a dual-cable twist-grip or modify the existing one to take two wires.

It is essential to adjust each control so that both cylinders accelerate absolutely in unison. The best method with standard Amals is first to slacken both cables and set the tick-over on each cylinder by means of the throttle stops and pilot screws until even running is obtained. Next set the cable adjusters so that the slightest grip movement causes a rise in engine speed with each cylinder firing equally.

If with both slides apparently moving simultaneously, firing is still uneven, try the effect of closing each air lever in turn. If, say, one cylinder is missing or does not fire at all yet cuts in when its air control is closed, less cutaway is required on the throttle slide. In border-line cases, an enriching of the pilot mixture may help the offending cylinder to come off the pilot at the expense of slightly erratic slow-running. It is, of course, essential to have an individual air control to each cylinder; by intelligent use of that control equal carburation can be obtained throughout the entire range.



As there are no throttle-stops on racing-type carburettors, this system cannot be used; the idling speed can only be adjusted by varying the cable lengths but, if cut-aways on both slides are equal it is essential to verify that they both disappear simultaneously at the tops of the choke as the grip is opened, otherwise acceleration will not be good. If there is a big difference it will be necessary to check for air leaks or fuel blockage which may have affected the idling speed. Incidentally, it is quite useless to attempt to adjust the idling setting on one cylinder only with the other cut out by shorting the plug. The amount of throttle required to pull the engine round with only one lung operating is so much greater than with both in action that the scheme gets you nowhere.

Accuracy of the main jet settings must be verified finally by the appearance of the plugs, in exactly the same way as for a single-cylinder machine. Even if both carburettors are of exactly the same type and size, there is no certainty that the main jets, throttle cutaways and needle settings will all be exactly the same, because these are likely to be affected to varying extents by the air-pressures existing at the intakes and which are almost bound to be different in each case according to the local air-flow or the localities in which the intakes are placed. The position of the float-chambers in relation to the jets has a marked effect on mixture-strength during acceleration as described in Chapter XVIII, and this may be another cause of a difference in setting between the two instruments for optimum results.

## TWO-STROKE ENGINES

TWO-STROKE engines differ fundamentally from four-strokes insofar as the charge is not induced directly into the cylinder, but is transferred to it from the crankcase, or from a separate pump which may be either a rotary blower or a cylinder-and-piston mechanism. Either of the two last-mentioned devices are permissible for record-breaking but, under the rule that prohibits the use of superchargers in road-racing, an auxiliary pump which has a greater swept volume than that of the cylinder is definitely "out" and so are additional pumping pistons which act to increase the volume drawn into the crankcase.

Of recent years, therefore, the tendency has been to discard supercharged, or augmented-induction, designs which were in vogue before 1939 and to concentrate on the much simpler forms, relying solely upon straightforward crankcase compression such as are commonly used for touring work. Some really amazing results have been achieved by working upon these engines, speeds of 85 m.p.h. and over being obtained by private owners from 125 c.c. B.S.A. "Bantams" without recourse to fuels such as nitromethane and good results can also be obtained from Villiers engines, which are used in many current machines, and the more complex but still relatively simple split-single E.M.C. and Puch designs.

The basic line of development remains the same as for the four-stroke, namely to get the maximum quantity of fresh charge into the cylinder at the highest possible r.p.m., but it is more difficult to carry out because there are two sources of loss in volumetric efficiency—one in getting the fresh mixture into the crankcase and the other in transferring it to the cylinder. Consequently every endeavour must be made to reduce any losses in breathing ability to a minimum.



Dealing with the crankcase first, the initial step is to reduce the clearance space within it as much as possible by using disc flywheels or by building up existing flywheels or crankwebs until they almost fill the surrounding space and only just clear the connecting rod. This procedure may introduce some problems of balancing which may have to be solved later, but, at the outset, filling must be done in the lightest fashion either by using magnesium blocks or by sheet metal work, though in the latter case all joints communicating with cavities must be air-tight, otherwise the value of the scheme is largely lost. There must however be some clearance between moving and stationary surfaces. A certain amount of gas movement takes place therein, so it is advisable to polish both the flywheel assembly and the whole interior of the crankcase to assist this movement.

When, as is usual, the inlet port is controlled by the piston skirt, the time available for induction is extremely short, less than  $1/300$  sec. at 6,000 r.p.m., which can be considered a comparatively low speed for this type of engine in highly-tuned form. The ability of the port to pass gas depends upon its "time-area integral"; that is the combination of the area open at any given instant and the total amount of time elapsing between opening and closing points. Widening the port around the circumference gives a greater area, without altering the timing; deepening the port increases both the area and the time of opening, but must be done with great care, because the effective compression stroke of the piston in relation to the crankcase does not commence until the inlet port closes, unless the induction tract is of such a length that an appreciable ramming effect is generated by the fast-moving column of gas. To some extent, this ramming action is always present, but unless it comes at the right time, which is just towards the latter end of the closing period, it will be of little assistance. Actually, the combination of the crankcase and induction pipe (including the carburetter) constitutes a resonant system equivalent to a closed vessel with extension pipe and has a natural vibration frequency which can be

used over a limited speed range to augment the ramming effect very considerably. Whether the problem is amenable to calculation with any accuracy is rather doubtful because of the number of variables involved, so the solution is best arrived at by trial and error.

Widening the port around the circumference is quite permissible and has little adverse effect because the piston-rings do not usually traverse this port, and naturally all the interior surfaces must blend well into each other and be polished. Provided that one is prepared to sacrifice low-speed torque, relatively enormous carburetters can be used;  $1\frac{1}{2}$  in. or  $1\frac{3}{8}$  in. type 29 Amals give very good results, and are less expensive than the R.N. or T.T. patterns.

From the crankcase, the gas, after compression, enters the cylinder through the transfer ports, and the angle of entry of these into the cylinder is of great importance in all flat-top piston designs. The early versions used two opposed exhaust ports and four transfer ports, the latter being so arranged that the streams of gas from each impinged at the centre-line and deflected each other up towards the top of the cylinder whence they curled downwards towards the exhaust ports. In later versions with only a single exhaust port there are only two transfer ports, which direct the fresh gas across the piston crown almost in the opposite direction to the outgoing exhaust gas. The streams then combine, travel upwards towards the head and then downwards towards the exhaust ports, the principle being referred to as loop-scavenging.

The angles of the entering gas streams are most important to obtain best scavenging with least loss of fresh charge out through the exhaust port and, though the ports can with advantage be enlarged a little, this process is best done in stages. In some designs, small cover-plates are provided through which access to the ports can be obtained but in others the casting is solid. In the case of the latter it is possible to build up around the elbow of the transfer passage with bronze, then machine the surface off level and break into the passage. The holes can finally be closed by fitting



screwed-on cover-plates with internal extensions shaped to conform exactly to the interior contour. Any obstruction to gas-flow in this region is extremely detrimental, so that the whole of the transfer passages must be smoothed off and particular care taken to improve the entry into them from the crankcase by rounding off all sharp corners.

Exhaust ports in touring engines are usually made with the top edge inclined or angular; this gives a gradual opening and, by taking some of the "crack" out of the exhaust noise, simplifies the silencing problem to some extent. The inclined edge gives the rings an easier passage as they pass over it on the upward stroke. For racing, it is desirable to obtain the quickest possible rate of opening in order to make use of what is termed the "Kadenacy effect." Kadenacy discovered that, if the ports are large enough and are opened with sufficient rapidity, escaping gas rushes out with such vim that the cylinder pressure drops several pounds below that of atmosphere, and, by adding an exhaust pipe of the appropriate length, fresh gas is drawn into the cylinder through inlet ports even without the aid of crankcase compression or an external blower. In fact, one industrial engine, the Petter "Harmonic," after being started on crankcase compression, is run thereafter purely on air induced through the agency of tuned exhaust and intake pipes. This engine, being a diesel, operates all the time with a full air supply to the cylinder and is thus not directly comparable with the carburettor-type engines under discussion, but it is mentioned to indicate the great influence and importance of the exhaust system.

During the period of port-opening we have a cylinder, with its capacity continually varying as the piston moves towards b.d.c., connected by an orifice (the exhaust port), also of varying area, to an exhaust pipe, which is the only non-variable item in the system. As a further complication, the natural frequency of any system depends upon the speed of sound in the gas contained in it, and though it is usual to take 1,500 ft. per sec. as being a reasonably close figure for a normal four-stroke engine, sound-wave velocities of up

to 4,000 ft. per sec. may be attained in a short exhaust system with a rapid discharge of high-pressure gas through piston-controlled ports. This factor alone would make calculation of suitable exhaust-pipe length difficult, to say the least, and may, in conjunction with the other variables, account for the wide diversity of systems in use. The Lambretta, for instance, had, in effect, a curved megaphone which starts right at the port and widens to about 4 in. diameter in 16 in. length, whereas the Eysink Villiers had two pipes, of more normal length, exhausting into fish-tailed expansion boxes. B.S.A. "Bantams" operate well with a 1½ in. pipe about 20 in. long, but for r.p.m. in excess of 8,000 the pipe can be shortened to about 10 in. if the regulations will permit a system of that length.

The functioning of a two-stroke exhaust is more complex than that of a four-stroke; it must first assist in evacuating the cylinder rapidly and in so doing is almost bound to draw a certain amount of fresh charge out through the port. By correctly proportioning the exhaust system, this charge can not only be retained in the pipe but can be rammed back into the cylinder by a wave of positive pressure arriving at the port towards the end of its closing period. Conversely, if the system is incorrect or the relationship of the transfer and exhaust closing points is wrong, a considerable portion of the charge may be lost irretrievably. Poor power development with a high fuel consumption and the strong smell of unburnt fuel in the exhaust are all signs that serious charge-loss is taking place. If the work is being conducted under laboratory conditions, an Orsatt gas analyser is an invaluable instrument for determining charge-loss as, by its use, an accurate measurement of the amount of free oxygen in the exhaust is obtained. There is bound to be some free oxygen, up to perhaps 2%, due to combustion not being absolutely complete, but anything above this figure (provided the mixture is correct) represents an actual charge-loss of *five times* the amount of excess, because, of course, only ½ of the atmosphere is oxygen. With the aid of the Orsatt device, therefore, the effect of port- or exhaust-



pipe modifications can be determined quickly, but without it, recourse must be had to the time-honoured "cut-and-try" method which, fortunately, is not too difficult or costly with a small single-cylinder engine.

A general scheme which has possibilities is to raise the upper edge of the inlet port until it is  $\frac{1}{8}$  in. or so higher than the edge of the piston crown at b.d.c., so placing the carburetter in direct communication with the cylinder for a few degrees of crank-travel when the exhaust port is almost fully open. The reduction of cylinder pressure which comes about by the Kadenacy effect then causes fresh mixture to be induced through the inlet port at the same time that the normal supply is being fed into the cylinder via the transfer ports. There will be no supercharging created thereby, but a gain in power will result because of improved scavenging of the residual exhaust products. This device is essentially one which will work well only over a limited speed range and may result in very poor power output at other speeds.

When modifying port heights, the simplest method for trial purposes is to alter the cut-off edges of the piston instead of altering the barrel. The effect is not exactly similar because whilst the port timing is altered the actual area is not, but it will at least give an indication, and if a mistake is made, it is cheaper to replace a piston than to start all over again with another barrel. One of the most exasperating things about two-strokes is the difficulty of duplicating results; two barrels with ports which appear to be identical will rarely give equal power, possibly because small differences in port contour, which are difficult to measure, create differences in the direction of gas-flow and modify the degree of scavenging to a much greater extent than might be expected. The moral of this is that if you do get a good barrel, look after it; do not think that all you have to do when the experimental one is worn out is make another which looks the same.

One of the difficulties with a two-stroke is cooling; the cylinder does not have the benefit of the idle inlet stroke, nor is the fuel vaporized actually in the cylinder as in a four-

stroke. Consequently the heat-loss to the walls is high, yet a large part of their area is occupied with ports which cause unequal temperatures and local distortion. Some riders go to the trouble of turning off the fins above the ports and shrinking on a finned aluminium jacket, but this does not really get down to the root of the matter and it is doubtful if it is worth doing on an engine used on road circuits where the greater cooling required by the increase in power is automatically supplied by the increase in air-speed.

Cylinder-heads with larger fins and a smaller combustion space giving compression ratios of about 12 to 1 have become commercially available for B.S.A. "Bantams"; as the pattern-work is relatively easy these are not difficult to make, though trouble with cracking round the plug-boss may be experienced unless the material is good. Heat-treated Y-alloy or R.R.53 will be perfectly satisfactory. In designing such a head it is easy to obtain an impressive appearance without much gain in cooling; to be of real value, fins must spring directly from the actual metal around the combustion space, there being little or no virtue in putting a fin on a fin. The old Villiers system of radially-disposed fins is excellent as the positioning of the fins is good and the air-stream has a clear path past each one, which is not always the case with deep parallel fins.

Flywheel magnetos appear to be able to run satisfactorily at any speed which can be got out of the engine but some experimenters prefer to utilize a 180° twin-cylinder magneto, running at half-engine speed with both leads going to one plug, which was standard practice on the E.M.C.-Puch, or with each lead going to its own plug in a special two-plug head, a scheme which gives the plugs an easier life. The reason for using a half-speed magneto is simply to reduce stresses in the instrument, as a normal magneto is not ordinarily called on to run at more than 4,000 r.p.m. It is essential, however, to have the magneto checked electrically to verify that the sparks do occur precisely at 180°, otherwise correct ignition cannot be obtained as a one-degree variation on the magneto means 2° variation on the engine.



It is general practice to use uncaged roller big-ends, there being insufficient lubrication to permit the use of cages. In some Villiers engines, the steel rollers are alternated with bronze rollers of slightly smaller diameter, the idea being that the bronze rollers do not carry any load but merely separate the steel rollers and rotate backwards; other models use the conventional crowded-row system. The big-end assembly of the B.S.A. was enlarged in 1954 and the later pattern should be used as a replacement on early models.

Interference-fit crank-pins without retaining nuts are commonly employed and can be assembled either in a press or a stout vice with parallel jaws. As there is some loss of interference-fit caused by removal and refitting, the replacement pins should be .001 in. larger than the originals. Villiers spares are supplied as standard with this amount of oversize. Great care must be taken to start the pin squarely in the hole to avoid any chance of damaging the surface, and a piece of shim steel .010 in. thick should be interposed between the rod and one crank-web to limit the final position of the crank-web and provide the necessary amount of side clearance. For finally truing-up the shafts, the easiest way is to hold the assembly in one hand and knock one crank-web in the required direction with a copper hammer; this is more effective with small assemblies than the "bumping" method employed with much heavier four-stroke flywheels.

Although seemingly simple, the design of a twin two-stroke crankshaft is not easy, because of the necessity to provide a sealed bearing between the crank chambers. On the Excelsior "Talisman" the right-hand assembly is built up and fitted to the case; one-half of the left-hand assembly follows and the central retaining nut fully tightened. Then the con-rod is fitted to this assembly and the last crank-web is placed in position. To facilitate this part of the work, the crank-pin is not a very tight fit in the web, the necessary rigidity being provided by supporting the main-shaft in two relatively widely-spaced bearings.

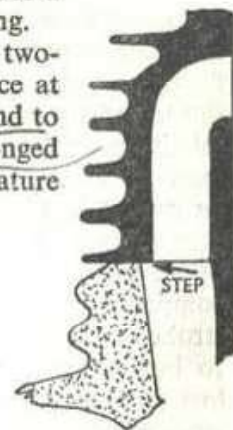
Crankcase pumping efficiency depends largely upon the seals provided at the main bearings, and which vary from

make to make. The system common to all modern Villiers engines consists of flanged bronze bushes which are a sliding fit in the crankcase and have a slight clearance on the shafts. The flanges are pressed lightly against the crankcase by star-washers; none of these parts can be examined unless the case is heated and the bearing tapped out but they give little trouble unless the engine has been run with loose main bearings.

On B.S.A. "Bantam" engines there is a number of shims and one collar on the mainshafts; these must be replaced in exactly the same order, so their initial position should be remembered or noted down.

Mention was made in an earlier chapter of the necessity for accuracy of fit of two-stroke pistons, which have to be reasonably gas-tight on the skirt and must not run with excessive clearance. One line of attack is to fit the piston deliberately a little on the tight side, and give the motor a short burst on full throttle after sufficient running to warm everything up thoroughly. Then strip down and very carefully ease off piston areas on which excessive pressure is evident; a fairly rough file is better than a smooth one for this work, as the file marks act as oil reservoirs and may save a bad seizure. This process should be repeated several times until the motor will hold full throttle for a couple of miles with no sign of tightening.

It is useless attempting to run-in a two-stroke piston by covering a long distance at part-throttle openings; it is almost bound to seize the first time it is given a prolonged burst at full bore, because the temperature



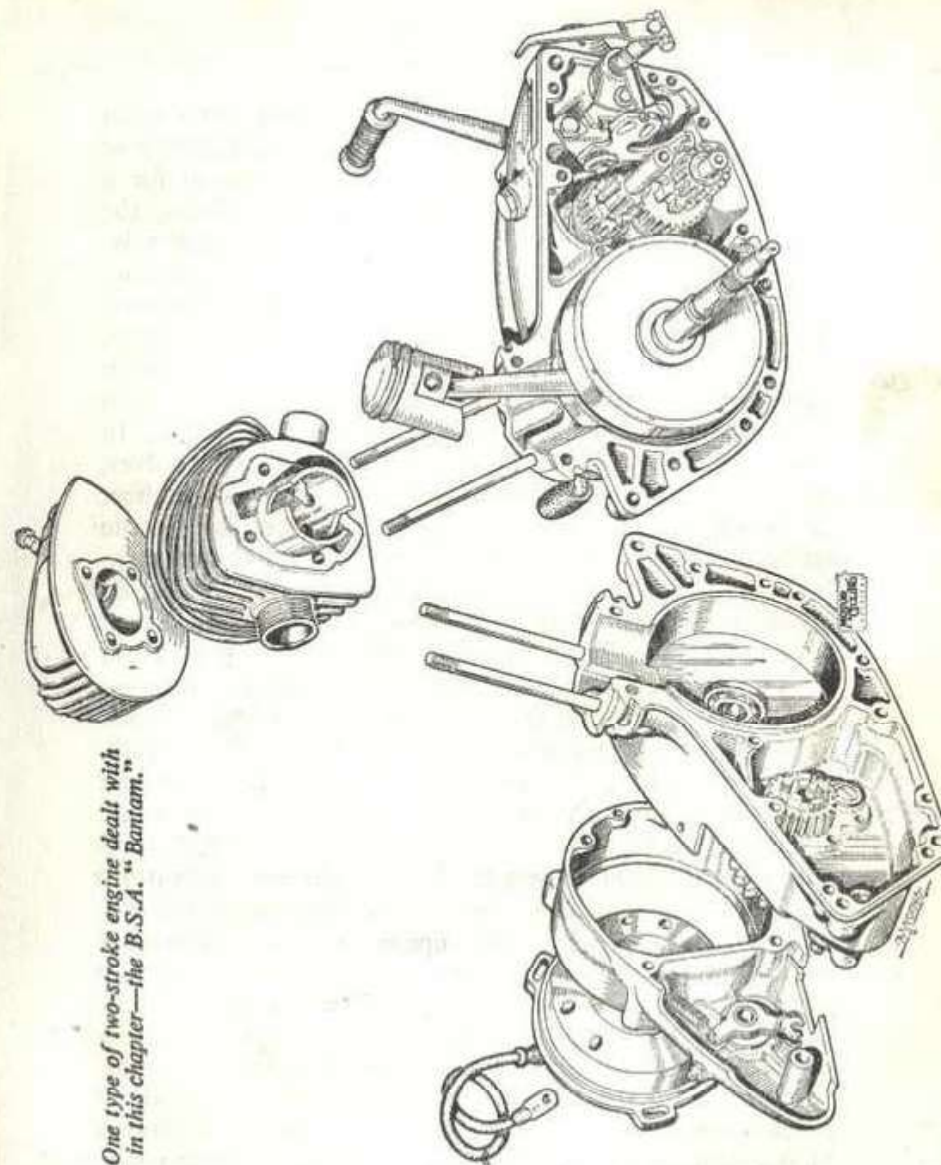
*A step in the crankcase-transfer port joint hinders gas-flow and, therefore, impairs performance.*



distribution and, consequently, the shapes and dimensions of piston and barrel are then quite different from what they were during the light-load running period.

Obviously, the lower the coefficient of expansion of the piston material, the less will be the amount of distortion for the same temperature difference, and this has led to the introduction of pistons made from 16 per cent or 22 per cent silicon alloy, which are much better in this respect than any of the alloys regularly employed in quick four-strokes. The 1959 A.M.C. two-stroke piston is in 22 per cent silicon which enables the skirt clearance to be reduced to  $1\frac{1}{4}$  thou. without fear of seizure. This alloy is also said to be less prone to burning of the top land at the danger-zone opposite the exhaust port: in an endeavour to eliminate this trouble completely some designers have experimented with chrome-plating applied to the crown and top land, but this is by no means a general practice and severe trouble would follow if the plating ever decided to flake off.

Another line of attack against the distortion problem pursued on the 125 and 250 twin M.Z. racing engines is to fit a relatively thick austenitic cast-iron liner, in which the ports can be accurately machined, into a very rigid finned sleeve made from high-silicon alloy and therefore closely matching the liner in its rate of expansion. Before fitting the liner, the transfer passages can be fully machined in the sleeve and given a high degree of surface finish; when finally assembled, the liner forms the dividing walls between the transfer passages and the bore. Incidentally, whilst a relatively small crankcase volume is desirable in order to increase pumping efficiency, it appears that there is more to be gained in the long run, by using transfer passages of generous cross-section and reduced resistance to flow, than by restricting their size in order to raise the crankcase compression ratio. Positive pressure existing on the down-stroke and negative pressure on the up-stroke both give rise to internal power loss and should therefore be kept as low as is possible, consistent with maintaining pumping efficiency.



*One type of two-stroke engine dealt with in this chapter—the B.S.A. "Bantam."*



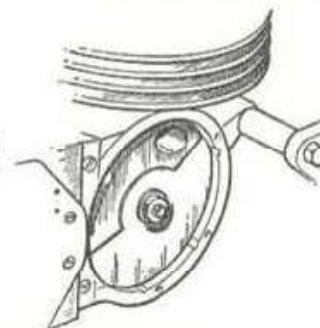
### Mechanical Inlet Valves

Piston-controlled inlet ports are only open for a short while at the top of the stroke, and a negative pressure of five or six pounds per square inch will be present for a considerable portion of each up-stroke; in addition, the very short time of opening must limit the amount of mixture which can be induced, even with the aid of a long induction pipe to give a ramming action, as used on the Adler. Lengthening the port increases the time element, but reduces the effective piston stroke, a condition which can only be eliminated by the use of some sort of valve which may be either automatic or positively operated. In the former class come the "reed" or diaphragm valves, once employed by D.K.W. and used with good effect today on some American outboard engines, and in the latter come the rotary valve built into the centre main bearing of the British-Anzani twin, and the disc valve used in some model aircraft engines and in the M.Z. In this specific instance, the disc is made of sheet steel, only half a millimetre thick, attached to a splined boss which floats on the mainshaft so that the disc is free to centralize itself in a narrow space provided between the crankcase wall, which is ported, and a cover plate which carries the inlet pipe and carburetter. Portion of the periphery of the disc is cut away so that it uncovers the port for a period of about 200°, but in order to maintain crankcase pressure during the transfer period, the inlet port opening does not commence until the transfer ports are nearly closed. One great advantage of this form of valve is that it is quite simple to alter the duration of opening by cutting away more of the disc or to alter the timing by re-positioning the boss on its splines, so it offers a fruitful field for investigation.

In the British-Anzani layout, the ports are formed by holes drilled in the centre journal, but are necessarily limited in size compared to the areas possible with a disc valve. Nevertheless, this engine gives a very good performance,

which could perhaps be improved by careful work in the port-drillings to improve their shape and surface finish.

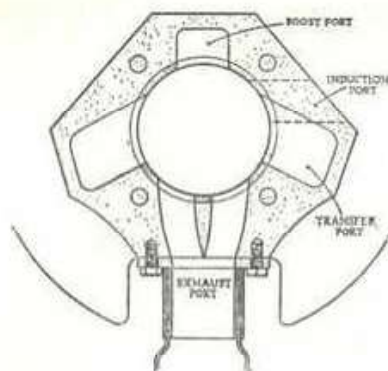
*M.Z. disc valve, with cover plate removed to show port.*



### Additional Transfer Ports

An idea used on D.K.W. and M.Z. engines and on which J. Ehrlich, the designer of the E.M.C. has done a great deal of work culminating in the granting of patents, is to utilize one or more additional transfer ports in the cylinder communicating with passages termed "regenerative" or "booster" chambers though neither term is an exact description. In the M.Z. version, a single chamber is located immediately opposite to the exhaust, and it opens to the cylinder via a port slightly lower than the transfer ports. At the inner end, there is another port in the cylinder-liner which is closed for most of the time by the piston, but is opened to the crankcase by a port cut in the piston wall just below the rings. During this "open" period compressed mixture is forced into the boost-chamber and retained there by the further descent of the piston until the boost-port into the working cylinder is opened by the top edge of the piston. A "puff" of mixture is thus discharged across the crown and clears away any dead gas which has not been scavenged by the streams from the two transfer ports. An additional and very real benefit of the scheme is that the mixture inside the piston does not remain stagnant, but is moved on at each stroke, thereby ventilating and cooling the underside of the crown.





*Boost port. A third transfer, or "boost" port situated opposite to the exhaust port is used on several high-performance two-strokes.*

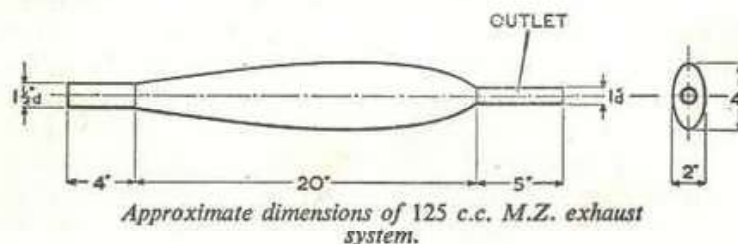
One Ehrlich design utilized two boost-ports, each located between the exhaust port and the transfer port, but the water-cooled version used on the E.M.C. from 1960 on has only one boost-port located, as in the M.Z., opposite to the exhaust. The Japanese Suzuki also contains this feature which, in conjunction with the rotary disc-valve, produces a power output of over 180 b.h.p. per litre at 11,000 r.p.m. Since reduction of internal friction is most important at these high revs, roller-bearing small-ends and chrome-plated steel rings are used.

In the  $54 \times 54$  mm. 125 c.c. M.Z. the divided exhaust port is 25 mm. high and opens and closes  $80^\circ$  each side of b.d.c. The transfer ports are 15 mm. high, opening and closing at  $68^\circ$ , while the boost-port is 12 mm. high, opening and closing at  $64^\circ$ . The disc-valve opens at  $45^\circ$  after b.d.c. and remains open for  $205^\circ$ .

### Exhaust Systems

Earlier in this chapter, mention was made of the wide diversity of exhaust systems which have been tried, but of recent years the principle of exhausting into an expansion chamber with a very small outlet is universally used on Continental two-strokes. The details differ between make

and make and are finally settled by experiment, but the general scheme is to connect an expansion chamber of twelve to fifteen times the cylinder volume to the exhaust port with a very short pipe of large diameter, and to restrict the outlet either by using a very-small-diameter pipe (D.K.W.) or a larger pipe possibly with a variable method of reducing its area. On the 250 c.c. twin racing Adler, for instance, a washer with a hole about 18 mm. diameter is brazed into each tail-pipe, but on the M.Z. the pipe is clear, the whole proportions being approximately as shown in the sketch below. The "washer" method of varying the restriction would enable a series of experiments to be made very quickly and obviously there is much to be gained by arriving at a correctly-proportioned system of this nature.



### Combustion-chamber Shape

The latest tendency in combustion-chamber shape is to employ a certain amount of "squish" by machining a portion of the surface so that it conforms to the section of the crown when the piston is at t.d.c. with 30 to 40 thou. clearance; with any amount appreciably less than this, the piston may touch at maximum revs.

The most favoured crown shape is a flattened segment of a sphere which provides the best combination of strength with absence of distortion, and provides good flow-lines for the working gas as it enters or leaves the ports.

The compression ratio can be as high as 14 or 15 to 1, even with petrol if of 100 octane, and the required volume



is obtained by machining a depression into the squish area; a position slightly off-centre and towards the front of the engine is used on the M.Z. but the optimum position and also the location of the plug may vary, as both depend on the direction taken by the gas at the end of compression under the influence of the turbulence created during the scavenging period. As the cylinder-bolts are usually disposed in a square formation, it is a simple matter to try the effect of reversing the head to alter the position of the combustion space in relation to the cylinder centre line. Sometimes, it is found that a plug-position which gives the best power is one which is prone to oiling or "drowning" the plug when starting or when opening-out after shutting-off for a corner and in that case a compromise may have to be effected.

The foregoing remarks are mainly of a generalized nature and many apply to engines which are unlikely to come into the hands of private owners, but they do show the lines along which the ultra-high-speed two-stroke is developing and indicate ways in which an earnest seeker after power may attain satisfactory results, if he has the right facilities. Most people will, however, have to be content with extracting more urge from commercially-obtainable motors which usually have piston-controlled inlet ports and are commonly though not quite correctly, referred to as the "flat-top piston" type. Of these, the B.S.A. "Bantam" has often been selected as a basis for the tuner's art, and the following remarks are based on the experience of A. E. Rose, who had a considerable degree of success with this engine. Although dimensional details would naturally differ, the same procedure would apply to other makes such as C.Z. or Jawa which operate on similar design principles.

The standard B.S.A. port timing, according to Rose, is:

Exhaust opens 70° b.b.d.c.  
Transfer ports open 60° b.b.d.c.  
Exhaust fully open 10° b.b.d.c.

Transfer ports fully open 10° b.b.d.c.  
Transfer ports close 60° a.b.d.c.  
Exhaust closes 70° a.b.d.c.

Opening period slightly later on recent engines.

The carburettor inlet aperture has a V-notch at the bottom, and for this port the timings are:

Closes to bottom of rectangle 50° a.t.d.c. (130° b.b.d.c.)  
Closes to bottom of V-notch 60° a.t.d.c. (120° b.b.d.c.)  
Opens to bottom of V-notch 60° b.t.d.c. (120° a.b.d.c.)  
Opens to bottom of rectangle 50° b.t.d.c. (130° a.b.d.c.)  
Fully open but shrouded by  $\frac{1}{32}$  in. of piston skirt at t.d.c.

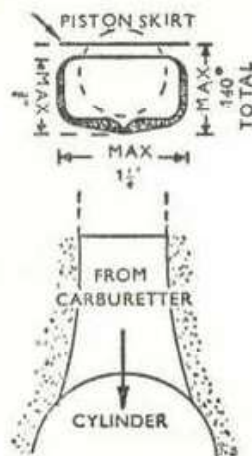
To obtain better breathing it is necessary to extend the ports both in height and width. In order to minimize the amount of grinding work and filing, and to improve access to the port, the inlet stub should be sawn off to within  $\frac{1}{16}$  in. of the rear fin tips. Three first-class riffle files, double-ended and of assorted shapes, should be obtained by people not in possession of a grinding shaft. The standard port outline, where it enters the cylinder, is shown overleaf with the new shape superimposed. Gradually open out the port and blend it into a  $\frac{7}{8}$  in. diameter orifice at the cut-off stub.

No more than  $\frac{1}{32}$  in. should be removed from the top of the inlet port. The sides may be swept out up to  $\frac{1}{8}$  in. and the base curved around to about  $\frac{1}{16}$  in. below the original "V." Do not exceed a total depth of  $\frac{3}{4}$  in. or the width shown, for the piston-ring ends lie comparatively near to this port at the bottom of the stroke. Owing to the size of the "Bantam" cylinder casting at the cut-off stub, it may not be possible to attain a truly circular port of  $\frac{7}{8}$  in. diameter but this may be overcome by introducing a slight ovality, blending into the increasing width of the port.

A piece of  $1\frac{1}{8}$  in. O.D.,  $\frac{7}{8}$  in. I.D. tube should be counter-bored at one end to accept the  $\frac{3}{16}$  in. stub (here only about  $\frac{1}{32}$  in. thick) and bronze-welded (not brazed) to the cylinder



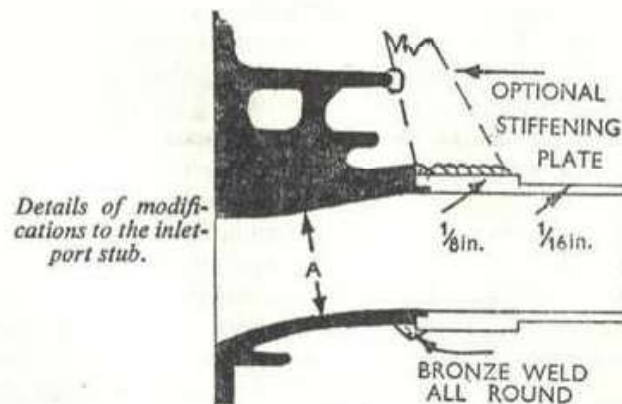
## TUNING FOR SPEED



*The original inlet-port profile and (shaded) the area of metal to be removed to procure a modified outline suggested to improve "Bantam" induction.*

barrel. The bore of the tube may then be blended into the port so as to leave no step. A further sketch illustrates this job. Alternatively a flange may be welded on.

The shape and position of their upper orifices at the cylinder bore determine the "opening" and "closing" of the transfer ports, and the behaviour of the jets of mixture issuing from them. In standard form they are not capable of passing the greatly-increased quantities of mixture now induced into the crankcase and it will be necessary to modify



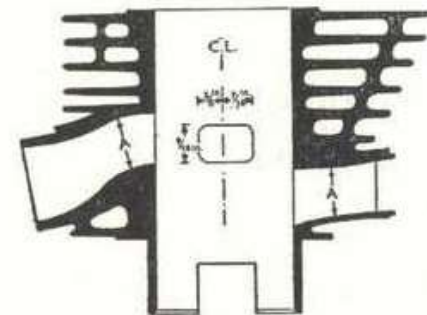
*Details of modifications to the inlet-port stub.*

226

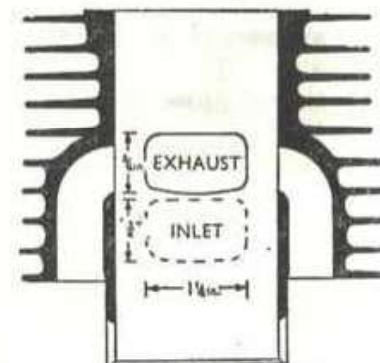
## TWO-STROKE ENGINES

them. This can be affected in two ways, (1) by greatly increasing the width circumferentially, or (2) by moderate increase of the whole cross-section of the ports, very slightly altering the height of the cylinder orifices and reshaping the width to a streamlined form. The latter course is preferable.

Be careful not to leave a lip at the rear of the transfer port exit; this will cause the incoming column of gas to swing to the front of the cylinder, which is undesirable. It should be noted that the orifices are offset to the rear of the bore centre line for a distance of  $\frac{1}{8}$  in. This assists the new gas to move toward the rear of the cylinder to displace the residual gases. One further point remains. The cylinder



*Modified port dimensions. The reference "A" denotes the point of commencement of swell.*



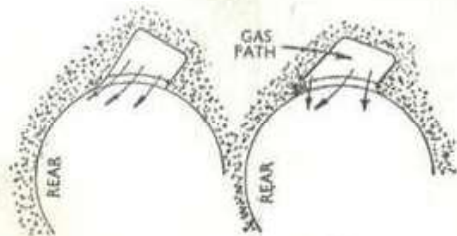
227



cutaway at the spigot should be streamlined where it is bevelled at the lower entrance to the port. Later "large fin" barrels have transfer ports adequate in size.

The timing of the exhaust port, as determined by its upper boundary, presents a problem. If the port is opened considerably earlier, its closing is similarly delayed, with the result that, when using a highly extractive exhaust system, a proportion of the hard-won fresh mixture may follow the exhaust gases. No doubt this does occur, anyway, at certain engine revolutions, but this is of little moment, provided the effect occurs outside the speed-range which will be employed. It has already been mentioned that in the ideal exhaust system, the rapid drop in pressure which follows the initial discharge is followed by a high-pressure pulse which is able to prevent any further flow through the port and may even reverse the flow, thus returning some of the lost charge to the cylinder. This enables a very early opening of the exhaust to be used, at the cost of reducing power at low speeds, and at the outset it is suggested that the exhaust port be modified only to the extent shown in the sketches by straightening the top edge and radiusing the sides. It is most important not to exceed the widths shown otherwise ring-breakage may follow through the tips being able to spring out into the ports. This is a point which must be watched most carefully in any other engine, with respect to all and not just the exhaust port only.

Attention can now be given to cleaning-up the transfer ports and their junction with the crankcase to remove any steps as indicated on page 217. The simplest method is to deal with each crankcase-half separately, applying a smear



*Removal of the lip of metal facilitates gas-flow from the transfer port.*

of bearing-blue to the flange-faces to show up discontinuities. At the same time, the slots cut in the cylinder skirt should be matched up with the ports and the inner edges of the slots rounded off to assist in smooth gas-flow. A small additional improvement may be made at this stage by opening out the entrance to the main bearing oil-feed holes a trifle.

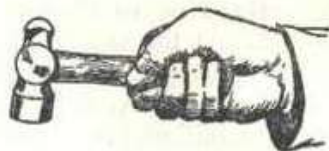
Where the last ounce of power is desired it is possible to fill up the crankcase a little. Two aluminium discs should be made to fit inside each crankcase half to reduce the volume. These should be a little less than  $\frac{1}{8}$  in. in thickness in order to clear the flywheels and they should be cut away in the centres to permit main bearing replacement and also at the top so as to coincide with the "step" in the crankcase.

Fix the plates to the crankcase walls by drilling in suitable positions, tapping and using  $\frac{3}{16}$  B.S.F. set-screws. The discs should be drilled and countersunk; screws with countersunk heads should be used. The heads should be filed flat if they protrude in order to clear the flywheels. Coating the threads with jointing compound will eliminate the chances of leakage should the walls be penetrated when drilling, and also help to secure the screws.

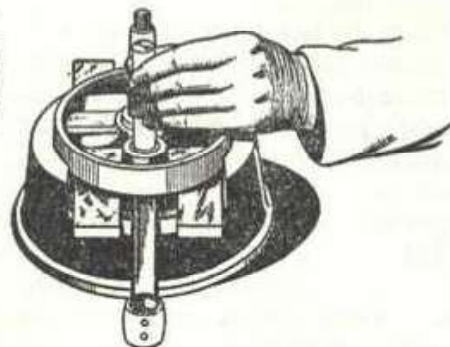
It may appear that these discs have little practical value. Calculation of their volume will show that they account for several cubic centimetres and an appreciable percentage of the crankcase and under-piston volume. The general effect will be to maintain higher crankcase pressures, possibly increasing carburettor blowback at low engine speeds, but assisting proper transference of the charge at useful racing speeds.

Piston shape determines, with the port positions, the timing of the engine, and perhaps the most important dimensions are those which operate over the inlet tract. It will be seen that the longer the rear skirt is, the later it will uncover the inlet port when ascending and the sooner it will close it when descending. Provision has already been made for "longer" timings by bringing down the bottom of the inlet port and a further improvement may be made by filing





*Driving out the crankpin and parting the flywheels in order to get access to the big-end bearing is simplified by Service Tool 61-3206.*



away the bottom of the piston skirt at the rear. It can be seen that, in a standard engine, the skirt does not clear the port when the piston is at t.d.c. This factor, in addition to reducing the effective area of the port, also causes turbulence around the skirt, reducing the port efficiency.

If necessary, therefore, the skirt should be cut away  $\frac{1}{8}$  in. at the rear or  $\frac{3}{16}$  in. (up to the base ring of aluminium) for racing. This alteration will have the following effects: (1) to open the port earlier and close it later. (2) to eliminate the port obstruction. (3) to allow a "dwell" period of full open-

	NEW	OLD	INCREASED PERIOD
Exhaust opens, b.b.d.c.	75°	70°	—
Exhaust closes, a.b.d.c.	75°	70°	10°
Transfer ports open, b.b.d.c.	65°	60°	—
Transfer ports close, a.b.d.c.	65°	60°	10°
Inlet port opens, a.b.d.c.	115°	120°	—
Inlet port closes, a.t.d.c.	65°	60°	10°

ing of the port as the crankpin swings across its top arc, the total alterations giving this timing.

The next operation is to build up the flywheel assembly, piston and cylinder on one crankcase half and mark the piston so that it may be radiused at each cut-away to blend with the cylinder and crankcase ports which have already received attention. Repeat on the other side and then bevel the front of the piston skirt slightly and put a  $\frac{1}{8}$  in. radius on the base of each cut-away corner. Some authorities prefer to use second-hand pistons, suitably polished; they have settled down and, therefore, have little tendency to seize.

Coming now to compression ratio, this can be increased to about 9 to 1 for fast road work, but for racing on 80 octane petrol, 11 to 1 may be used with confidence and for 100 octane or alcohol fuels up to 13 to 1 is feasible. The volume of the raised crown of the 125 c.c. Bantam is 3 c.c., so the actual volume of the head for any required ratio should be 3 c.c. bigger than that obtained from the graph on page 54. One way of reducing the head volume is by building-up the existing chamber by argon-arc welding and then re-machining the sphere and plug-hole as depicted, as the central position is said to increase maximum speed slightly and the plug does not oil-up. Other workers have also reported good results with squish-type heads, particularly at very high compression ratios.

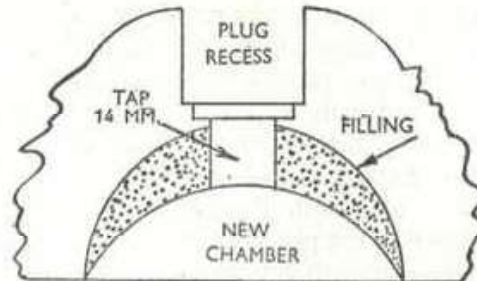
The improved volumetric efficiency and higher compression pressures call for a reduced ignition advance, though the exact amount depends upon many factors, including the type of fuel; a figure around 20° is a good starting-point with ratios of 11 or 12 to 1. With the B.S.A. flywheel magneto it is necessary to cut away the contact-breaker housing in the region of the three slots and clamp down by means of short distance-pieces and washers and the original screws. Measured on the stroke, the breaker-points should open at .110 in. before t.d.c. for 21° or .096 for 19°, according to the amount of advance required. Some users of two-strokes with flywheel magnetos prefer to substitute a lighter, plain wheel, merely retaining the con-

NOTE



## TUNING FOR SPEED

*A sectional impression of a modified high-compression cylinder-head showing filling material machined to the required hemisphere.*



tact-breaker to operate a battery and coil ignition system. One popular arrangement is to use a 6-volt battery, and a  $4\frac{1}{2}$ -volt coil as fitted to Ford side-valve V-eight engines, the reason being that this combination will deliver the right number of sparks, despite some degree of voltage-loss inseparable from the high speed of operation.

### "Bantam" Carburetter Tuning

Assuming that a T.T. Amal of  $\frac{7}{8}$ -in. bore is used, the setting to start off with, for petrol, is No. 4 slide, needle 2nd groove from top, main jet around 300, but this varies according to the exhaust system used. For alcohol it may have to go up to 650 or 800. Still larger carburetters can be used for fast, open circuits, but whatever the size, the object when tuning is to select a main-jet which will just cause the engine to begin to four-stroke when the air-lever is moved back to three-quarters open from the full-open position; the air-lever should always be mounted where it can be conveniently operated at speed. New checks should always be made whenever the inlet or exhaust lengths are altered, however slightly, and a highly-tuned two-stroke should always be run on a slightly rich mixture rather than a slightly weak one, as the former keeps the engine cooler and an occasional stutter through richness will not reduce speed appreciably. After determining the main jet size, experiments must be made with the throttle valve and needle position in order to get the cleanest acceleration, as outlined in Chapter XVIII. This is especially

## TWO-STROKE ENGINES

important when standing-starts have to be made, as otherwise the engine may die immediately the clutch is let in.

When alcohol fuel is used, Castrol "R" or a similar type of oil must be employed, because it is soluble in this fuel whereas mineral oil is not. Castor oil is also able to dissolve in some modern petrols and should be used whenever possible. Some riders prefer to increase the proportion from the normal 16 or 20 to 1, to as much as 10 to 1, in the interests of mechanical reliability, but the presence of oil in the mixture in effect lowers the octane number of the fuel and detonation may rise from this cause. The latest Continental two-strokes appear to function quite well on a proportion of 25 to 1. The amount of oil in the fuel affects the main-jet size appreciably, and once determined the proportion should be adhered to.

Experiments have been carried out with a 12-in.-long megaphone and a 12° pipe divergence, the exhaust pipe being shortened gradually. Sometimes a length which gives good power at high revolutions introduces a "miss", or flat spot, in the carburation at other speeds. A fairly sound scheme is to have an exhaust pipe which terminates at the "Bantam" front chain-cover studs, or up to 3 in. shorter if a megaphone is attached.

If the big-end is in good order, there is not much more to do. An improvement to the lubrication of the "Bantam" big-end is possible, however, by bellling out the slots of the connecting-rod eye on the outside. This can be done by

*Belling out the oil slot in the connecting-rod big-end eye of pre-1953 model "Bantams", tends to improve lubrication at this highly stressed point.*

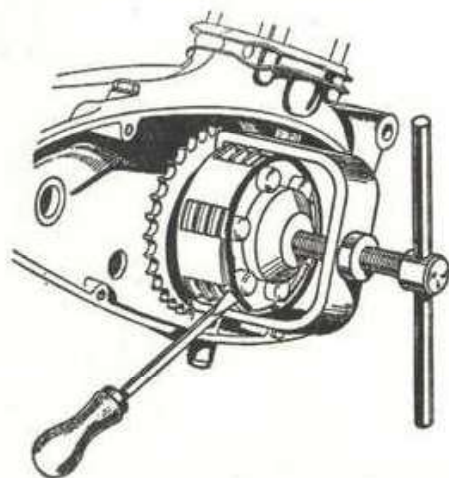




## TUNING FOR SPEED

running the lip of a grinding wheel around the edges. The 1953-type rod is cut away at the big-end faces to achieve the same result. This, with the more liberal oil supply coming from the new, larger jets, results in bearing reliability. Attention is drawn to the fact that the "Bantam" 1953 big-end bearing is  $\frac{1}{8}$  in. wider than its predecessors and, therefore, has a better load capacity.

An improvement to magneto-side main-bearing lubrication can be carried out by drilling the duralumin distance collar, which fits against the bearing, so that the holes line

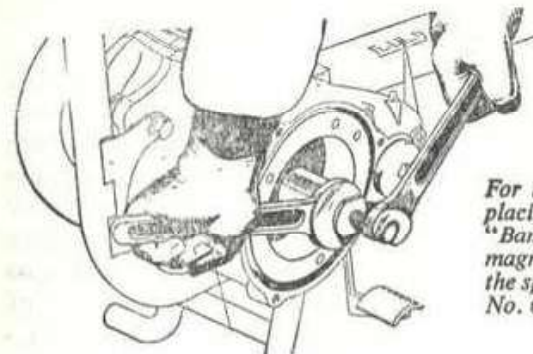


*Special tool No. 61-3191, in conjunction with a screwdriver facilitates the dismantling of the "Bantam" clutch-retaining circlip and main clutch assembly.*

up with the balls in the bearing. Ten  $\frac{1}{8}$  in. holes will do. If, when reassembling the crankshaft and case, there is a sense of stiffness, or a tight spot, suspect your initial inspection of the bearings. New bearings will be slightly stiffer in rotational movement. Or possibly the shafts are not in line; they should be tested in the usual way between centres. True them by tapping—before the covers are put in position—and re-test.

Obviously, gearbox bearings and bushes should be checked; the splines and dogs should be inspected because the increased power it is hoped to attain may reveal weakness in the transmission component. Irrespective of condition,

## TWO-STROKE ENGINES



*For removing and replacing the B.S.A. "Bantam" flywheel magneto components the special factory tool No. 61-3188 is useful.*

the following improvement is suggested. The second-gear notch in the selector should be deepened to a  $\frac{3}{16}$  in. "V" and the selector spring shimmed to a depth of  $\frac{1}{8}$  in. in its housing. The gear selector-arm retaining spring, located by a cup and secured by a split-pin, should be reinforced by adding a short spring inside. This spring should be about  $\frac{3}{8}$  in. O.D. and care should be taken that the coils do not bind at normal movement of the selector mechanism.

Alternatively, a heavier main spring may be fitted but this "mod" must not prevent the change mechanism from functioning properly. These improvements to a "Bantam" box are necessary to avoid the tendency for second gear to be forced out of engagement due to the selector mechanism side-thrust under racing loads.

The clutch on the average two-stroke unit is quite up to its job and will perform well under racing conditions without any increase being made in spring pressure. Inspection should be made, of course, to ensure that the friction inserts are not worn. Running under oil-bath conditions, these clutches are most reliable and heat-resistant.

The magneto flywheel is fitted with the engine installed; care should be taken to see that it is well home on its taper and tightly locked up. The Wico-Pacy instrument will give very good service and the engine can be revved to well over 7,000 r.p.m. without misfiring.



## Close-ratio Gears

The inevitable effect of increasing the top-end power and speed is to diminish the engine's ability to pull at low speeds, and consequently five- or six-speed gear boxes are now the accepted thing. These are not normally obtainable on English models, although a five-speed Albion box is now available; four-speed boxes are fitted as standard to many English and continental units, and it is usually possible to obtain special close-ratio sets for racing. In the case of the "Bantam", the parts required for conversion to close-ratio gears are:

- (1) Gearbox main shaft and fixed pinion with 19-t. B.S.A. Part No. 90-445.
- (2) Layshaft pinion 23-t. meshes with (3). Part No. 90-447.
- (3) Mainshaft primary gear (sleeve gear) with 24-t. Part No. 90-474.
- (4) Layshaft sliding pinion (C.R.) 25-t. Part No. 90-449.
- (5) Layshaft gear—meshes with (1)—28-t. Part No. 90-448.

Use the standard layshaft but remove the 19-t. gear, observing correct way round, and press on the 23-t. gear pinion.

The ratios of the two sets are as below:

STANDARD "BANTAM" RATIOS

	SLEEVE	SLIDING	INTEGRAL	RATIOS
Mainshaft	28-t	22-t	15-t	Top 7:1
Layshaft	19-t	25-t	32-t	2nd 11.7:1
				1st 22:1

*All ratios with 15-t gearbox sprocket and standard 47-t rear wheel sprocket.*

CLOSE RATIO

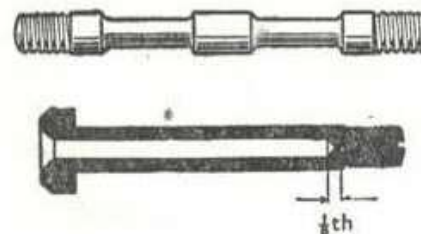
Mainshaft	24-t	22-t	19-t	Top 7:1
Layshaft	23-t	25-t	28-t	2nd 8.4:1
				1st 10.78:1

## ASSEMBLING THE ENGINE IN THE FRAME

ASSUMING that the frame of the machine is not suspected of having been damaged or is not out of line in any way, the next step is to assemble the engine into it; this is usually a fairly straightforward matter calling for little comment in itself.

If the design is such that the unit can be built up as a whole on the bench and then dropped into the frame, that is the best sequence to employ, because it permits the rear engine plates and bolts to be fitted without any juggling or forcing into line. Whatever the method, however, it is important that those bolts and others which connect the plates to the frame lugs really *are* a good fit in their respective holes, and that all mating holes are accurately in line. Slack-fitting bolts are very prone to work loose, however tight the nuts may be, and if the holes are out of line to an extent, calling for a considerable amount of levering (and perhaps a few hammer blows) to work the bolts into position, peculiar internal stresses are created, which accentuate any tendency there may be for vibration to be set up at certain speeds.

Commercial engine bolts are generally made from mild steel, but for serious racing something better is worth while, particularly in view of the fact that the bolts will have to be



*Waisting bolts down to a diameter equal to that of the bottom of the threads, leaving a central parallel portion for registering purposes, or, if a bolt has a solid head, drilling down the centre makes for lightness without sacrificing strength.*



## TUNING FOR SPEED

taken out and replaced several times in a season. For such bolts, heat-treated K.E. 805 steel cannot be bettered; this material is very strong and tough, and is not liable to strip the threads if tightened by an over-zealous hand. A less expensive grade, such as 3% nickel steel, is a good substitute.

A not inconsiderable saving in weight can be effected by waisting the bolts down to a diameter equal to the bottom diameter of the threads, leaving a short parallel portion at each end to centralize the bolt or act as a register for the engine plates if required; alternatively, if the bolts have

TABLE OF DRILL SIZES

BOLT SIZE	CORRECT HOLE DIAMETER	DRILL SIZE	NEAREST FRACTIONAL SIZE
1/4 in. B.S.F. (26 T.P.I.)	0.151	No. 25	9/64 in.
5/16 in. B.S.F. (22 T.P.I.)	0.185	No. 13	3/16 in.
5/16 in. × 26 T.P.I.	0.174	No. 17	11/64 in.
3/8 in. B.S.F. (20 T.P.I.)	0.208	No. 4	13/64 in.
3/8 in. × 26 T.P.I.	0.185	No. 13	3/16 in.
7/16 in. B.S.F. (18 T.P.I.)	0.239	Size B	15/64 in.
7/16 in. × 20 T.P.I.	0.228	No. 1	7/32 in.
7/16 in. × 26 T.P.I.	0.202	No. 7	13/64 in.
1/2 in. B.S.F. (16 T.P.I.)	0.271	Size 1	17/64 in.
1/2 in. × 20 T.P.I.	0.244	Size C	1/4 in.
1/2 in. × 26 T.P.I.	0.217	7/32 in.	7/32 in.

solid heads they can be drilled down the centre, as shown in the diagram, the hole (which must be accurately concentric with the outside) extending very nearly to the beginning of the thread. Far from weakening the bolts, this treatment actually makes them less prone to fatigue-failure and, if carried out consistently throughout the whole machine, a perceptible mass of excess metal will be eliminated.

The table above gives the right drill sizes to employ for the varying diameters and thread pitches which are commonly used for engine and frame bolts.

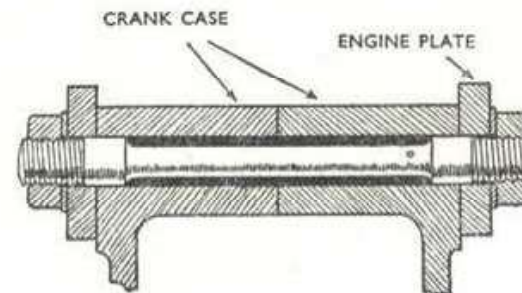
When making up those engine-securing bolts which also

## ASSEMBLING THE ENGINE IN THE FRAME

*Spigot nuts made to fit enlarged engine-plate holes cure wear and obviate the risk of vibration.*



pass through the engine plates, the length of the thread should be such that the plates have at least half their thickness bearing on the full bolt diameter and not on the crests of the thread, otherwise the latter will rapidly hammer down and looseness will develop. Occasionally in old machines some of the plate holes will be found to be oval or enlarged, in which event they can either be reamed out to  $\frac{1}{8}$ -in. over-size, and bushed to bring them back to original size, or else special spigot nuts can be made up to fit the enlarged holes. The latter is a good idea, particularly if the plates are thin, because the bolt threads are then protected from further damage from the action of the plates.



*Whether or not a bolt, or stud, is waisted, be careful to see that at least half the width of the engine plates bear on the full bolt diameter—not wholly on the thread crests which will rapidly wear.*



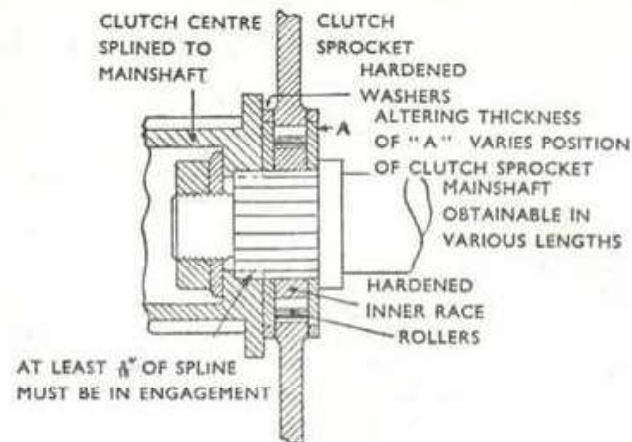
## TUNING FOR SPEED

With all engine and gearbox bolts tight, the chain alignment should be carefully checked. Two straight edges should be used in preference to one, since this method shows up any errors more clearly. The most common error is for the sprockets, though parallel, to be displaced sideways relative to each other; in some designs this may be caused by clutch inserts of incorrect thickness, for which the remedy is obvious, but in other patterns (such as the Burman, in which the insert thickness cannot influence the chain-line) the spacer-washers between the clutch body and the shoulder on the gearbox mainshaft may be of incorrect thickness.

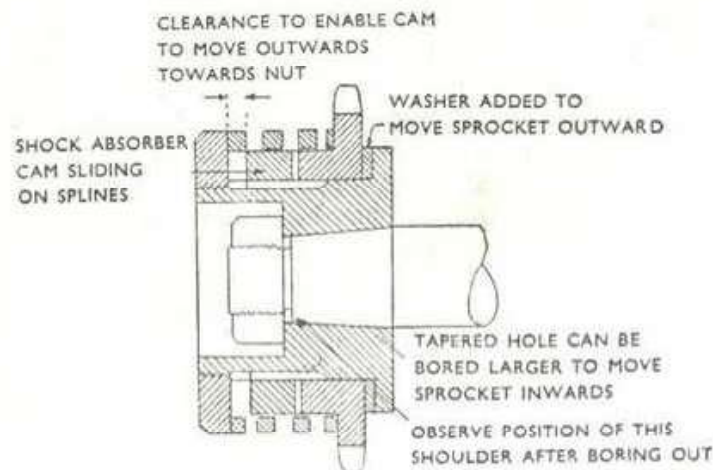
Another cause met with when an engine-shaft shock-absorber is fitted, is wear on the inner face of the engine-sprocket, but in any event this form of error is usually not difficult to eliminate. Lack of parallelism is rare, which is fortunate, as it is more difficult both to trace the source and effect a cure.

If the engine plates are interchangeable a quick check can be made by swopping them over, side for side, and if the error has vanished, or is still there but in the opposite direction, obviously the plates are at fault. If this course cannot be adopted, or yields negative results, the faces of the box lugs may not be square to the mainshafts, which can be verified by bolting the box down on to a block on a surface plate and checking the parallelism of the sprocket with the plate. Should an error thereby be brought to light, the lug faces will have to be carefully filed up to eliminate it.

When under load, the pull of the chain has a tendency to deflect the shafts towards each other and, consequently, a slight *outward* divergence of the shafts is not necessarily detrimental. But if the shafts are inclined *towards* each other when at rest, both sprockets will be badly out of line when under load. It is not sufficient merely to check the alignment along the top run of the chain; the bottom run should also be checked, and any noticeable difference between the two is evidence that the box is twisted relative to the engine. This can be caused by lack of flatness of the engine plates, or, in designs where the box is mounted on a bracket separated



*Self-explanatory methods of (above) adjusting chain line from the clutch end, and (below) making similar alteration to the engine sprocket assembly.*





from the engine, there is a possibility that the frame itself is twisted.

Accuracy of alignment is particularly important at speeds of 6,000 r.p.m. and over; for instance, on A.J.S. racers of recent manufacture, provision is made to obtain almost dead accuracy by the addition of shims behind the sprocket. It is somewhat confusing, therefore, that speedway J.A.P. engines are fitted with floating sprockets with about  $\frac{1}{4}$ -in. lateral movement, but this is to allow for the flexure which is inherent in the very light frames employed for this work. So long as the chain does not come off the sprockets, it does not matter much if its life-span is short in this instance.

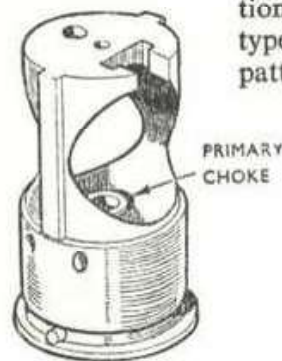
Chain lubrication is vital for high speed; the best scheme with an open chain is to adopt the Norton practice of feeding mineral oil through a suitable jet to a divided pipe from which the oil drips on to the side-plates just before the chain starts to run on to the clutch sprocket. The aim is to get the lubricant directed into the actual joints, as the rollers themselves need little lubrication externally.

When adjusting a primary chain in a layout employing aluminium engine plates, or in a unit-construction plot, allowance must be made for thermal expansion, by permitting about 1 in. up-and-down play; a little too much slack does not harm a chain nearly so much as too little. The best plan is to adjust a little on the tight side then to operate the adjuster to bring the box slightly forward: the adjuster then holds the box forward against the pull of the rear chain, and there is no possibility of the box being shifted rearwardly.

## MAINTENANCE OF RACING CARBURETTORS

THE standard needle-jet Amal carburettor is a very simple instrument which, owing to its "straight through" construction, gives a higher power output, size for size, than other types which make use of butterfly throttles. From the standard pattern there have been developed a number of special racing editions with the object of obtaining maximum power and acceleration throughout the whole of the speed range. Of these the best known are the T.T. type and the remote-needle or R.N. type. In addition, non-needle models have been produced for track and dirt racing, where engines are operating at full bore most of the time, and in 1952 the GP types appeared.

In the T.T. pattern the needle—which meters the fuel at certain throttle openings—is suspended from the top of the throttle slide, and thus hangs across the centre of the choke. In the R.N. pattern—as its name implies—the needle is suspended from an extension on the throttle slide and is housed within a chamber cast on the side of the carburettor body; thus the needle offers no obstruction to air-flow, and as a result the R.N. type is slightly faster than the T.T. pattern for any given choke size.

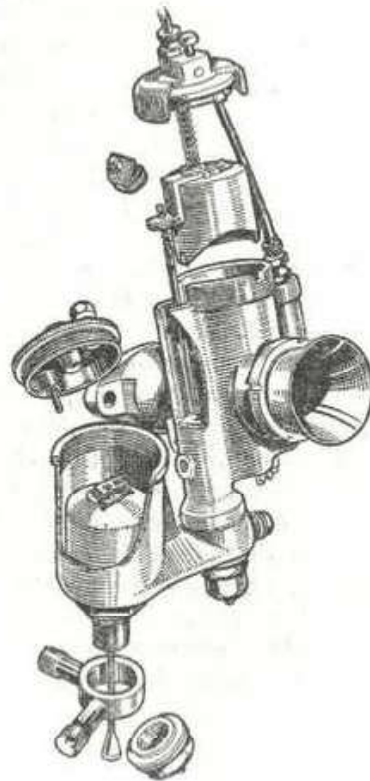


*The main choke, showing the small primary choke protruding into the airstream. The block is of the standard type but T.T. and R.N. types are similar in principle.*



#### TUNING FOR SPEED

The more devious fuel passages in the R.N., however, render it rather more sensitive at low engine revs., which means that the T.T. pattern offers more foolproof carburation and is a little easier to tune than the R.N. The latter also occupies more space (due to the presence of the remote-needle chamber), which sometimes makes it difficult to house.



*The position of the taper needle, which is "remote" from the main intake tract, gives the initials R.N. to this type of racing carburettor. Being carried in a separate chamber off-set from the main barrel of the instrument, the needle offers no obstruction to the air-flow.*

In all types the diameter of the main choke, at the point where the small primary choke protrudes upwards into the airstream, is usually bigger than the bore of the instrument immediately on the engine side of the throttle. This latter and smaller diameter is actually the choke size and corres-

#### MAINTENANCE OF RACING CARBURETTERS

ponds to the figure stamped on the outside of the mixing chamber for identification.

The increase in diameter referred to is provided (a) to minimize the obstruction to air-flow caused by the uprush of liquid fuel from the primary choke, and (b), in the case of the T.T. type, to help overcome the obstruction caused by the needle. Therefore, when discussing choke sizes, it is important to remember that the choke size refers to the diameter of the bore just past the throttle valve, and not to the larger diameter at the point where the throttle-valve works. In  $1\frac{3}{8}$  in. and 32 mm. carburetters, however, this reduction in diameter is not present.

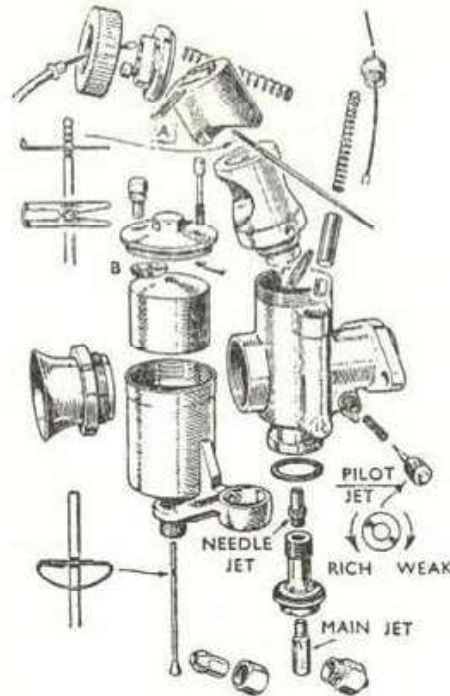
The normal R.N. type is essentially a petrol-benzole instrument, and cannot be used with a main jet size larger than 900, whereas simply by fitting a larger size of needle jet, the T.T. type can be used with straight alcohol up to a jet size of around 1500. The needle jet is formed by the hole in the upper end of the brass component into the lower end of which the main jet is screwed, and is varied for different sizes of carburetters.

Up to 1-in. choke diameter, the standard size is .107 in., and over 1-in. diameter it is .109 in. For alcohol fuel the .107-in. jet must be replaced by one of .113-in. diameter, while the .109-in. jet should be replaced by one of either .113 in. or .120 in., according to the size of main jet which the fuel necessitates. Incidentally, alcohol fuels vary considerably in their chemical make-up and some require a much larger jet than others; one characteristic of these fuels is their relative insensitivity to jet size as compared with petrol, and excessively rich mixtures can be handled without the loss of power which would be associated with petrol.

A special type of R.N. carburettor has been evolved for use with alcohol, primarily for use on the dirt; this has a .160-in. needle jet, as opposed to the .109-in. size fitted in the petrol-benzole edition. But other considerations are involved, and if the two sorts of fuel are to be used, then two R.N. carburetters will be required, whereas a single



*On the T.T. pattern carburetter it is the air slide only which operates in a separate chamber; the taper needle is centrally disposed as in the standard instrument.*



T.T. type will suffice for both fuels with the requisite change in needle and main jets.

To obviate this situation, and also to incorporate some design features which render the carburetter less sensitive to "megaphoning" and yet retain an unobstructed choke, the GP type has been evolved. The salient feature of this type is that the needle is suspended from the throttle-valve so that it is just to one side of the choke, and the fuel enters the choke through a spray-tube inclined at an angle instead of directly underneath as in the R.N. and T.T. types. The passage from the needle jet to the end of the spray tube is, therefore, quite short and direct, a most desirable feature. To furnish a more precise tuning control, the metering jets are supplied in two tapers but there are only five adjusting grooves in each, instead of seven, and there is an additional

mixture control provided by an air jet, situated between the air control slide and the needle-jet passage. This jet, which can be unscrewed after removing the hexagon plug at the base of the air slide, is made in two sizes, .100 in. and .125 in. bore, and the appropriate size is fitted to each carburetter by the manufacturers. Normally no alteration is necessary, but if difficulty is found in obtaining absolutely precise mixture control, it may be advantageous to alter the size from that already fitted. Increasing the size weakens, and decreasing it enriches the mixture (see page 260).

The main jets are interchangeable with those used in R.N. and T.T. carburetters, but the needle jets, throttle valves and needles are different. There are three basic models in the GP series, the 15 GP in choke sizes from  $\frac{7}{8}$  in. to  $1\frac{1}{8}$  in., 10 GP from  $1\frac{1}{8}$  in. to  $1\frac{7}{8}$  in. and 5 GP from  $1\frac{7}{8}$  in. to  $1\frac{3}{4}$  in., which is considerably larger than that of any R.N. or T.T. previously obtainable commercially. The two smaller groups utilize similar needles, designated and marked GP (standard) and GP6 (weak) and are flange-mounted with bolts at the regular 2 in. centre distance, but the 5 GP range uses its own needles, marked 5 GP (standard) and 5 GP6 (weak) and has flange bolts spaced at 65 mm. or 2.56 in. Hence, while all the other types are interchangeable, so far as the flanges are concerned, the 5 GP cannot be fitted directly unless the head has been drilled especially for it, as in the case of the short-stroke "Manx" Nortons. GP instruments are fitted with long air-intakes and although less bulky laterally than the R.N. types, may be difficult to instal on account of extra length. In this connection it is essential to place the mouth of the intake in a position where carburation cannot be upset by stray air currents blowing across or into it. Some odd effects can be created by, for instance, the flat surface of an oil tank within an inch or so of the intake; carburation trouble was prevalent on the early A.J.S. "Porcupine" owing to the air whistling at high velocity through the tunnel in the saddle tank in which the carburetters were placed. (See pages 260 and 276 for more notes on GP, GP2 and Monobloc carburetters.)

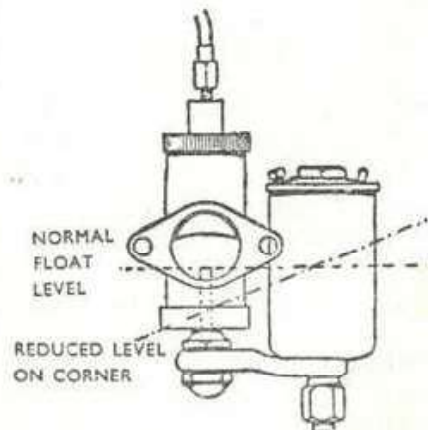


## Float Chambers and Location

Both T.T. and R.N. carburettors use orthodox single float chambers, with enlarged passages capable of passing as great a flow of fuel as the double float chambers which were popular some years ago, and which are still used at times with alcohol fuel.

With twin floats, the fuel lines also can be duplicated and there is less risk of complete fuel supply failure. But it must be remembered that, if one side blocks, the other cannot pass enough methanol to supply the engine and a hole may be burnt in the piston crown very rapidly, this being the inevitable outcome of running with a weak mixture at ratios of more than 12 to 1.

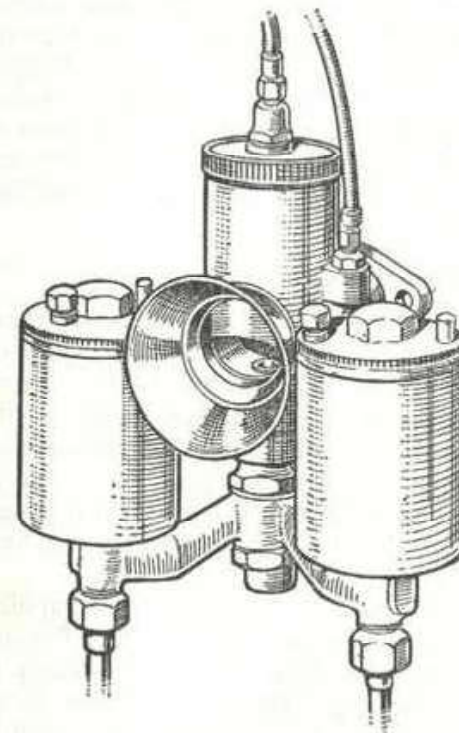
*Variation in fuel level, caused, for example, by cornering with a sidecar, gives this effect, and causes "cutting-out."*



Amal top-feed float chambers, type 302 or 504, fitted to all GP carburettors can handle all the fuel required for the largest jets ever used, and are also equipped with an anti-frothing baffle. Frothing can be a puzzling reason for loss of revs., owing to the rich mixture which it causes, and to minimize this possibility, it is advisable to mount the float chamber on an adjacent part of the frame connecting it to the jet block by a flexible petrol-proof hose, or a piece of

P.V.C. plastic tubing, making sure that it is not bent so sharply that it may kink and shut off the flow. A bronze spring pushed inside the tube will obviate kinking.

It is preferable to place the float chamber fairly close to the carburetter, but the hose must be long enough to absorb engine vibration without undue flexing and possible damage



*Twin-float carburettors offer advantages where alcohol fuel is used.*

to the lining; the relative movement which occurs between the cylinder head and frame on some machines is much larger than might be imagined.

One method of remote mounting is to clamp the bowl bodily to an object such as the tank; another, and very neat system is to screw a rod into the chamber top or union nut and hang the chamber from a flexible rubber mount bolted to some convenient spot such as a tank bracket.



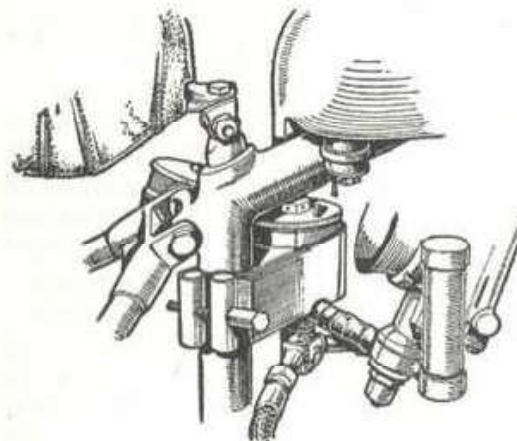
The float should not be placed appreciably to the rear of the jet, as the fuel will tend to lag behind during violent acceleration, and this may cause the engine to hesitate when opened up on the road. Somewhat the same effect occurs when cornering sharply on a sidecar outfit, and can be entirely eliminated only by using a double float chamber, or by the addition of a "swill pot" (which resembles a small float chamber, minus the float, needle valve and petrol union) placed at the side of the mixing chamber farthest from the main float chamber, the latter being remotely mounted in the approved fashion. Under the action of centrifugal force the fuel level drops in one chamber and rises in the other; thus the level of fuel at the jet remains approximately constant.

#### Carburettor Tuning

On page 276 is given a range of carburettors and approximate main-jet sizes which have been found suitable on various machines, but it must be understood that any such list can form only a very rough guide. So many varying characteristics are involved that hard-and-fast rules cannot be laid down, and every racing engine must be treated as an individual unit when final tuning is being performed. Note that the main jet sizes for GP carburettors are very much smaller than for any of the other types.

The fuel used, the nature of the course and its geographical position, all have their effects, which can be allowed for only by experience coupled with trial-and-error methods of obtaining the optimum settings. The exhaust system in use also exercises a profound effect on carburation; as will be seen later, a larger jet is required with a megaphone than with a straight-through pipe, whilst a smaller jet is required if any form of silencer is fitted.

The best method of determining the correct main jet is to drive the engine (after it is fully warmed up) "flat out" over a distance of at least a quarter-mile, and at the finish to slam the throttle shut, cut the engine dead and immediately lift the clutch, thus avoiding any alteration to the sparking-



*The tendency mentioned above may be corrected by fitting what is known as a "swill pot."*

plug condition caused by the slow-running mixture or excess oil. The sparking plug is then removed and examined. If the end of the body is of a grey or lightish colour, then the mixture is weak and a larger jet is required. If it is heavily coated with soot, the jet is too big and a smaller one is required. With a correct mixture, the end of the plug should have the appearance of polished ebony; but, of course, this condition will be achieved only if the plug was clean (and preferably lightly polished) on the face before fitting. "Reading" a plug is quite an art which takes a little experience to acquire. Also, modern plugs with ceramic insulators are less easy to read than the now almost obsolete mica patterns. The former exhibit a rather harsher-looking brown colour with a correct mixture instead of the ebony of the mica type, but, in either case, any suspicion of greyness on the electrodes or body is a sign of weak mixture and the jet size must be increased. Some people prefer to go up one size further as a safety precaution if the race is on a course with long sections calling for full-throttle work unless they are able to check the plug reading on the actual circuit during practice.

On T.T., R.N. and GP types the air-lever—or, more accurately, the mixture control—operates an air-valve on the side of the carburettor, and so controls the depression

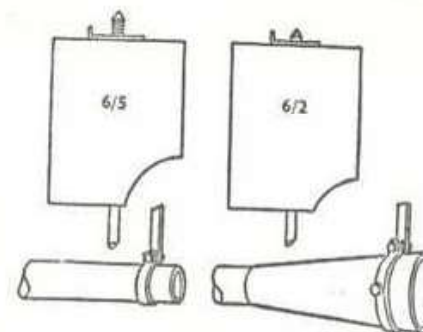


on the main jet via the primary choke tube; consequently, this control can be used as a mixture-correcter without reducing the effective bore of the carburetter. The effect on mixture-strength between the fully open and fully shut positions is approximately equal to three or four jet sizes. It can therefore be used as a rough indication of correctness of jet size; if the engine runs happily with the control shut, the jet is too small, whereas if it sounds woolly or misfires with the control fully open, or nearly so, the jet is too large.

This, however, is not in any sense a true test of mixture correctness, for which the sparking-plug test is the only safe method. The real value of the air control is in its use as a correcter for temporary conditions. In the Isle of Man, for instance, it can be used for the first mile or so on the first lap, when both the motor and air temperature are definitely cold. To permit of such use the main jet should be of a size which provides a correct mixture with the air control fully open when the engine is hot.

Having satisfied oneself that the main jet is correct, attention must next be given to obtaining clean running and acceleration throughout the entire throttle range. The three factors which affect this are: (1) the pilot adjusting screw, situated on the side of the mixing chamber; (2) the cut-away of the throttle valve; (3) the adjustable taper needle.

The pilot adjuster screw on the T.T., R.N. and GP instruments controls the amount of *fuel* supplied to the engine for slow running; screwing the pilot adjuster clockwise weakens the mixture, and, conversely, screwing it out enriches it. This is the opposite way to the adjustment of the pilot screw on standard, Monobloc and GP2 carburetters, which controls the amount of *air* to the jet. The adjuster should be set as rich as possible consistent with a good tick-over; the *rate* at which the engine turns is controlled by adjusting the throttle-cable stop to give the necessary small opening of the throttle valve when the twist-grip is shut, but as the position of the valve also affects the slow-running mixture strength, the cable adjustment and the pilot adjuster position are to some extent inter-related.



*A diagrammatic explanation of the tuning rule referred to in the text. With (left) a straight-through exhaust pipe, the jet needle is fitted higher and the throttle-slide cut-away is greater than when (right) a megaphone exhaust is employed.*

Eventually, a setting for both can be arrived at which gives a reasonably good tick-over, erring, if anything, on the fast side, and with the mixture strength slightly rich. If pains are taken to obtain a really super tick-over, it is sometimes found that snap acceleration suffers; clearly, it is essential that the engine should never fail to respond instantly if the throttle is slammed open.

The throttle valve cut-away (i.e., the bevel milled on the slide on the side farthest from the engine) controls the mixture strength from low openings up to about half-throttle. The cut-aways vary in steps of  $\frac{1}{16}$  in. in height; the No. 5, for instance, has  $\frac{5}{16}$ -in. cut-away, and consequently allows more air to enter than a No. 3, which, as its number indicates, has a  $\frac{3}{16}$ -in. cut-away. Generally speaking, the lower the cut-away can be kept, consistent with clean running, the better will be the snap acceleration, and this should be borne in mind when tuning. It is advisable, therefore, to have on hand a range of slides varying from, say, No. 2 to No. 6, and to select from them the one which is the most suitable.

These racing slides are very delicate, and a close fit in the body; if they are handled roughly or assembled in a dirty



condition, binding may occur, and it may be impossible to get even running at small throttle openings.

#### Position of the Jet Needle

The taper needle is adjustable to five or seven positions in the valve by a spring clip; the lower the needle the weaker the mixture, and the higher the richer, for any particular throttle position. Any flatness or tendency for the engine to eight-stroke at over half-throttle can always be eliminated by needle adjustment. A very common position on both T.T. and R.N. types is 4: that is, the fourth groove down from the top; the final setting should, if anything, be on the rich side.

By systematic attention to the adjustments described, a setting can be obtained which furnishes clean and lively running from tick-over right up to full-throttle, particularly when the engine is being used with a straight-through exhaust system.

As is well known, the fitting of a megaphone exhaust affects the running of the engine adversely at low and medium speeds, a disinclination to run at anything below 3,000 r.p.m. and a tendency to cut right out when the throttle is opened rapidly from low speed being typical symptoms. Almost invariably the carburetter is blamed unfairly for these troubles—a change-over to a straight-through pipe will prove otherwise—but it is necessary to alter the carburetter tuning in order to minimize the ill-effects as much as possible.

A megaphone exhaust system usually requires a main jet about 40 c.c. larger than that used with a straight pipe—incidentally, Amal jets are calibrated in c.c.'s., the numbers stamped on them indicating how much fuel the jets will pass under a given standard set of conditions, although not necessarily in the carburetter. To counteract the tendency to cut out, it is necessary to fit a throttle-valve with greatly reduced cut-away. Very often a No. 2 slide has to be used, but whilst this helps appreciably in getting the engine to open up from low r.p.m., it may introduce a rich spot at small throttle openings. To attempt to balance out this

excessively rich setting it is usually necessary to employ a very weak needle setting; for instance, in conjunction with a No. 2 slide one may have to lower the needle to position 1, i.e. with the clip in the topmost groove.

Unless the increase in lap speed gained by fitting a megaphone really warrants its use, the carburetter people advocate straight-through pipes every time; but if a megaphone has to be used the foregoing instructions should be followed, otherwise a chaotic setting will result.

Racing carburetters are made in brass or light-metal, the latter being most usual. When used with alcohol, a precipitate in the form of white powder gradually accumulates; thus it is very necessary frequently to dismantle and clean the whole instrument, paying particular attention to the small fuel passages and preferably blowing them clear with compressed air.

Alcohol fuel is prone to leave a deposit of jelly which can block the fuel flow completely, and it should be a firm rule to drain the carburetter by loosening the jet-plug after every race meeting.

Throttle slides in light-alloy carburetters are very sensitive to grit and before a slide is pulled right out, the interior of the mixing chamber must be cleaned, for if a grain of sand does get in and jam the slide, the only safe course will be to dismantle the whole thing, including removing the jet-block from the body, in order to remove the offending particle from the narrow annular gap in which the throttle valve works. For very dusty conditions brass bodies are superior to those made of light alloy, but, in either case, efficient air filters should be made up, not only to prevent premature wear, but as a safety precaution to obviate a jammed throttle. To be fully effective, this filter or a small auxiliary one, must also cover the slot in the side air-control.

On light metal carburetters the threads of screwed parts tend to seize rather easily, and any brute force exerted will result in a scrapped component. The correct treatment, when partial seizure is detected, is to dose the thread with penetrating oil, leave it for a few minutes and then carefully



ease the component out in a series of quarter-turns. Before replacement, the threads should be smeared lightly with graphite grease. To conclude these remarks on racing carburetters, it cannot be emphasized too strongly that they are precision instruments which will not perform properly unless well cleaned and nicely adjusted; it is, unfortunately, a common sight at race meetings to see riders putting jets, throttle slides, etc., down on a dusty road, forgetful of the fact that any dirt or grit picked up may, a few minutes later, be the cause of trouble in the race.

### Tuning a Standard Carburetter

In the absence of one of the racing carburetters described, a standard model can be made to perform quite well if it is sufficiently large in the bore. Most sports engines will accept a choke size slightly larger than that normally fitted for touring use, particularly if a lot of work has been done to improve the breathing ability of the engine. (An indication of the sizes which can be employed was given in Chapter II of this book.) On the other hand, it is easy to make the mistake of fitting too large a carburetter; this will result in an engine which can only scream around at high revs., lacking bottom-end power and acceleration.

A small amount of hand-work in smoothing and polishing the bore and carefully blending-in the junction of the air-intake with the carburetter body will improve the air-flow, but care must be taken to see that the tiny slow-running mixture hole (drilled at an angle into the bore immediately on the engine side of the throttle) is not blocked in the process. As with the racing instruments, it also pays to dismantle the mixing and float chambers completely to make sure that the small fuel and air passages which control the idling mixture are free from dirt or sediment.

Basically, the principles of operation and tuning are similar for racing and touring models, thus the tuning sequence already described holds good for the latter. It is important to remember, however, that the pilot adjuster on

the standard models is screwed *in*, not out, to enrich the mixture; as a general rule a setting of one to one and a half turns "out" is about right. The tick-over speed is first set by means of the throttle-stop on the side of the mixing chamber, after which the throttle cable is adjusted so that the engine responds immediately to grip-movement. It is surprising how much a well-adjusted and smooth-acting throttle control helps in improving the general "feel" of the carburation.

The standard air control differs from the racing type in that, except when fully open, the air slide obstructs the air-flow, and thus simultaneously enriches the mixture and throttles the engine. It is, therefore, not of the same value as a mixture-corrector, but it can be used as a guide to jet-size when tuning. If the speed increases when the lever is closed slightly, the jet is too small; if the speed falls off, the jet is either correct or too large.

Here, again, the appearance of the sparking plug is the only reliable guide to correct jet size, but when conducting plug tests it is essential (unless a magneto cut-out is fitted) to have the carburetter so adjusted that the engine stops *instantly* when the throttle is snapped shut; even a few explosions with the engine idling are sufficient to mask the true appearance of the plug.

Plug tests *must* be conducted with whatever exhaust system is to be used when racing; the removal of even an efficient silencer usually necessitates an increase in jet size, a point which is sometimes overlooked to the great detriment of the engine.

### Tuning for Alcohol Blends

Standard touring Amals are not suitable as they stand for use with alcohol blends, for none of the passages is large enough to cope with the increased fuel flow, but it is quite simple to modify them to use any blend containing up to 60% alcohol such as Shell "X", "Y", or "M". Fuels such as 811 or "A", containing much methanol, are rather beyond the capacity of these instruments.



The first step in conversion is to enlarge the area of every fuel passage from the connection to the float-bowl right up to the jet. Double the number of drilled holes in the banjo connection and remove all sharp corners of the holes. Bottom-feed float chambers need the float-needle hole enlarged to a size only .030 in. smaller than the head of the float needle; drill out the hole in which the head is housed about .030 in. larger. Careful lapping-in of the head and seat may be necessary after this operation.

Enlarge the holes leading from the needle valve seating into the float chamber, being very careful not to damage the seat, as there is not much room in this region. Remove the small hexagon screw in the banjo end of the float bowl and open out the hole right through about .005 in. larger, which will still leave enough thread for the screw, and double-up the number of holes in the jet-plug. A good check on the efficacy of the work can be made by rigging up a pint tin with a piece of hose to couple up to the float bowl, and timing the interval required to pass a pint of fuel: it should eventually be reduced to less than 40 seconds.

With top-feed bowls, the rate of flow is limited by the feed hole which obviously must be smaller than the needle. It can be enlarged a little, but the best scheme is to make up a large head for the needle and drill the needle seating to suit.

The next step is to remove the needle jet and jet block from the mixing chamber. Enlarge the idling jet, which is the tiny hole at the bottom of a larger hole in the engine side of the jet block, to about twice its present diameter and drill out the needle-jet to .113 in. for "X" or "Y" fuels and .118 in. or .120 in. for "M"; it may be necessary to experiment later on with these sizes if tuning proves difficult.

Main jets of the size required are not very readily obtainable and, although the makers hate the idea, the only scheme is to drill out some standard jets, the size for "M" being in the region of .068 in. (No. 51). For this work, a set of number size drills (ranging from No. 48 to No. 55) is necessary, and a set of jet reamers (which can be obtained from suppliers of tools to the watchmaking trade) is also

advisable. The trouble with drilling is that jets opened out with the same drill do not necessarily pass the same quantity of fuel owing to variation in the diameter, length, and roughness of the hole and the only course open is to make up a number of jets and go through the tuning sequence previously described, starting first with the largest sizes to avoid premature damage to the engine.

Other factors, however, may be limiting the flow of fuel, and the precautions to be observed if the engine does not respond to variation in jet size are dealt with in the chapter on Testing the Finished Work (page 267).

#### Additional Notes on GP Carburetors

To fill a demand for chokes larger than  $1\frac{1}{8}$  in., the 3 GP type was introduced, ranging from that size up to  $1\frac{1}{2}$  in. The tuning sequence is exactly as for other GP's; main and needle jets are interchangeable, but the needles are different and stamped 3 GP (standard) and 3 GP6 (weak). In all sizes, a small circle is engraved on the air-jet plug adjacent to the air-slide; the *bottom* of this circle indicates the correct fuel level, and the right height of the float-chamber if this is remotely mounted is best checked by means of a transparent flexible stand-pipe showing the height of the fuel. The maximum allowable down-draught is  $20^\circ$ .

Although the largest air-jet supplied is .125 in., better results are sometimes obtained by drilling out to a larger size and increasing the main jet to suit. On the 7R A.J.S., for instance, the most suitable air-jet is .136 in. The air coming through the jet compensates the mixture-strength for the varying air-speed through the choke if the engine-speed rises or falls on open throttle, and difficulty in tuning GP instruments may be due to an incorrect relationship between the sizes of the air and main jets. As the air-slide overrides the action of the air-jet, intelligent use of the air control will indicate whether the air jet is too large or too small, but once established, there should be no need to alter this jet again.



### Type GP2 and Monobloc Carburetters

To obviate pilot-jet weep at steep down-draught angles type GP2 was introduced in 1961. The pilot jet and adjusting screw are located on the atmosphere side of the mixing chamber and the latter is locked by a nut instead of being spring-loaded. Tuning is exactly the same as for the GP, except that the pilot adjustment screws *in* to richen and *out* to weaken the mixture. Also, the pilot jet is removable from the side away from the adjuster and is made in sizes from 20 to 35 c.c. flow-rate, so that it can be varied if the adjuster-screw alone cannot obtain the correct mixture-strength. These pilot jets are identical with those used in the Monobloc. Main jets, needle jets and needles are as for the corresponding sizes of GP instruments.

A flat float chamber, type 510, was also introduced. This takes up less space than the round types and can conveniently be mounted between a pair of carburetters on a parallel twin.

The Monobloc design, which has replaced most other types on standard models, has an integral float chamber with pivoted float, a variable pilot jet, and it can be mounted at up to 15° down-draught. Tuning for petrol follows normal lines as described on pages 253 and 257, but for use with alcohol a larger float needle (part 376/161) and seat (376/118) are essential, also a needle jet of .120 or .125 in. diameter. Also, fit a larger pilot jet and if necessary slightly enlarge the small hole leading from the pilot jet to the cross-bore, just inside the throttle. When mounted in pairs when space is restricted, one float chamber can be milled off and the lid refitted. The two mixing chambers can then be connected with flexible tubing, joining two banjoes fitted under special long main-jet holders (376/140).

### CHAPTER XIX

#### "HARD" AND "SOFT" TYPES OF SPARKING PLUG

THE correct selection of sparking plugs is of the utmost importance in high-speed work; the time has long passed when one could stop for a plug change and still have a sporting chance of winning. Prior to 1939, special racing plugs were produced which were able to stand up to continuous full throttle with unfailing regularity, and in most cases they were of the non-detachable type, with mica insulation. Then "hardness," or ability to withstand heat, was conferred by the use of heavy, high-conductivity, central electrodes, by reducing the internal area of the plug body and by minimizing the amount of insulation exposed to flame round the central electrode. The insulation was also protected to some extent by almost closing the end of the plug body, and the path by which heat could escape from the central electrode to the plug boss was kept as small as possible.

Unfortunately, all such measures designed to keep the plug cool also have an adverse effect in that they make the plug much more sensitive to oil, which may either lodge between the points, and so prevent the passage of a spark, or may give rise to a conducting coating on the insulation, through which the current may leak in preference to jumping the gap.

Speaking very broadly, a hot-running engine is not usually oily, but there are always occasions, such as when starting or at the end of a long downhill stretch, when the engine is not really hot, and it is at these times that the susceptibility of the plug to oil is an important factor.

Each plug maker produces a range of plugs which can cover almost all known engine types and, in addition, supplies what are known as warming-up plugs, for use when riding to and from the start or when not actually racing. These



types are usually detachable, and so can be taken apart for cleaning, whilst the non-detachable racing versions can only be properly cleaned by the makers.

Just prior to 1939 plugs with sintered aluminium-oxide insulators, known under various trade names, made their appearance and have since completely supplanted the older types. These usually had their insulation made of mica, a material which, besides being more expensive and liable to unpredictable electrical breakdown under stress, cannot be used for any length of time with fuels containing much tetra-ethyl lead; the lead salts formed during combustion react with the mica to form a conducting coating which it is almost impossible to remove. Apart from this, it may be taken that, provided the selection of type is correct, any of the known makes of plug will function satisfactorily. However, although various grades are supplied in each make none are directly comparable with each other in their characteristics, and it may be found that a change of make is beneficial (or otherwise) merely because the heat value of one type is intermediate between two types of another make.

There may be a few engines still using 18 mm. plugs, but the 14 mm. size is at the moment almost universal; a description of the method of conversion from 18 to 14 mm. was given in Chapter I. In addition, 14 mm. plugs are available with two lengths of thread, known as "short-reach" and "long-reach," the latter being generally employed in engines having aluminium heads. Short-reach plugs should not on any account be used in long-reach heads, except as a purely temporary measure, otherwise the exposed threads in the hole become filled with carbon. When next the plug is changed there is every chance, then, of the threads becoming seized, and it may be very difficult to extract the plug without ruining the thread in the hole.

Conversely, long-reach plugs should never be used in short reach heads; even the expedient of using several plug-washers to shorten the effective length is bad, because they are seldom gas-tight and, besides causing loss of power, gas-leakage heats the plug excessively.

# "HARD" AND "SOFT" TYPES OF SPARKING PLUG

The accompanying tables give various types made by Lodge, Champion, and K.L.G., arranged in order of hardness—i.e., those at the lower ends of the tables will stand the most heat and the least oil; the grades of different makes are, however, not necessarily equivalent and when

## RACING SPARKING PLUGS

The plugs shown below are graded, in each size group, according to their resistance to heat and oil. Those at the top of each group are the "softest" and withstand least heat and most oil, those at the bottom are the "hardest" and withstand most heat and least oil.

The comparisons between the products of the various makers are intended merely as a guide, and may not be strictly accurate for all engines and conditions.

Thread Diameter	Thread Length (reach)	Lodge	K.L.G.	Champion	Bosch
10 mm.	12 mm.	10R47	T240		
10 mm.	12 mm.		T260		
10 mm.	12 mm.	10R49	T280		
10 mm.	12 mm.	10R51	T300		
10 mm.	12 mm.	10R53	T320		
10 mm.	18 mm.	10RL47	TE240		
10 mm.	18 mm.		TE260		
10 mm.	18 mm.	10RL49	TE280		
10 mm.	18 mm.	10RL50			
10 mm.	18 mm.	10RL51	TE300		
10 mm.	18 mm.	10RL52			
10 mm.	18 mm.	10RL53	TE320		
14 mm.	12 mm.	R47	F250	LA10, L-58R	W260T1
14 mm.	12 mm.		F260		
14 mm.	12 mm.	R49	F280	LA11, L-55R	W 275/300 T2
14 mm.	12 mm.	R50	F290		
14 mm.	12 mm.	R51	F300	LA14	W380/400 T2
14 mm.	12 mm.		F310	LA15, L-53T	
14 mm.	12 mm.	R53	F320		
14 mm.	18 mm.	RL47	FE250		W 260 T2
14 mm.	18 mm.		FE260		
14 mm.	18 mm.	RL49	FE280	NA12, N-58R	W 275/200 T2
14 mm.	18 mm.	RL50	FE290		
14 mm.	18 mm.	RL51	FE300	NA14	
14 mm.	18 mm.	RL52	FE310	NA18, N-55R	
14 mm.	18 mm.		E258/2		
14 mm.	18 mm.	RL53	FE320	NA19, N-53T	W 440/480 T2



a specific type is claimed by the engine maker to give the best power, that recommendation should be adhered to.

The symbol denoting the plug-type indicates whether it is of short or long reach, but when ordering it is essential to quote the full symbol or to mention which length is required.

For warming-up, any of the softest plugs in the ranges quoted, i.e., those at the upper end of each group, are very suitable, but one of the "hard" varieties of sports plugs, such as the Lodge HN or HHN, can be used quite satisfactorily. For actual racing, the correct plug is that which will stand up to the engine under the conditions prevailing, which in this sense embraces the length of race and severity of the course, and the fuel in use. There is nothing to be gained by using a harder plug than is necessary, and by so doing one runs the risk of the plug failing through oiling-up halfway through a race, when, perhaps, the piston rings are not as good as they were when they started.

Courses where full throttle is held for long stretches will call for harder plugs than short courses with many slow corners; but the most careful choice has to be made for long-distance races, such as the T.T., where it is not unusual to hear engines missing or cutting-out entirely after shutting off for Creg-ny-Baa corner. This happens less often to top-flight riders who come down the mountain really hard than to those who start to ease back a trifle up near Kate's Cottage. As an initial choice, a plug near the middle of the range should be selected, such as K.L.G. 250 or 280, or Lodge R49. It has been found, however, that, owing to their construction, the latest Lodge plugs with Sintox insulators have a very wide heat range, and even the hardest, R51 or RL51, will deal with a range of conditions which once had to be covered by four types, BR48 to 54, in the old mica series. On the other hand, when conducting mixture tests, as described in the part dealing with carburation, these plugs will appear, even with a correct mixture, to be running much hotter than would be considered advisable with the mica variety. This point must be borne in mind when tuning,

otherwise too large a jet may be fitted in an attempt to attain the correct appearance of the plug.

A few engines are fitted with 10 mm. plugs, usually because there is insufficient room for a larger one, and not from reasons of superiority of the smaller type. In some cases when two plugs have been fitted in one combustion chamber, one may be a 14 mm. and the other one 10 mm., purely because of this difficulty of fitting the second one into the space available. Two-plug heads were tried around 1937 in large single-cylinder engines and the general experience was that little extra power resulted except at moderate speeds when gas-turbulence is too low to afford rapid combustion, and the scheme lost favour, not being considered worth the extra complication.

Lately, however, it is coming back into use, especially on very high-revving Continental engines possibly because the time-element involved in lighting up the charge is so incredibly small at around 12,000 r.p.m. The position of the plug relative to the inlet port also has a bearing on power output, and it may well be that the second plug is simply in a more advantageous locality. Firing both plugs simultaneously can be achieved either by using a specially-wound magneto with two high-tension leads, or by employing two coils with their primary circuits connected in series and opened and closed by a single contact breaker.

Having settled by experiment on the most suitable plug, it is not a bad scheme to carry, as the spare, another which is one grade softer, for obviously if the hard one oils up, another of similar grade is likely to do the same thing. Spare plugs should be looked after very carefully both when in store and when being carried, and always kept with their threads and open ends protected, to prevent damage to the threads or the ingress of dirt or fluff, which may get up inside the plug and cause an internal short. Spare plugs should always have fresh washers in place, the best form of washer being the solid copper type, as this provides the best heat conductivity, and does not pack down and allow the plug to loosen, as either the copper-and-asbestos or rolled-



copper varieties can do; a loose plug will overheat very rapidly.

Racing plugs are invariably supplied with the correct gap, and in many instances the gap cannot be altered; in others this is not the case, but if the gap has to be set, it must be done by bending only the *earth* electrode, for attempting to bend the *central* electrode is almost certain to damage the insulation. Non-detachable racing plugs are not easy to clean: the best method, if convenient, is to return them to the factory for reconditioning, which is done for a moderate fee. They can also be cleaned by sand-blasting in an ordinary garage plug cleaner, using a low air pressure to avoid damage to the surface of the insulator and subsequently taking particular care to see that every vestige of sand is removed from the thread and from the small annulus between insulator and body.

For most work, there is nothing to beat the K.L.G. spring-wire clip for retaining the high-tension lead, but when excessively wet conditions are the rule it is better to use one of the proprietary types of umbrella terminal which protect the plug from water or mud.

Another point is the plug spanner; it pays to obtain a really good example, which fits the hexagons properly and with the shank bent to the minimum amount necessary to clear the adjacent head fins. A badly fitting spanner can easily cant over in use, until it bears on the upper end of the plug, and may then either bend or crack the insulated core.

## TESTING THE FINISHED WORK

THE ideal method of testing an engine is, of course, on a brake. With just a small Heenan and Froude brake, type DPX1 for preference, an accurate rev.-counter and an Amal flowmeter, much information can be collected, and the effects of varying induction and exhaust pipe lengths, ignition and valve timing and compression ratios can be studied and tabulated. Naturally, to do this sort of job thoroughly takes a fair amount of time, and is rather beyond the scope of this book, but apart from such development work, a brake is invaluable simply for determining whether an engine is, or is not, developing its correct horse-power, and for finding out its peak r.p.m.—that is to say, the speed at which it gives off its maximum power, not necessarily maximum speed at which it will turn over.

Very briefly, the method is to run the engine against a light brakeload until it is thoroughly warmed up, and during this period the lubrication system can be checked to ensure that oil is reaching everywhere it should. Then the throttle is opened up and the load on the brake increased until the engine is pulling the maximum load it can at a medium speed of, say, 3,000 r.p.m. (most racing engines do not like full-throttle at anything below this speed).

Readings are then taken of the r.p.m., brake load, and fuel consumption, and by means of the air control an indication can be obtained as to the correctness of the jet size. Obviously, if closing the air-control causes the speed to rise, the jet is too small and should be increased before proceeding further, as much damage can be done by running for long on a lean mixture.

Likewise, the ignition setting can be checked by operating the ignition control: if the speed rises as the spark is retarded



the advance is too great, but if it falls there is insufficient advance. The ideal setting for bench-testing is one which gives maximum power when the lever is pulled back just a trifle, as then you can always check the effect of giving a little extra advance at any time. It is, however, not a good plan to run continuously with the magneto considerably retarded, as the spark then being delivered is not of maximum intensity; it is wiser to stop the engine and re-set the timing to the correct figure. Incidentally, when observing the effects of adjustments to ignition or mixture, it is better to watch the pointer on the load dial of the brake rather than the tachometer, because the former is far more sensitive to a slight variation in speed than the latter.

The brake does not give a direct reading of the horse-power; this has to be calculated, but the arithmetic involved is very simple. All one has to do is multiply the load in pounds shown on the dial by the r.p.m., and divide by a fixed figure, called the "constant," which depends upon the type of brake in use; on modern DPX1 brakes the constant is 4,500. Put into the shape of a formula:

$$\text{B.H.P.} = \frac{\text{R.P.M.} \times \text{load}}{4,500}$$

As an example, if the engine is pulling 28 lb. at 5,000 r.p.m., it is then developing  $\frac{5,000 \times 28}{4,500} = 31.1$  b.h.p.

It will be seen that the accuracy of the b.h.p. figure depends upon the accuracy of the rev. counter; should this be at fault, the whole business is worse than useless, it is most misleading, and it is easy enough to delude oneself into believing an optimistic result without the added complication of inaccurate instruments. Consequently it is best always to have at least two separately driven rev-counters, one which, preferably, will be used on the motorcycle itself. This serves as a check on the proceedings and avoids subsequent discrepancies in readings when the model is tested on the road. A simple method of checking the speed indication stroboscopically consists of painting a white mark on

the drive coupling, and observing this solely by the light from a neon lamp. In England and all countries using 50-cycle mains frequency, the mark will appear stationary at 3,000 and 6,000 r.p.m., while in the U.S.A., Canada and all countries with 60-cycle current, the stationary speeds will be 3,600 and 7,200 r.p.m. There are, of course, more accurate methods of checking such as by Strobotac or the Ashdown Rotoscope, and these devices can also be used for observing the behaviour of valve springs, etc.

By manipulation of the brake, the horse-power developed at various speeds at intervals of, say, 500 r.p.m. can be determined, and from the figures so obtained the power curve can be plotted on graph paper. Readings should always be taken after the engine has settled down to a steady speed; "snap" readings can be deceptive and are usually optimistic. It may be found that the ignition timing and jet size which are correct at low speeds are not necessarily correct at high r.p.m., and if so, it is best to re-set them to suit the latter condition. The effect of varying the induction and exhaust pipe lengths can be determined very easily, and sometimes quite surprising results can be achieved by obtaining just the right combination. Sometimes, of course, such a combination cannot be used on the eventual installation, either for structural reasons or because of regulations which restrict or define the layout of the exhaust system, in which case the next best combination which does comply with the rules must be sought. In this connection, builders of 500 c.c. racing cars are somewhat better off than motorcyclists, because they have room to use almost any conceivable arrangements, and are not restricted by regulations regarding pipe lengths.

Usually a gain in power at the upper end of the speed range, where it is mostly required, is accompanied by a drop in power somewhere lower down, although this is not of much consequence on a racing motorcycle, where the revs. are always kept up by use of a close-ratio gearbox. There are, however, conditions where it may be better to sacrifice a little at the top end in order to get more power low down, or



to eliminate a flat spot in the power curve; for instance, on dirt-track machines, which are single-gear, it is essential to have plenty of power available to give traction and keep the rear wheel spinning when coming out of the corners, and, therefore, good pulling power at around 3,000 r.p.m. is a vital necessity. Then, again, the comparatively great weight of a 500 c.c. motor car demands wider gear ratios, particularly for standing-start hill climbs, and if the engine torque is inclined to fall off rather badly with a diminution in speed, the acceleration following a change to a higher ratio may suffer.

Mention was made earlier of taking fuel consumption readings on a flowmeter, of which the Amal pattern is the one most frequently used. This instrument indicates the rate, in pints per hour, at which the engine is consuming fuel and from this the "specific consumption" can be found by dividing the consumption by the power being developed. For short-distance racing, a lavish consumption does not matter greatly, but if the specific consumption, when using petrol, works out at much more than .8 pint per b.h.p. per hour, there is something wrong somewhere. Probably the jet is too large; but if reducing the jet also reduces the power or causes overheating of the plug, then either the valve timing is not correct, the exhaust system is of the wrong length, or the compression ratio is too high necessitating a large flow of fuel to keep the internal temperature down.

The flowmeter may show also that an increase in power gained by some alteration has been accompanied by a disproportionately great increase in consumption, this being particularly liable to happen with two-strokes. That may be a serious matter for long-distance work, as the increase in performance may, or may not, be nullified by the necessity for carrying more fuel, or perhaps making an extra pit stop. In such cases the alternative settings should be carefully noted, so that subsequent experiments can be conducted in practice on the course, to determine which is the better of the two over the distance of the race.

Arguments frequently arise as to the power which various

engines can or have developed. Few motorcycle engines are subjected to officially observed tests as aero engines are, and such arguments are likely to continue. It may be said, however, that anything over 50 b.h.p. for a 500 c.c. engine and proportionately less for other capacities, is very good indeed. Atmospheric conditions exert quite a large effect on power, which increases when the barometer reading is high, or when the air temperature is low, and vice versa. For that reason, it is always advisable, particularly in a changeable climate, to correct all power readings to N.T.P.—i.e. normal temperature and pressure, which are respectively internationally fixed at 15° C. and 29.92 in. mercury. Correction is made by the following formula:

Corrected B.H.P.=

$$\frac{\text{Observed B.H.P.} \times 29.92}{\text{barometer reading}} \times \frac{400 + \text{air temp. (degrees C.)}}{415}$$

From this it will be seen that a drop of 1 in. in the barometer reading lowers the horsepower by approximately 3%, and although the effect of temperature change is less important over the ranges normally met with in an enclosed room with the engine running, it is certainly advisable to correct the readings fully to avoid subsequent confusion.

Other figures, such as the torque of the engine and its B.M.E.P. can also be obtained from the brake test, and for those who are interested the relevant formulae are included in the Appendix.<sup>1</sup>

### Power Requirements

The power required to propel a vehicle depends upon two main factors—rolling resistance and air resistance. The former depends upon the gross vehicle weight, the state of the road surface and the size and inflation pressure of the tyres, but it remains substantially constant over a wide range

<sup>1</sup> For those who are really interested in this work, A. W. Judge's book *The Testing of High-speed Internal-combustion Engines* (Chapman and Hall Ltd.) provides a wealth of information.



of speed provided the factors stated do not vary. Air resistance depends upon the frontal area and the aerodynamic shape, and increases as the square of the speed, and, therefore, the power absorbed in overcoming it increases as the *cube* of the speed, whereas that required to overcome rolling resistance increases only directly as the speed. In consequence, on a good surface, the power absorbed by wind pressure at speeds in the region of 100 m.p.h. is nearly 10 times that absorbed by rolling resistance, hence the necessity for adopting the most compact riding position possible. The air resistance can be found from the formula:

$$\text{Resistance (pounds)} = A \cdot V^2 \cdot C$$

where A = the frontal area in square feet.

V = velocity in feet per second.

C = a constant called the "drag coefficient."

A, the frontal area, is difficult to determine accurately for such an irregular shape as a motorcycle and rider, but it can be taken as being about 5 sq. ft. for a rider of the average physique, lying down on a Senior T.T. machine; it would obviously be less, perhaps  $4\frac{1}{2}$  sq. ft. for a small man on a 125 c.c. model. For the factor C, a figure of .0008 has been found by experiment to give results which tally reasonably well with known performance figures, though it may be slightly less for a clean design of machine with the rider exceptionally well tucked in.

Using these figures, the air resistance of a Senior T.T. model, without fairing, at 124 m.p.h. (182 ft. per sec.) is

$$5 \times 182^2 \times .0008 = 132 \text{ lb.}$$

Taking the total all-up weight as 520 lb. and rolling resistance at 2 per cent, which is a good average figure to employ, then the rolling resistance is approximately 10 lb., a very small figure compared to the air resistance. The total resistance is therefore 132 plus 10 = 142 lb.

The horsepower required to overcome a known resistance at a known speed is found from the expression

$$\text{H.P.} = \frac{\text{resistance} \times \text{feet per second}}{550}$$

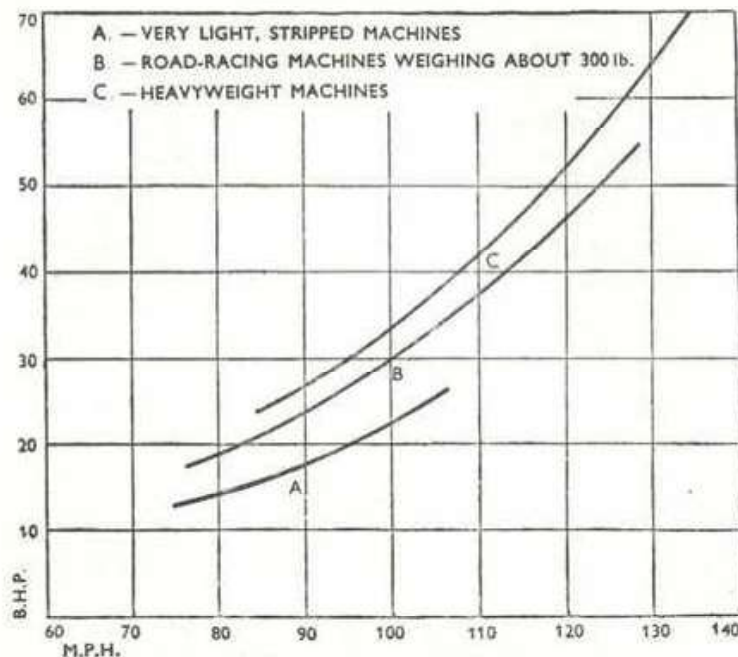
$$\text{therefore, in this instance, H.P.} = \frac{142 \times 182}{550} = 47.$$

This is the power required at the rear tyre; assuming 96 per cent transmission efficiency, 49 B.H.P. would have to be developed by the engine to achieve the quoted speed of 124 m.p.h. This result agrees very closely with the power development and maximum speed of known racing machines, particularly the 1938 Senior Norton, but there is always bound to be a certain amount of guesswork in estimating the factors involved. The drag coefficient "C" for instance can be brought down to .0003 by full streamlining and though this usually results in a simultaneous increase in frontal area, the net gain is very considerable. To avoid much laborious calculation, the accompanying graph has been compiled from a number of known performance figures and gives a fairly accurate guide to power requirements; the lower curve is for light, small-capacity machines in track condition, the middle curve being for heavier machines in road-racing trim, and the upper curve is for large, heavy mounts in the big-twin class. These curves do not appear to rise so rapidly as might be expected with increase of speed, but this is because in the examples chosen greater efforts had been put into reducing wind resistance by streamlining, and rolling resistance, partly by reducing weight, and partly by using special tyres. With the aid of this graph and the performance figures of the engine an estimate of the speed attainable and of the gear-ratio to employ can be obtained which will be sufficiently close to serve as a "jumping-off place" pending final settlement on the course.

Most amateurs will, of course, have to do what they can without the advantage of knowing the actual developed horsepower, unless they can get their engines tested for them. In any case, even if the figures are available, final tuning has eventually to be done under race conditions, as



## TUNING FOR SPEED



*In graph form this approximate guide to the tie-up between b.h.p. and m.p.h. gives the tuner an idea of the results his efforts should produce—always bearing in mind wind and other resistance factors.*

it is almost impossible to duplicate them on the bench. For this work a correctly geared *front-wheel-driven* speedometer is a great help, although so far as the indirect gears are concerned it is not so useful as a rev. counter, because it is difficult at racing speeds to convert m.p.h. into r.p.m. The rev. counter is also considerably more sensitive to small variations in speed.

Runs can be made over a stretch of the course, or a similar piece of road, using the correct exhaust system and the "official" fuel, if any is stipulated. First of all, the correct jet size should be determined by the method described in Chapter XVIII and experiments made, when the jet is approximately correct, to check the accuracy, or otherwise, of the ignition timing.

This is done when running on full throttle by moving the

## TESTING THE FINISHED WORK

lever back a very small amount in progressive stages, and noting the rise or fall of the r.p.m. Alternatively, the timing can be altered very easily over a range of about 4 degrees by opening or closing the contact-breaker points, remembering that one-sixth of a turn on the adjustment is roughly equivalent to two degrees. Care must be taken to see that the effect of one adjustment does not mask the effect of the other; for instance, if the advance is too great, the plug tends to run hot and an over-large jet will have to be fitted in an attempt to obtain the right appearance of the plug-face; speed will suffer, both on account of the excessive advance and the over-rich mixture.

When using alcohol fuels, it is not uncommon to find that an increase in jet size has no effect on the mixture strength. This is a sure sign that the fuel flow is being restricted elsewhere; possibly the needle-jet is not large enough, or else the float chamber or fuel pipes cannot deliver a sufficient quantity to keep the jet supplied. If this symptom is noted it is useless to continue going up in jet sizes. The real cause of the trouble must be found and cured, because, although an engine runs very cool in a correct alcohol mixture, piston failure will take place with great rapidity if the mixture becomes weak through any cause, and there may be no premonitory symptoms of engine distress to warn the rider of the impending trouble.

Modern testing plant and the wealth of technical data now available do much to obviate trial-and-error methods of eliminating the minor snags enumerated in this concluding chapter. Such facilities, however, must be regarded as going hand in hand with the skill of the man doing the job, and in putting finishing touches to work assembled carefully down to the finest detail, it is patience and perseverance which ultimately produce the desired result.



# APPENDIX

## REPRESENTATIVE CARBURETTOR TYPES AND SETTINGS (Petrol unless otherwise specified)

MODEL	CARBURETTOR		JETS		REMARKS
	TYPE	BORE	MAIN	NEEDLE	
A.J.S. 498 c.c. o.h.v.	T10TT9	1 1/8 in.	1250	.120	Alcohol
A.J.S. 7R, pre 1955	10RN	1 1/8 in.	320	.109	
A.J.S. 7R, 1955-59	T10GP	1 1/8 in.	270	.109	GP6 needle
A.J.S. 7R, 1959-62	T5GP	1 1/8 in.	310	.109	5 GP6 needle .136 Air jet
Ariel 350 c.c. "Red Hunter"	15TT38	1 in.	260	.107	
Ariel 500 c.c. "Red Hunter"	10TT38	1 1/8 in.	320	.109	
B.S.A. 350 c.c. "Gold Star"	T10TT9	1 1/8 in.	360	.109	
B.S.A. 350 c.c. "Gold Star"	T10TT9	1 1/8 in.	1200	.120	Alcohol
B.S.A. 350 c.c. "Gold Star"	T10GP	1 1/8 in.	280	.109	.125 Air jet
B.S.A. 500 c.c. "Gold Star"	RN	1 1/8 in.	520	.109	
B.S.A. 500 c.c. "Gold Star"	T10TT9	1 1/8 in.	360	.109	
B.S.A. 500 c.c. "Gold Star"	T10TT9	1 1/8 in.	1700	.120	Alcohol
B.S.A. 500 c.c. "Gold Star"	T5GP	1 1/8 in.	330	.109	.125 Air jet
B.S.A. 500 c.c. "Gold Star"	T3GP	1 1/8 in.	350	.109	
B.S.A. 500 c.c. Scrambler	T10GP	1 1/8 in.	240	.109	.100 Air jet GP6 needle
B.S.A. 650 Road Rocket	T10TT9	1 1/8 in.	340	.109	
B.S.A. 650 Super Rocket	T10TT9	1 1/8 in.	410	.109	
Indian, 700 c.c. Apache	T10TT9	1 1/8 in.	480	.109	
J.A.P. Speedway; 8-80 twin	27/013	1 1/8 in.	960	NIL	Alcohol
Matchless 500 c.c. G45	10GP	1 1/8 in.	240	.109	.125 Air jet
Matchless 500 c.c. G80R	T3GP	1 1/8 in.	450	.109	3GP6 needle
Matchless OHC G50	T3GP	1 1/8 in.	450	.109	3GP6 needle
Norton 350 c.c. "Manx"	10TT38	1 1/8 in.	350	.109	
Norton 350 c.c. "Manx"	10TT38	1 1/8 in.	500	.109	
Norton 350 c.c. "Manx" 1946	T10TT	1 1/8 in.	350	.109	
Norton 350 c.c. "Manx" 1946	T10RN	1 1/8 in.	500	.109	
Norton 350 c.c. "Manx"	T10GP	1 1/8 in.	230	.109	
Norton 350 c.c. Short stroke	T10GP	1 1/8 in.	350	.109	
Norton 490 c.c. "Manx"	10TT38	1 1/8 in.	460	.109	
Norton 490 c.c. "Manx"	10TT38	1 1/8 in.	600	.109	
Norton 499 c.c. "Manx"	T10RN	1 1/8 in.	560	.109	
Norton 499 c.c. "Manx"	10RN9	1 1/8 in.	560	.109	
Norton 499 c.c. "Manx"	T10GP	1 1/8 in.	260	.109	
Norton 499 c.c. Short stroke	T5GP	1 1/8 in.	310	.109	
Norton 499 c.c. Short stroke	T5GP	1 1/8 in.	390	.109	5GP6 needle
Norton 500 c.c. Dominator	T15GP	1 in.	190	.107	
Royal Enfield Constellation	T10TT9	1 1/8 in.	480	.109	
Triumph "Cub" Racing	376	1 1/8 in.	140	.106	
Triumph T100 straight	376	1 in.	210	.1065	
Triumph T100 megaphone	376	1 in.	540	.120	Alcohol
Triumph T100 megaphone	15GP	1 in.	220	.107	GP6 needle
Triumph T110 straight	376	1 1/8 in.	250	.106	
Triumph T110 megaphone	376	1 1/8 in.	340	.106	
Triumph T100	76	1 in.	190	.109	
Velocette KTT	T10tt	1 1/8 in.	400	.109	
Velocette Viper	376/61	1 1/8 in.	270	.1065	
Velocette Viper straight	TT9	1 1/8 in.	360	.109	
Velocette Venom silencer	389/15	1 1/8 in.	330	.1065	
Velocette Venom straight or megaphone	389/15	1 1/8 in.	370	.1065	
Velocette Venom straight or megaphone	10TT9	1 1/8 in.	390	.109	GP6 needle
Vincent Black Lightning	T10TT9	1 1/8 in.	360	.109	
Vincent Black Lightning	T10TT9	1 1/8 in.	1400	.120	Alcohol
Vincent Black Lightning	T10TT	32 mm.	1400	.120	Alcohol
Vincent Black Lightning	T5GP	1 1/8 in.	800	.120	Alcohol .150 Air jet

For an increase in altitude the percentage decrease in jet size required is approximately 1 1/2% per 1,000 feet above sea level.

# APPENDIX

## Power Calculations

$$B.M.E.P. = \frac{B.H.P. \times C}{R.P.M.}$$

C depends upon size and type of engine, though not upon the number of cylinders, and is as follows:

For 250 c.c. 4-strokes	..	..	52460
350 c.c. 4-strokes	..	..	37480
500 c.c. 4-strokes	}	..	26230
250 c.c. 2-strokes			
350 c.c. 2-strokes	..	..	18740
1,000 c.c. 4-strokes	}	..	13125
500 c.c. 2-strokes			

For other capacities the value of C can be obtained by direct proportion.

$$B.H.P. = \frac{PLAN}{33,000}$$

where P = Brake Mean Effective Pressure (B.M.E.P.).

L = Stroke in feet.

A = Area of one piston in square inches.

N = Number of explosions per minute.

$$\text{Torque} = \frac{B.H.P. \times 5,250}{R.P.M.}$$

Therefore at 5,250 r.p.m. the torque in lb. per ft. is numerically equal to the b.h.p.

## Froude Brake Calculations

$$B.H.P. = \frac{P \times R.P.M.}{C}$$

where P = Pull in pounds shown on dial.

C = A constant, usually either 4,500 or 5,500, depending on type of brake.

To find Torque or B.M.E.P. directly from the "pounds-pull" reading, multiply by the following factors, irrespective of the number of cylinders in the engine:

$$A = \frac{G \times H}{L}$$



# TUNING FOR SPEED

	ENGINE SIZE	BRAKE CONSTANT	
		4,500	5,500
TORQUE	ALL CAPACITIES	1.167	.965
B.M.E.P.	250 c.c. 4-str.	11.56	9.46
	350 c.c. 4-str.	8.26	6.76
	500 c.c. 4-str.	5.78	4.73
	250 c.c. 2-str.	4.13	3.38
	350 c.c. 2-str.	2.89	2.36
	1,000 c.c. 4-str.		
	500 c.c. 2-str.		

*Example:* If a 500 c.c. four-stroke engine is pulling 30 lb. at 5,000 r.p.m. on a 4,500 constant brake, it is developing

$$\frac{30 \times 5,000}{4,500} = 33.3 \text{ b.h.p.}$$

Its torque is  $30 \times 1.167 = 35$  pounds-feet.

Its B.M.E.P is  $30 \times 5.78 = 174$  pounds per sq. in.

## Engine Calculations

$$\begin{aligned} \text{Mean Piston Speed} &= \frac{\text{R.P.M.} \times \text{stroke in inches}}{6} \\ \text{(ft. per min.)} & \\ \text{or} & \frac{\text{R.P.M.} \times \text{stroke in mm.}}{152.4} \end{aligned}$$

$$\text{Mean Gas Velocity Through Port} = \frac{\text{Piston speed}}{60} \times \frac{D^2}{d^2}$$

(ft. per sec.)

$$\text{Mean Gas Velocity Through Valve Seat (ft. per sec.)} = \frac{\text{Piston speed}}{60} \times \frac{D^2 \times 22}{d_v \times L \times 7}$$

where

D = Diameter of piston.  $d_v$  = Diameter at throat of valve.

d = Diameter of port. L = Lift of valve.

(Either in. or mm. can be used on the right-hand side of these equations, provided the same scale is used for each of the dimensions used.)

# TRIUMPH

## Holder of the world motorcycle speed record

achieved by a  
650 cc Triumph streamliner  
ridden by Bill Johnson  
at Bonneville Salt Flats  
on the 5th September 1962

## 224.57 m.p.h.

\* Subject to confirmation by the F.I.M.

# TRIUMPH

TRIUMPH ENGINEERING CO., LTD., MERIDEN WORKS, ALLESLEY, COVENTRY





50 years experience in the supply of high speed and competition clothing are at your service. Order from us and be certain your clothing and equipment will be 100% correct. Send today for NEW 20 page catalogue.

27 Carburton Street, London, W.1 Tel: EUSton 4793

**HOW TO  
GET THE MOST  
OUT OF  
MOTOR CYCLING**

**MAKE SURE THAT  
YOU ARE WELL  
PROTECTED BY  
EQUIPMENT AND  
CLOTHING FROM**

**(S) LEWIS**

**THE LEADING COMPETITION SPECIALISTS**



**BACKED BY A FULLY EQUIPPED RACE DEPARTMENT**

*All leading agencies and competition accessories*

**GREEVES · COTTON · BULTACO · DOT  
METISSE · AERMACCHI-H/D**

**LAWTON & WILSON**

**264 Millbrook Road · Southampton 27744**

## APPENDIX

### Inertia Load of Reciprocating Parts

Load at T.D.C. (in pounds) =  $.0000142WN^2S \left(1 + \frac{S}{2L}\right)$

Load at B.D.C. =  $.0000142WN^2S \left(1 - \frac{S}{2L}\right)$

where W = Weight of components in pounds.

N = R.P.M.

S = Stroke in inches.

L = Length of connecting-rod in inches.

### Centrifugal Load on Crankpin

Load =  $.0000142WN^2S$

where W = Weight of big-end and rollers.

N = R.P.M.

S = Stroke in inches.

To calculate speed in m.p.h. when time and distance factors are known:

$$\frac{\text{Distance (in miles)}}{\text{Time (in seconds)}} \times 3,600$$

For speed over quarter mile, divide 900 by the time in seconds.

### Conversion Factors

To convert—	Multiply by—
Miles to kilometres .. ..	1.609 (or 8/5 approx.)
M.p.h. to k.p.h. .. ..	1.609 (or 8/5 approx.)
Kilometres to miles .. ..	.621 (or 5/8 approx.)
K.p.h. to m.p.h. .. ..	.621 (or 5/8 approx.)
Gallons to litres .. ..	4.536 (or 4 1/2 approx.)
Litres to gallons .. ..	.2205 (or 2/9 approx.)
Pounds to kilograms .. ..	.4536 (or 9/20 approx.)
Kilograms to pounds .. ..	2.205 (or 11/5 approx.)
Cubic centimetres to cubic inches ..	.061 (or 1/16 approx.)
Cubic inches to cubic centimetres ..	16.39 (or 33/2 approx.)
M.p.h. to feet per second .. ..	88/60 (or 1 1/2 approx.)



## TUNING FOR SPEED

### COEFFICIENTS OF THERMAL EXPANSION

(per ° Centigrade for temperatures between 0 and 200° C.)

MATERIAL	COEFFICIENT
" Invar "	.000009
Cast Iron	.000011
Mild Steel	.000011
Hardened Steel	.000012
Phosphor Bronze	.000018
Aluminium Bronze	.000018
Austenitic Valve Steel	
(D.T.D. 49b, K.E. 965. Jessops G2)	.000018
Austenitic Cast Iron	.000019
Austenitic Stainless Steel (18/8 grade)	.000020
<hr/>	
" Lo-Ex " Aluminium Alloy	.000019
L 33 High-silicon Alloy	.000019
R.R. 53 B	.000022
Y-alloy	.000022
D.T.D. 424 Aluminium Alloy	.000022
Most Wrought High-strength Aluminium Alloys	.000024
8% Copper Aluminium Alloy, L8	.000026
<hr/>	
Heat-treated Cast Magnesium Alloys	.000024/26
Magnesium Alloys as Cast	.000029
Wrought Magnesium Alloys	.000029

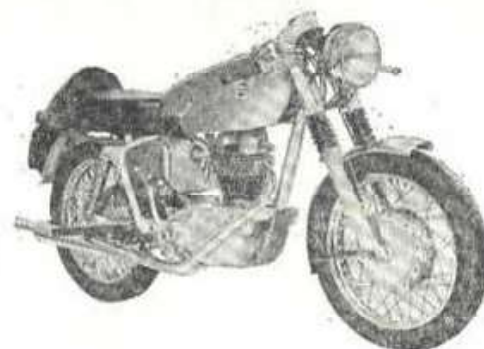
Multiply coefficient by 5/9 to obtain coefficient per degree F.

Example: To find expansion at 200° C. of a 3 in. bore in a jacket cast in L33 material.

Assuming the bore to be measured at 15° C. (normal atmospheric temperature) the temperature rise is 185° C. Expansion therefore is 3 in.  $\times$  185  $\times$  .000019 = .1015 in.

## DEEPROSE CONVERSIONS

for  
your  
250 c.c.  
Crusader  
Sports



Glass fibre 3-gall. Tank	8. 15. 0	Thruaxton Bars	1. 5. 0
Glass fibre 5-gall. Tank	9. 15. 0	Back Set Foot Rests (Plate Type)	3. 19. 6
Tank-Fixing Bracket	7. 6	Back Set Foot Rests (De luxe Type)	7. 19. 6
R.E. Stick-on Tank Badges (pr.)	2. 6	Gold Star Type Exhaust Pipe	2. 19. 6
Red or Gold Crossed Flags (pair)	2. 6	Silencer End Replacement	1. 9. 0
Chrome Head Lamp	5. 19. 6 & 6. 4. 0	Megaphone with Detachable Baffles	1. 9. 6
Head Lamp Stays (Black)	15. 0	Gold Star Type Silencer	2. 19. 6
Head Lamp Stays (Chrome)	1. 5. 0	Exchange Cylinder Head Mod.	5. 0. 0
De Luxe Chrome Clip-on Stays	1. 15. 0	Carburettor 1.1/16 Monobloc	5. 5. 0
Front Brake Air Scoops (set)	19. 0	High Lift Camshaft	4. 19. 6
Front Fork Gaiters (pair)	15. 6	9.75 to 1 Piston (complete)	2. 15. 0
Gaiter Clips (set of 4)	8. 0	Chrome Rear Springs (pair)	1. 10. 6
Fork Top Speedo Head Panel	5. 7. 0	Prop Stand	1. 2. 6
Alloy Mudguard (front with Stays)	1. 9. 6	Anti-Thief Lock '61 onwards	1. 2. 6
Alloy Mudguard (rear with Stays)	1. 7. 6	Oil Warning Light	1. 19. 6
Sports Rear No. Plate	8. 6	Bucket Seat	4. 15. 0
Clip-on Bars (pair)	2. 2. 6	Seat Fittings	19. 6
Adjustable Clip-ons	2. 10. 0	Close Ratio Gears	2. 18. 8
Ball End Levers	13. 6	Standard Tank Covers	3. 4. 0
Blade only	6. 3	Twin Leading Shoe front Brakekit	5. 16. 6
Wal Phillips Fuel Injector	6. 17. 6	Exchange version with new linings	7. 16. 6

Postage and packing extra

Spares Stockists for

MATCHLESS • A.J.S. • ROYAL ENFIELD • SUZUKI • N.S.U.  
DOUGLAS • LAMBRETTA • AMAL • LUCAS • VINCENT  
ARIEL • FI-GLASS • SKEFCO • HOFFMAN • ETC.

**DEEPROSE BROS. LTD.**

178-184 Brownhill Road, Catford, London S.E.6

HIT 8888 • Telephones • LEE 7777



# T. W. KIRBY LTD

Entrant of riders at National and International race meetings

for Speed for Touring

Special parts and service, fairings, etc. For information and help write or telephone

We have a full range of new and used machines of all popular makes, spares, accessories, and fairings. Terms, exchanges, insurance plus enthusiastic service

Contact Tom at

RONEO CORNER · HORNCURCH · ESSEX

Telephone: Hornchurch 48785

# 1<sup>ST</sup> for SPARES

## BLAY'S OF TWICKENHAM

Largest stock in Country

also ACE CLUBMAN racing handlebars and clip-ons, A.C.U. helmets, goggles, etc.

Write, 'phone or call for immediate service

192 HEATH ROAD, TWICKENHAM

Tel: TW1 2103

SPEED AND R.P.M. FOR 3.25-IN. AND 3.50-IN. × 19-IN. RACING TYRES

		R.P.M.												Difference Table											
		30	40	50	60	70	80	90	100	110	120	130	140	1	2	3	4	5	6	7	8	9			
m.p.h.	Gear ratio	3.5	1.382	1.844	2.304	2.765	3.226	3.688	4.146	4.608	5.068	5.530	5.990	6.452	46	92	138	184	230	277	323	367	415		
3.6	1.422	1.895	2.370	2.843	3.316	3.790	4.264	4.740	5.210	5.686	6.160	6.632	7.104	47	95	142	190	237	284	332	379	426			
3.7	1.462	1.948	2.436	2.910	3.384	3.858	4.332	4.806	5.280	5.754	6.228	6.702	7.176	48	97	146	195	244	293	341	390	439			
3.8	1.500	2.000	2.500	3.000	3.500	4.000	4.500	5.000	5.500	6.000	6.500	7.000	7.500	50	100	150	200	250	300	350	400	450			
3.9	1.540	2.054	2.566	3.080	3.593	4.108	4.620	5.134	5.646	6.160	6.673	7.186	7.700	51	103	154	205	257	308	359	411	463			
4.0	1.580	2.106	2.634	3.160	3.687	4.212	4.740	5.266	5.794	6.320	6.847	7.374	7.900	53	105	158	211	263	316	367	421	474			
4.1	1.620	2.160	2.700	3.240	3.780	4.320	4.860	5.400	5.940	6.480	7.020	7.560	8.100	54	108	162	216	270	324	378	432	486			
4.2	1.660	2.212	2.764	3.318	3.870	4.424	4.978	5.530	6.082	6.638	7.188	7.736	8.284	55	111	166	221	276	332	387	442	498			
4.3	1.698	2.264	2.830	3.397	3.963	4.528	5.096	5.660	6.227	6.794	7.360	7.926	8.492	57	113	170	226	283	340	396	453	510			
4.4	1.738	2.318	2.896	3.476	4.055	4.636	5.214	5.792	6.372	6.952	7.530	8.108	8.686	58	116	174	232	290	348	405	464	521			
4.5	1.777	2.370	2.964	3.555	4.148	4.740	5.332	5.925	6.520	7.110	7.700	8.290	8.880	59	118	178	237	296	355	415	474	533			
4.6	1.817	2.422	3.030	3.634	4.240	4.846	5.450	6.057	6.664	7.268	7.870	8.472	9.074	61	121	182	242	303	363	424	485	545			
4.7	1.856	2.476	3.094	3.713	4.332	4.952	5.570	6.188	6.807	7.426	8.042	8.658	9.274	62	124	186	248	309	371	433	495	557			
4.8	1.896	2.528	3.160	3.792	4.424	5.056	5.688	6.320	6.952	7.584	8.216	8.848	9.480	63	126	190	253	316	379	442	506	569			
4.9	1.935	2.580	3.226	3.871	4.515	5.161	5.805	6.452	7.096	7.742	8.386	9.030	9.674	65	129	193	258	323	387	452	516	580			
5.0	1.975	2.634	3.292	3.950	4.608	5.267	5.925	6.584	7.242	7.898	8.556	9.214	9.872	67	134	201	269	336	403	470	537	604			
5.1	2.015	2.688	3.360	4.030	4.702	5.372	6.045	6.716	7.387	8.057	8.726	9.396	10.066	68	137	205	274	342	411	479	548	616			
5.2	2.054	2.738	3.424	4.108	4.793	5.476	6.162	6.847	7.532	8.216	8.900	9.584	10.268	70	140	209	279	349	419	488	558	628			
5.3	2.093	2.792	3.490	4.187	4.885	5.584	6.280	6.979	7.677	8.372	9.066	9.760	10.454	71	142	213	284	356	427	498	567	637			
5.4	2.133	2.844	3.556	4.266	4.977	5.688	6.400	7.110	7.822	8.532	9.242	9.952	10.662	72	145	217	290	362	434	507	579	650			
5.5	2.172	2.896	3.620	4.345	5.070	5.792	6.517	7.241	7.965	8.688	9.412	10.136	10.860	74	148	221	295	369	442	516	590	664			
5.6	2.212	2.950	3.686	4.424	5.160	5.898	6.636	7.372	8.108	8.844	9.580	10.316	11.052	75	150	225	300	375	450	525	600	675			
5.7	2.251	3.002	3.752	4.503	5.245	5.986	6.734	7.480	8.226	8.972	9.718	10.464	11.210	76	153	229	305	382	458	534	611	687			
5.8	2.291	3.054	3.804	4.582	5.345	6.109	6.873	7.636	8.400	9.164	9.928	10.692	11.456	78	155	233	311	388	466	544	622	699			
5.9	2.330	3.108	3.884	4.661	5.438	6.216	6.990	7.768	8.546	9.324	10.102	10.880	11.658	79	158	237	316	395	474	553	632	711			
6.0	2.370	3.160	3.950	4.740	5.530	6.320	7.110	7.900	8.690	9.480	10.270	11.060	11.850												

Note.—For 3.25-in. and 3.50-in. × 20-in. tyres subtract 4% from r.p.m. figures. For speeds intermediate between even tenths, odd r.p.m. from "Difference Table." For ratios lower than six select any sub-multiple and multiply r.p.m. accordingly, i.e. For 12 to 1 select speed for 6 to 1 and multiply by 2.



## FUEL ANALYSIS TABLE

This table gives the symbols and approximate composition of proprietary racing fuels; their availability may vary from country to country.

The increases in jet-flow are approximate only, and are given merely as an indication.

FUEL	APPROXIMATE COMPOSITION (PERCENTAGES)	MAXIMUM COMP. RATIO	JET FLOW INCREASE (per cent)
Shell TT	Lead-free petrol 50, benzole 50	9.5	5
Shell 100	100 octane leaded petrol, plus I.C.A.	9.5	Nil
Shell 115/145	Aviation petrol with T.E.L.	12.0	Nil
Shell X	Ethyl alc. 30, petrol 40, benzole 30	10.5	40
Shell M	Methanol 60, petrol 20, benzole 20	12.5	75
Shell Y	Ethyl alc. 75, benzole 14, acetone 5, water	14.5	100
Shell 811	Methanol 80, petrol 10, benzole 10	14	150
Shell A	Methanol 96, acetone 4, + trace castor oil	15	200
Shell AMI	Methanol 94, acetone 6, + trace castor oil (AMI may be used for blending with petrol or benzole)	15	200
B.P. W	Petrol 50, benzole 50	9.5	5
B.P. 100	Aviation petrol, 100/130 octane	9.5	5
B.P. 115/145	Aviation petrol with T.E.L.	12.0	Nil
B.P. M	Ethyl alc. 20, petrol 40, benzole 40	10	25
B.P. K	Methanol 50, petrol 35, benzole 10, acetone 5	12	60
B.P. JA	Methanol 50, 115/145 petrol, benzole	12	75
B.P. JB	Methanol 60, 115/145 petrol, benzole	12	80
B.P. F	Methanol 80, B.P. Super 10, benzole 10	14	125
B.P. A	Methanol 95, acetone 4, castor oil 1	15	200
Vac. 1	Aviation alkylate (lead-free)	8.5	5
Vac. 2	Petrol plus alkylate; leaded, 100/130 octane	10	5
Vac. 3	Petrol plus alkylate; heavily leaded, 115/145 oct.	12	5
Vac. 4	Methanol 30, ethyl alc. 20, alkylate 40, petrol 10	12	60
Vac. 5	Methanol 80, ethyl alc. 10, alkylate 5, acetone 5	14	125
Vac. 6	Methanol 100	15	200
Vac. 7	Methanol 50, alkylate 10, petrol 25, benzole 10 Acetone 5 (Maserati formula)	12	60
Esso Racing Spirit No. 1	Aviation alkylate (lead-free)	8.5	5
Special 100/115/145	Avgas + 3.5 c.c. T.E.L. per gallon, 100/130 oct.	10	5
No. 4	Avgas; heavily leaded, 115/145 octane	10.5	6
Cooper	Methanol 30, ethyl alc. 20, alkylate 40, petrol 10	12.5	70
No. 5	Methanol 50, benzole 10, acetone 5, 100/130 petrol 35	12	60
	Methanol 80, ethyl alc. 10, acetone 5, petrol 5	13.5	125

The limiting compression ratios quoted apply to well-designed, air-cooled cylinders of 500 c.c. capacity. Engines with smaller cylinders may utilize higher ratios; those with larger or less well-cooled cylinders may need up to one ratio lower.

AVON

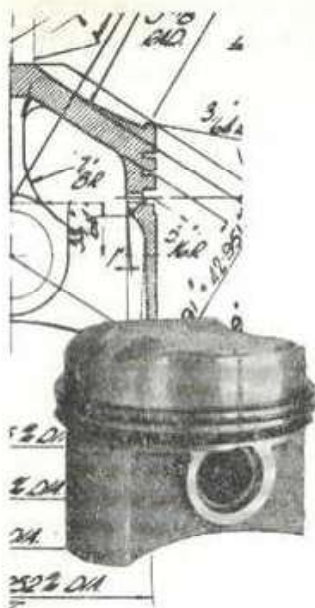
## TYRE PERFORMANCE DATA

MAX. SPEED MPH	RECOMMENDATION	
90 <i>sustained</i>	SOLO	COMBINATION
	AVON PAIRED TYRES Rear AVON S.M. Mk II Front AVON SPEEDMASTER Mk II	AVON TRIPLE DUTY on all three wheels
	PRESSURES: Follow your machine manufacturer's recommendation. Load normally.	
110 <i>sprints</i>	As above, but only intermittent sprints are permissible, and these should not exceed 1 minute's duration.	
110 <i>sustained</i>	As above but pressures should be raised 5 lbs. per sq. in.	
120 <i>sprints</i>	SOLO Rear: AVON G.P. (3.50-19 only) Front: AVON SPEEDMASTER (as above) PRESSURES: For sustained speeds up to 110 mph and short sprints of up to 120 mph, follow machine manufacturer's recommendations. For sustained speeds of 110 and above increase pressures by 5 lbs. per sq. in.	
WHEEL BALANCING. For normal running it is not necessary to have wheels balanced, but for all speeds above 80 mph it is considered essential and will greatly improve handling qualities of the machine.		

Ride safe on

AVON  
TODAY'S *leading* TYRES



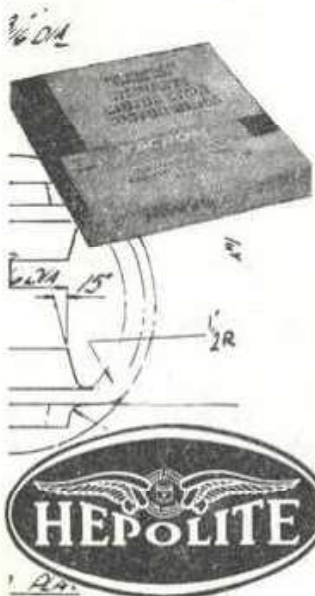


### FIT HEPOLITE

and be confident that your engine is at its peak. Fit Hepolite and know that you couldn't have anything better. There is no greater assurance than Hepolite—nothing more reliable, no surer way to top performance.

### HEPOLITE PISTONS

Hepolite manufacture pistons for every type of machine, from the latest model to older machines of even pre-war vintage. Manufactured in a wide range of forged or cast aluminium alloys, each of which has been selected specially to give maximum results and long trouble-free service under every kind of engine usage.



### PACKAGED RING SETS

The complication of piston ring lay-outs has often caused difficulty where spare rings have to be ordered and fitted. Hepolite Packaged Ring Sets overcome this difficulty—rings are packed in individual envelopes showing full ring details and fitting instructions. They can be supplied with the famous long life Vacrom Chromium Plated Top Compression Ring.

*The obvious choice of all winners!*

**HEPWORTH & GRANDAGE LTD.**  
St. John's Works, Bradford 4  
Telephone : 29595 (17 lines)

## CONWAY MOTORS

**The VINCENT  
Specialists**

Comprehensive range of spares for all post-war models in stock. C.O.D. orders promptly dispatched

**AUSTIN, FORD AND MORRIS STOCKISTS**

ANY MAKE OF NEW CAR OR MOTORCYCLE SUPPLIED

HIRE PURCHASE AND EXCHANGES

Open daily (including Saturday) 8.30 a.m.—6 p.m.

Early closing Thursday 1 p.m. Closed Sundays

**299-309 Goldhawk Road, Shepherd's Bush, W.12**

Telephones: Sales—Riverside 4872/3; Spares—Riverside 5725

## L. W. E. HARTLEY

The well-known engine development consultant, is able, as the result of over 45 years' ungrooved experience, with many engine types, to offer unbiased advice on preparation and modification, for all racing, trials, or touring purposes.

### HARTLEY "GROOMED" ENGINES

have taken over 6,000 awards. Varying from fastest time Brooklands Hutchinson Hundred (twice) and over 100 other Brooklands awards, to records and awards at Cadwell, Oulton Park, Alton Towers, Brands Hatch, Osmaston Manor, Warminster, Silverstone, Shelsley Walsh, Ramsgate and Brighton Sprints, and S.E. Centre Grass Track Championships (solo and sidecar).

### HARTLEY FUEL USERS

have taken over 7,000 awards.

**WHY NOT GET IN TOUCH NOW?**

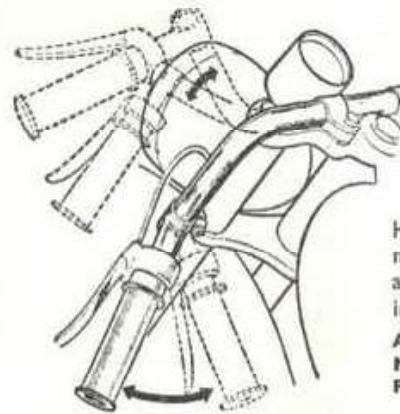
**289 Plumstead High Street, London, S.E.18**

Telephone : Woolwich 1895



## INDEX

- Adler, 223  
Air resistance, formula and calculations, 272, 273  
A.J.S., 18, 82, 98, 110, 119, 163, 260; bearing crankshaft, 195, 196; Vernier device for ignition timing, 188  
Alcohol, 52, 233, 245, 257  
"Alfin", 64  
Allen screws, 34  
"Alpax", 60  
Amal, types of carburetter, 243, 252  
A.M.C., 218  
Ariel, 163  
Austenitic valve inserts, 12, 64; cylinder liners, 218  
Big-end bearings, various modifications, 87, 88, 199  
Big-end cage, 85, 201  
Big-end repairs, 82  
Binks "mouse-trap" type of carburetter, 173  
"Birmabright", 60  
Black and Decker tools, 12  
"Black Lightning", 203-5  
Booster ports, 221  
Brake horse power, co-ordination with m.p.h., 274; formula and calculations, 268, 272, 277  
Brake mean effective pressure, 277  
"Brightray", 30  
British Anzani, 220  
B.S.A., 163  
B.S.A. B32/B34, 190  
B.S.A. "Bantam", 209, 217; carburetter tuning, 232; conversion to close-ratio gears, 236;  
exhaust-pipe length, 211; modified cylinder head, 232; modified exhaust arrangement, 233; port modifications, 226-30; standard port timing 224-5; various compression ratios, 231; work on the transmission, 234, 235, 236  
B.T.H. type of magneto, 185, 186  
Cam-followers, 149; offsetting to minimise wear, 151  
Cams, basic design and modifications, 177  
Cams, Vincent "Black Lightning" type, 198  
Camshaft drives, correct meshing, 148, 149  
Carburation, Amal GP models, 246, 247, 252, 259; modifications for "Twins", 207, 208; R.N. and T.T. Amal instruments, 243, 251; representative settings, 276; "Swill pot", 251; various modifications, 250, 259  
Champion plugs, 263  
"Cobalite", 30  
Coefficients, comparison in varying materials, 282  
Coil springs, 35, 39  
Compression ratio, adjustment, 48, 49, 54; measuring procedure, 2, 5; ready reference table, 54, 286  
Conversion factors, 281  
Craig, J., I.A.E. paper, 174  
Crankcase details, the use of test mandrels, 11, 114



## MOTORCYCLE ENTHUSIASTS

Get the 'Lowdown'  
with the patented

## STEERWELL CONVERSION

Handlebars. Easily fitted to most machines. Adjustable to any position and style. Finished in heavy bright chrome.

ARIEL, B.S.A., A.M.C., HONDA, NORTON, TRIUMPH, JAMES, FRANCIS-BARNETT, ARIEL ARROW.

TRIUMPH 52/6  
OTHERS 50/-

Trade enquiries invited  
24-Hour C.O.D. Service

**BILL SELBY**

HONDA MAIN AGENTS

179 ST. SEPULCHRE GATE  
Phone: DONCASTER 4354

## Other Books by P. E. Irving

### TWO-STROKE POWER UNITS

A new book describing the development of the modern small two-stroke engine as applied to touring and racing motorcycles, and in addition covering cars with capacities up to one litre, outboard motors and various other portable power units.

8½ x 5½ in. 224 pp. Illus. 25s. net

### AUTOMOBILE ENGINE TUNING

Written for the keen private motorist who carries out his own tuning, no less than for those who build and prepare sports and racing cars for competition work, this book explains, in practical detail, how the performance of any car can be substantially improved without detracting from its reliability or handling.

4th Imp. (Revised) 8½ x 5½ in. 224 pp. Illus. 25s. net

### MOTORCYCLE ENGINEERING

A full and authoritative treatment of motorcycle engineering dealing with all aspects of the subject from steering geometry to balance and torque reactions. An essential reference for all design office staff concerned with motorcycles and their power units.

3rd Imp. (Revised) 8½ x 5½ in. 336 pp. Illus. 25s. net

From all booksellers or by post from the publishers

## TEMPLE PRESS BOOKS

42 Russell Square London WC1



# INDEX

- Crankpin, centrifugal loading, 281
- Crankpin modifications, 96, 199
- Crankshaft, checking end-play, 127, 128
- Creg-ny-baa, 264
- "Cromard", 59
- Cylinder filling, improved induction, 170, 173
- Cylinder-head joints, 138; copper gasket annealing, 140; elongation of bolts, 140; the "Double-ground" principle, 138
- C.Z., 224
- Detonation, a limiting factor, 55, 56
- Disc valve, 220
- D.K.W., 220-3
- "Dope" fuels, 44
- D.T.D. steel specifications, 30, 50, 140, 150
- Duralumin, light-alloy, 41, 94, 147, 150
- Dykes type piston ring, 76
- Ehrlich, J., 221
- "Elektron", 60
- E.M.C. Puch, 215
- EN, steel specifications, 83, 94
- Engine assembly, 237, 242
- Engine balance, calculating balance factors, 107, 110
- "Enots" adjustable drip-feed, 120
- Excelsior "Talisman", 216
- Exhaust port, modifying work, 26, 28
- Exhaust systems, 211, 212
- Eysink, 209
- "Fabroil", 186
- Firth-Brown N.M.C. material, 141
- Float chambers, 248, 249
- Flywheel alignment, methods of checking, 102
- Fuel analysis table, 286
- "Gasket-goo", 130, 169
- Gilera, 18
- Gudgeon circlips, 79, 80
- Gudgeon-pin, 78
- Guzzi, 18, 164
- Heenan and Froude, 267
- Hepworth and Grandage Limited, 76
- "Honeychrome", 59
- Ignition advance, co-ordination with compression ratio, 190; "hard" and "soft" plugs, 261, 266; timing, 163-5, 231; timing, an electrical check, 187
- Induction improved o.h.v., 21, 22
- Inlet port, modifying work, 18, 19
- J.A.P., 16, 35, 63, 91, 96, 163, 164
- J.A.P., model "8.80", 120; split-type collet, 40; type of fuel, 44, 197
- Jawa, 224
- Jessop G2 type of steel, 8, 9, 140
- Judge, A. W., *The Testing of Highspeed Internal-combustion Engines*, 271
- "K-Monel", 11
- Kadenacy, 210, 212
- KE965 type of steel, 8, 9, 29, 140
- K.L.G., 263, 264, 266
- L.33 alloy, 60, 64
- Lambretta, 213
- Laystall Engineering Co. Ltd., 59
- Levis, 164
- "Limalloy" piston rings, 76
- "Listard" chrome process, 59
- Lister, R.A., and Co. Ltd., 59
- Lodge, 263, 264

# INDEX

- "Lo-Ex", 60, 64, 70
- Lubrication, gear-type oil pumps, 120; plunger-type oil pumps, 122; reciprocating plunger-type oil pumps, 122; total-loss type, 120; Velocette double-gear oil pump, 121
- Lucas types of magneto, 185
- Main bearings, removal and refitting, 99, 100
- Mainshaft alignment, method of checking, 103, 104, 105
- "Manx" Norton, 18, 91; Vernier settings, 160; valve timing, 164
- Matchless, 119, 164; G3L main bearings, 125; G3L main bearings, method of assembly, 125; three-bearing, 195, 196
- Mean gas velocity, formula, 278
- Mechanical condition, an initial check, 6, 7, 8
- Methanol, 51, 200, 286
- Monobloc, 260
- Monochrome Limited, 59
- Morrison, J. G., 173
- M.Z., 218, 221, 222, 223
- New Imperial, 164
- "Ni-Resist" nickel-iron alloy, 10
- Nitro-methane, 52, 53
- Norton, 164, 165; "Dominant" crankshaft details, 194; 30/40M models, 190; main-bearing details, 126
- Octane ratings, 50, 200
- Orsatt type of gas analyser, 213
- Petrol lubrication, 233
- Petrol-benzole, 7, 49, 50, 286
- Petter "Harmonic" type of engine, 212
- "Picador", 205
- "Pickavant" hydraulic extractor, 81
- Piston alloys, 69
- Piston, alignment, 69, 132; essential clearances, 70, 71; speed formula, 278
- Piston crown, detail modifications, 135, 137
- Piston ring sizes, 75, 76
- Plug readings, 250, 251
- Power corrections, 271
- Proceedings of the I.A.E., 174, 175
- "Progress in Motorcycle Engines", I.A.E. paper by J. Craig, 174, 175
- Puch, 209
- Push rods, 147, 148
- Python engines, 29
- Rev.-counter drive, 186
- Rocker mechanism, Rudge type, 144, 147
- Rocker modifications, lightening o.h.v. mechanism, 145, 146
- Rockwell diamond hardness numbers, 83
- Rolling resistance, 272
- Rose, A. E., 224
- Rowntree, W.B., 53
- R.R. light alloy specifications, 11, 12
- R.R.50; alloy 60, 64
- R.R.56 alloy, 41, 94, 150
- R.R.77 type of alloy, 41
- Rudge, 144, 165, 166
- S.A.E., American steel specifications, 84
- Schweitzer, Dr., 173
- Seat grinders, Black and Decker, 12
- "Seeger" type of piston circlip, 79, 80
- Side-valve engines, 16, 19, 25, 166
- Sifbronze welding, 13, 225
- Silicon piston alloy, 218



# INDEX

- Small-end bearings, modification to con-rod eye, 91
- Sparking plugs, 263
- Steels, expansion coefficients, 141
- "Stellite", 30, 151
- Sunbeam, 165
- Tappet adjustments, 41, 42
- Tappets, direct-impact type, 153
- Tapping drill sizes, 238
- Tetra-ethyl lead, fuel additive, 50
- "Thermochrom", piston rings, 76
- "Top-half" modifications, 134, 137
- Transfer ports, 217, 218, 228
- Transmission, lining up chain drive, 241
- Triumph, 165, 206, 276; crankshaft assemblies, 194; G.P. conversion kit, 192, 206; special extractor tools, 192; splay head, 206
- "Tufnol", 186
- Twin-cylinder engine tuning methods, 191, 208
- Two-plug heads, 265
- Two-stroke engine tuning modifications, 209-36
- Useful formulae, 276-86
- Valve gear adjustments during reassembly, 156-161
- Valve lift, 14; seat inserts, 12; seat mandrels, 11; stem caps, 147
- Valve modifications, 9
- Valve-seat angles, 31
- Valve settings, representative data, 163, 165
- Valve springs, an auxiliary return device, 38
- Valve springs, hairpin type, 36, 37
- Valve timing, degree/linear conversion, 161
- Van der Horst chrome process, 59
- Velocette, 7, 82, 97, 102, 147, 165, 276; KTT head studs, 141; types of rocker box, 37; MSS crankpin details, 202
- Villiers, 97; radially disposed finning, 215
- Vincent, 91, 119, 165, 190; adjusting the compression ratio, 200; big-end details, 87; "Black Shadow" and "Black Lightning", 199; camshafts, 166, 203; details of a modified crankpin, 201; main bearing details, 126; rocker lubrication, 120
- Wellworthy Limited, 64
- Wellworthy pistons, 10, 76
- Y-alloy, 11, 12, 64, 69, 215





A. CLOUGH



D. BICKERS

*Greaves*  
GREAT BRITAIN

**Winners of ALL the  
1964 Lightweight**