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AERO ENGINE ENCYCLOPAEDIA

MODEL AERO ENGINE ENCYCLOPAEDIA

COMPILED BY
R G MOULTON

THE miniature two-stroke Model Aero Engine has revolutionised model flying in the last decade and the rate of development has outpaced full-size practise in many respects. We have glowplug and compression ignition, pressure fed induction and reed valves, useful revolution ranges from 3,000 to 22,000 and power/capacity ratios in the region of 100 B.H.P./Litre. Such advances are fully covered in the 208 pages of this fine volume.

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MODEL AERO ENGINE ENCYCLOPAEDIA

Compiled by
R. G. Moulton

A comprehensive study of the miniature two-stroke power unit in its many and varied forms. Dealing with home construction, mass production, design and operation, it provides a wealth of information on allied subjects and summarises the performance data and specifications of the world's most popular commercial diesel and glow-plug engines.

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Contents

CHAPTER ONE

Types of Engines

THE ultimate performance of a miniature two-stroke engine is governed largely by its porting—the disposition and timing of those passages which permit transfer of the fuel mixture from tank to crankcase, thence to the top of the cylinder and, after firing, out through the exhaust. Usually the "timing" employed has to be a compromise. It can only be absolutely right for one particular speed, which means that it is less efficient at others. Timed correctly for maximum speed, the engine may be difficult to start because the porting is too "open" for low speed running. Timed for easy starting, the same porting arrangement may "strangle" the gas flow at a fairly early period on the speed curve, so the engine will not run very fast.

Manufacturing technique also enters the picture, and physical design limitations. What may approximate to "ideal" timing for a particular design may be costly to make and is ruled out in favour of a less efficient compromise so that the selling price of the engine can be held to a reasonable level. Or perhaps the "ideal" leaves the cylinder too weak so that it can distort, or even break.

Side Port Induction

Prior to about 1948 almost all production engines were of the side port type—the classic three port system where all the ports are formed in the cylinder walls, opened and closed directly by movement of the piston—*Fig.* 1. The intake tube supplying the fuel mixture *via* the needle valve and spraybar assembly (standard "carburettion" on model engines) is therefore attached to the cylinder, either centrally or to one side. "Side" port does not necessarily mean that the intake tube is attached to the *side* of the cylinder, although this is the more usual arrangement, for reasons which we will explain.

Simplifying the engine to just a cylinder, piston, shaft and con rod and intake tube, as in *Fig.* 2, the "timing" of the engine can be expressed in terms of crankshaft rotation (which is more usual) or vertical piston movement (which is a more correct geometric diagram). Either are quite easy to understand, and both are called timing diagrams

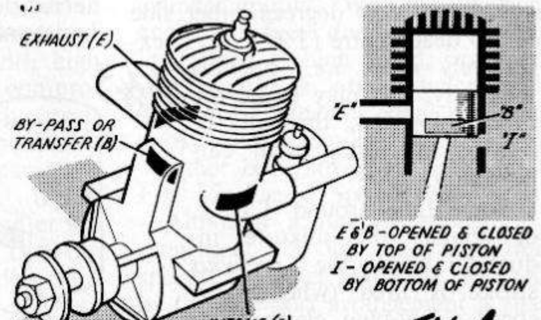
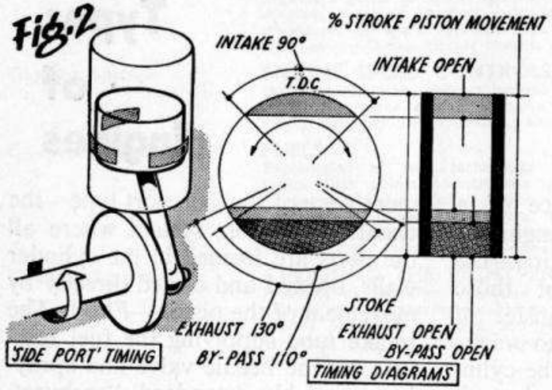


Fig. 1

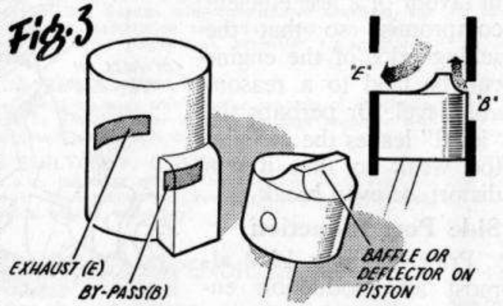


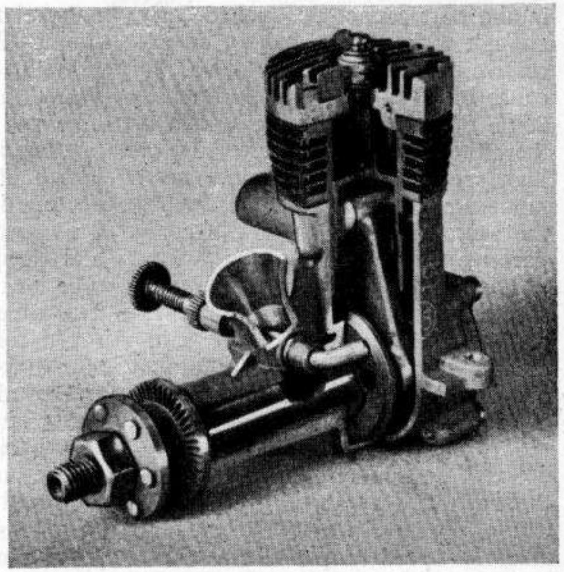
How It Works

Approaching the top of its stroke the piston must uncover the intake opening or port so that the lowering of pressure produced in the crank-case can draw in the fuel mixture, *i.e.*, the intake port is so positioned that it is uncovered so many degrees of rotation, or a specific fraction of the stroke, before top dead centre. In the case of the side port engine this timing must be symmetrical; *i.e.*, the same opening before and after top dead centre. It cannot be anything else. Also the actual "open" time is governed by the depth of the port opening in the cylinder uncovered. A typical design value is 90 degrees either side of top dead centre (T.D.C.). If excessively deep, *i.e.*, opening too early, this may cause too much blowback through the intake, interfering with carburettion.

Some time around T.D.C., of course, the mixture inducted from the *previous* stroke is fired (whether by spark, glow-plug element, or self-ignition is immaterial) and the push for the down stroke is provided by the rapidly expanding gases. Before the piston reaches bottom dead centre (B.D.C.) it must open an exhaust port for these burning gases to escape and a transfer port to transfer the fresh inducted mixture from underneath the piston (where it is being pushed down and compressed into the crank-case) in the upper cylinder.

The exhaust port opens first—a "standard" value being about 115 degrees (crankshaft rotation) past T.D.C.—and again it is obvious that the "timing" will be symmetrical, the exhaust staying open until the piston reaches a corresponding point on the next up-stroke (in this case 130 degrees exhaust opening). The transfer port opens some 15 degrees after the exhaust and therefore for a period of some 100 degrees about bottom dead centre both exhaust and transfer are open. Gases are free to flow both out and in off the top of the cylinder. To prevent the fresh gases flowing straight in and out again is largely a matter of internal design arrangement. With exhaust and transfer ports diametrically





CUTAWAY JAPANESE O.S. MAX-15 reveals flat top piston with straight baffle, crankshaft port, venturi section carburettor and forged connecting rod.

opposed, for example (known as cross scavenging) a baffle or deflector fitted to the top of the piston (in practice a shaped piston top) will direct the incoming gases up and out of the way of the expanding, outgoing gases—*Fig.* 3. A certain outflow of the incoming gases is not undesirable as this promotes proper "scavenging" so that the remaining mixture trapped in the top of the cylinder as the up stroke closes both ports is all fresh fuel-air mixture.

As before, timing is controlled by the depth of the ports. Extending the exhaust port (upwards) gives an earlier opening, but means that the burnt gases are free to escape whilst still highly compressed, hence some of the power available to push the piston down is wasted. The designer aims to delay the exhaust opening until most of the useful power in the expanding gases has been extracted, but, particularly with high speed engines, is forced to compromise, *i.e.*, between the early opening in order to get the necessary time for transfer and maximum utilisation of gas pressure. If the transfer is opened too soon after the exhaust there is a danger that the burnt gases in the cylinder, still under pressure even if they are now escaping through the exhaust, will tend to blow down through the transfer, retarding the transfer of the fresh charge and producing very poor scavenging.

Some of the limitations imposed by timing can be offset by increasing the *width* of the ports, *i.e.*, increasing their actual area. It does not necessarily follow, however, that this will automatically improve the efficiency. Excessively large port widths may also weaken the cylinder unduly. With the side port engine, in fact, due to its inherent limitations at high speed, optimum port width is about twice port depth for engines of equal bore and stroke and a similar effective area of other bore/stroke ratios.

For Moderate Speed Only

Although a flexible enough arrangement for low and moderate speeds, the chief limitation imposed by the side port arrangement is that it cannot induct enough fuel for high speed running. In other words, the intake port cannot be opened early enough without also having an excessive opening time *after* T.D.C. to cause blowback through the intake. Some other intake timing system is therefore invariably preferred for high speed engines and since most modern engines are high speed types the sideport engine is now a comparative rarity.

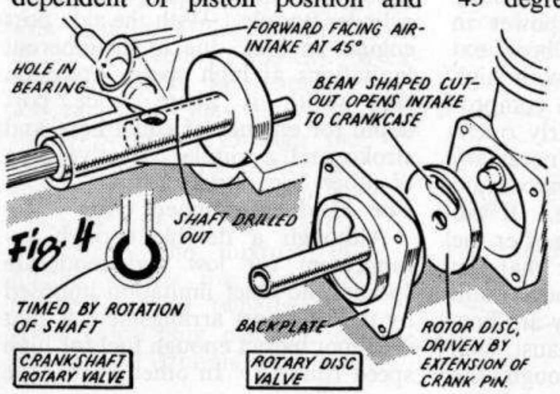
The advantage of a high operating speed is primarily that the efficiency of a two-stroke engine tends to increase with speed and if a torque output can be maintained then the corresponding power output (being the product of torque and speed) will be that much greater. Thus the early pre-war types of engines which, perhaps, developed comparable low speed torque had a maximum speed of 5-6,000 r.p.m. could only develop a maximum power output of about one-quarter to one-third of its modern counterpart peaking at some 14,000 r.p.m.

Rotary Valve Induction

The two standard methods of providing a symmetric induction timing are the crankshaft rotary and crankcase disc type valves in which opening and closing points are independent of piston position and only related to it for the purpose of timing—*Fig.* 4.

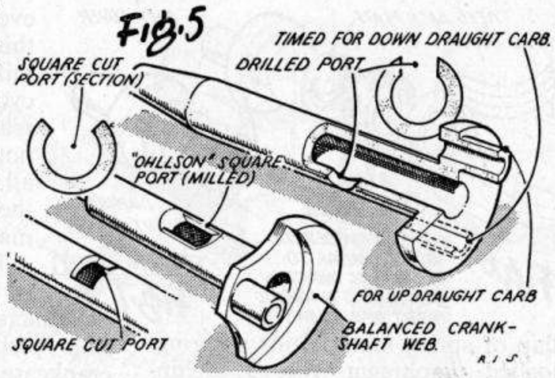
With the CRANKSHAFT ROTARY VALVE the port is a round or square hole cut in the crankshaft itself, opening into a hole drilled along the length of the shaft (and thus connecting directly with the crankcase). This port is timed by its appearance and disappearance past the intake tube let into the crankshaft bearing. This tube or carburettor is normally raked forwards, but not invariably so, although if a vertical tube is employed the end is nearly always cut off at an angle to produce a forward-facing entry.

The amount of forced draught produced by a forward-facing entry is quite small, as also are any improvements in induction resulting. It can be shown, however, that with a vertical squared-off tube, holding a piece of flat material above the end of the tube to deflect air down into it can result in improved induction, so some degree of forward entry would appear worthwhile. Most designers adopt a forward rake of about 20-25 degrees for the intake tube and then angling the top so that the actual entry is at about 45 degrees. A definite forward-stream, tends to make needle valve setting extremely critical and has little to recommend it.



The choice of circular or square port entry in the crankshaft is arbitrary, especially as the end of the intake tube is almost invariably circular. Since a square port gives maximum area for a given width it is often preferred

from the design point of view, when it can also be claimed that the type of port entry produced is more efficient in accelerating the gas mixture into the hollow portion of the crankshaft—*Fig.* 5. About the only objection which can be raised is that the form of stress raiser produced by "stepping" or notching the shaft weakens it more than a circular drilled hole. But as generous crankshaft diameters are common with this type of engine, overall strength is seldom a problem.



Induction port timing is now limited by the size of the "bite" the designer is prepared to take out of the crankshaft ; also, to some extent, by the size of the induction tube. Average figures for high-speed engines are about 150 degrees total opening, positioned 116 degrees before and 34 degrees after top dead centre. These figures are measured off Frog 2.49. *See Fig.* 8.

ROTARY DISC type of induction is virtually unlimited as regards timing at the expense of being a more critical proposition mechanically. It is quite obvious that to increase the opening it is only necessary to increase the length of the slot in the rotor disc, without any resulting weakening of stressed parts. In such cases extremes of timing may be encountered, such as the intake opening as much as 130 degrees before top dead centre, or with the piston only 17 per cent of its stroke up from the bottom dead centre and closing 52 degrees after T.D.C. These figures measured off E.D. 2.46 Racer diesel. More significant from the development point of view is that port timing is readily modified during testing simply by working on the rotor disc with "cut and try" methods and without having to alter any major feature of the engine. Another advantage is the shorter gas passage with this form of induction.

On the debit side is the fact that the rotor disc *must* provide a good seal between the crankcase and the backplate (which generally means hand lapping the two surfaces) ; mounting is a major problem since it has to be driven at very high speed ; and wear is inevitable. In the main, therefore, production-minded designers are more favourably inclined towards the crankshaft rotary valve.

"Flutter" Valves

An attractive alternative which has been exploited on model engine designs quite recently is the reed valve, which appears to have achieved for itself other designations of "Flutter", "Feather" or "Clack" valve—*Fig.* 6. This acts in essentially the same manner as the rotary disc valve, but without rotating parts ; the opening and closing action being provided by a

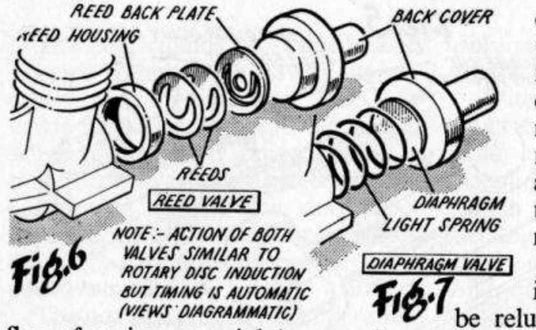


Fig.6 Fig.7

flap of spring material (or a spring-loaded diaphragm—*Fig.* 7. Timing is controlled automatically by the differential pressure between crankcase and induction tube, tending to pull the flap open for mixture to be inducted all the time there is suction in the crankcase and closed when crankcase pressure is higher than intake tube pressure. If spring inertia is discounted this must provide ideal induction timing—a valve open for induction for the whole period there is suction in the crankcase and closing immediately the piston starts its downward travel and begins to build up blowback. In practice, with the right choice of spring material, this ideal timing does in fact appear to be approached closely.

Limitations are the fatigue life of the spring material under operation stresses and the inertia of the system. The conventional reed valve consists of a flap of beryllium copper or phosphor-bronze of about .002-.004 in. thick (depending on size). How greatly this is stressed can only be guess estimated. But both beryllium copper and phosphor bronze are materials with continually diminishing strength subject to fatigue cycles and so eventually must fail under any vibratory load, however light. Since however, this should be measurable in millions and millions of cycles of reversal there is no reason why valve life should not be quite long enough for all practical purposes, with the correct initial choice of materials.

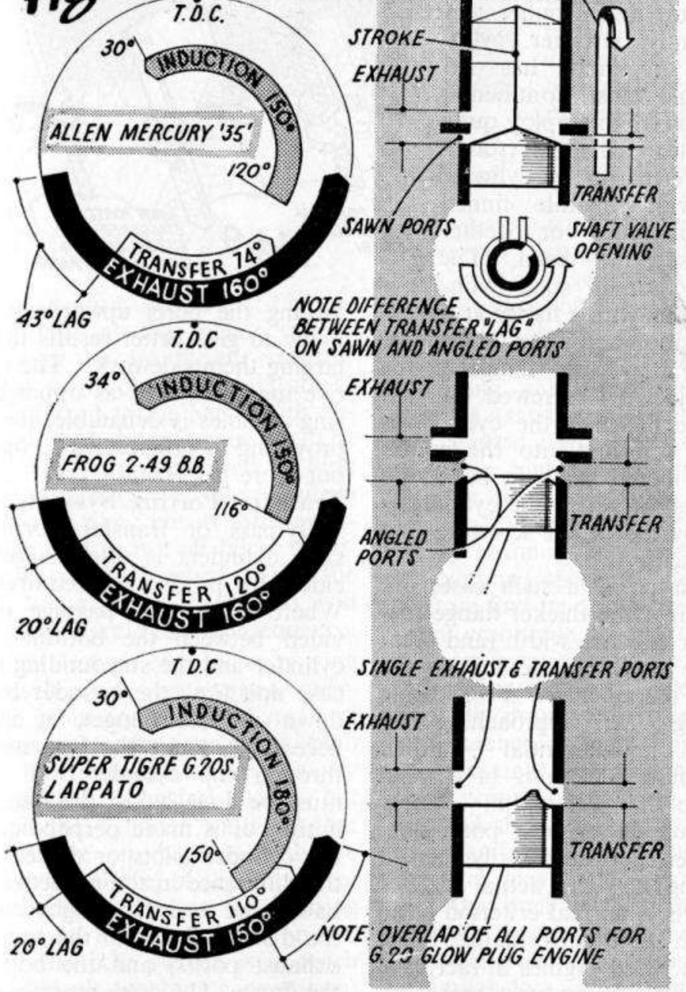
If the inertia of the valve is too great it will probably be reluctant to open at high speed (the predominant pressure in a crankcase being positive), or it may tend to "float" in a partly-open position with a considerable amount of blowback on each revolution. It is still possible, however, that an engine could continue to run, and run quite well, under such conditions. Largely, however, reed valve design is governed by "cut-and-try" methods—both the type employing a clamped reed unit and that employing a spring-loaded diaphragm.

360° Porting

Pushing up the engine speeds beyond the limits reached by side port layouts also has the effect of making the other ports more critical. The faster the speed the less time there is for the mixture to transfer from crankcase to cylinder and for scavenging to be completed. The apparent solution is an increase in port areas all round to maintain a similar volume-time or flow rate figure. Hence the appearance of the so-called 360 degree porting where the ports are cut all the way round the cylinder wall, with only relatively narrow columns of material between to maintain the strength and rigidity of the cylinder.

Here one must pay tribute to the original Arden engine which ap-

Fig.8



peared on the American market in 1946. The use of a steel cylinder with almost 360 degree exhaust porting cut in the walls (the top of the cylinder being carried by only three small columns of metal remaining) and similar 360 degree bypass transfer formed by cutting out passages in the bore at the lower end of the cylinder, set a new standard which has been copied, modified and improved upon throughout the world, but more particularly in Great Britain, beginning with the first of the modern diesels, the Elfin 1.8.

"Diesels"

Concentration on compression ig-

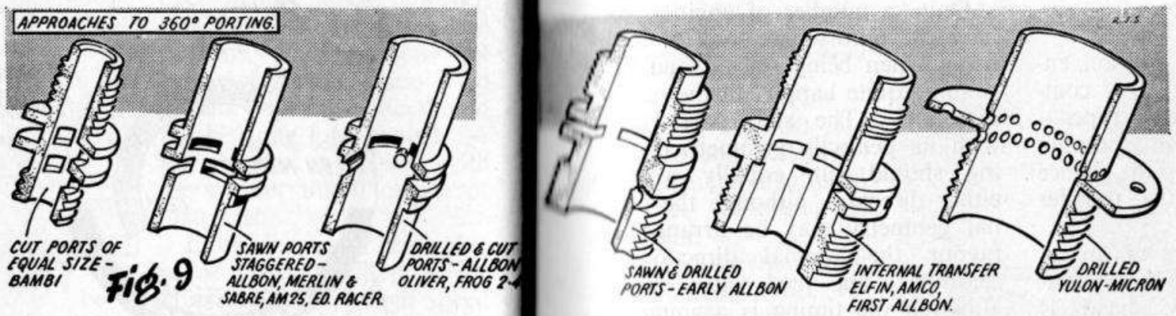


Fig. 9

nition design where cylinder stresses are much higher and consequently stronger cylinders are required, has led British and most Continental manufacturers to employ quite rugged steel cylinders (sometimes referred to as cylinder liners) with separate finned jackets combining—or together with a separate—head. The common form of such cylinders is plain, with a flange at the exhaust position and external surfaces threaded above the flange if the cylinder jacket is screwed on, and below the flange if the cylinder is fitted by screwing into the crankcase—or merely plain in bolted up assemblies, or where the cylinder is located by the jacket screwing into the crankcase.

Exhaust ports in such cases are then cut into the thicker flange section, their effective width (and area) being the resulting opening on the inside of the cylinder. From being quite large, *i.e.*, approaching 360 degrees circumferential porting, there is now something of a trend to reduce the port width. Simply opening up the exhaust ports sideways does not necessarily benefit scavenging and the actual size of these ports is no real criterion at all as to high speed performance. Some very high speed engines in fact have quite small exhaust port areas, although frequently in such cases it is to be found that the opening time has been advanced to something like 110 degrees past top dead centre. Timing, of course, is governed by the depth of the ports (and determined by the position of the flange). On a design where scavenging appears to be incomplete, enlarging the ports upwards is more likely to give better results than enlarging them sideways. The respective merits of slots as opposed to a ring of holes is debatable, the latter providing "progressive" opening, but more gas friction.

Transfer Porting Systems

By-pass or transfer porting on such cylinders is often largely decided by production requirements. Where a free gas passage is provided between the bottom of the cylinder and the surrounding crankcase unit (*i.e.*, the cylinder bedding down on the flange), it is only necessary to cut the by-pass ports through the cylinder wall. This must be done below the flange and if the cut is made perpendicular to the cylinder (slots or drilled holes) the difference in timing between exhaust and transfer is governed by the distance between the top of the exhaust port(s) and the bottom of the flange. The same practice can be followed when the cylinder is screwed into crankcase by milling by-pass passages down the threaded portion of the cylinder—see *Fig.* 9.

If the flange is thick and the stroke of the engine short, this frequently implies a restricted transfer opening time and the higher the bore/stroke ratio the more significant this feature becomes. The result is that at the higher speeds there may be insufficient transfer time for a full charge of mixture to be transferred to the head, with appreciable reduction in power. The engine may run quite well at high speeds, and may even have quite a high peak r.p.m., but its torque output will tend to fall off fairly rapidly as the speed is pushed up and the brake horse power developed will be only moderate as a consequence.

With this type of transfer porting it is nearly always necessary to overlap the geometry of the exhaust and transfer ports for sustained high speed performance, such as by angling the transfer ports cut through the cylinder walls so that they emerge on the side with the top above the level of the bottom of the exhaust ports; or by forming passages to a similar level on the side of the bore (which is a much more difficult machining operation) and so on. The main trouble is in achieving this without unduly weakening the cylinder. One cannot cut into the flange area, either from the outside or the inside, without weakening it. This, in fact, is a point in favour of employing minimum size exhaust ports so that more metal is available in the walls for forming the transfer passages to a greater height. But restricted exhaust port areas will be equally effective in reducing peak performance by preventing proper scavenging although the size of the transfer ports is far more significant, as regards performance.

The shape of the transfer passages themselves is also a matter of some significance for peak performance and satisfactory high speed running (satisfactory in the sense that a good torque output is maintained at high speed; speed, with little or no usable torque is of little practical value). Hence again an engine cannot be judged on its "maximum speed". Letting the mixture find its own way, as it were, from crankcase to the top of the cylinder is satisfactory up to a point, but an increase in gas velocity will normally improve starting and peak performance. Increasing transfer passage area can only be useful up to a point. After that it can be harmful since the mixture may tend to expand into these areas and slow up after leaving the crankcase unless the area is progressively decreased, bottom to top. Starting characteristics then deteriorate and a similar loss of performance may be experienced as that due to insufficient transfer timing. Individual "tuning" techniques frequently refer to polishing the inside of the transfer passages with a view to reducing gas friction and hence minimising any tendency for the flow to slow up, but this, in general, has very little effect if the transfer passage is already of adequate size.

"Glow-plug"

The timing for glow-ignition engines is different from that of compression-ignition (diesel) types—generally arrived to give optimum timing at very high r.p.m., hence tending to exaggerate the transfer period. (*Fig.* 8.)

Thus the performance of an individual engine fitted with alternative "diesel" and "glow" heads is usually quite different. There may be exceptions to the rule, but in general the torque output as a diesel is markedly better than with glow ignition, although with the latter higher r.p.m. figures may be obtained.

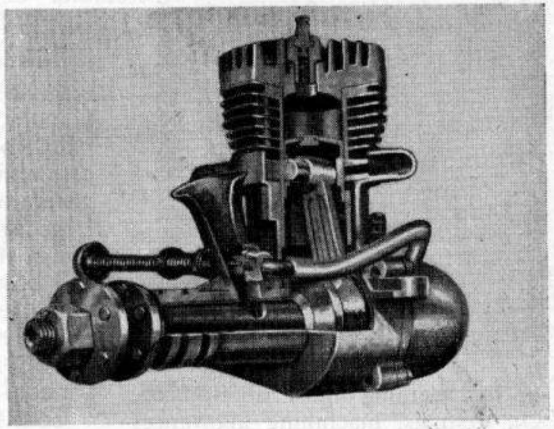
Starting Technique

All engines with "high speed" timing, too, are quite prone to blowback when turned over by hand—a feature which often makes it difficult to draw the fuel line full of fuel by finger choking. (The secret here is to remove the finger when the piston reaches top dead centre and so cut off the blowback effect.) Partial plugging of the induction tube (*e.g.*, with interchangeable venturi inserts), or even special valves on the intake (*e.g.*, the McCoy .049) are features sometimes adopted by manufacturers to promote better starting characteristics on engines timed for high speed running. The point to bear in mind is that for starting high speed engines the faster the propeller is flicked over the better. Quite a number of such engines will not smooth out and run satisfactorily on their own below about 8,000 r.p.m. and so they can hardly be expected to start merely by pushing the propeller over.

Quite a number of engines, too, have the unfortunate trick of backfiring when being started and then running quite happily in the reverse direction. The side port engine with its perfectly symmetrical timing should run equally well in either direction, although the internal geometry may be arranged to favour the normal direction of rotation. The reed valve engine, although the timing is asymmetric, will adjust this asymmetry to either direction of rotation and will again run equally well either way. But the same characteristic is often present with rotary valve engines, particularly when being started with small propellers. Quite often this feature is associated with restricted transfer timing, making the mixture a little late in getting to the top of the cylinder. There is not much one can do about it with a particular engine, except to be prepared for it to happen and avoid starting with the engine over-compressed, which is the condition most favourable to backfiring. Also the sharper the flick-over in the right direction the better.

Modification

It should be emphasised, in fact, that without the proper tools and equipment, and the necessary skill, the average user will probably do more harm than good in attempting to modify the timing of a standard commercial engine. Most British and Continental engines have hardened cylinders to start with, which necessitates either grinding of the ports or softening of the cylinder line and re-hardening after working. This would also mean a new piston would have to be lapped in to fit. Also most crankshafts (including



CUTAWAY JAPANESE O.S.29 includes sectionalised glowplug and shows matched shaped cylinder head and piston crown.

American engines) are hardened and will probably be ruined if softened as exact tempering is important.

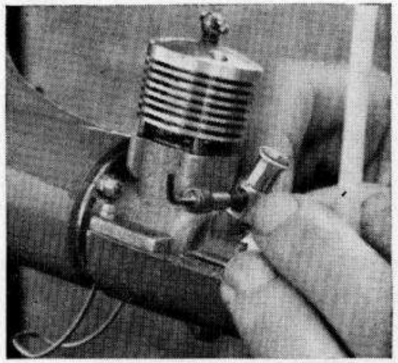
But for the man with the necessary skill, reworking an engine in search of that little extra in performance can often be an attractive and interesting, proposition. The mass-produced commercial engine, is, after all, a compromise between design for performance and design for production, with particular emphasis on good starting characteristics in the more popular ranges.

CHAPTER TWO

Manufacturing Fits and Tolerances

SINCE it is a practical impossibility to produce any machine parts to absolutely *exact* dimensions, it is necessary in manufacture to specify limits of permissible differences or tolerances corresponding to the margin of error which is permissible to give the required degree of practical uniformity. Such tolerances will vary according to the class of work, and the capabilities of the machinery used to manufacture the parts. The skilled individual with fine equipment can work to much finer or closer tolerances on a "one off" job than the normal machine operator on a mass production line. Yet commercial engines have, of necessity, to be tackled as a mass production item to keep the price within reasonable limits. The result of a particular manufacturer's solution is largely passed on to the customer either as a definite characteristic of a particular engine or can be responsible for a considerable difference in performance and handling qualities between individual specimens of a certain engine design.

With first rate machines and a competent operator a practical tolerance figure for turned work is about plus or minus .002 in. Boring can be held to about the same limits. Drilled holes (or bored) followed by reaming can be held to plus or minus .0002 in. although a



ONE OF THE FINEST EXAMPLES OF excellent manufacturing fits is the individually produced Oliver Tiger Mk. III diesel. Radially mounted version on left has a tank built around the crankcase, supported under engine bearers.

normal reaming limit is about .0005. On castings, machining allowances of the order of .030 to .040 usually have to be allowed for on gravity castings in light alloys, whilst with good quality pressure die castings where the molten metal is forced under pressure into metal dies this is reduced to about .005 in. and in some cases nil.

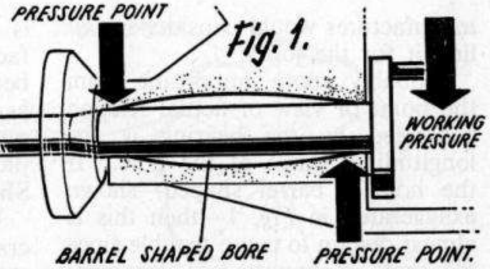
It is now interesting to compare these practical tolerances for *production* against the sort of limits which can be accepted for satisfactory model engine performance on mating parts. The *fit* between mating parts is simply the amount of play or interference between them when they are assembled together. There are three general classes of fits in engineering—clearance fits where there is a positive allowance between the largest possible shaft or sliding member and the smallest possible hole or bore; interference fits where the smallest shaft is smaller than the largest bore; and transition fits where the production tolerances may produce either clearance or interference fits between any two mating components selected at random.

The mating fits we are most closely concerned with in model engine manufacture are the crankshaft-main bearing and piston-cylinder assemblies. These are the main generators of friction which to a large extent govern the power output of the engine. The big and little end bearings on the connecting rod (and the timing disc in the case of crankcase rotary valve engines) contribute negligible friction by comparison.

Considering the main bearing first as the simpler of the two cases, virtually the sole purpose of this bearing is to provide alignment of and support for the crankshaft. The degree of friction or braking effect it produces in so doing will be dependent on the mating materials, the fit, lubrication, r.p.m. and load —and also the surface finish of the shaft and bearing in the case of plain bearings.

The choice of materials is important since this governs the frictional *rate* or coefficient of friction, and also the wear. The general rule is that similar metals in contact generate high friction and high rate of wear (such as the same metals in contact, or two hard or two soft surfaces in rubbing contact). The crankshaft is invariably of steel, usually hardened, and so the bearing surface with a plain bearing is best relatively soft. It has been found, in fact, that the light alloy used for crankcase castings is quite satisfactory as a bearing material and so a lined bearing surface is not strictly necessary.

There are, however, certain advantages in using a lined bearing



such as cast iron, bronze, bearing alloy, etc., principally lower friction. After machining the bearing bore to size such liners are pressed into place (or in some cases shrunk in) and then finished to give the required fit. Amongst the latest practice in this country is to use split sleeve bearings of sintered metal (*e.g.*, Vandervell bearings which are actually produced from flat material consisting of sintered bronze welded to steel sheet. The bearing is finished by wrapping around a former and then tumbled to remove sharp edges).

To make a sleeve it is generally sufficient that the bore of the crankcase casting be bored out to size in a single operation. The outer diameter of the sleeve can be similarly machine finished to a tolerance of about plus or minus .002 in. to ensure a definite interference fit.

Finishing the actual bearing surface is rather a different matter, considerations being the same whether the material is "plain" or the inner surface of a tightly fitting sleeve. A drilled hole is quite unsatisfactory and reaming out to final size is the least of the additional operations required to ensure tolerances and surface finishes of the order required for fit, and also the degree of trueness throughout its length. To reduce the tolerance still further, and to improve the surface finish, honing may be resorted to as a further operation. There is no definite agreement on this point. Some manufacturers adopt honing as standard practice for finishing the bearing bore on plain bearing engines (*e.g.*, Davies-Charlton, E.D., Allen-Mercury, Elfin): Frog engine bearings are currently reamed to size; some American engines are broached.* (Reaming, theoretically at least, results in a hole which is always out of round to some degree, with as many circumferential high spots as the reamer has flutes, the sharper the reamer the less noticeable this effect. A spiral fluted reamer produces spiral high spots which are less significant, but in any case such high spots are extremely small and do not normally cause trouble. Honing after reaming will not necessarily remove all the high spots, but ideally should produce a " cross batched " pattern. Much depends on the skill of the operator in getting a first class finish. Probably broaching is the nearest approach to finishing the ideal round hole, although the necessary equipment is very expensive, and it is doubtful if any British

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* *BROACHING.—A broach is a metal-cutting tool having a series of teeth formed round it, in individual rows. The teeth increase in size slightly from one end of the tool to the other and are also staggered from one row to the other. Thus when the broach is pushed or pulled through a hole the teeth successively cut the hole to the required form, removing metal evenly over the whole of the bore.*

manufacturer would consider installing it for the job.

Probably more important from the point of view of actual friction generated by the bearing is the longitudinal shape of the hole. If the hole is barrel shaped—shown exaggerated in *Fig.* 1—then this is almost certain to cause trouble since the shaft is supported by line contact at each end. In a two-stroke the web end of the crankshaft is always loaded in the downward direction and so the shaft will tend to run on the two point contacts as shown, considerably overloading the bearing at these points. The bearing, as new, may appear to be very nicely fitted with very little play, but in this case will soon score and wear and run hot, denoting excess friction, at the effective contact points.

A bell-mouth bore, on the other hand—shown exaggerated in *Fig.* 2—will allow the shaft to be wobbled up and down in the hand and *appear* very poorly fitted. In practice it may well give excellent performance, even with excessive clearance, simply because there is far more bearing area at the effectively loaded points when the engine is running. As a generalisation, in fact, it can be said that a plain bearing engine (two-stroke) is only as good as its bearing, and the quality of the bearing cannot be judged on apparent *fit* alone.

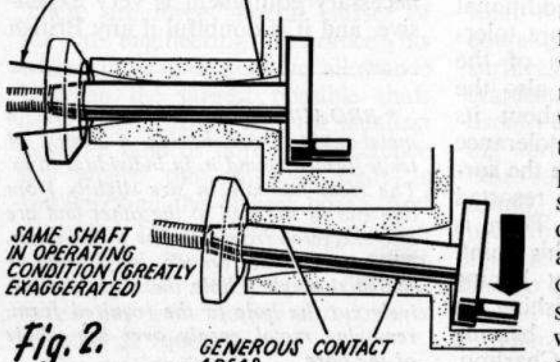


Fig. 2.

Shaft Fit Selection

Usual practice in fabricating the crankshaft to fit the bearing is to machine to normal allowances oversize to harden and grind to a finished size some .0005 to .001 in. above the nominal size. Crankcase bearings are then individually honed to fit a particular shaft, the degree of interchangeability, if any, then depending on the grinding limits and the degree of fit obtained by the honing operator. Thus it is largely improbable that a replacement shaft could be bought to fit an engine manufactured with a honed bearing, since it is generally held that the best fit is of the order of .0002 to .0003 in. Hence it would be necessary to have the crankcase as well to select a shaft giving the desired fit.

The same is true of typical American engines where production practice differs slightly in that finished crankshafts are usually graded in batches to within .0001 in. size and shafts selected from appropriate batches with similar limits for the required fit. Thus the working tolerances on the two mating components produce transition fits and so must be selected individually to match up as clearance fits of the required order. This is not necessarily a disadvantage for where replacements are called for in such cases, if the bearing is available for matching a "good as new" fit is obtained regardless of uniform wear, provided the bearing surface is undamaged.

The main objection to a bearing with a generous clearance fit is that it tends to destroy the seal on the crankcase. The crankcase is effectively the casing of a pump with a predominant positive pressure inside it when the engine is running. Hence a generous amount of oil is likely to be pumped out through the front end of a loosely fitted main bearing. Only if the leak is excessive is the efficiency of the pump action of the engine likely to be seriously affected. In such cases also the necessary lubricating film of oil between the shaft and the bearing surface may not be maintained resulting in excessive friction and wear.

Bearing Tolerance

Thus there is a limit to the amount of clearance which can safely be allowed on a main bearing, again depending on the bearing material. With a clearance much in excess of about .003 in. loss of power may result. On the other hand, a fairly free bearing is to be preferred to a tight one. The latter is likely to pick up on localised high spots, and at the effective loaded areas, which effect can be exaggerated if the shaft is not finished true. Centreless grinding, for instance, will normally finish to a constant diameter but the actual *shape* may not be truly circular.—*Fig*. 3. Slight chatter or vibration will result in a series of very shallow hills and valleys, always an odd number so that diametrically a "valley" always comes opposite a "hill" (hence the constant diameter). So if the operator is in a hurry, or the machine is in need of attention, the actual shaft section may be anything but truly circular. Faults can also occur grinding between centres, depending on how the shaft is held for grinding, so that it is possible to produce (accidentally) barrel-shaped or waisted shafts, and in some cases even out-of-round shafts, although the latter are relatively uncommon. A barrel-shaped shaft is not necessarily objectionable if this is only slight, but a waisted shaft will again produce line contact and highly loaded localised bearing areas.

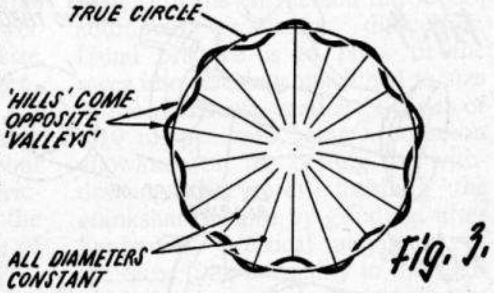


Fig. 3.

It could also be mentioned at this point that since the shaft loading is the direct result of pressure on the piston, the greatest pressure is produced on the down stroke and proportional to the mean effective pressure in the cylinder. Since this pressure and torque follow an identical pattern, as the r.p.m. of the engine *increases* the actual bearing loading *decreases*. Hence, taking an extreme case, it is possible to have a bearing which would seize if run at a moderate speed, but not be loaded to such a dangerous level at a higher running speed. In other words, such an engine might damage a bearing if run in at a low

Fig. 4.

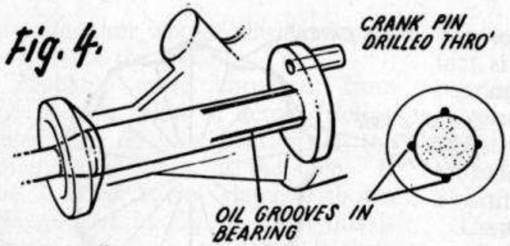
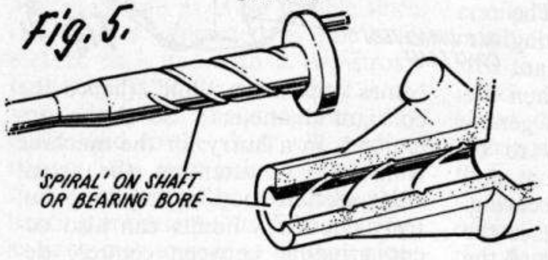


Fig. 5.



moderate speed, but not if run straight away at a much higher speed.

Lubrication Methods

Detail modifications are sometimes incorporated to improve the lubrication of plain bearings, such as grooves cut along the length of the bearing—*Fig.* 4—to distribute the oil; or a spiral formed along the length of the shaft (or bearing surface) to "pump" oil along the length of the bearing—*Fig.* 5—or circumferential grooves in the shaft to retain oil at certain points along the bearing length. The method of *Fig.* 5 can be used to pump oil back into the crankcase on a "leaky" bearing, if the pitch of the thread is reversed. None of these devices, however, is commonly employed in engine design.

A fair test of a plain bearing is that the bearing should feel relatively cool as compared with the cylinder, touching this with the fingers, as the engine is running or immediately after it has been stopped after a run. If the bearing feels excessively hot, it has a high spot or is too tight, which, to the average engine owner, means simply that he must give it more running-in time, preferably at fairly high r.p.m. If necessary, the bearing may be doused with fuel when running to cool it down and prevent local seizure. An engine with a tight main bearing or a tight spot on the bearing will never develop maximum power. A normal well-lubricated bearing will warm up until the heat generated by friction is equal to that dissipated by radiation when it will remain at a constant temperature unless the speed or load changes, and this temperature should be quite moderate. The temperature will *increase* on stopping the engine due to conduction of heat to the bearing area from hotter parts of the engine.

Frictional Values

Friction (and heat) will increase with increasing r.p.m. and, in general figures, frictional values tend to become excessive at speeds of 14,000 to 15,000 r.p.m. although at such high speeds it is usual that the piston-cylinder friction becomes the governing factor. Thus a plain bearing engine generally reaches its peak somewhat below this r.p.m. figure. This is not necessarily true of all plain bearing engines and is tied up with the fit and shape of the bearing. Thus the onset of excessive friction may be delayed by using a more generous fit or more accurate bearing surfaces. Few British plain bearing engines, however, peak above 14,000 r.p.m. and most peak at an appreciably lower figure—the larger the engine size the lower the peak r.p.m. as a generalisation. With glow motors, a higher operating r.p.m. is desirable since the torque figure is lower, but here the reduction in internal friction is generally achieved on the piston-cylinder fit at the expense of some loss of pumping efficiency.

Friction is generally lower when the shaft is mounted on ball bearings—*Fig.* 6—with the additional advantage that the frictional coefficient of a ball race is independent of speed. Thus ball races become an essential feature for the high speed racing type engines which may have to operate at speeds of up to 20,000 r.p.m. and peak at figures of from 15,000 to 18,000 r.p.m.

Ball races themselves are examples of precision engineering, but again produced on mass production lines. Hence they, too, are subject to normal tolerances. The outer ring of a ball race, for example, is never perfectly circular and is generally reliant on a true and substantial housing to minimise distortion (which is seldom available on an engine casing, particularly for the front bearing). For minimum friction, too, it is necessary with most single row ball races to ensure that the races are mounted truly at right angles to the shaft and with the right fit between shaft and ball race bore to prevent distortion of the inner ring. Further, ball races are a relatively expensive item so their use both adds to the cost (and weight) of the engine and introduces additional production difficulties. Usual practice is to press fit the races into housings machined to size in the crankcase casting (a taper of .010 in. per inch being a common allowance on the casting for withdrawing the core), finishing the crankshaft to size by grinding, after hardening. Vertical alignment of the races (*i.e.*, normality to the shaft axis) is ensured by grinding the crank web (front) face and/or the step on the shaft in the case of engines with two diameters on the shaft. It is surprising that so far no manufacturer has employed self-aligning ball bearings on model engine production, which are available in single row bearings in suitable miniature sizes, although probably too expensive for any but the specialised engines.

Main limitations of ball race mounting of the shaft are the susceptibility of the front bearing to pick up dirt and grit and the general "porous" nature of the bearings as seals. The latter can be minimised by the use of suitable cover plates or seals. The Frog 2.49 B.B. utilises a synthetic rubber cover which both keeps out dirt and acts as a seal to prevent the escape of excess oil through the races (in the latter respect a pair of ball races is worse than a loosely-fitted plain

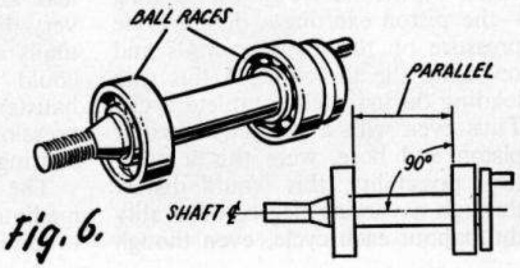
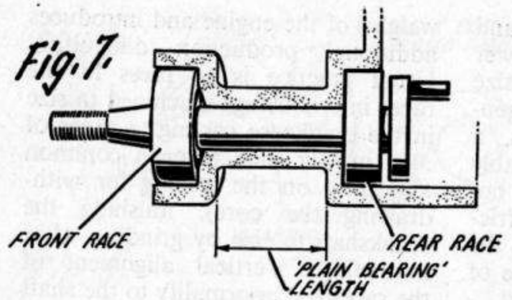
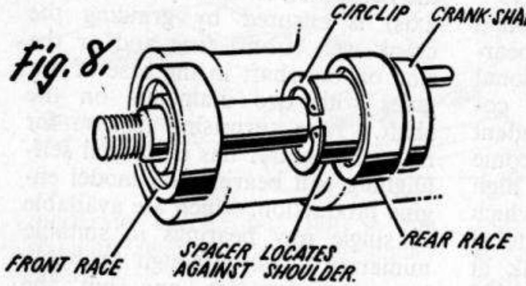


Fig. 6.





bearing, unless steps are taken to trap the oil flow in some way. This can be done by incorporating a section of "plain bearing" length between the two races effectively as a capillary seal without actual metal-to-metal contact (and therefore minimum added friction)—*Fig.* 7—or with a fitted spacer which serves the same purpose, as in the Elfin 2.49 B.R.—*Fig.* 8.

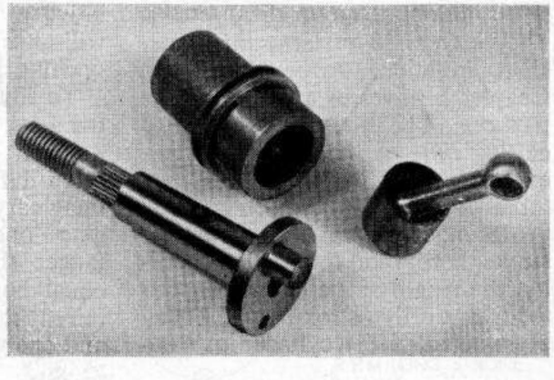
With the piston cylinder fit the question of the seal between the two sliding surfaces is somewhat more important. This problem is aggravated by the nature of the loading—the piston exerting a definite side pressure on the cylinder walls and reversing the direction of this side loading during each complete cycle. Thus even with a perfectly circular piston and bore, were this a practical possibility, this would distort through a varying degree of ovality throughout each cycle, even though the actual change may be microscopic. Such changes may well, however, be within the "fit" limits recognised as necessary for optimum performance, particularly if the cylinder walls are thin. In other words, thin-walled cylinders can be a source of trouble, even distorting when screwed down by bolts through the head.

Contrary to popular opinion, an extremely close fit between piston and cylinder is not necessary for good performance. In fact, it is more probably true to say that the looser the fit the better provided the pumping action of the engine is not impaired and that an oil film is still maintained between the piston and cylinder walls. On a test conducted with a typical commercial 1.5 c.c. diesel, the piston fit was reduced to the order of .0005 in. (as opposed to the more usual average of .0002 in,) with the result that with the same propeller loads corresponding r.p.m. figures were increased by some 10 per cent at the upper end of the speed range. This gain can be attributed entirely to reduced internal friction. Against this was the fact that the excessive compression leak at low speeds made the engine very difficult to start—requiring virtually a temporary seal of oil (which could be injected through the exhausts) to get the necessary compression and crankcase pumping for starting.

The ideal is obviously an intermediate fit. A piston loose enough to minimise friction at the operating

speed required, and yet sufficiently closely fitted to preserve the necessary compression seal. Glow motors are far less stringent than diesels in this respect, requiring far lower compressions to start. Hence it is possible to get away with the lower pumping efficiency produced by a more generous piston-cylinder fit. Thus although the geometric compression ratio may be fairly high, glow motors commonly appear to lack compression when turned over by hand, a feature which cannot be tolerated to anything like the same extent on diesels, if they are to be easy to start. On the other hand, an apparent compression leak past the piston on a diesel when turned over cold is no indication at all of its potentialities.

With relatively few exceptions the cylinders on model engines are designed so that they can be bored right through, then reamed. Normal production technique in Gt. Britain where hardened cylinders are used, employs internal grinding which corrects any out-of-roundness which occurs when the bore is reamed, or any distortion following hardening. Final finishing is then done by honing (although this is not universal), or honing may be used in place of internal grinding. It is a characteristic of honing that the stone will tend to "dig in" at cut-outs, thus tending to form slight depressions in the region of ports formed in the cylinder walls. Being towards the bottom of the stroke this feature has little significance. It is, in fact, becoming increasinly common practice to deliberately increase the diameter of the bore at the bottom of the cylinder thus producing a slightly tapered bore. This, effectively, gives a loose piston fit at the bottom of the stroke and a relatively tight fit at the top for maximum compressoin seal where it is most required.



KEY TO MUCH OF THE success of D. Allen's A.M.10 diesel is the perfect mating fit of the piston in a heavy, distortion-free cylinder. Crankshaft is also burly for 1 c.c., giving excellent bearing surface in the crankcase.

In the case of engines with a blind bore (*e.g.*, the original Ohlsson and Rice engines), the cylinder is made from special steel with an integral head and finished to size on the bore with a fine grinding machine. The cylinder was left soft and a hardened piston employed.

In Gt. Britain the production method commonly adopted for pistons is to turn these .010 to .015 in. oversize and grind down to final size to limits of the order of plus or minus .001 in. It is usual to make in addition batches of pistons .0005 to .001 in. oversize to allow for cases where the honing operator has been rather too generous in the amount of stock removed from the bore of a proportion of the batch of cylinders. American practice is then to grade finished pistons in sizes to within .0001 in. for selective fitting. The more usual practice in this country is to hone the cylinder to fit a particular piston, often using a comparatively coarse hone. One or two manufacturers still persist in lapping pistons in to individual cylinders (*e.g.*, E.D. and J.B. Products). In general, a lapped piston will be much tighter in a finished engine and require a longer running-in time than one in which the piston has been ground to size and the cylinder bore honed to fit.

It will also be appreciated that any of the techniques described can result in appreciable differences in the actual bores (and thus displacement) of a batch of engines of the same nominal size. Usually the greatest differences are found in those engines whose bores are not ground after hardening and may vary as much as .003 to .005 in., depending on hardening techniques and accuracy of machining in the first place. In some cases the differences possible within the manufacturing tolerances accepted, and the production technique and jig limits, can be responsible for an engine of nominal "class" size being found slightly oversize on subsequent checking.

From the previous comments regarding centreless grinding, it will also be seen that a ground piston is not necessarily truly circular, although would indicate so with micrometer measurement (the jaws spanning the minute "hollows"). Pistons ground by other methods should be circular as finished (provided the grinding wheel is reasonably true). This is quite distinct from any deliberate "waisting" of a piston attempted to reduce the the bearing area and so reduce friction.

CHAPTER THREE

Bore and Stroke

THE timing diagram of an engine—expressed in terms of crankshaft rotation, as explained in Chapter I—gives us only part of the picture. The actual opening and closing *time* of the various ports—expressed in fractions of a second (or more truly milliseconds)—will be dependent on the bore/stroke (or stroke/bore) ratio for a given capacity, the length of the connecting rod relative to the stroke, whilst any asymmetry of the cylinder axis relative to the crankshaft centre line will alter the relative speeds of port opening and closing.

To illustrate the effect of varying the sizes of the bore and stroke for a given capacity we can take the three different arrangements for an imaginary 0.1 cu. in. (1.6 c.c.) engine—one with a stroke appreciably longer than the bore; one with equal bore and stroke (usually referred to as a "square" layout); and one with the stroke much short-

er than the bore, or an "over square" layout. These are shown diagrammatically in *Fig.* 1.

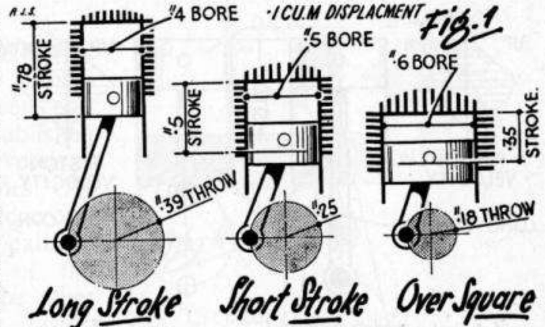


Fig. 1

Shortening the stroke (*i.e.*, increasing the bore/stroke ratio or decreasing the stroke/bore ratio for a given capacity) has two obvious effects. The distance travelled by the piston per revolution is *reduced*; and the load on the crankpin is *increased* for a given shaft torque (due to the reduced "throw"). Also the resulting engine is squatter, enabling its external dimensions to be reduced, with the possibility of an appreciable saving in weight. And for very high revving engines the reduction in friction and wear resulting from a lower piston speed makes the short stroke design more to be favoured than the long stroke counterpart. This advantage is gained at the expense of higher loads on the crankpin and main bearing for the same torque and a greater leakage path around the piston (due to the increased circumference). Although it was at one time held that the advantage of a short stroke for high speed engines was not so apparent in model sizes, with most standard engines having a normal operating speed of 10,000 to 12,000 r.p.m. and above, a near "square" arrangement is almost always adopted in modern designs. (Notable exceptions include the Oliver and Elfin 2.49 radial). This, too, is in direct contradiction to the early conception that a long stroke engine was best for high compression ratios and essential for model diesels. In general terms, the improved performance of model diesels has largely been due to "tailoring" them for high speed operation by increasing the bore-stroke ratio.

In a long stroke engine the piston has to be accelerated from zero at bottom dead centre (B.D.C.) up to a maximum one-quarter of a revolution later, then decelerated to zero again at T.D.C.—*Fig.* 2. The corresponding velocity gradient for a short stroke engine is appreciably flatter. This means that, apart from the mid position and B.D.C. and T.D.C., the piston is sweeping any other point on the cylinder faster with a long stroke than with a short stroke at a given r.p.m. If, there-

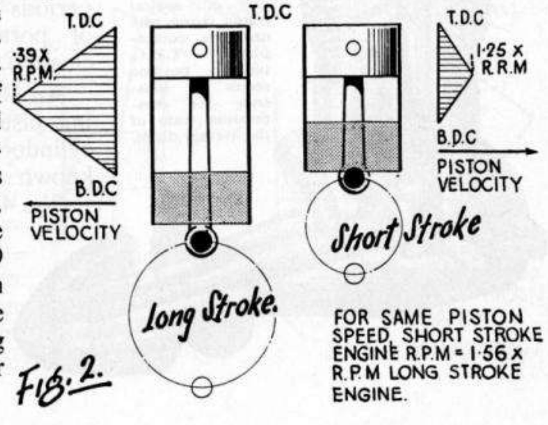
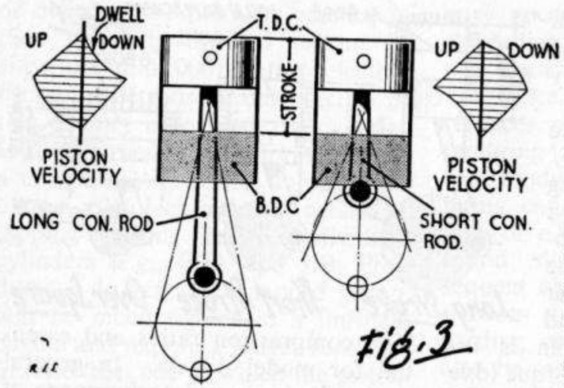
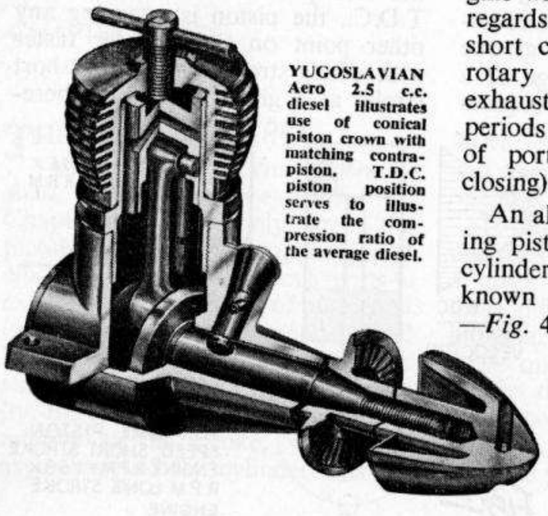


Fig. 2.



fore, the port depth is limited the gas flow will have to be correspondingly faster, a feature which may not be clear from a study of a timing diagram alone. For the same opening period, compared with a short stroke engine, port depth would have to be increased to correspond to the same percentage length of stroke in each engine. Thus the only way to compare port timing without taking the bore/stroke ratio into account is to express it in terms of percentage stroke.



YUGOSLAVIAN Aero 2.5 c.c. diesel illustrates use of conical piston crown with matching contra-piston. T.D.C. piston position serves to illustrate the compression ratio of the average diesel.

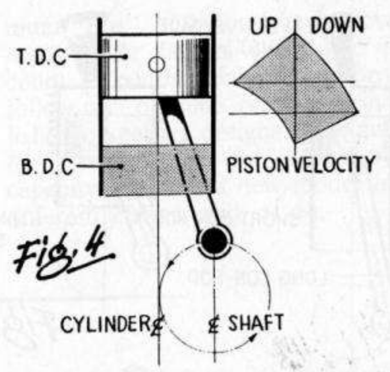
The timing period is also modified by the length of the connecting rod, relative to the stroke. Lengthening the con. rod (for a given stroke) will modify the "velocity gradient" of the piston so that it tends to dwell about T.D.C. and accelerate more rapidly through B.D.C. Similarly, shortening the con. rod will have the opposite effect—the piston tending to dwell about the B.D.C. and accelerate more rapidly through T.D.C.—*Fig.* 3.

Thus, con. rod length can be an important factor in engine design, although usually once the prototype has been made it cannot be changed without a major re-design. It will be appreciated, however, that a relatively long con. rod length could be an advantage in a sideport engine with its inherent limitations as regards induction timing; and a short con. rod an advantage with rotary valve induction to increase exhaust and transfer port opening periods for a given physical depth of ports (strictly, retarding their closing).

An alternative method of promoting piston "dwell" is to offset the cylinder relative to the crankshaft, known as the Desaxé arrangement—*Fig.* 4. If the cylinder is offset in the direction of rotation the piston accelerates faster away from B.D.C., promoting quicker opening and slower closing, giving in effect a larger opening for a given size of port.

Such an arrangement is relatively uncommon on present-day model engines although it has been employed on a number of published British designs and in some versions of the K & B series of engines. As a general rule it is applied to cross-scavenged engines with the exhaust on the "displaced" side of the cylinder. Compared with the other methods of promoting piston "dwell", too, the system is essentially uni-directional. It cannot operate with equal efficiency if the direction of rotation is reversed. Hence a sideport engine with a Desaxé cylinder would have a preferred direction of rotation (the sideport engine having previously been described as the only layout which would run equally well in either direction).

Actually the timing feature of a Desaxé cylinder is not necessarily the reason for its adoption. It may be employed for mechanical reasons in that it greatly reduces the side thrust of the piston during the power stroke. As *Fig.* 5 shows, once the piston has moved away from T.D.C on its power stroke, in a symmetrical cylinder design, the pressure is being transmitted at an angle via the con. rod, thus causing the piston to bear against one side of the cylinder. For a given stroke, the Desaxé cylinder of *Fig.* 5 reduces this angular thrust effect to the minimum possible, at the expense of increased piston side thrust on the up or compression stroke, this being almost negligible by comparison since the load is very much lower.



Normally, however, side thrust loads on pistons do not appear to be a critical problem with model engines. The side thrust generated will be proportional to the stroke, but independent of bore/stroke ratio if the ratio of con. rod length to stroke is the same in each case. The longer the con. rod (for a given stroke) the lower the side thrust because of the reduced angular displacement—*Fig.* 6. Likewise, the greater the stroke, the longer the con. rod required to maintain the same value of side thrust, which means that the overall height of the engine tends to become still further exaggerated, if this particular

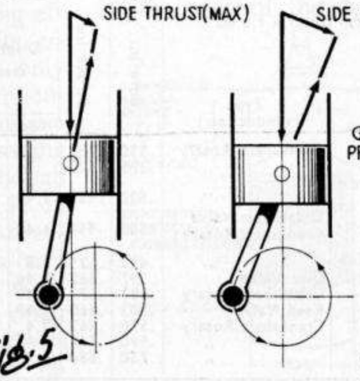
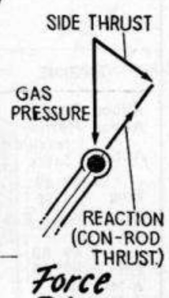
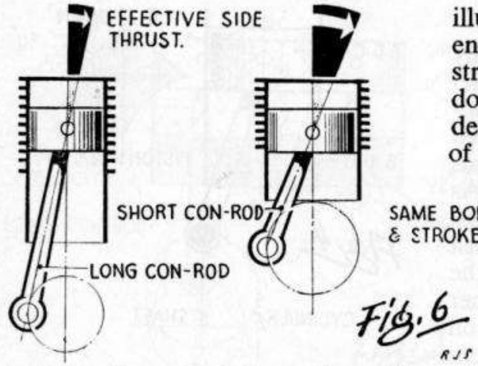


Fig. 5



Force Diagram.



feature is pursued. Generally speaking, the only troubles which are likely to arise, within conventional design proportions, is if the piston depth is greatly reduced, relative to its diameter. It is normally considered inadvisable to make the depth of the piston less than ¾ x diameter.

As a matter of interest, the table summarises measured data extracted from a number of typical engines

TECHNICAL DATA ON BORE/STROKE OF BRITISH ENGINES

ENGINE	Type (Induction)	Bore in.	Stroke in.	Displacement c.c.	Displacement cu. in.	Bore : Stroke	Stroke : Bore	Piston Speed Factor*	Con Rod Length in.	Con Rod : Stroke
Allbon Dart	Crankshaft Rotary	.350	.350	.55	.0036	1.0	1.0	.0583	.58	1.66
Allbon Merlin	,, ,,	.375	.420	.76	.0464	.89	1.12	.07	.58	1.38
Allbon { Javelin / Sabre }	,, ,,	.520	.420	1.49	.0909	1.24	.81	.07	.718	1.71
Frog { 1.49 / 1.50 }	{ Diaphragm Valve / Crankshaft Rotary }	.500	.460	1.48	.0903	1.09	.92	.076	.844	1.83
Oliver Tiger Cub	,, ,,	.432	.625	1.5	.092	.69	1.45	.104	1.062	1.7
ED. 2.46	Disc Valve	.591	.562	2.46	.150	1.05	.95	.094	1.062	1.89
Frog 2.49 BB	Crankshaft Rotary	.581	.574	2.49	.152	1.01	.99	.0956	.969	1.69
Elfin 1.49 BB	Reed Valve	.503	.460	1.49	.0910	1.09	.92	.076	.92	2.0
A-M "25"	Crankshaft Rotary	.570	.562	2.4	.15	1.02	.98	.0936	.88	1.57
A-M "35"	,, ,,	.590	.567	3.44	.210	1.59	.63	.0936	.88	1.57
Frog "500"	,, ,,	.750	.680	4.93	.30	1.1	.91	.113	1.375	2.02

* Piston Speed = $\frac{\text{stroke x r.p.m. ft./min.}}{6}$ = column factor x r.p.m.

illustrating the range of proportions encountered in practice. The short stroke or near square engine is undoubtedly the present standard for design with a connecting rod length of 1.7-1.8 x stroke (average), resulting in a relatively squat, compact layout as compared with earlier engines.

Increasing the bore/stroke ratio, however, is no cure-all for high performance design problems. In fact a large bore can become something of a disadvantage with high-compression ultra-high performance engines where particular attention is paid to the design of the combustion chamber of optimum flame propagation. Clearances within the head are very small and the relatively large piston and head areas exposed may tend to reduce thermal efficiency. Invariably, however, all high speed engines are of short

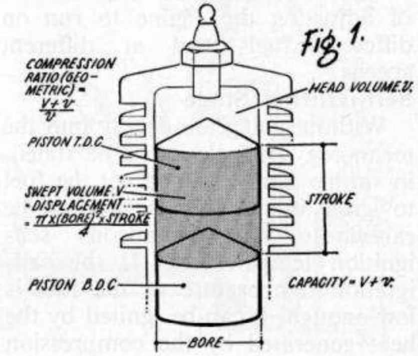
stroke design; and long stroke engines, where still made, designed for generating high torque at low or moderate speeds. The true "general purpose" engine, it has been suggested, should have a stroke slightly greater than the bore, this being what we would classify as a "sports" type engine with a maximum life. Obviously, however, many other factors come into account in commercial productions—following proven practice established by earlier designs; designing for "reworking" to a different capacity later for a new model in a different class; and so on.

CHAPTER FOUR

Compression Ratio

THE effective compression ratio of an engine is determined both by the geometry of the engine and its efficiency as a pump. The latter feature is commonly overlooked and the *geometric* compression ratio taken as the absolute criterion for performance. (For example, full size engines normally have the compression ratio *lowered* when supercharged, except the low pressure blowers fitted to series production cars to improve performance.) But, for example, suppose the (geometric) compression ratio of any particular engine is 10 : 1 and its pumping efficiency 60 per cent. Then its *effective* compression ratio is only 6 : 1—or the same as, say, an engine with an apparent compression ratio of 6 : 1 with a pumping efficiency of 100 per cent (although the latter would be impossible in practice). This, in part at least, explains why some "hot" engines have (apparently) a relatively low compression ratio, compared with other racing designs which do not perform so well. It also explains why increasing the pumping efficiency of an engine can considerably improve its effective compression ratio and performance, although physically the engine is apparently unchanged.

Compression ratio is defined as the ratio of the total volume within the cylinder above the piston bottom dead centre position to the unswept volume, *i.e.*, the small volume left in the head when the piston is at top dead centre—*Fig.* 1. The two "volumes" involved are the head volume (v) and the swept volume (V). The latter is readily calculated as the area of the bore *times* the



stroke and is also called the *displacement*. The full capacity of an engine, on the other hand, is equal to the swept volume or displacement *plus* the head volume, *i.e.*, V + v. For "class" assessment, it is the swept volume that is always accepted as the c.c. or cu. in. figure.

The actual value of the head volume (v) is often difficult to calculate and, in fact, the actual volume required for a particular design is usually "guesstimated" rather than calculated, or arrived at by trial and error methods in altering the shape of the head, or the top of the piston, or both. Thus the only true "size" rating of an engine is *displacement* and to speak of the capacity of an engine without knowing the head volume or compression ratio is quite wrong. In fact, nearly always when an engine is stated as "X" c.c. capacity, when it is meant that the swept volume or displacement of that engine is X c.c.'s.

In the case of diesels, of course, the head volume and thus both the capacity and compression ratio is made variable. Although a few fixed capacity diesels have been made, it is now universally recognised that a variable compression ratio is the most satisfactory method of adjusting the engine to run on different fuels and at different speeds.

Self-ignition Stage

Without going too deeply into the technology of fuels it can be stated, in simple terms, that to get the fuel to ignite in the cylinder it must be raised to its spontaneous self-ignition temperature. If the self-ignition temperature of the fuel is low enough, it can be ignited by the heat generated by the compression of the fuel-air mixture in the head, this heat of compression being directly related to the effective compression ratio of the engine. If the self-ignition temperature of the fuel is too high for this to be realised employing practical compression ratios, then some other method of supplying the heat must be provided, such as a spark plug or heated element. In the latter case, as in the glow plug, the fuel also has a catalytic action on the element, tending to heat it to red heat (*e.g.*, as in the simple science experiment where a piece of platinum wire held in alcohol vapour will heat up to incandescence and set the vapour alight). The actual temperature at which the element will be maintained, however, is greatly influenced by the compression ratio, which has led to the variety of so-called "hot" and "cold" glow plugs.

The significance of compression ratio with diesels need not be discussed in detail since the working ratio is readily adjustable to give optimum running conditions. The main criterion, in fact, becomes the fuel. It can be mentioned, however, that conventional compression-ignition fuel oils, which are mainly paraffin-type oils, have a self-ignition temperature too high to be ignited by the maximum heat of compression normally generated in model engines. Hence they have to be mixed with a substance which has a lower self-ignition temperature (usually ether) and is a relatively poor fuel, as such. Add necessary lubricant and you have the basis of all diesel fuels—a paraffinic oil (which is the base fuel), ether to promote easy starting and lubricating oil.

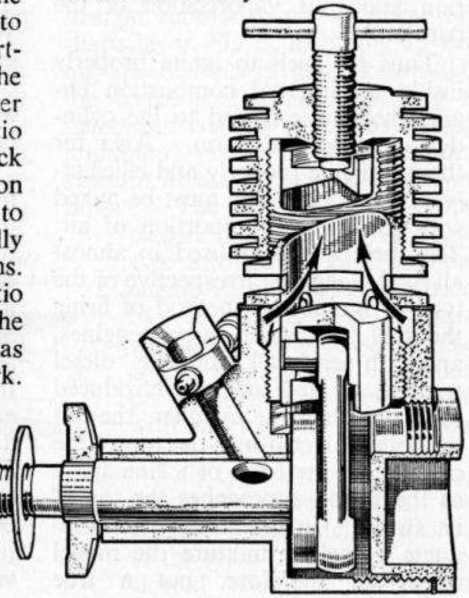
In the case of glow motors, the position is rather different. The base fuel is methanol, which has a self-ignition temperature nearly twice that of diesel fuel and nearly three times that of ether. Hence a heated element is required to ignite it. (It is an interesting fact that an ether-methanol mixture *can* be used to start and run a diesel, although such a fuel mixture is not recommended! Similarly a "diesel" fuel can run a "glo" motor, with the same reservation!)

Chapters 11 and 14 detail the fuel requirements for self-ignition engines.

The significance of compression ratio can thus be summarised in this form. For use with a "straight" glow fuel (*i.e.*, no added nitromethane), a glow motor must be made with a high compression ratio. The geometric compression ratio required will depend a lot on the pumping efficiency and may need to be as high as 14 : 1 for good starting and continued running with the lead disconnected. On the other hand, such a high compression ratio will tend to make the engine kick back, particularly hand starting on small propellers, and be prone to start and run backwards, especially on "symmetrically" ported designs.

Decreasing the compression ratio will improve starting and enable the engine to go faster, because it has to do less internal pumping work. But unless nitromethane is added to the fuel it will be prone to stop, or run inconsistently, once the starting battery is disconnected. With a lower compression ratio still and a "straight" fuel, starting may be difficult even with the battery lead connected and the plug just goes out once the engine is running and the lead disconnected.

The average commercial design will therefore probably aim at a compromise compression ratio where a relatively inexpensive fuel can be used, *i.e.*, with about 5 per cent nitromethane. The use of a fuel with a higher proportion of nitromethane will still improve performance, because this is a characteristic of such a dope, but probably at the expense of shortened element life on a standard plug, if this were originally correctly matched to the design. A change of plug to a "hotter" type might well be advisable, or even necessary.



DUTCH TYPHOON 2.5 c.c. DIESEL drawn to illustrate piston at bottom dead centre when transfer passages around the lower cylinder are opened and fresh gases are forced into the upper cylinder, helping to expel remainder of the expanding exhaust gases.

CHAPTER FIVE

Carburettion

IT is a fact that liquid fuels in liquid form are reluctant to burn. To render them combustible, they have to be in finely divided or vaporised form, mixed with air. Ordinary paraffin provides a good example of this. It does not vaporise at room temperatures and so a match plunged into a tin of paraffin would merely be doused, almost as if you had plunged it into water. Yet gently heated so that the surface of the liquid was covered with a film of vaporised paraffin and a match brought near it would readily set it alight. (You get a similar effect with a wick used with a paraffin lamp, the wick promoting evaporation and thus vaporisation of the paraffin.)

Thus for fuels to ignite properly inside an internal combustion engine they must be fed to the cylinder in vaporised form. Also for them to ignite properly and efficiently, the vaporised fuel must be mixed with the correct proportion of air. This principle is utilised in almost all model engines, irrespective of the type of ignition or method of firing the fuel. In some larger engines, and with nearly all "full size" diesel engines, air and fuel are introduced separately into the cylinder, the latter being injected into the top of the cylinder in the form of a fine spray as the piston approaches the top of its stroke. In the sense that it inducts a fuel-air mixture the model diesel is, therefore, not a true "diesel" in the accepted sense (a true "diesel" employing "solid" fuel injection) and is more correctly, a compression-ignition engine. In model engine sizes, and particularly because light volatile fuels are used, supplying a fuel-air mixture to the cylinder is by far the simplest solution and gives quite satisfactory results.

The part of an engine concerned with metering and atomising the fuel and mixing it with air is generally termed a carburettor. Again, in model sizes, the type of carburettor used is about the most elementary form that it can take—again because of simplicity and the fact that it will do the job satisfactorily. Whereas the carburettors on larger engines have to incorporate throttle controls, model engines are, largely, one speed engines with any particular load.

Because of its simplicity the model engine carburettor is seldom termed as such, although it performs the basic function of "carburettion"—*i.e.*, metering and mixing the fuel and air supplies to the engine. Nearly all forms are basically similar and consist of a metering jet inset in a tube, the latter called either the choke tube or induction tube. The metering jet is either of fixed size (comparatively rare) or with variable size of orifice brought about by means of an adjustable needle which can be advanced into or withdrawn from the

orifice and so vary its effective opening or area.

The needle valve and jet assembly is usually of one of two forms. The jet opening can be located in one side of the intake tube with the needle valve entering it; or the jet tube can be extended across the width of the intake tube with a hole (or holes) at its centre, the effective jet orifice area being varied by adjustment of a needle valve running inside the tube—*Fig.* 1. A majority of modern engines employ the latter type, the extended jet tube being known as the spraybar. It is far less critical and rather more efficient (for most purposes) than the jet in the side of the tube. It is also less sensitive to changes in fuel level due to better capillary attraction between needle and spray bar.—*Fig.* 2.

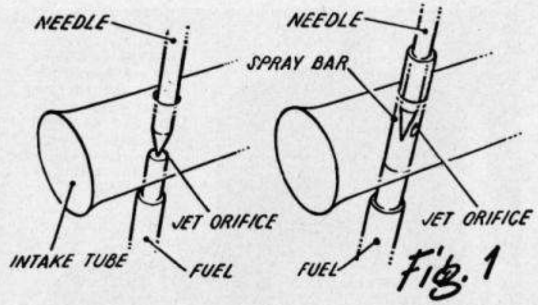


Fig. 1

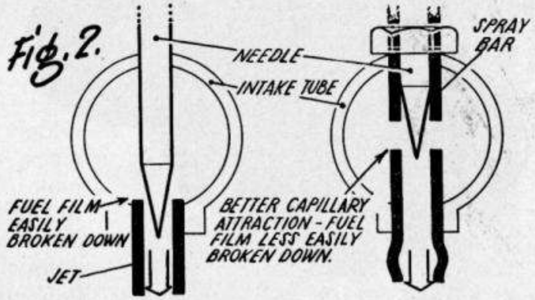


Fig. 2.

Whichever design is employed the principle involved is that of creating a reduction in pressure within the intake tube at the region of the jet, thus producing a suction effect to lift the fuel out from the jet in the form of a spray. The simplest way to ensure a suction effect in a straight tube is to give it a venturi shape, as in *Fig.* 3, incoming air being speeded up in passing through the convergent section, reaching a maximum velocity (and thus having minimum pressure or maximum suction) at the narrowest section or throat. The jet orifice is thus placed at this point. Air passage through the tube into the engine is, of course, produced by the reduction in pressure within the crankcase of the engine during the induction period of the timing cycle, and with the induction port open.

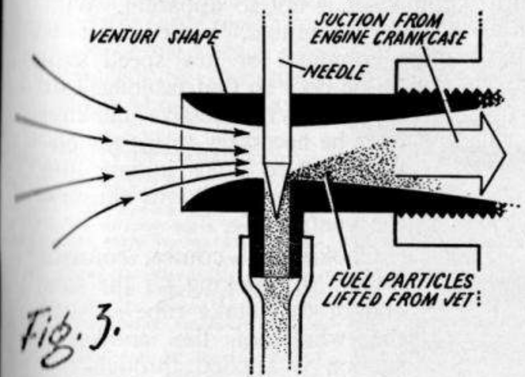
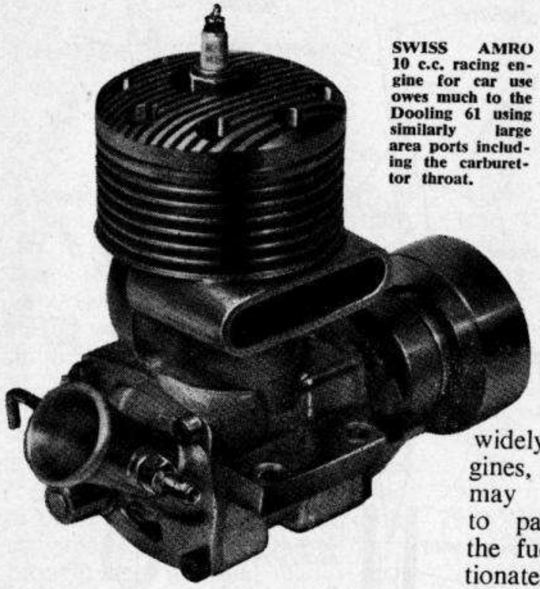


Fig. 3.

This type of carburettor has a number of limitations. The airflow immediately adjacent to the walls of the tube will be slowed down by friction,



SWISS AMRO 10 c.c. racing engine for car use owes much to the Dooling 61 using similarly large area ports including the carburettor throat.

hence the actual suction effect will be less on the walls than at the centre of the tube—*Fig.* 4. Thus the fuel will be less ready to emerge in the form of a fine spray and also the size of the orifice will tend to be very critical. In other words, even with a finely tapered needle valve the setting for correct fuel proportions will tend to be extremely critical, a fraction of a turn making all the difference between a mixture which is too weak or too lean.

This inherent disadvantage can be minimised by using a good venturi shape in the first place, which ensures that there is a reasonable amount of suction at the walls and a high finish on the walls to minimise gas friction. Also it becomes less important where relatively large quantities of fuel are involved, with proportionate large air volumes and high velocities. This type of carburettor is still widely used on the larger racing engines, where the intake diameter may be quite big in order to pass the necessary air and the fuel flow rate is also proportionately high. Engines of this type, too, usually have rotary disc induction, which itself induces swirl and "chops" any solid fuel particles into more finely divided form. All two-strokes inherently tend to have good atomisation characteristics, due to the heat of the combustion chamber, swirl induced by crankshaft rotation, etc. And because of the large quantity of fuel passing the sensitive nature of the needle valve control it is not so apparent. With such carburettors, however, it is quite common for low speed suction to be poor so that prolonged or (apparently) excessive choking may be necessary to get the engine started and initiate the proper flow conditions through the venturi.

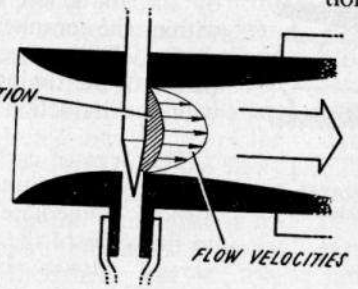
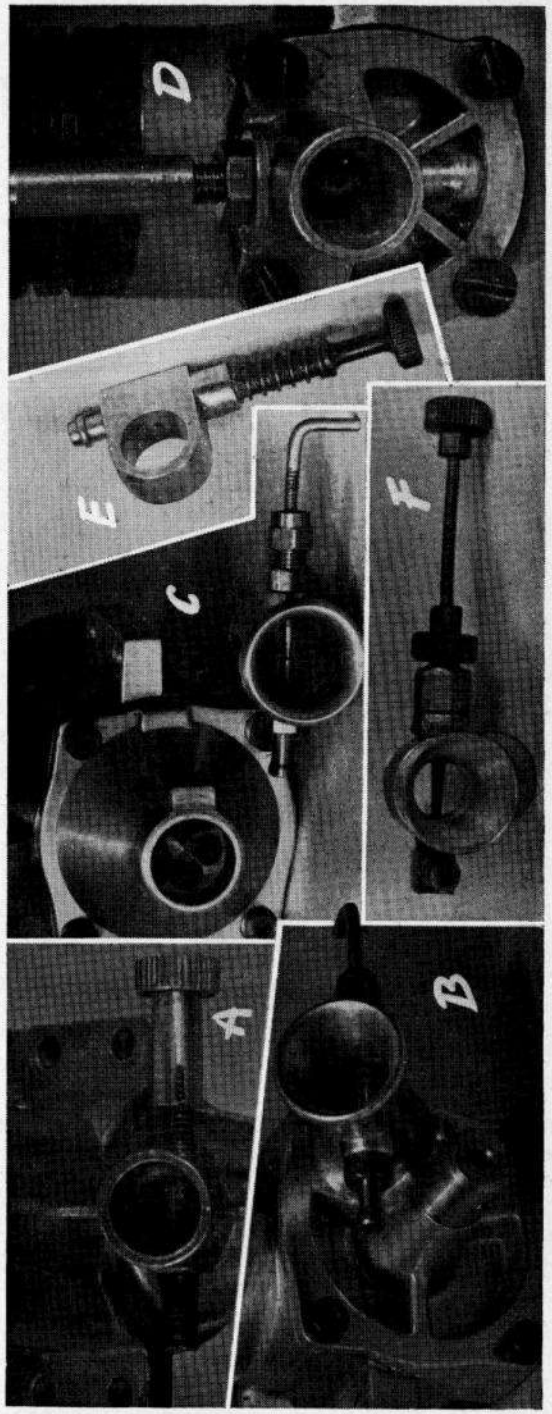


Fig. 4.

Choking, of course, consists merely of blocking off the free end of the intake tube so that the whole of the crankcase suction is applied through the fuel system, thus providing an extremely powerful suction lift to draw raw fuel up into the intake tube—*Fig.* 5. It necessarily results in an over-rich mixture because of the absence of air. Partial choking implies leaving the intake tube partly open so that a very rich mixture is produced (*i.e.*, limited air induction)—*Fig.* 6, and may be used to promote firing and initial running after full choking. Partial choking will not, normally, draw in raw fuel unless the fuel level in the tank is at the same horizontal height as the jet.

With the spraybar type carburettor—*Fig.* 7—the spraybar itself acts as a constriction in the intake tube and therefore a venturi shape is not essential to promote suction at the jet hole. It is also apparent, looking at the assembly from the top, that there are three regions of suction—one at each side produced by "venturi" effect, and one behind the



VARIETY IN CARBURETTORS. A—An American K & B 15 with straight spraybar. B—Shows Amro 10 with Dooling type needle entering separate valve body. C—Flutter valve version of the same engine using a valve gear after the style of the Dyna-jet and shows needle valve orifice near mid-point of carb. D—Webra Mach 1 with spraybar using flattened sides to increase throat area. E—American Thimbledrome or Thermal Hopper remote needle assembly which has multiple jets around inside face of collar. F. The American McCoy surface needle valve with 90 deg. fuel line connection.

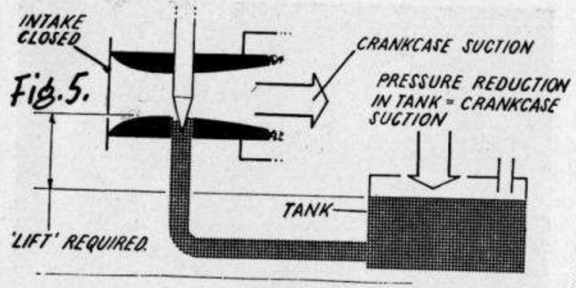


Fig. 5.

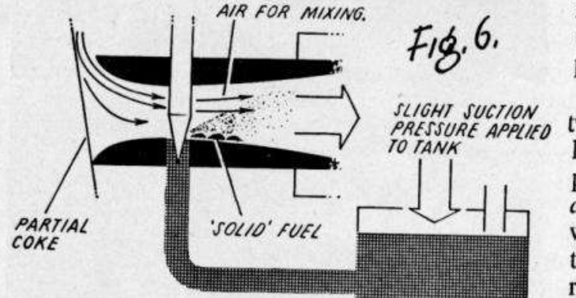


Fig. 6.

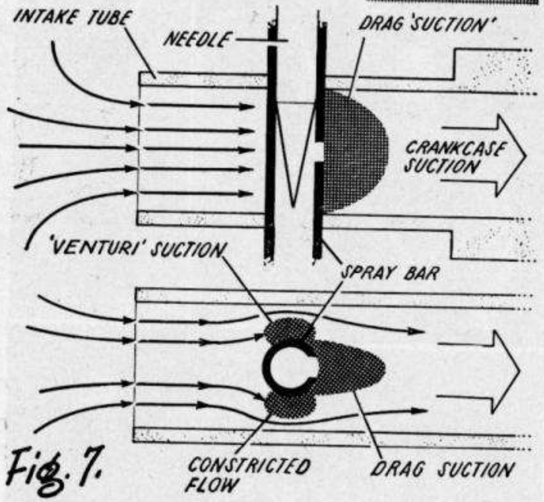


Fig. 7.

spraybar due to its drag effect. Also these effects will be maximum in the centre of the tube.

If the spraybar has a single jet hole, then the best effect is produced if this is faced downstream—*Fig.* 8(a). On some designs with this type of spraybar, the engine will only run consistently with the jet hole so located. On others consistent running can be obtained with the hole at one side, but where this is so a slightly coarser needle valve setting is usually required to supply the same mixture.

If the spraybar has two diametrically-opposed holes, then it will usually perform satisfactorily in *any* position. In other words, you do not have to worry about how you reassemble the spraybar, if removed but again the most economical position (*i.e.*, as reflected by the leanest needle valve setting) will be specific—with the holes in the position shown in *Fig.* 8 (b).

Locating the spraybar with one hole facing forward and the other back—*Fig.* 8 (c)—will mean that only one hole is effectively spraying, but this action will be assisted by a positive pressure build-up through the other hole. In any other position—*Fig.* 8 (d)—either one or both holes may spray, but in all cases adequate carburettion is produced.

The actual bore of the intake tube must be proportioned so that it is large enough to pass enough air to

produce the required volume of mixture at the highest design speed of the engine. In simple terms, the faster the engine is to run the larger the intake tube required. Coupled with the increased gas velocities resulting from high speed operation, adequate suction will be maintained.

At lower speeds, however, the large bore intake will prove an embarrassment. Increasing the bore means a marked reduction in suction effect with lower gas velocities so that even if the engine can be started initially by choking, suction in the intake during the period when it is running on its own and building up speed, may be unsatisfactory and so the engine does not get the proper mixture to sustain running. This is the reason why some high speed engines are hard to start and often difficult, or even impossible, to run at low or moderate speeds (although other design factors also enter into the question of low speed running, of course). The solution may be a definite venturi shape for the intake or, more usually, interchangeable venturii which can be inserted into the throat of the intake tube—*Fig.* 9. A whole range of such venturii may be used, the one with the smallest opening making for good starting and low speed running (but starving the engine and thus limiting its high speed performance), and so on up to no insert for maximum performance. Plugging the intake with a piece of balsa or hardwood, was at one time a common method of "taming" high speed flow motors for moderate speed performance—*Fig.* 10.

An apparent solution to getting more air through the intake tube is to have the intake facing forwards into the airstream and thus utilise

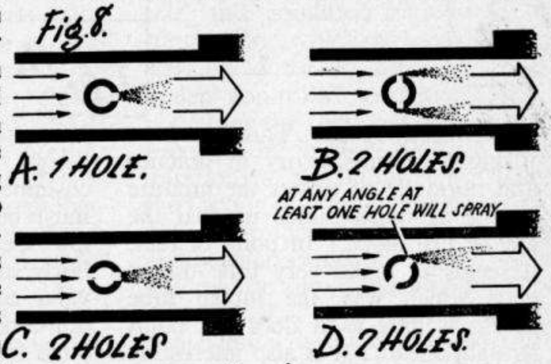


Fig. 8.

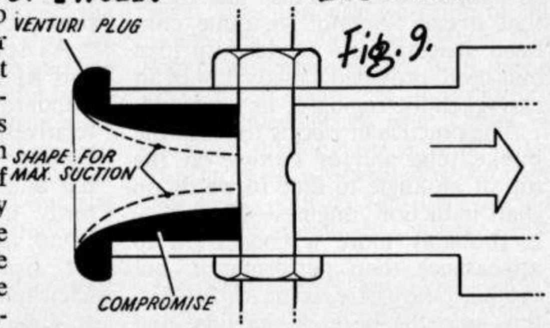


Fig. 9.

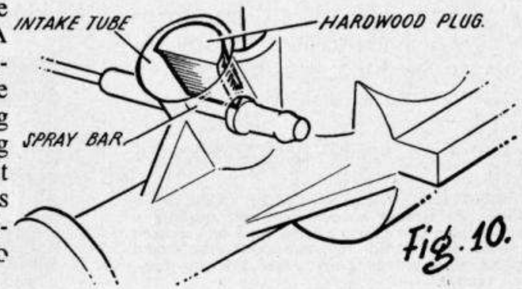
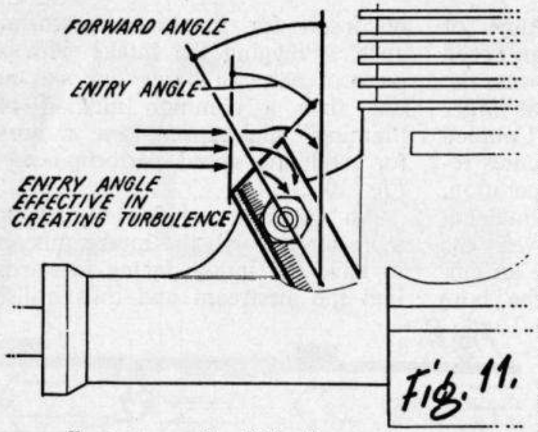


Fig. 10.



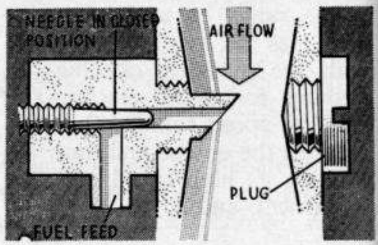
ram effect at speed. This, however, is highly unsatisfactory in practice and normally so upsets the mixture setting as speed builds up that the engine just stops. In point of fact, it seems to make very little difference which way the intake tube faces, as long as it does not point straight ahead. It is also interesting that it can "breath" in quite confined spaces, such as close up to a bulkhead, provided a ready flow of air can reach the region of its open end.

The practice of raking forward the intake tube and/or cutting off the top of an angle to face forwards on shaft-induction engines—*Fig.* 11—is probably more a concession to appearance than performance, but in this case there is no *direct* airflow into the bore of the tube and the adverse ram effects just mentioned are not present. There appears to be little possible gain in experimenting with rake angles, as such.

The high speed performance of some engines can, however, be improved by increasing the bore of the intake tube and giving the entry a bell-mouth shape. It may well be that production demands a rather generous safety margin on wall thickness in the first place and perhaps adherence to an original crankcase design for which the moulds have not been fully utilised.

The manufacturer of standard commercial engines has nearly always, of necessity, to produce a compromise design. To satisfy a majority of his customers, starting characteristics must be good (which means good low speed suction and therefore a fairly small bore intake); the needle valve control needs to be relatively non-sensitive, but still positive enough for accurate setting at around peak r.p.m.

As such, the spraybar type with two jet holes has become almost a standard and as an example of how relatively non-critical such a carburettor control is, the same spraybar unit will often perform satisfactorily in a range of engine sizes from, say .5 c.c. up to 2.5 c.c. (as for example the entire Davies-Charlton range) even if in the former case it appears to be almost blocking the intake tube.



A MODIFIED NEEDLE VALVE ASSEMBLY designed to replace normal spraybar, employs a blanking plug and uses a needle valve with shaped jet orifice and 90 deg. fuel line attachment. Could also be employed in pairs either side for two-speed control.

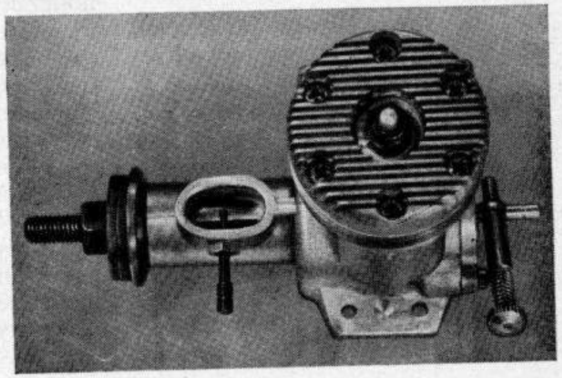
CHAPTER SIX

Cooling

WITHOUT exception, model aero-engines are designed for air cooling and are seldom critical about this particular requirement. That is to say they will run satisfactorily over a wide range of temperature with little difference in performance, provided the "standing" heat is not so high as to cause distortion of the cylinder or burn or "carbonise" the oil in the fuel mixture so that its lubricating properties are destroyed. High surface speed will also break down castor oil, but not Castrol 'M'.

There are three sources of heat generation when the engine is running—rubbing friction between the moving parts, heat generated by compressing the gas mixture and the heat given out by the mixture when fired. Theoretically, at least, this flame temperature is subject to cooling as the gases expand and escape through the exhaust, but the residual heat is still quite high and the overall heating effect pronounced.

Thus in the complete engine the cylinder, and particularly the top of the piston and upper part of the cylinder, is subject to the most heating—*Fig.* 1. Friction heat should be tolerably low. Excessive friction means bad running fits and these are to be avoided. In the case of new engines set up on the "tight" side, this condition is relieved by running in, which is a process of wearing down to size and "fit" under controlled conditions, *e.g.*, not letting the frictional heat become excessive by limiting the speed and duration of the initial runs. Thus the friction of a main bearing should always be low so that the bearing continues to run cool, and thus does not require any particular form of cooling. If it does run hot on any one point it is quite likely to burn away the lubricating oil film at this point, increasing local friction (and local heating) still more until partial seizure can occur. If this condition is suspected when the engine is running, dousing the outside of the bearing with a liberal dose of coolant (*e.g.*, pouring fuel over it) will often momentarily relieve the



FOX 29R ILLUSTRATES THE need for deep section fin area on a cylinder head in a high speed engine of this type of design employing voluminous carburettor intake.

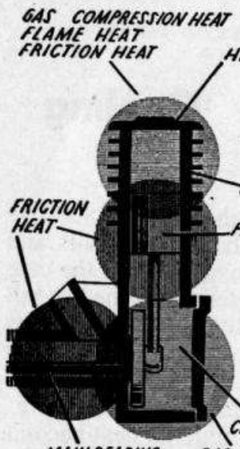


Fig. 1.

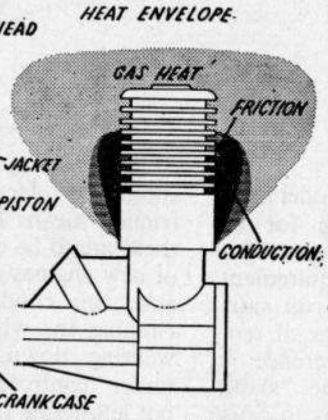


Fig. 2.

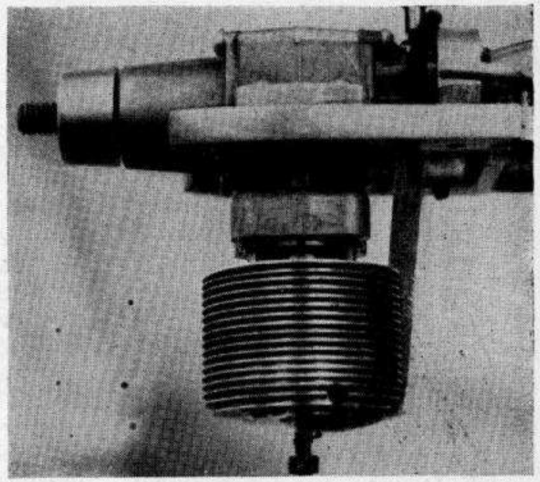
trouble. But the real cure in this case is not improved cooling but a better running fit (see previous chapter on fits and tolerances).

The cylinder, on the other hand, normally receives unequal heating. Frictional heat, again, should be quite low and normally a negligible part of the total heating effect, provided there is adequate lubricant in the fuel. The practice of relieving the cylinder bore at the bottom part of the stroke to reduce friction is far more concerned with reducing power losses than with reducing heating.

Thus the cylinder is heated, mainly, by the compression and firing of the fuel mixture at the top of the stroke. The top of the cylinder is heated directly by this means whilst the lower portion receives heat indirectly through conduction of some of this heat through the cylinder walls. The final heat "envelope" is of the form shown in *Fig.* 2 with the top of the cylinder receiving by far the most amount of heating. And since metals expand on heating to a degree proportional to the temperature rise, it is fairly obvious that distortion of the cylinder can take place. such distortion can have several effects. It can obviously affect the piston-cylinder fit at the top of the stroke, perhaps to a point where piston friction does become excessive, so resulting in loss of power —and still more heating to make matters worse. If the temperature reaches the point where the oil itself is carbonised lubrication will break down and the piston will soon seize. Distortion can also lead to gas leakage, further affecting efficiency, and is a problem which engine designers are always up against. It is more apparent in diesels than in glow motors, largely because of the higher working pressures and "tighter" piston fits, which is the main reason why the cylinder liner or cylinder of a diesel is usually much thicker in the wall and much more robust than that of a glow motor of similar size. The faster the engine is made to run the hotter it is likely to get (due to the increased rate of "heat" cycles) and the bigger the problem. In the end the "best" engine is usually the one which experiences minimum cylinder distortion and it is significant that some engines with exceptional performance for their size—like the A.M. "10" — have exceptionally robust cylinders.

Fortunately only a relatively moderate amount of cooling is necessary to restrain the heat "envelope".

EXTRA LARGE FIN DIAMETER and increased number of fins on this ETA 15D Mk. II is the work of Mans Hagberg (Sweden). The lever is a method of remote compression control for team racing.



Finning

The ideal arrangement is a series of very thin fins formed integral with the cylinder walls, their individual lengths corresponding approximately to the shape of the aforementioned "heat envelope"—*Fig.* 3.

American manufacturers commonly do adopt this method, machining thin fins directly on to the cylinder barrel. British and Continental engine design is more or less standardised around the use of of a separate cylinder jacket screwed on to or bolting down against a hardened steel cylinder which becomes, in effect, a sleeve or liner—*Fig.* 4. It is not practicable to reproduce the jacket as a series of very fine fins. Equally it is impractical to form fine fins on a *hardened* steel cylinder as these would be extremely brittle and readily broken. There is also the point that the cylinder steels used on American engines are not readily available in this country.

This cylinder jacket is nearly always made from light alloy, to save weight. Aluminium, too, is a very good conductor of heat, so that the whole of the finned area readily heats up and dissipates engine heat to the cooling airstream. This more than offsets the inherent disadvantage that the fins cannot be made so thin, and so closely spaced, as would be possible with steel.

Nearly all aluminium alloys, however, have the characteristic of relatively high expansion with heat. If the jacket is, at first, a tight fit against the cylinder it will tend on heating up to expand away from the cylinder walls and so leave a definite air gap between—*Fig.* 5.

Air is a very poor conductor of heat and so this gap is effectively a "heat dam" or barrier preventing free transfer of heat from the cylinder to the outer jacket. Thus the cooling effect of the airstream is not readily passed back to the cylinder.

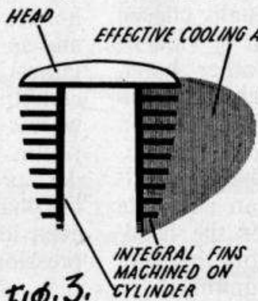


Fig. 3.

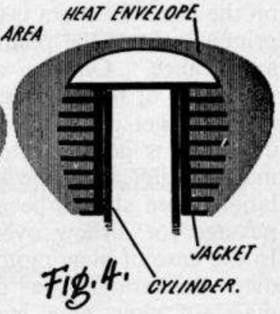


Fig. 4.

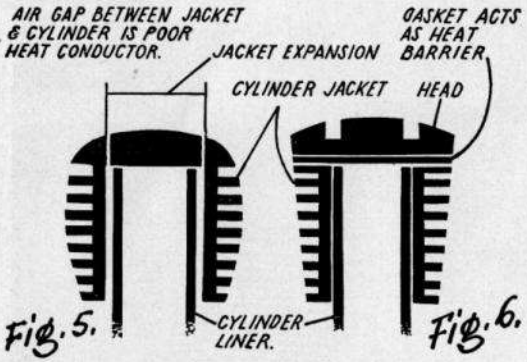


Fig. 5. Fig. 6.

The most satisfactory way to minimise this is to make the cylinder jacket a really snug fit to start with. On such a layout, too, the head may become the most effective cooling area since it receives its heat by direct conduction through the metal-to-metal contact. The problem is not necessarily as serious as would appear, and in fact, most "jacketed" cylinders are quite satisfactory from the point of view of adequate air cooling. They may, however, show signs of overheating when this cooling airflow is restricted, such as when the majority of the slipstream is blocked off.

Head cooling seldom appears to be critical. A plain, hemispherical shape generally provides adequate cooling area and where fins are used on the head these are usually chosen primarily from the point of view of appearance. On the other hand, the heads of modern high speed engines do get extremely hot and on diesels it is generally to be recommended that in a cowled-in installation there should be an adequate passage for airflow over the head. In the case of glow motors the head may deliberately be unfinned to maintain glow plug heat when in the air (*e.g.*, Dooling and Carter) with cowled-in glow engines no cooling flow over the head should be necessary.

A gasket will act as a "heat barrier" where fitted under a detachable head to act as a gas seal—*Fig.* 6. If the actual combustion space is distant from the head, as in a diesel, this would probably make the head much cooler without affecting the running of the engine. On a glow motor where the flame plays directly on the underside of the head the gasket may play an appreciable part in determining the working temperature of the plug. With complete cooling in flight, *i.e.*, a slipstream all over the engine, the head may be too cool if gasketed. If in direct contact with the cylinder (no gasket) excessive cooling would be offset by a transfer of cylinder heat by conduction.

The unequal expansion rates of light alloys and steels mitigates against the use of the former material for contra-pistons, although this is quite common practice on certain Continental engines. The top portion of the cylinder is nearly the hottest point of the whole engine and so there exists in this region the largest expansion differential. As a consequence, as soon as the engine warms up, the light alloy contra-piston virtually seizes in the cylinder, providing an excellent gas seal but making it extremely difficult, or even impossible, to adjust the compression setting from that point on. It is usually possible to increase the compression with the contra-piston seized, but it will not blow back on its own if the compression screw is backed off.

The higher rate of expansion common to light alloys also affects choice of this material for pistons, the top of the piston being the hottest part of the working engine. Where light alloy pistons are employed they are not used to provide a gas seal, so need never reach the condition of being a "seize" fit. The necessary seal is produced by fitting the piston with rings so that the piston itself need only be a relatively slack fit in the cylinder to start with. In such cases, too, it is general to use a low-expansion light alloy (a standard "full-size" piston alloy).

Alloy Pistons

A number of engines have been tried with plain aluminium pistons (and one, the American "Thor", even had an aluminium cylinder to go with it), but no such combination has worked out successfully in practice. There may, however, be possibilities here in using deep anodised aluminium as anodised light alloy surfaces have been used with considerable success for gears in the engineering world. Thus the use of a plain aluminium piston is not entirely ruled out.

The cooling effect of a propeller slipstream under static conditions is somewhat different to that in flight. In the latter case cooling should be much more effective and may even affect engine layout at high speeds—*e.g.*, the plain head on a glow motor, as mentioned previously. Where the engine is completely cowled in it should be satisfactory to provide a flow of air to the depth of the cylinder, and over the head in the case of a diesel, but not necessarily so with a glow motor—*Fig.* 7. The small amount of heating received by the crankcase should normally be nothing to bother about as this will be dissipated by conduction through the rest of the engine. Crankcase cooling can, however, be important on an engine where the bearing is not too good. Often an engine with a main bearing a little on the tight side, or with tight spots, will run much better in the air than "static" because the bearing is receiving continuous cooling in the former case but not in the latter.

The out-in-the-open engine will always receive adequate cooling in flight. The completely cowled-in engine will receive adequate cooling provided there is a good air entry and *exit* to the cooling. A failing on some free flight installations is to provide an air entry into the cowling space and the engine may overheat, although this is unlikely on a short run. The main objection is the high drag of such an installation.

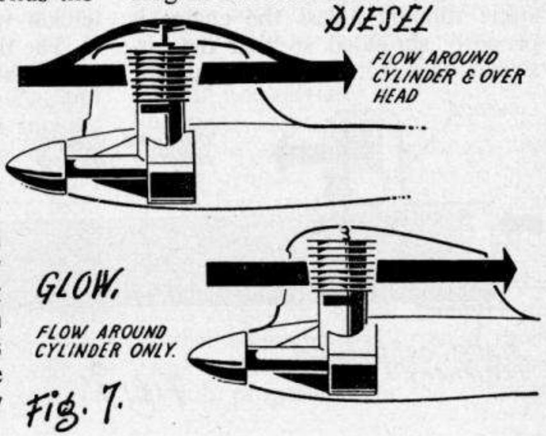


Fig. 7.

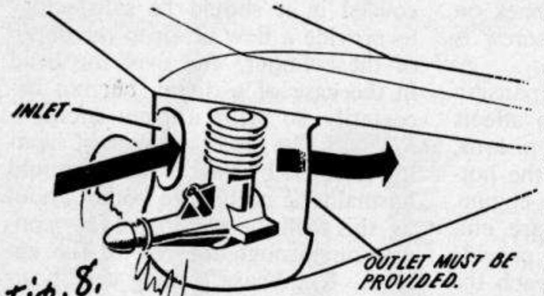


Fig. 8.

Most engines will run satisfactorily at quite high speeds for limited periods without any cooling at all, other than radiation of heat to the lower ambient temperature of the surrounding air. On the *Aeromodeller* dynamometer tests, for example, where the engine is driving a rotor with no generated air blast or slipstream, cooling is provided by a separate centrifugal blower mounted by the engine and blowing a constant stream of air over the engine. Without this, the cylinder of the engine on test will quickly "fry".

Fan blades incorporated on a flywheel are not a suitable means of providing a cooling airstream for static running unless the engine is properly shrouded so that the airstream is directed up and past the cylinder—*Fig.* 9. Otherwise what slipstream was generated by the flywheel periphery would be directed only along the bearing and crankcase. Any type of "blower" driven directly by the engine will, of course, absorb power, which means that unless useful power can be extracted from the engine crankshaft.

For Model Boats

Water cooling does not suffer from this limitation and is the logical choice for prolonged static running with no fan-type loads, or for marine installations. Most air-cooled engines are readily converted to water cooling by replacing the cylinder jacket with a hollow jacket (usually of brass) through which water can be circulated. Circulation can be achieved by thermo-syphon action provided the pipes are of generous diameter (usually at least ¼ in. bore is required for satisfactory circulation), the main reservoir or water holder also acting as a cooling tank and thus constantly feeding the cylinder jacket with cooled water—*Fig.* 10.

The thermo-syphon is particularly adaptable to static running, but marine units usually draw in a supply of water by means of a scoop under the hull and discharge it overboard again after circulation through the jacket. Thus the engine is fed with a constant stream of cool, fresh water without any mechanical pump being involved. Mechanical pumps driven off the engine are, however, employed on some types of marine engines, or where a closed circuit is preferred. The latter must have a cooling tank. The main limit to any water cooled system is that water boils at 212° F. (100° C.) and this is the maximum fluid temperature which can be realised in the cooling system. Flow rates must, therefore, be adjusted accordingly.

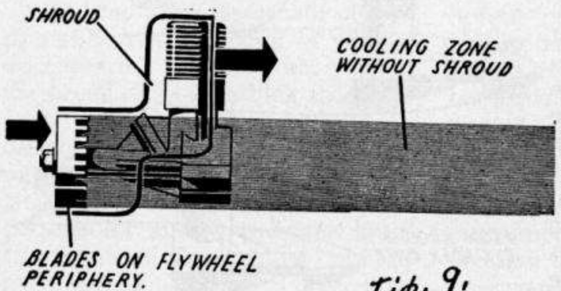


Fig. 9.

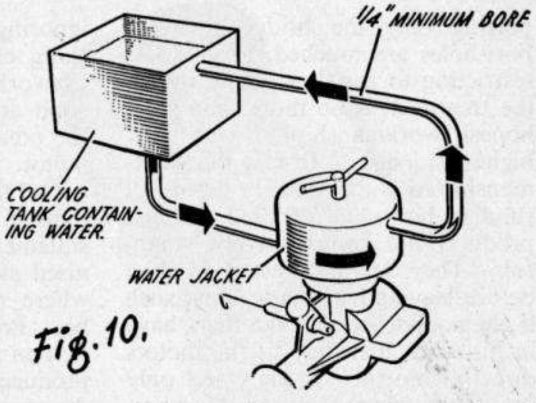


Fig. 10.

CHAPTER SEVEN

Tuning for Speed

IN discussing engine tuning for speed, we can do no better than to study the World Record breaking products of F. E. Carter, a watch repairer by trade and himself at one time a pretty good control line flyer. His speciality, and indeed his sole practical interest in the movement, is in making motors which will go faster than anyone else's. Starting point may be a "recognised" racing motor, like a stock McCoy, Dooling or Eta, but the end product may well include only the crankcase casting of the original unit.

Let us examine the Carter technique in altering a new McCoy 29 for team racing. The McCoy is favoured for this duty, as it has the stroke to deal with the large pitch and blade area (compared with speed toothpicks) needed, and it offers the best speed/range ratio of up to 112 m.p.h. over 24 laps. Its crankcase will not shatter, and it lends itself to the Carter treatment without complaint.

McCoy Modifications

What happens to this new motor is roughly as follows. The alloy induction disc and steel spring shim, the rear ball race, and the head gasket are consigned to the waste bin. A Tufnol disc is made, with the same timing (provides better running fit and obviates worn alloy disc scrapings from going through the engine to spoil other bearings) and a new cylinder head turned to give a leak-free metal-to-metal fit on the cylinder.

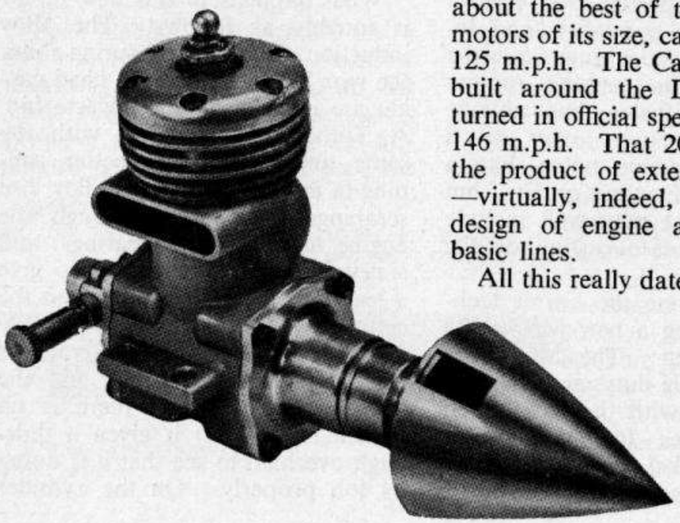
The short-life rear ball race is exchanged for a Hoffman and the front race (for which there is no British equivalent) is given a thorough overhaul to see that it is doing its job properly. On the cylinder porting only the bridges between port-holes are touched, to give less restriction to gas flow. The rest of the treatment is no more than good honest workmanship of the very highest standard. In fact the workmanship is such that in terms of running hours, all of Fred Carter's products are comparatively youthful. They are not tested by him before leaving his charge—and such is the confidence that the fliers have in his work that they fit the motors directly into their models and only use them when required for competition.

Speed performance—and that includes team racing—is ultimately dependent on how much power you can get out of a particular motor. Right from the start, tuning and "hotting up" standard motors has been established practice, the only trouble being that tuning up a motor is a rather intangible subject. As soon as one authority lays down a set of general rules, another comes along with equally good results by ignoring these rules and doing something else. A lot of the so-called "reworking" of an engine does no good at all, and may even produce the potential performance of a stock motor.

On the other hand Fred Carter himself makes the rather sweeping statement that none of the recognised stock racing engines are anywhere near as good as they could be. Presumably this could be re-written as a statement that no mass-produced engine can be built with the precision necessary to get the maximum B.H.P. per c.c. out of it. Further, world class performances in speed have now quite surpassed speeds which can be achieved with stock engines and that a "special" is essential to compete on equal grounds with the state and manufacturer-sponsored teams of other countries.

Put it down in facts and figures and you begin more and more to respect Fred Carter's views. A standard Dooling 29 which is just about the best of the stock racing motors of its size, can achieve about 125 m.p.h. The Carter Special, re-built around the Dooling 29 has turned in official speeds in excess of 146 m.p.h. That 20 m.p.h. gain is the product of extensive reworking—virtually, indeed, a complete re-design of engine along the same basic lines.

All this really dates back to about

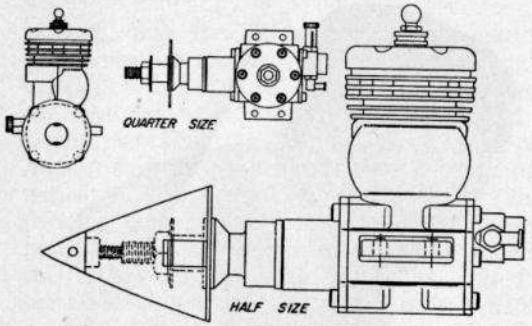


THE FAMOUS Carter Special 5 c.c. based on the Dooling 29 as described in this chapter.

FRED CARTER AT work on a McCoy 29 he is preparing for Team Racing

1950 when Fred Carter was flying control line stunt, was being tempted by speed and had the feeling in the back of his mind that having found out a little about commercial engines and their limitations that he could do a bit better. Perhaps the contrast between a precision watch mechanism and the precision standards adopted for engine manufacture was too much! At any rate, his first serious attempt to rework an engine was on a Nordec 10 c.c. (virtually a copy of the American McCoy 60), which he accomplished with no mean success. At a time when the other speed flyers were still finding the "ton" elusive, Fred's reworked Nordec carried his "Little Rocket" well past the 100 m.p.h. mark and was fairly consistent at about 116 m.p.h. Modifications in this case consisted mainly of a new piston and new head.

From Nordec to McCoys to Etas and Doolings, Fred Carter's fame was soon established, locally, at least. But speed flying itself was in the doldrums and not until the first British Nationals at Cambridge did people begin to sit up and realise that there was something very special about a Carter reworked engine. Davenport turned in a speed of over 160 m.p.h. with a Carter-Dooling which was more "Carter" than "Dooling" and set many a person arguing that the timekeepers had missed counting a lap. But that performance was real enough, as Gibb's record speeds with both 5 c.c. and 2.5 c.c. engines have shown. They are performances which can be duplicated under similar conditions. The real point is that the jump in performance is so startling that it confounds people's previous ideas of control line speed standards. It also makes some of those *apparently* fantastic American speed



THREE VIEWS OF THE CARTER Special illustrates the method of spinner attachment, short carburettor over-hang and "solid" head.

claims quite logical, especially as the Gibbs-Carter combination have bettered some of them!

As far as the suitability of the engine is concerned, there is only one *type* of motor which Fred Carter considers worthwhile for speed. That is the cross-scavenged glow motor with rotary disc induction and ball bearing crankshaft—as exemplified by the McCoy Dooling, and the British Eta. Fred will just not consider diesels in any form, plain piston engines, plain bearing engines or those with circumferential porting. He is interested only in getting maximum peak performance, and that is his starting formula.

Reworking is essentially an application of a "basic" modification plus any little extras which he thinks worth trying out—a sort of calculated guess as to whether or not a small alteration here and there will improve performance or not. And that is something which cannot be put down in words! The reworked engine invariably ends up with the same crankcase, but that may be the only original part. In other cases the original crankshaft and bearings may be used, but inevitably there is a new cylinder liner, piston, rotor and back cover assembly.

The obvious question at this point is, why not start right from scratch and make the crankcase too? The answer here is that the crankcase casting is usually quite intricate on a motor of this type and to have this unit as a starting point saves a lot of time and trouble. That is the only reason; and we should not be too surprised if one day Fred does start from "basic stock".

The Carter "5"

The particular motor we can review in detail is the 5 c.c. reworked Dooling which retained the standard Dooling crankcase, crankshaft connecting rod and front ball race (this only being because of a size difficult to duplicate in Britain). Everything else is new, although externally the only noticeable difference with the finished engine is in the head with centrally-located plug, plus the extremely high finish on all the new parts. Internally however, it is quite a different story.

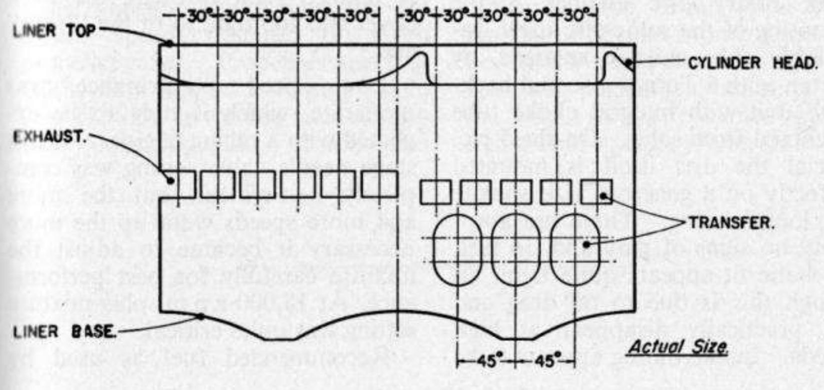
The original shrunk-in liner was removed and replaced with one made from Meehanite bar stock carried down to the full depth of the casting. Ports were filed by hand, duplicating the layout of the original but with a slightly later exhaust opening and slightly later transfer. (Shallower ports providing *less* area than on the original Dooling.)

The piston is a new casting in standard piston alloy, cut with ports in the wall in Dooling fashion but with a curved skirt the full depth of the liner each side. Some 120

degrees of the piston on the exhaust side is generously relieved and a single ring is used just below the crown. The deflector is straight but sharply peaked, the head machined away to a matching shape. Compression ratio, by rough estimate, is about 8 : 1.

The top of the piston and the inside of the head are highly polished, this being the only internal polishing done. Here, in fact, is one of those contradictions in engine tuning. Nearly everyone who has written anything about the subject emphasises the importance of polishing and careful shaping of the ports to minimise gas friction with fillett radii on "square" corners, etc. Carter does not consider this necessary, or worthwhile. About the only concession in this direction is filing down the vertical columns across the exhaust port to minimum size (about $\frac{3}{64}$ in. wide). As to the importance of plug location, Carter holds here that if it gives the expected performance with the plug in the middle, then leave it there.

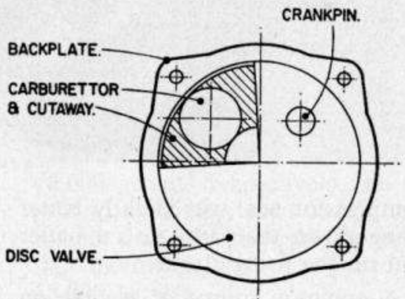
"UNWRAPPED" CYLINDER FOR CARTER Special shows how the Dooling ports are actually reduced in area to a degree, while transfer is opened up. This is a characteristic of most Carter engines.

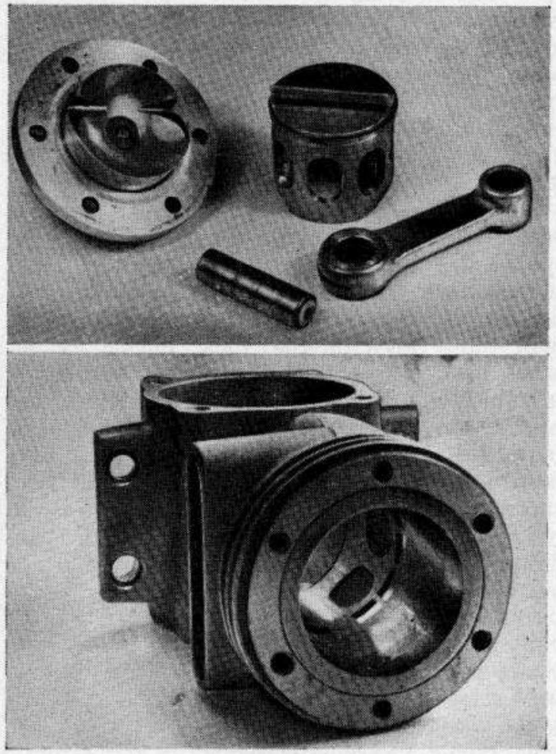


If performance does not come up to scratch, a new head and piston would be an obvious "second try" We get the impression, however, that with Fred Carter, second tries are a bit of a rarity.

"One-way" Fits

Undoubtedly particular care is taken over the piston-liner fit to get optimum compression seal and piston support with minimum friction. It makes a noticeable difference on the 5 c.c. engine for instance (and the same would apply to similar types reworked in the same way) if the cylinder unit is rotated through 180 degrees to bring the exhaust to the left hand side, *i.e.*, transposing the side thrust to the other side of the piston. Another interesting fact was that the





INTERNAL DETAIL OF THE Carter Special helps to impart some idea of the magnificent internal finish of these engines which are produced with all the skill of an experienced watchmaker and which have achieved considerable international success in the speed field and dominated team race Class B in Great Britain.

compression seal was slightly better going down than up, *i.e.*, a better seal on the firing stroke.

A common source of trouble on stock rotary disc engines is the mounting of the rotor disc itself. Invariably this unit is re-made by Carter, with a Tufnol disc and backplate unit with integral choke tube machined from solid. On the 5 c.c. special the disc itself is mounted perfectly on a generous size spindle and long bearing. There are absolutely no signs of play and, in fact, the static fit appears quite tight, although this is due to oil drag and will practically disappear at high speeds. Intake timing appears to be reduced at both ends, compared with a standard Dooling, this being 180° induction period.

The front end of the engine is quite conventional, beautifully made and with perfect shaft support. The only modification to the crankshaft is a reduction in diameter of the end of the chrome plated pin which engages in the rotor disc. The front bearing remains the standard ball race by Doolings but the rear ball race is of British origin. The unit, like the backplate, is assembled without gaskets. It is perhaps an indication of the accuracy of fitting that although the standard Dooling crankcase unit is reamed right through, the back cover was too tight to fit the "front" end.

Low speed performance was moderate, which is only to be expected with a racing engine. At this stage needle valve setting was completely non-critical, but the more and more speeds went up the more necessary it became to adjust the mixture carefully for best performance. At 18,000 r.p.m. plus mixture setting was quite critical.

Recommended fuel as used by

AEROMODELLER ENGINE ANALYSIS DATA

Specification

Bore : .792 in.

Stroke : .594 in.

Displacement : 4.8 c.c. (.293 cu. in.).

Bore/Stroke ratio :

Weight : 7 ounces.

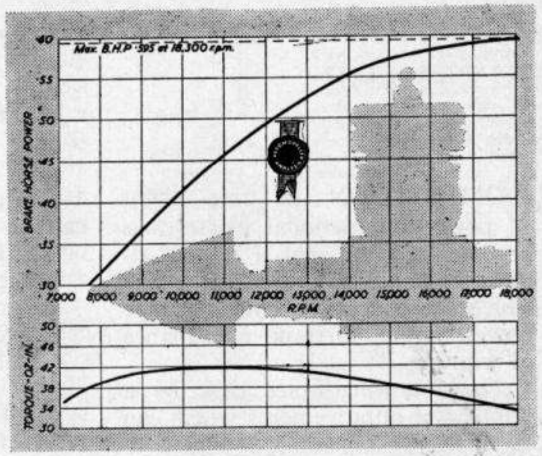
Max. B.H.P. : .595 at 18,000 r.p.m.

Max torque : 42 ounce-inches at 11,000 r.p.m.

Power rating : .125 B.H.P. per c.c.

Power/weight ratio : .086 B.H.P. per ounce.

Availability : Special racing engine by F. Carter. Not available commercially.



PROPELLER R.P.M. FIGURES

Propeller	r.p.m.
dia. pitch	
11 x 8 (Whirlwind)	6,300
9 x 4 (Stant)	13,600
9 x 8 (Stant)	10,600
9 x 10 (Stant TR)	10,500
8 x 4 (Stant)	17,800
8 x 8 (Stant)	13,800
8 x 9 (Stant)	12,500
7 x 10 (Reworked Stant)	15,000

Fuel: Mercury No. 7 plus added 25 per cent Nitromethane.

Gibbs is equal parts of nitromethane and methanol, plus about 20 per cent Castrol "R". The mixture used on the test runs was obtained by adding 25 per cent nitromethane to standard Mercury No. 7 to arrive at a similar nitro content. No increase in performance was apparent with more nitromethane.

The torque is well sustained and the engine, as tested, peaked at just over 18,000 r.p.m. at an equivalent B.H.P. of almost exactly 0.6. The actual power figure is a little lower than that which could be achieved in the air, although the peak r.p.m. point appears about right for the official speeds obtained on given propeller sizes. Propellers used by Gibbs were Stant 7 x 10 and Truflo 7 x 11 with re-worked blades, giving something like 15,000 and 14,000 r.p.m. respectively, on the ground. Gibbs and Carter estimated that actual flight r.p.m. of the motor at record speed was 18,000-18,500.

Summarising we can only say that the Carter Special is one of the most beautifully made motors it has ever been our pleasure to examine. For the rest, its performance on the flying field speaks for itself.

CHAPTER EIGHT

Tuning a diesel for Free Flight

COMPETITION in free flight power international classes has become so keen, that it is now almost essential to have an engine which produces above average power output, to stand any chance of winning.

The following notes describe a number of improvements which can be made to a standard engine to improve performance using a minimum of tools. The description applies in particular to the E.D. 2.46 c.c. in which a particularly high standard of workmanship is maintained, and which Peter Buskell has used to such good purpose in many British Power Teams.

Notes on other types are included where relevant.

The points at which improvement can be expected are as follows: —

(1) To increase the charge induced into the crankcase by removing obstructions in the induction system.

(2) To improve burning by mixing fuel and air more fully.

(3) Reduce obstructions in transfer system.

(4) Reduce wear and friction by attention to bearing alignment and lubrication.

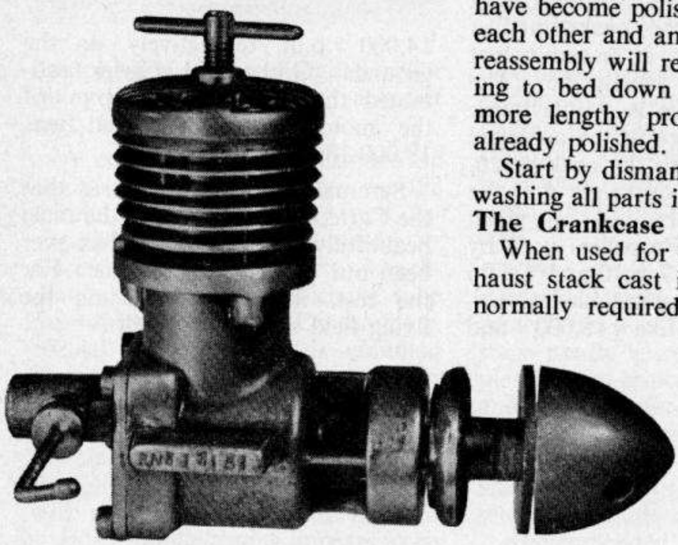
(5) To decrease vibration by improving the balance of reciprocating parts.

For optimum results, "tuning" should be carried out before the motor is run at all. Should running-in be completed, the bearings will have become polished and mated to each other and any dismantling and reassembly will result in parts having to bed down again. This is a more lengthy process as they are already polished.

Start by dismantling the unit and washing all parts in petrol.

The Crankcase

When used for free flight the exhaust stack cast into a 246 is not normally required, as the cylinder

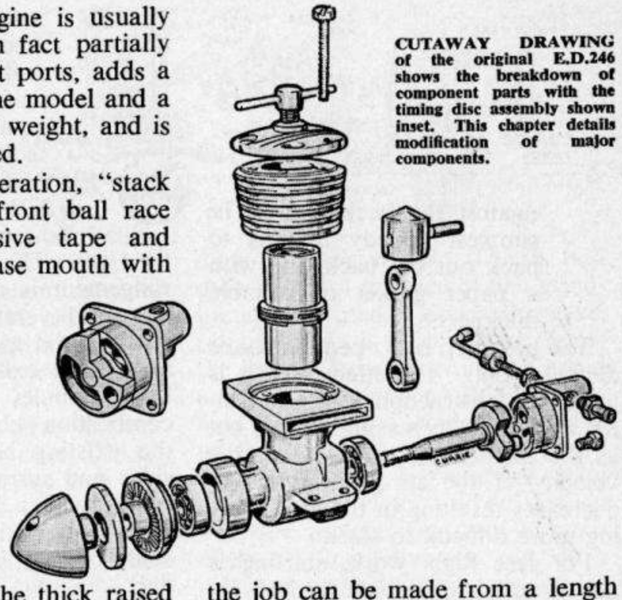


THE E.D.246 AS modified by P. Buskell. The only external difference is complete lack of exhaust stack—compare with drawing opposite.

if not the whole engine is usually exposed. It does in fact partially block off the exhaust ports, adds a fraction of drag to the model and a fraction of unwanted weight, and is therefore best removed.

Before starting operation, "stack removal", seal the front ball race housing with adhesive tape and block up the crankcase mouth with clean rag rammed in tight to obviate the entry of metal dust and filings.

Clamp the crankcase in a vice under the bearers, using soft vice clamps and not too much pressure, the stack can then be sawn off carefully, using the thick raised portion into which the cylinder holding-down bolts screw, as a guide line. File off to this line and finally emery-cloth to a smooth finish. Clean off all metal dust and remove protective cloth and tape.



CUTAWAY DRAWING of the original E.D.246 shows the breakdown of component parts with the timing disc assembly shown inset. This chapter details modification of major components.

The Back Plate

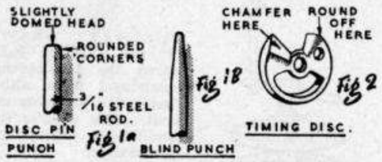
Begin by checking the end play on the timing disc. On a new component the ideal figure is about 1 to 1$\frac{1}{2}$ thou. inch, *i.e.*, so that the disc appears to rub on the back plate slightly; this will wear off with running-in to a free close fit. A maximum of about 2 thou. is permissible in a new component after which serious loss of crankcase compression and poor starting will result. To remedy, open the vice jaws just sufficient to pass the disc pin, carefully position the backplate and tap the pin through gently, checking frequently until the desired fit is obtained. A suitable punch for the job can be made from a length of $\frac{3}{16}$ in. rod, the face being filled to a slight dome—*Fig.* 1*a.*

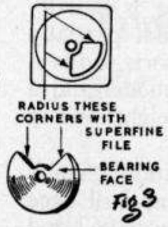
Afterwards mount the punch upright in the vice and get a friend to support the backplate with the disc pin head resting on the punch; then, using a blind punch—*Fig.* 1*b*—spread the end of the pin so that it cannot shift in future.

Next fit the backplate to the crankcase and check that the disc does not bind. There are two possible faults: —

(1) the register hole in the disc does not line up with the crankshaft so causing the disc to be pulled against the pin. The remedy is to scrape out the hole slightly, a suitable tool being a small screwdriver filed and stoned on one edge.

(2) the crankshaft is too long or the hole in the disc not deep enough, so pushing the disc





against the backplate. The simplest remedy here is to pack out the backplate with a paper gasket of suitable thickness.

The next step is to open the choke diameter out. The effect of this is to reduce obstruction to the incoming charge so increasing power, but as the diameter is increased so the velocity of the air passing the jet decreases resulting in the motor being more difficult to start.

For free flight work, starting is not critical and the standard $\frac{3}{16}$ in. choke can be opened to $\frac{1}{4}$ in. in safety. Mount the backplate in the vice using soft clamps and drill through from the disc side using light pressure and high revs.

The choke can now be faired into the timing cutaway on the backplate face. A power drill and rotary files or burrs ease this work considerably but it can be done just as well with a 4 in. coarse cut rat tail or needle file. The intake should also be filed out to trumpet shape and the whole finally smoothed with emery. Chamfer off the square edges of disc timing cutaway on the outside—*Fig.* 2, and radius the corners of the cutaway on both disc and backplate on the working face with a superfine file—*Fig.* 3. Finally clean thoroughly with petrol until all traces of metal dust are removed.

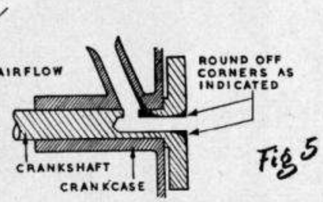
The next item for attention is the spraybar, the sides of which are waisted by filing so as to reduce obstruction in the choke to a minimum. Do not carry this too far otherwise there is a danger of it breaking in two when the fixing nut is tightened.

Another worthwhile modification is to the jet. The standard E.D. arrangement is a $\frac{1}{16}$ in. hole facing toward the crankcase, this does not give a good mixture and gives rise to a hard exhaust note indicating large globules of fuel reaching the combustion chamber. Solder up the existing hole by scraping the sides and surround, then insert the needle with a blob of grease on it, and apply a dab of acid flux and solder with a *hot* iron in one quick dab.

The new jet arrangement consists of two rows of holes at the "guesstimated" point at which airflow breaks away from the spraybar—*Fig.* 4—about 60° apart around the section.

These are best drilled on the opposite side to the original hole. Clamp the bar lightly in the vice and scribe two lines $\frac{1}{16}$ in. apart along it. You can now drill either four holes with a 68 drill or 5 with a 75, depending on your patience (!) along each line. A pin vice is a necessity and the drill should be chucked so that only about $\frac{1}{16}$ in. is exposed. It may be necessary to waist the needle to allow fuel to reach all holes. Check this before assembly.

On shaft valve type motors any work on the induction system is severely limited by consideration of crankshaft strength. Generally it is safe to open the port in the shaft



and choke up to square section and round off any shaft bends or corners—*Fig.* 5. Also check that the choke tube cross-section minus the spraybar cross-section is not less than that of the shaft. If it is, open out the choke tube if sufficient wall thickness is available. Check the timing by mounting a degree marked disc to the crankshaft (the piston can be refitted to find T.D.C.). A suitable timing for free flight is: —

Inlet opens 50 deg. after B.D.C.
Inlet closes 50 deg. after T.D.C.

If the open period is less than this it is usually possible to file out the crankcase port by the required amount. Finally radius all port edges with an oil stone slip.

Cylinder

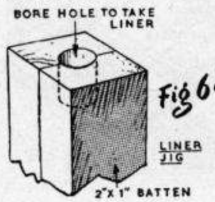
The transfer system of the E.D. 246 consists of a gap left between the outside wall of the cylinder and the inside crankcase wall. Work on the cylinder is directed towards enlarging this passage and smoothing the entry and exit.

Since the same method of production is employed on many other engines, Davies Charlton, Allen-Mercury, Zeiss, etc., the following is a general improvement for a wide variety of engines.

First a means of holding the cylinder firmly in the vice is required. Two lengths of 2 in. by 1 in. batten are required. Place these together in the vice with the edges flush and bore a hole of cylinder diameter into the ends centred on the joint—*Fig. 6a* and *Fig. 6b* show the work to be done on the cylinder. This is best done with a 4 in. bastard file, grinding can be used if done very carefully but has a tendency to distort the liner. Take your time to avoid any chance of distortion. The bottom end of the liner may be found to be hardened, this is of small depth and can be removed with coarse emery cloth. When dealing with the transfer port, cut a semicircle of liner diameter out of a piece of tin plate and use this to protect the cylinder seating face.

Work on the Piston

This requires the use of a lathe and $\frac{1}{4}$ in. power drill or a flexible shaft set. If you cannot get the use of these items do not attempt the work. The idea is to remove as much metal from the inside of the piston as possible, so decreasing mechanical losses at high r.p.m. and improving the balance of the engine. The piston is the heaviest reciprocating component of the engine and the more we can remove from its interior, the greater our chance of a more powerful engine.



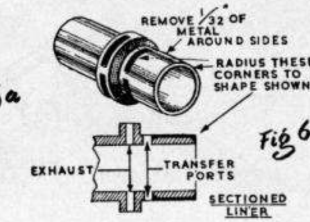
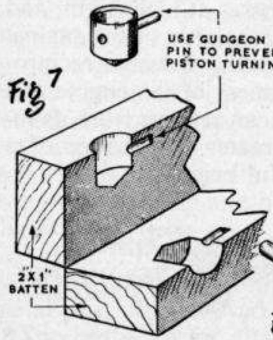


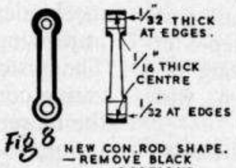
PHOTO SHOWS THE METHOD OF RELIEVING the rotor load as detailed on page 53. The back-plate is positioned so that the rotor pin is mounted over a punch and is now being rivetted securely.

In the motor cycle engine world an approximate balance factor of 60 per cent of the reciprocating weight is used and it is worth noting that using a cast iron piston and normal manufacturers' techniques it is not possible to achieve this factor in a model diesel engine. If the flexible shaft set only is available the work of lightening is best done by grinding.

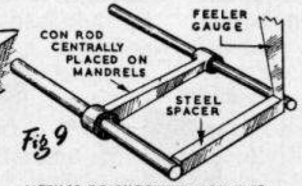
Make up the jig as shown in *Fig.* 7 for holding the piston, and work slowly and carefully. The piston walls can be tapered off to about $\frac{1}{32}$ in. thick at their lower edge and an appreciable amount of metal can be removed from the crown and around the gudgeon pin bosses. If a lathe is available for re-lapping the piston, advantage can be taken of lightening operations to improve the piston fit. The materials used in the 246 liner piston set are such that the liner expands more than the piston when hot so causing a loss of power. If the lightening is done with rotary files and a fair amount of pressure the piston walls are expanded, particularly on the working faces. The piston is then lapped to be a tight push fit in the top portion of the barrel. The writer has found it preferable to use a coarse paste for initial lapping rather than a fine one all through. This leaves scratches of slight depth on the piston surface which retain oil and hence decrease wear and friction. Radius the top and bottom edges of the piston with an oil stone slip.



PISTON JIG (OPENED UP)



NEW CON.ROD SHAPE. —REMOVE BLACK PORTIONS.



METHOD OF CHECKING CON ROD BEARING ALIGNMENT

Con-rod

Considerable lightening can be carried out on this component—*Fig.* 8 gives details. When finished, drill oil holes through big and little ends (No. 68 drill) and file a narrower groove across the bearing surface where the hole

meets it. A fine half round file does the job nicely.

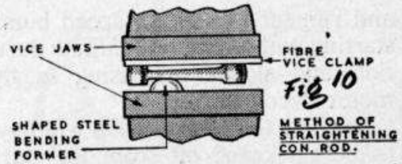
The alignment of big and little ends should also be checked. Purchase two lengths of ground silver steel rod (obtainable from most tool shops) to the big and little end diameters; these should be a close fit. *Fig.* 9 shows how to check using a length of steel bar and feeler gauges. If slight errors are present they may be corrected by bending—*Fig.* 10—but if they are more than about 2 degrees out of line or if the bearings are slack a new rod should be purchased or if a lathe is available, made.

Mark off the centres on dural plate of the required thickness and mount on the face plate, drill slightly undersize and finish with a reamer or "D" bit.

The rod can then be sawn out and filed to shape by hand. A square or oblong cross-section rod is, of course, preferable to a round one.

Assembly and Running

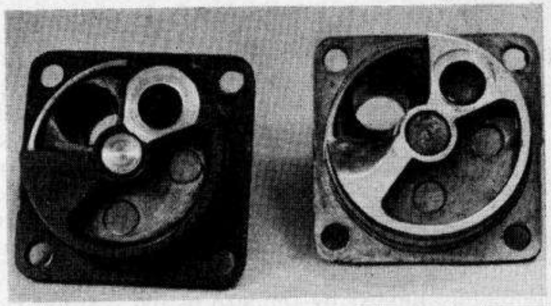
Clean off components in petrol and assemble the disc and big and little end bearings with graphite. Smear cylinder and piston with light machine oil.

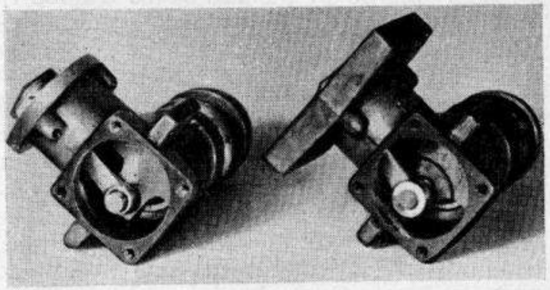


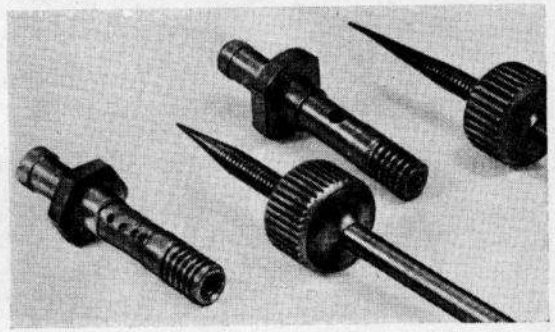
METHOD OF STRAIGHTENING CON. ROD.

For initial runs keep the speed down and the mixture rich. A good running in fuel is: 35 per cent Castrol M 35 per cent Ether, 30 per cent Derv. (Road Vehicle Diesel Oil)

When the initial stiffness has worn off, change to smaller props







BEFORE AND AFTER PHOTOS show the Buskell modifications. Top, to the rotor disc with modified version at right, centre, the crankcase, con. rod and exhaust stack, etc., revision at left, and spraybar and needle valve assembly at bottom.

and run for short high speed bursts starting with about one minute duration and slowly increasing as the motor becomes free.

Initially the compression will have to be slackened off from the starting position as the motor warms up but this will become less and less necessary as the motor runs in. A fully run-in motor with the correct piston fit *should* start and run on the same setting.

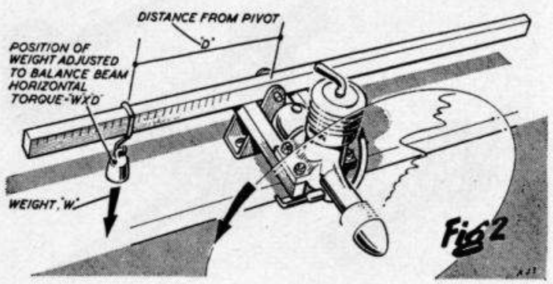
CHAPTER NINE

Horsepower and Torque

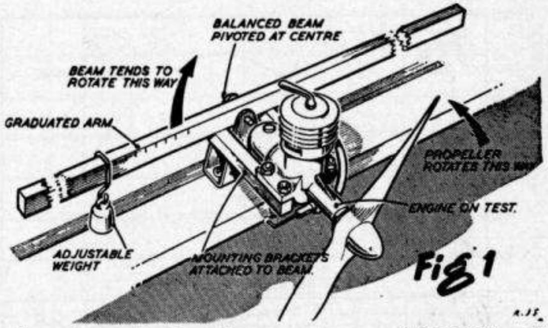
ALTHOUGH the basic results are the same, *i.e.*, determination of brake horse power, the technique of testing model engines is considerably different from that of most "full size" internal combustion engines. The chief difference lies in the fact that a model engine has no throttle, as such, and running speed is mainly limited by the size of the load being driven. Fitted with a given load (*e.g.*, a certain size of propeller), and the controls set for most efficient running, there is only one speed at which that propeller (load) will be driven. Any variation in speed obtained by further manipulation of the controls can only result in less efficient operation and loss of speed. In other words, provided the design is sufficiently flexible, the engine can be slowed by making it run less efficiently, such as by making the mixture excessively rich (opening the needle valve), but such a control is never as positive as the normal throttle control on a larger internal combustion engine.

The only positive control over speed is to vary the size of the load using a smaller propeller to increase speed and a larger one to decrease speed. The size of the propeller then limits the power which that particular engine can develop. To make this clear, we must study the relationship between horse power and speed in more detail.

Horse power is a measure of the work done by an engine. It is a derived function. That is to say, it cannot be measured directly. The two factors we can measure are the speed or revolutions per minute of the engine crankshaft, with a tachometer or stroboscope; and the torque or turning moment imparted to the crankshaft.



Work, basically, is the product of a force and the distance through which the point of application of the force moves in the direction of the force. In the case of a simple torque testing rig—*Fig.* 1—the counterweight on the torque arm is subjected to gravity, but the gravitational effect is overcome by the torque imparted by the motor. The "force" is thus just the value of the weight W. The corresponding "distance" is 2 π D times the rate of revolutions of the crankshaft.



To reduce this to practical figures:

$$\text{Horse power} = \frac{2\pi DNW}{33{,}000},$$

where N is revolutions per minute. Now "DW" is the actual torque—balancing weight multiplied by distance from the axis. In working units, expressed the other way round as "WD", this is the measured torque in ounce-inches—*Fig.* 2. To correct the basic formula for these units:—

$$\text{Horse power} = \frac{2\pi NQ}{33{,}000 \times 12 \times 16}$$

where Q = WD

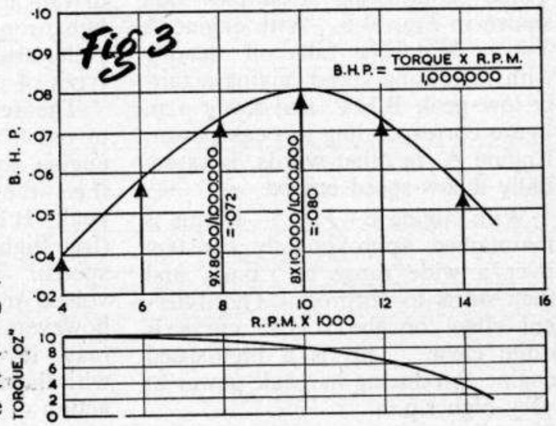
If this equation is simplified, we find that:—

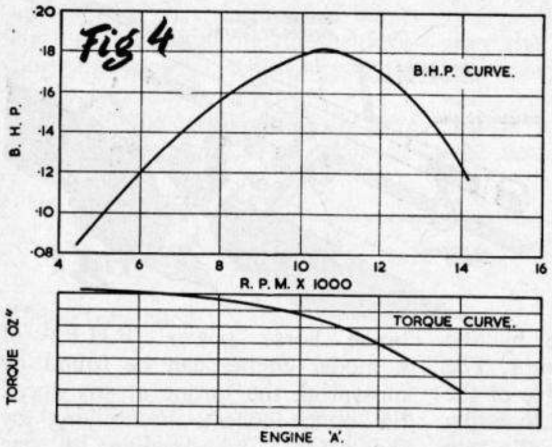
$$\text{Horse power} = \frac{NQ}{1{,}008{,}000}$$

In other words, with sufficient accuracy for most practical purposes, the brake horse power (B.H.P.) of a model engine can be found by measuring the torque at any speed (in ounce-inches), multiplying by the r.p.m. and dividing by one million.

i.e., $\text{Torque} \times \text{r.p.m.} = \frac{\text{B.H.P.}}{1{,}000{,}000}$.

The manner in which the brake horse power curve of an engine on test is built up should now be clear. The tests are concerned with fitting the engine with different loads, in turn, and measuring the corresponding r.p.m. and torque figures. The product of these related readings then lie on a more or less smooth curve—the B.H.P. curve—which is also plotted against r.p.m.—*Fig.* 3.





Now one of the facts which the average person finds most difficult to understand is that the torque tends to *decrease* with increasing r.p.m. This is because he confuses torque with horse power. Although the engine may produce more power at higher speeds, its capacity to turn a load at these higher speeds is reduced, hence calling for lighter loads, *i.e.*, smaller propellers.

The manner in which torque drops off with increasing r.p.m. varies with individual engines. Three contrasting examples are shown in *Figs*. 4-6. With engine A—*Fig*. 4—torque falls off steadily with increasing speed, giving a fairly low peak B.H.P. and low r.p.m. figure corresponding to peak power. Engine A, in other words, is essentially a low-speed engine.

With engine B—*Fig*. 5—torque is maintained approximately constant over a wide range of r.p.m., and then starts to drop off. The different effect on the B.H.P. curve is quite clear. This is a high-speed engine, producing its peak power at very high r.p.m.

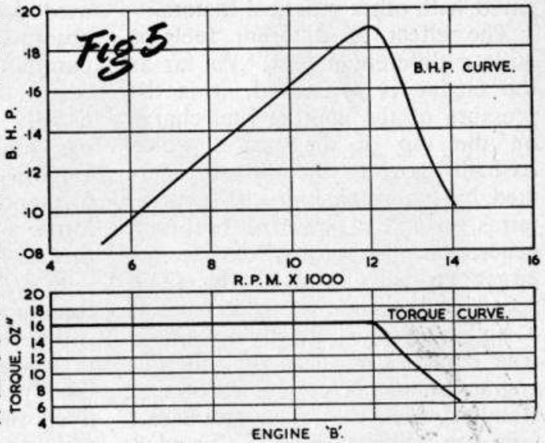
Engine C—*Fig*. 6—is relatively inefficient at low speeds and, again, essentially a high-speed type, peaking at a high r.p.m. value.

Quite a number of practical curves have "kinks" in them, *i.e.*, plotted torque readings do not lie on a smooth curve. This is largely due to varying engine efficiencies at different r.p.m. It may also be due to vibration setting in and the effect of other practical test conditions.

The "peak" previously mentioned refers to the B.H.P. curve—corresponding to the region where the B.H.P. output no longer increases with increasing r.p.m., but now starts to fall off again. The top of the curve represents the peak or maximum B.H.P. which that engine can develop. Operated at that speed it is developing all the work that it possibly can, on the fuel used. It will be appreciated, however, that the load capable of being driven at that r.p.m. may be quite light, representing too small a propeller size for practical use in many types of models.

Theoretically it may be advisable to use a propeller size enabling the engine to operate at peak r.p.m (*i.e.*, the r.p.m. corresponding to peak B.H.P.) for contest models (free flight duration and control line speed). The small propeller size involved may make starting difficult, however, and extremely high r.p.m. may involve trimming difficulties with the model due to the gyroscopic action of the propeller. There may

also be definite demands on the propeller characteristics desired, setting other limits.

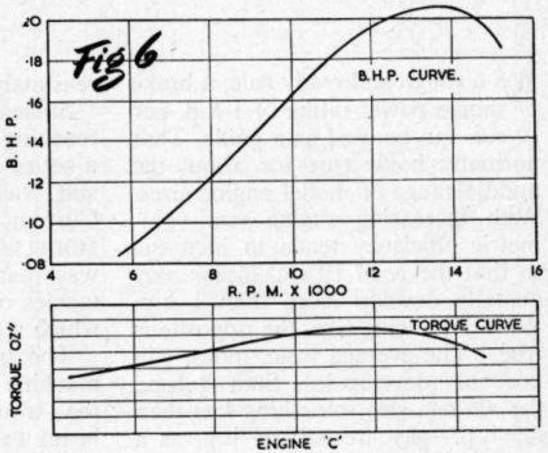


Important Points to Consider

Another point to consider is that the test figures represent static running conditions. Under flight conditions, propeller r.p.m. will tend to increase due to a change in the propeller load. Propeller load will actually decrease when it is both rotating and travelling forwards. equivalent, very roughly, to an increase in r.p.m. of about 10 per cent. To operate an engine at peak r.p.m. under flight conditions, therefore, the propeller load would be selected to give a static figure about 10 per cent below peak r.p.m., as determined for that particular engine.

Another danger with engine test data is that results from one particular specimen on test cannot be guaranteed as representative of all other engines of the same type. Even with the close tolerances held on production engines, there must be manufacturing differences and, with the small sizes involved, such differences between individual engines can be most marked. The smaller the engine size, the greater, proportionally, such variations are likely to be.

In addition there are numerous other possible causes of variation. Atmospheric conditions have a bearing on the efficiency of the fuel charge so that, strictly speaking, test data should be related to standard atmospheric conditions (*e.g.*, relative humidity, temperature and pressure). It is not considered worthwhile to introduce these complications in presenting test data, however, since their proportionate effect is not particularly great com-



pared with other practical factors.

The effect of different fuels is quite a different matter. As far as the engine is concerned, it is the pressure of the igniting fuel charge on the top of the piston which basically governs the torque generated by the crankshaft. In general terms, a more powerful fuel will generate more torque, to drive a larger propeller load at the same speed, or the same propeller load at a higher speed. Strictly speaking, true comparative tests on different engines should be conducted on one standard fuel, and AEROMODELLER tests use Mercury Nos. 5, 7 and 8 fuels exclusively for this purpose.

This is a reasonable compromise, since it does approximate to the operating conditions which the average user of the engine will follow. Comparatively few of the thousands of engines produced anually are used for specialised duties where absolute peak performance is of paramount performance and in such cases the engines concerned are usually subjected to considerable individual attention by their owners. Engine test data applicable to performance on standard fuel then still forms a useful background for further experimentation.

Probably the most interesting conclusion to be drawn at this stage is that the majority of model engines in use are operated at well below their peak power, *i.e.*, at comparatively moderate r.p.m., even in contest models. Peak r.p.m. has increased steadily with production engines over the past decade with the result that few peak at under 10,000. Glow motors tend to peak at somewhat higher r.p.m. than diesels, but both have outstripped the spark-ignition motor.

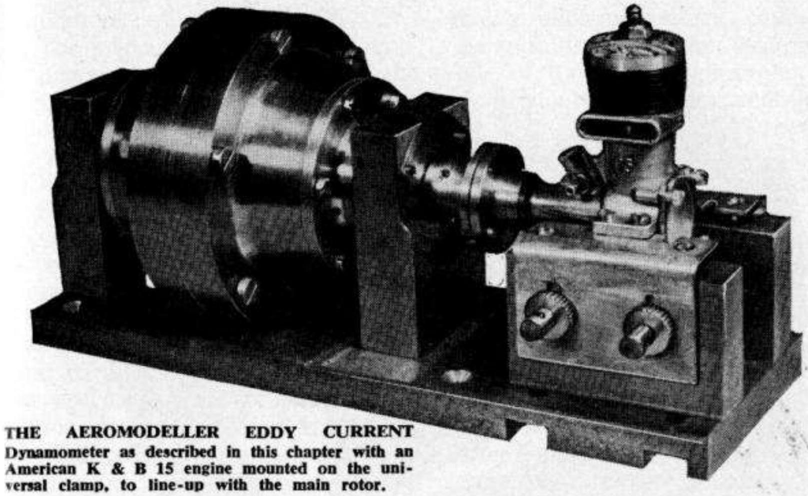
CHAPTER TEN

Test Apparatus

AS a rough and ready rule, a brake horse power rating of 1 h.p. per 10 c.c. can be used as a guide. That normally holds true for about the middle range of model engine sizes. With increasing engine size volumetric efficiency tends to increase, so that the *good* 10 c.c. motor *may* actually develop more than 1 h.p. With decreasing size, the opposite is true. The average 1 c.c. motor will probably develop less than .1 h.p., the .05 c.c. size something less than .05 h.p.—say around .04 h.p. as a reasonably good figure—see *Fig.* 1.

Some difficulty was experienced in reaching satisfactory conclusions on a series of tests and the eventual result was production of the Eddy-Current Dynamometer by AEROMODELLER magazine. The problem was that of replacing the inconsistancies of the torque reaction beam which was suspect.

The high mass or inertia of the machine was soon proved not to be the trouble. The original torque beam was built at a time when large

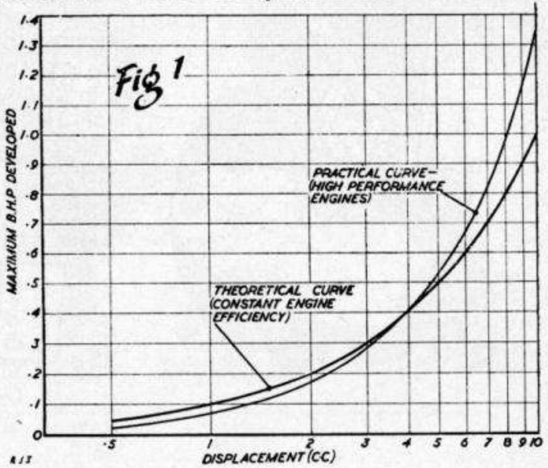


THE AEROMODELLER EDDY CURRENT Dynamometer as described in this chapter with an American K & B 15 engine mounted on the universal clamp, to line-up with the main rotor.

capacity engines were current. A baby half c.c. motor was virtually lost on it. A special lightweight beam built for these small motors produced no more flattering results.

About the only thing left to suspect was that the slipstream effect of the propellers used on the engines might appreciably modify the torque readings obtained. After the first two or three test efforts were, in fact, made, to shield the apparatus from the slipstream, with some noticeable improvement. Since such shields or baffles had to be mounted as close behind the propeller disc as possible, and certainly in front of the cylinder, such an expedient could only be regarded as a temporary one.

At about this time the opportunity arose to discuss the whole problem with Messrs. Heenan and Froude, undoubtedly world leaders in the design and manufacture of large dynamometers for brake testing all kinds of "full size" engines. The question of measuring fractional horsepowers was well outside their normal sphere but their comments on the types of dynamometers and their various limitations were most enlightening. Some typical forms of dynamometers and their



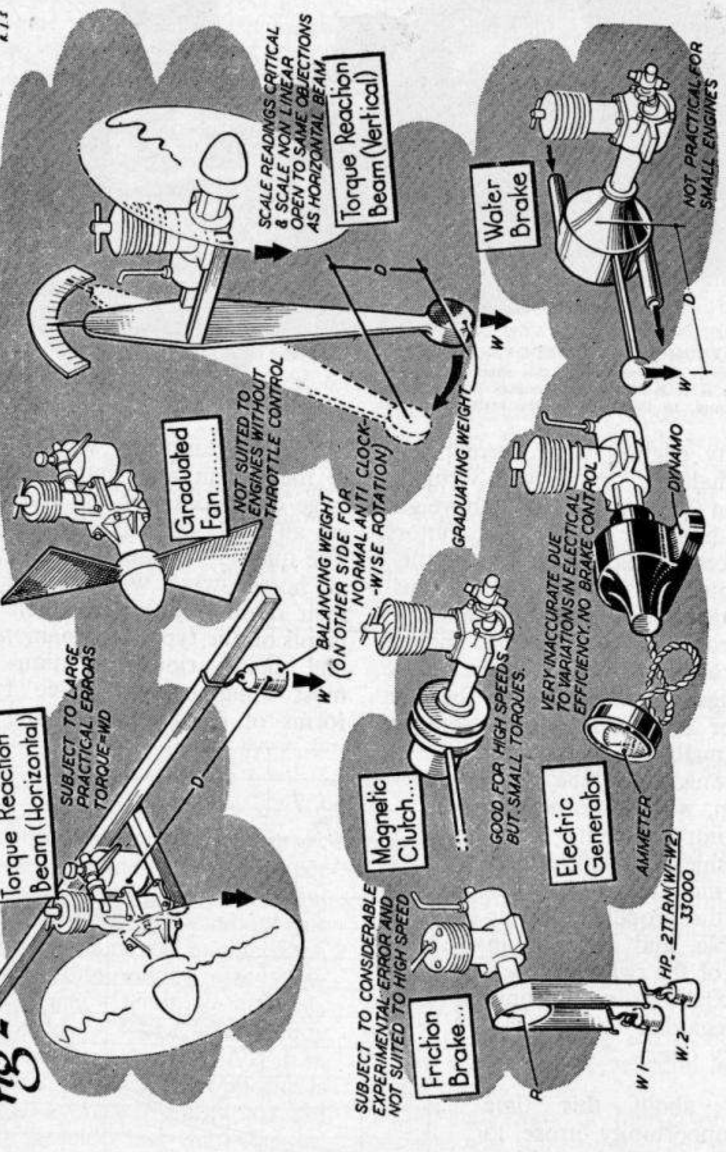


Fig 2

limitations are shown in *Fig.* 2.

The principle of the torque reaction beam, which is the basis of the original AEROMODELLER apparatus and probably the majority of other "power testing" machines, was described in the previous chapter. The engine is centrally mounted on a freely pivoted counter-balanced arm or beam. When the engine is running, driving a propeller, fan, airbrake or flywheel, etc., it is applying a torque at the shaft to drive that particular load at that particular speed. By the principle of reaction an equal and opposite torque is produced on the fixed part of the engine (*i.e.*, the crankcase-cylinder assembly), which will tend to rotate in the opposite direction. The crankcase being fixed to the beam, the beam tends to rotate in the opposite direction to the shaft.

The amount of reaction torque can now be measured, quite simply, by sliding a weight along the "high" side of the beam until it balances exactly horizontal. Reaction torque at this balance, which is equal to shaft torque, is then the product of the weight required to balance the beam, times its distance from the pivot point. The relationship between B.H.P. and torque is explained diagrammatically in *Fig.* 3.

Large Margin of Error

Now torque reaction apparatus of this kind was used in the 1920's for testing full size aircraft engines. It was fairly soon abandoned in favour of alternative means since the errors achieved were of the order of 30 to 50 per cent! Mainly this was due to slipstream effect, caused by the spiralling slipstream striking the side of the fixed engine, etc., and modifying the turning moment tending to tilt the beam. Slipstream effect tends to give a false *low* reading of torque, irrespective of the direction of rotation of the engine.

Other error-contributing factors are the jet effect of side-facing exhaust ports or exhaust stacks, the unbalance effect of an asymmetric engine immediately the beam is rotated away from the horizontal, resistance of any leads (*e.g.*, to plug) and so on. The smaller the force being weighed the greater the cumulative effect of such errors could be.

Now slipstream effect could be eliminated by using a load which does not produce a stream of air flowing past the engine, such as a flywheel. The whole principle of brake horse power determination, however, depends on measuring the torque produced over a range of speeds, from which a curve can be plotted either of torque against r.p.m. or, more usually, brake horse power against r.p.m. The former curve tends to decrease with r.p.m

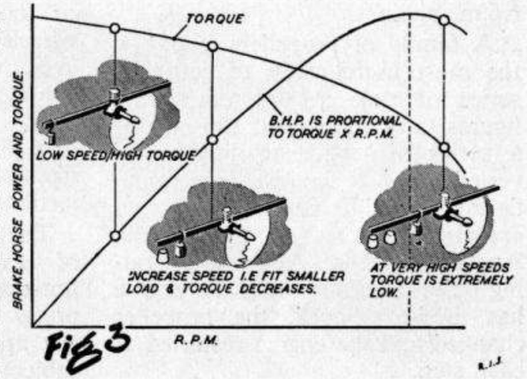
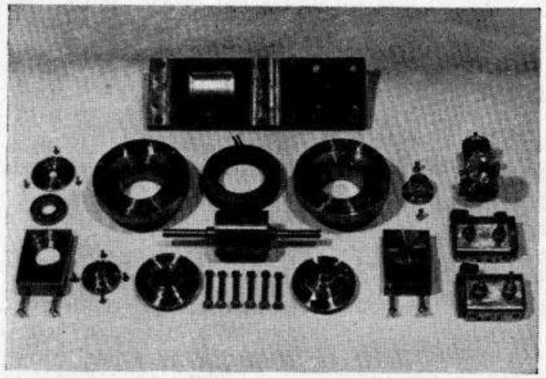
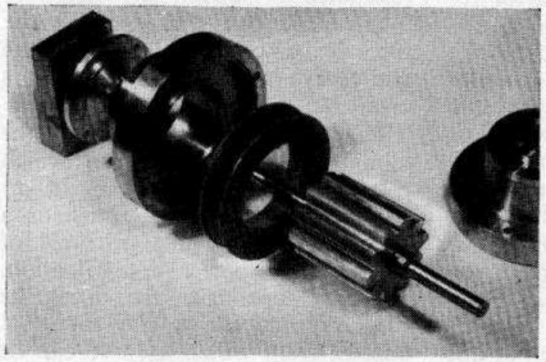


Fig 3





AT LEFT, THE COMPONENTS of the Eddy Current Dynamometer. Bottom photo shows the main rotor and its surrounding field coil.

The latter curve increases at first, reaches a peak of some high r.p.m. figure (maximum b.h.p. of the engine) and then drops with increasing r.p.m. again.

A family of propellers is one of the most useful ways of getting a series of runs at different r.p.m. figures (and equivalent, thus taking a series of torque readings, converted to b.h.p. figures) since r.p.m figures with different propeller sizes are also of interest of engine users. An inconvenience, from the operating point of view, is that the engine has to be stopped, the propeller changed and the engine restarted at each step.

Replacing the family of airscrews with an airbrake does not eliminate slipstream, since the blades of the airbrake have to be adjusted at each step, usually by twisting, to slow down or speed up the engine, as required. Again the whole test is conducted in a series of runs. This time there is no comparable propeller-r.p.m. data and also there is the very real danger of getting the brake dynamically unbalanced setting up vibration, is inevitably a source of loss of power.

Flywheels can be used for loads for high speed running — and in fact, proved the only satisfactory form of load for very high speeds. The equivalent size in "propeller load" renders hand starting too hazardous, or too difficult. A series of flywheels of different weights, however, would be impractical for a complete range of tests. Band brakes or similar devices operating on a flywheel to vary the speed are likewise subject to practical objections.

The best solution to the problem of *accurate* testing of fractional horsepower motors, was, therefore, pretty obviously another form of test apparatus which would not be subject to such working errors (of a

largely unpredictable value) or objections on the grounds of practicability.

Props and Power Absorption

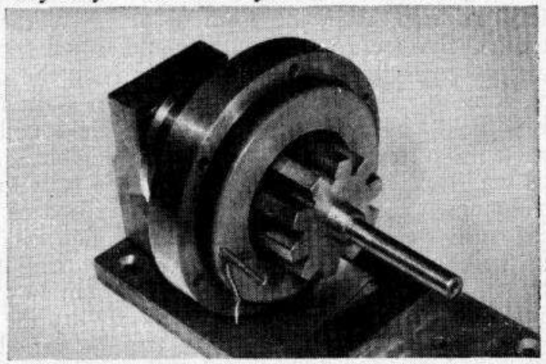
A possibility which occurred was to work strictly on a family of propellers of similar geometric proportions but different pitches and diameters, basing b.h.p. figures on power absorption figures to the r.p.m. at which individual propellers were driven. One major difficulty was how to calibrate the family of propellers. The other, more important, objection was that the high speed runs necessary to carry the tests past the maximum b.h.p. point on the performance curves would have to be attempted with very tiny propellers.

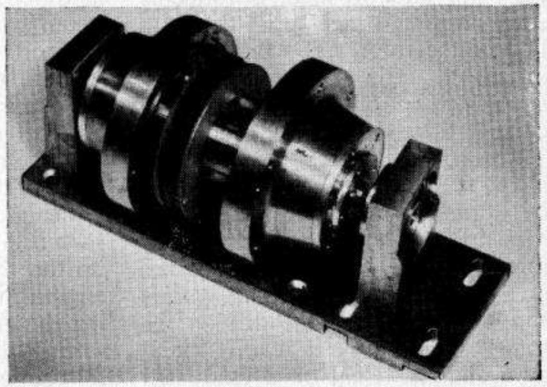
The first alternative investigated was a motor-generator or, virtually, a dynamo coupled to the engine shaft. Shaft torque driving the dynamo would then be measurable in terms of current produced. This method, however, did not lend itself too readily to accurate speed control and overall accuracy was likely to be of a low order. The heating effect in the coils and varying electrical efficiencies could play havoc with the final results. Although motor generators of this type are used for full scale testing, more often than not torque is determined by weighing the torque on the floating casing of the machine.

In full scale work there are two main types of dynamometers in current use—one a water brake and one an electric brake, working on eddy current principles. The former was ruled out as impractical for the size of dynamometer contemplated but the potentialities of the latter were investigated in some detail. A tentative design for the latter was suggested by Heenan and Froude with a nominal capacity of 1 h.p. at 10,000 r.p.m.

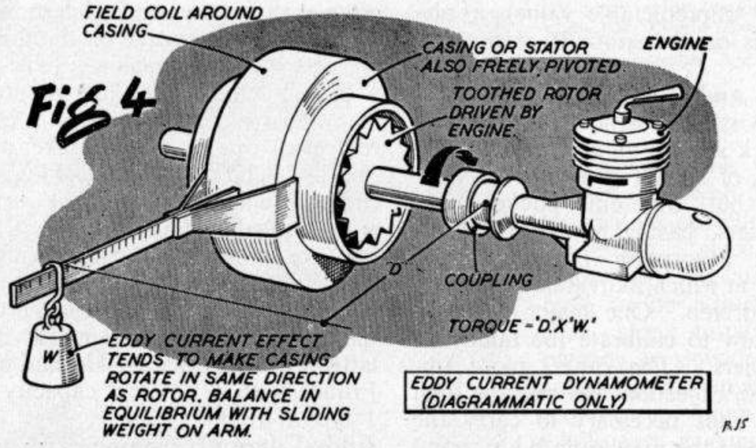
Eddy-Current Dynamometer

The principle of the eddy-current dynamometer can be ex-





STAGES OF ASSEMBLY ILLUSTRATE how the Eddy Current Dynamometer rotor is located within the field coil and casing, and in bottom view, other casing half and ball race bearings are shown fitted on to the solid base plate.



plained with reference to *Fig.* 4. The rotor constructed of a special form of iron, is in the form of a toothed flywheel rotating with small clearance inside a cylindrical casing or housing. Surrounding the casing is a field coil which can be connected to a source of electricity. The casing itself is freely mounted on trunnions.

In use, the rotor shaft is coupled to the engine shaft and thus driven by it. A direct current is fed to the field coil, producing a magnetic field within the housing. The teeth of the rotor cut this field, introducing eddy currents which resist the rotation of the rotor, *i.e.*, exert a braking effect. The amount of braking is readily controlled by adjusting the current flow through the field coil.

The rotor, in other words, is trying to drag the casing round with it by a magnetic coupling effect and measuring the turning force or torque on the casing is a measure of the shaft torque, *i.e.*, the torque supplied by the engine driving the rotor. This torque is readily found by a sliding weight on a graduated arm, as in the case of the torque reaction apparatus. This time, however, the only losses are the very small friction in the bearings carrying the outer casing and any stiffness in the lead connecting to the field coil. These should be negligible. At the same time there is a fully variable speed control. The speed of the rotor (and thus the engine on test) is dependent on the braking effect produced by the current flowing through the field coil. Vary the current and the speed can be varied at will. Torque readings at different speeds can therefore be taken without stopping the engine.

Construction of the AEROMODELLER eddy-current dynamometer has been carried out by E. Hook to Heenan and Froude's basic design and from start to completing took some 120 hours of careful work. Other problems had then to be met, such as a suitable means of coupling up the rotor shaft to the individual engines, and also a means of starting the engine once coupled up—to say nothing of a "universal" mount for the engine, capable of taking all

sizes likely to be encountered.

It became evident that even before the basic dynamometer was completed that the rotor size would be beyond the capacity of the smallest sizes of engines. Windage alone would limit maximum r.p.m. attained at too low a figure. Operating on exactly the same principle, alternative rotor designs, which can be interchanged, one of filled-in (plain periphery) and one of lightweight construction (plastic with inset slugs) are in the course of building to cover the full range of model engines from .05 c.c. up to 10 c.c. or larger with maximum test r.p.m. of the order of 20,000, if required. Interchange of rotors will not affect the accuracy of the results.

Considerable development work is still necessary to bring it to the final state required. Such developments will not affect the accuracy of the readings, or make the machine more efficient. It is being aimed at finding the best rotor designs for different sizes of engines and developing the control gear to the point where all the readings—torque r.p.m. and b.h.p. can be read off direct from dials, retaining means for a mechanical cross-check on both r.p.m. and torque.

CHAPTER ELEVEN

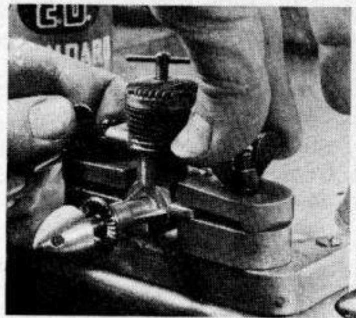
Operating a First Diesel

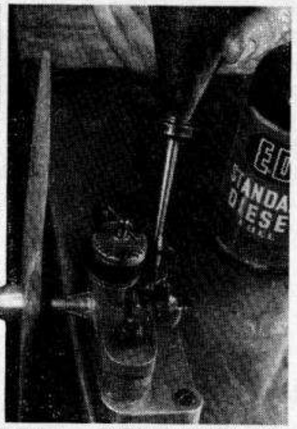
IT must be every young aeromodeller's aim and ambition to have a model engine to call his own—and these days it is inevitable that his first power unit will be a diesel of 1 c.c. or less. With this in mind, we intend to run through the process of purchasing and learning how to operate a small diesel. Really, we should be calling our engine "compression ignition" but the name of diesel is so widely used for these miniature two-strokes that it will now continue to be used for ever.

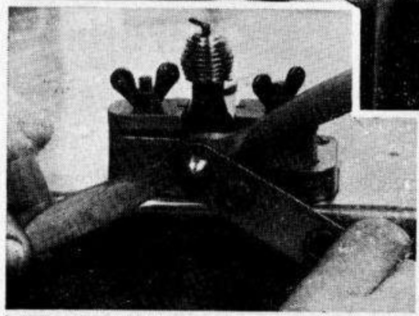
Mention of "two-stroke" recalls the method by which our engine will work. It will fire, or explode a combustible mixture every second stroke. (Unlike a four-stroke, where the firing is every fourth stroke and mechanical valves are employed to control intake and exhaust.) The fuel is drawn from the tank, through the carburettor where it mixes with air and into the crankcase via a rear disc, crankshaft or piston controlled valve. The descending piston compresses the mixture in the c/case, drives it up the transfer passages and through the transfer ports into the upper cylinder. As the rising piston compresses this mixture to the order of 20 : 1 it is self ignited and combustion gives a power stroke. The piston descends rapidly, and a further charge of mixture comes up from the c/case to help scavenge the burning gases out through the exhaust.

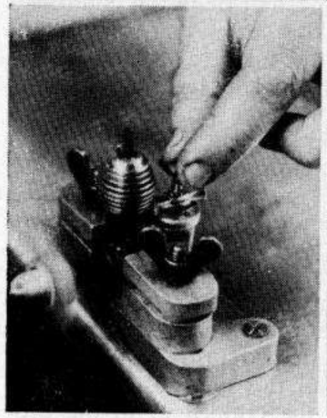
Now all this effort is transmitted through the crankshaft to the propeller, and according to the mixture we give the engine, and the compression ratio we can adjust, so can we vary the power and r.p.m. (revolutions per minute) of our engine. All we have to do is to learn the right way to go about the job.

With money in hand go to the model shop and ask the owner's advice on engine selection. You have a good choice, and for your part, you ought to have some idea of the kind of model you want this engine for, not only in the immediate future, but also for next year's building programme too. The engine you buy will last for years — again providing you use the right approach to it. Don't listen to the first modeller you hear in the shop; he is bound to say, "Oh, I wouldn't buy a so-and-so, the gudgeon pins are too weak", for he will be a classic example of one who has used the wrong approach Let's call him Compression Charlie —or "CC".



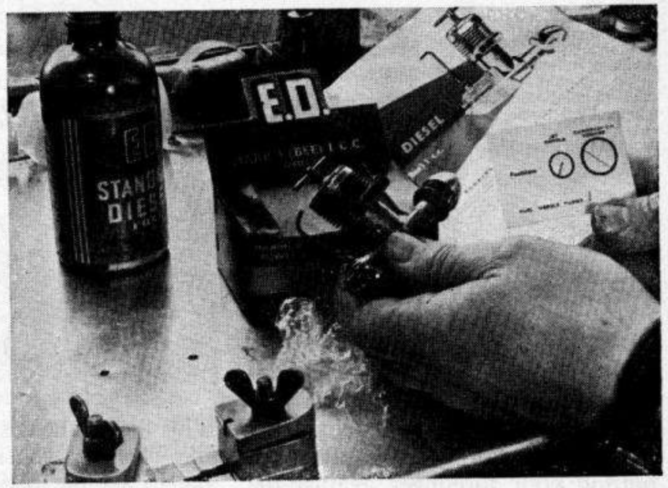






THESE PHOTOS show how one should follow the manufacturer's instructions. Mount the engine rigidly, fill up with the right fuel and set the propeller correctly for starting, using the proper spanner. Below right, how to choke carburettor; left, altering the needle valve.

Having made the big decision, take the engine home and read every single printed word in the instruction leaflet. Then fix yourself up with a suitable propeller — perhaps an 8 x 6—for low r.p.m. running-in on an engine of about



AFTER PURCHASE of the engine one should check the control positions against the manufacturer's setting card. Test clamp, diesel fuel and manufacturer's full instructions are ready for the first power run.

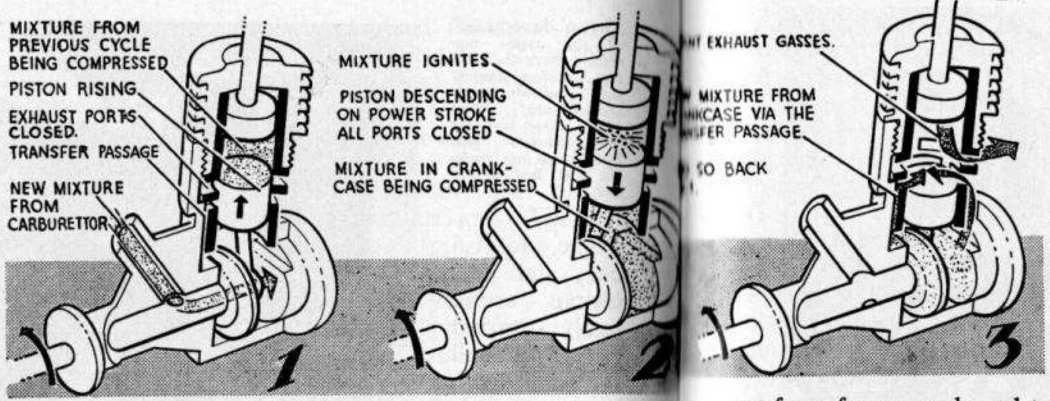
1 c.c. to 1.5 c.c., and purchase some ready - mixed fuel. We shall not dabble with home - made mixtures here, for the beginner can do no better than use a branded fuel with all its easy starting additives. Most engines have their own special brews, and the model shop will see that you are well equipped.

A test mount is a fine accessory you will be able to use over and over again, and if the pocket money will stretch that far, we suggest that you get one and mount it on the bench with long woodscrews or bolts. A facing of tinplate, available from most ironmongers for a shilling or two, will help keep the bench tidy and allow you to wipe it clean of exhaust. Now we are ready to start.

Mount the engine firmly and fit the propeller on the shaft by slipping it in position, turning the engine until the piston can be moved no further against compression, and tightening the prop nut with the propeller pointing at "twenty to two" as on a clock. Left handed people should treat this as "ten to four". Now swing the propeller over in an anti-clockwise direction. The engine is dry, stiff and there is little "feel" about it. Fill the tank, open the needle valve by unscrewing it the required number of turns from fully closed, and choke the engine. This means placing the spare forefinger over the carburettor and rotating the propeller one turn. Engines with the carburettor in front will indicate that fuel is entering the engine as you observe the flow through the tubing. Should there by a conglomeration of bubbles in the pipeline, then we must choke again to get the line full of fuel. Now try another swing at the propeller. Don't be afraid of it. it certainly will not fire, as all you are doing is filling the crankcase with a mixture, and creating a fine mist of fuel throughout the moving parts. This little amount of lubrication will change the engine from a lifeless object to something with the urge to "go", and as you continue to swing the propeller, you'll find there is an active "plop" as compression drives the propeller over.

Choke again, and repeat the swing

AT RIGHT, THREE sketches illustrate the complete two stroke ignition system as applied to the average modern miniature diesel engine. Particular subject detailed is the Allbon Merlin.



at the propeller, only this time putting a real effort into it. Start by putting your forefinger against the topmost blade, about halfway along and push the propeller over compression with a smart swing of both wrist and arm. At the same time take a firm grip on the compression screw at the cylinder head, and hold this set at the position indicated in the instructions. After a few sharp flicks of the propeller, there should be some reaction in the form of a mild firing stroke—or if you are extraordinarily lucky, the engine may burst into full song straight away.

If the engine refuses to show any inclination to work, look through the exhaust ports and see if the top of the piston is at all wet with fuel. If it is, then use the compression screw as though you have your hand on the pulse of the engine; treat it as you would a human, and raise the compression by screwing in the "vernier" as one maker calls the tommy bar or comp. screw. But do not be a "CC" for over-compression is dangerous, and is signified by a hydraulic lock when it is impossible to rotate the propeller.

This also indicates that our choking has been too generous, the cure being to set the piston at the bottom of its stroke, and to blow hard through the exhausts to clear the excess mixture, and to release compression.

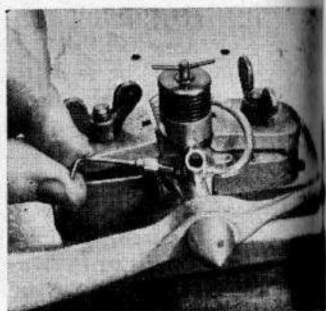
Should the piston be completely dry on inspection through the ports, then the choking has not been sufficient, or the needle valve setting is not open enough. It is better to err on the rich or "open" side for first starts—providing you release the comp. screw when compression seems too great.

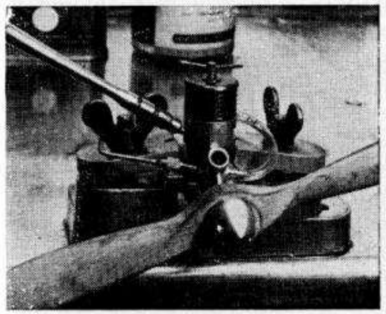
After a while, you get into the swing of things, and soon you are rewarded with a start. Once the engine has begun to run, leave it as set for a few seconds and take stock by watching the exhaust and listening to the note. Smoky, rich exhaust is cured by screwing in the needle valve, and a staccato misfire indicats the need for more compression. Most engines start for the first time in this condition, and will not harm themselves if allowed to run rich. Should the note sound laboured, gradually dying off in r.p.m., then the engine is over-compressed to some degree, and the comp. screw must be slackened off.

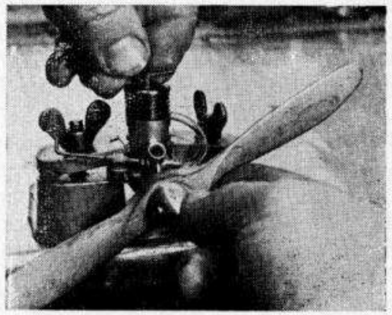
Now all of the foregoing has made one very critical assumption. That you have purchased a trouble-free engine (and it should certainly be so if brand new), and with it you have obtained some ready-mixed fuel of the advised brand for the particular engine. It may be that you do not have a local source of supply for commercial ready-mixed fuel, which cannot be transmitted by postal services. You may have purchased this manual for that very reason, and here is where we must apply special instruction for that most needy of modellers, the unattached man out in the bush of Africa, the Outback of Australia, or perhaps the Highlands of Scotland. All of you have one thing in common, an urge to build a successful power driven model: but lack of personal model shop demonstration and local advice throw one into the "pioneer" stage where to experiment is usually a waste of time and effort.

The procedure for mounting the engine, checking the fuel flow and getting oneself accustomed to the "flick" start as described in the first part of this chapter are simple enough to follow and will not need further enlargement: but let us presume that before we can start, we have to obtain the fuel.

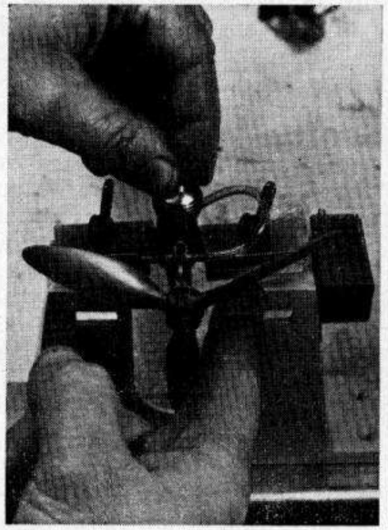
The model compression-ignition engine will not run on pure diesel fuel as used in tractors and road vehicles. This is a paraffinic hydrocarbon which has a spontaneous

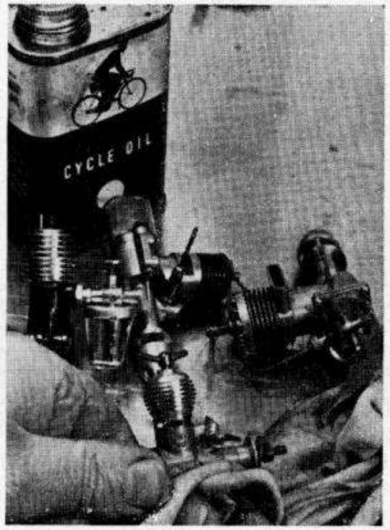






WITH FRONT rotary valve engines, check first that the needle valve can be altered with the engine mounted in the test clamp. Centre, prime exhaust with fuel and at right, prepare to relax compression as needed and flick start from near the boss of the airscrew.





UNUSUAL PROPELLER IS THE FROG PLASTIC three blade, here applied to a Frog 50, must be flicked very near the boss. At right, after running-in the engine, inject pure oil and wipe off excess to prevent any corrosion.

ignition temperature (known technically as an S.I.T.) of around 250° Centigrade, and when forced by injection into a large capacity engine cylinder under very high compression conditions, it will fire. Our model diesel takes in its mixture as described before, by the two-stroke principle and the degree of injection pressure is so negligible that such combustion is impossible. So we have to have "additives" in the fuel to reduce the "S.I.T." to a suitable level. This is quite simple. We add Ethyl Ether, having an S.I.T. of 188° C. and very wide explosive limits. We also have to add a lubricant as is customary with all two-strokes not employing a separate oil pump, and just one more component can also be mixed in the formula to ensure smooth even running. Such are usually referred to rather baldly as "dopes", for they do in effect, drug the mixture by preventing detonation and thus improve the maximum revolutions figure because the engine is allowed to run more consistently. These dopes are Nitrates or Peroxides, and the most widely used for diesel fuels are Amyl and Ethyl, Nitrites and Nitrates. It is emphasised that they are not essential for the formula: but offer a noticeable improvement of great benefit in competition flying.

Seldom is more than a drop required in a tankful of fuel, and if mixing by the pint or quart, then 3 per cent of the total volume is ample. The Amyl Nitrite has a terrific rate of expansion and is used medicinally as a heart stimulant so it should be treated with absolute caution. When the Nitrite or Nitrate is added to the mixture one can perceptibly feel the stimulant effect and it is recommended to mix fuel in a well ventilated area.

Thus we know that we want four components to make up our fuel. (A) Paraffinic base fuel, can be diesel engine fuel sold as D.E.R.V. in Great Britain, or straight paraffin sold as "petrol" on the Continent just to add to confusion (Petrol, Gas or whatever you call automobile engine fuel in your country, is not suitable and should not be used). (B) is Ether. This can be obtained in several grades, with consequent variety in cost. Ether sold in the chemists for surgical cleaning is quite good enough for us. (C) is the Lubricant, and here we have a wide choice. Castor based racing engine fuels for motorcycles is ideal, and Castrol M is most suitable. Medicinal castor oil can be used, but creates black sludge in the exhaust; Castrolite or any summer grade lubricant for big engines will suffice. (D) is the dope, either Amyl Nitrate or Nitrite advised, and purchased with a tactful approach at the dispensing chemists. A small 2 ounce quantity lasts for a long while if kept cool and in a dark area, but is expensive.

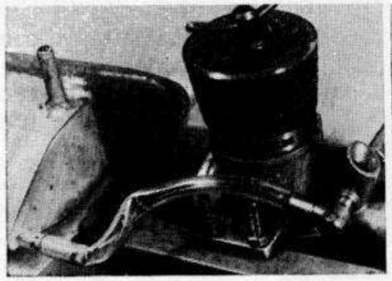
Quantities of the fuel components can fluctuate by wide margins. Simplest mixture, and one advised by Messrs. Electronic Developments Ltd. for their engines over a ten year period is equal parts (33 per cent) of each, Paraffin, Ether, Castor Oil plus additive. If you want to run faster then reduce the lubricant to a 20 per cent minimum, and increase the paraffin accordingly, keeping the ether a constant 33 per cent of the whole. A medicine bottle or graduated flask is a great asset when blending one's own brew.

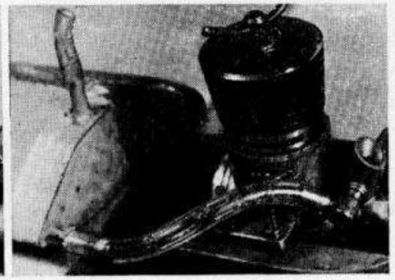
More on fuels will be found in chapter 17, but now we know the basic principles, we can assume that the first diesel is now ready to run on our own concoction.

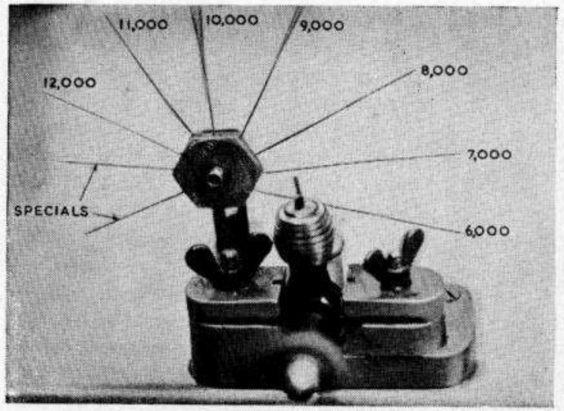
Let us make one more presumption for the "Outback" modeller—the engine arrives without any guide as to its working settings, or it does have such a guide, but the Customs & Excise people have been playing with the engine so much that factory settings cannot be identified on the controls.

To find the average carburettor setting, connect fuel tubing to the needle valve body and remove the needle valve itself. Blow through, and clear the line. Air should hiss through the jet and needle entry Close the needle entry with a forefinger and check that air flows freely through the jet(s). With this

A KINKED FUEL LINE AS AT LEFT OR bubbles in the line as at right, can prevent an engine from running. Bubbles sometimes signify an air leak at the fuel tube junction.







A REV. COUNTER WHICH CAN be made using 20 s.w.g. piano wire. Lengths are 82, 78, 73, 68, 65, 61, 58 mm. for the scale of 6-12,000 r.p.m.

point clear, no obstructions present, screw the needle fully home and try to blow through the fuel feed tube. It should be impossible for any air to pass the needle. If it does, check that the needle *is* fully home, and that it is soldered securely in its thimble or outer body. Sometimes this solder joint works loose with enthusiastic handling. To find the setting for running, which can be anywhere from one turn open up to eight turns on the modern fine control diesel, blow through the tube again, slowly opening the needle valve at the same time. The jet will soon be partially exposed and a gentle hiss observed. When the needle is about one turn open from the time the hiss first becomes noticed, should be our "average" setting.

So much for the mixture, now for compression. We should already have checked that all screws are tight and joints sealed so that there are no air leaks in the engine. Turn the motor over, after priming through the exhaust to make sure the piston has some lubrication, and at the same time, screw the compression adjustment tommy bar so that the contra piston is forced down. By rotating the propeller slowly or just oscillating it either side of top dead centre position, we can carefully ascertain the moment at which the main piston and contra piston contact one another. At this compression setting one should stop screwing "down" and noting the angle of the tommy bar relative to the engine, should "back-off" the compression one turn. This is usually a safe rule of thumb method of finding the correct setting. Now we have the settings, let's try to start just as described earlier. It won't go ? Well let us try to find out whether the fault is hidden within the engine or the operator.

Nine times out of ten, the "flick" is at fault. One must be quite vicious with a diesel, the propeller has to go over T.D.C. with a very sharp snap if ignition is to take place, and one must always be ready to adjust the compression ratio to suit the "feel" of the engine. Herein lies the secret of success in starting a diesel. The needle valve can be left alone providing it is open enough to supply fuel, all we need to worry about is the compression ratio.

With a prime in the exhaust ports of several fuel drops, start to flick. Should the prop go around to T.D.C. and suddenly freeze hard, as

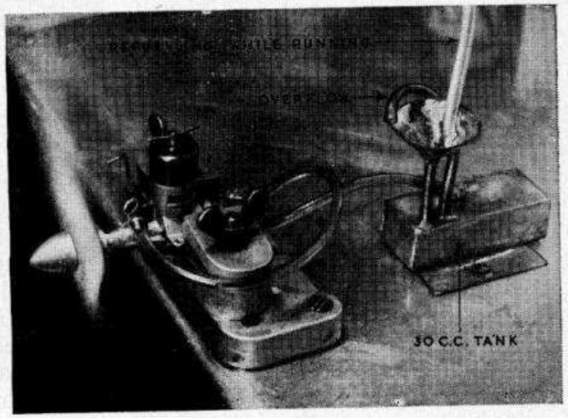
though against an obstruction, then you have primed too generously. The condition now created is known technically as a hydraulic lock, and the cure is to swing the engine backwards 180° so that the exhaust port is opened, and to blow into the port to distribute some of the excess mixture. One must also back-off the compression setting a half turn, then carefully rotate the engine using full prop leverage by turning with a prop tip. This ensures that there will be sufficient inter-piston clearance for the next flick. Try again. It will still be a wet motor, so one cannot expect a firing stroke for a few flicks. When it does fire, it will be at lower compression, so the spare hand must quickly react to any misfiring by screwing down the compression ratio until smooth running is achieved.

It may be that instead of running at all, the motor chooses to oscilate back and forth rapidly about T.D.C. This is due to a combination of too high a compression ratio and too rich a mixture in the cylinder, but the needle valve still does not have to be altered. Back-off the comp. screw a half turn and you will find that the engine stops oscillating, in some cases runs correctly and mostly stops, ready for an immediate start if flicked correctly.

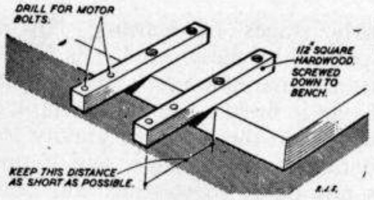
The tank position should not be any higher than the jet during these early stages of learning how to operate the diesel, and it is best to use the manufacturer's integral tank if one is fitted. Should the tank be high, then there will be gravity feed to the jet all the time, and if there is any delay in starting, this could completely spoil one's hopes of success by flooding the crankcase. This is a situation only remedied by removing the engine from the bench and turning it upside down, oscillating the shaft so that excess fuel flows from the exhaust ports.

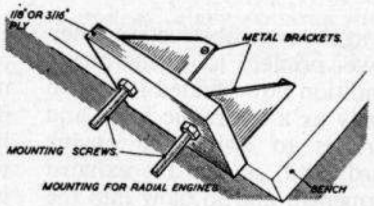
We have avoided touching the needle valve through the initial stages because, if near enough to average running setting by the "blow through" test, then the diesel can be fully adjusted to run continuously by means of the compression alone. For maximum performance the needle valve should be used to adjust consumption and here we begin to learn the idiosyncracies of the compression-ignition power unit.

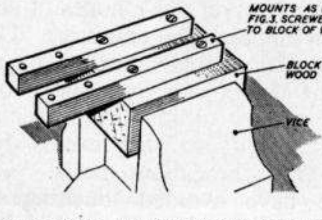
As the carburettor is "leaned-out" or made to run with a restricted fuel supply, then the engine will speed up at normal compression settings,



FOR LONGER RUNNING PERIODS, a standard tank can be fitted with filtered funnel which can be reloaded whilst the engine is still running.

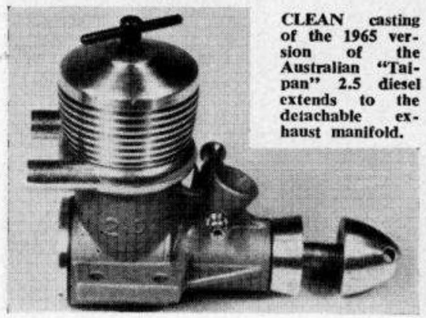






THIS SKETCH ILLUSTRATES THE USE OF hardwood engine bearers if a cast test mould is not available or desired.

then begin to misfire as the lean mixture takes effect. To compensate, the natural thing to do is to increase the compression ratio: but if the needle setting is too lean, then the reaction of increasing comp. ratio is to cause the engine to slow down, give the impression of drying up, and then to apparently seize. It does not actually seize (in all our experience with thousands of aero engines large and small, and hundreds of makes, we have only ever had one seizure and that was due to leaving an engine unwashed after testing a potent fuel—it seized through corrosion!)



CLEAN casting of the 1965 version of the Australian "Taipan" 2.5 diesel extends to the detachable exhaust manifold.

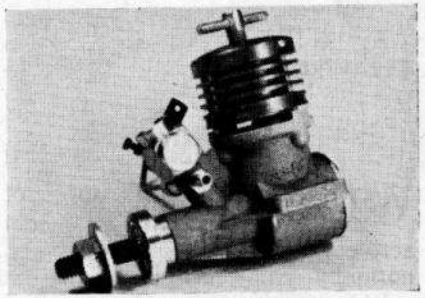
Obviously this is taking the leaning-out stage too far, we have to enrich the mixture and try again. So it goes on, turning first on the needle, then the compression until the sound of the engine is smooth, the exhaust clean, and the revs as high as possible.

During these stages of adjustment, one appreciates the feel of a lapped contra piston, and one can sense the necessary adjustment through the tommy bar. Some modern engines do not have lapped contra pistons, employing an alloy slug which expands rapidly to seal any fit, or what is known as an "O" ring or plastic insert to give a pressure seal between the contra piston and cylinder. The latter two systems do not unfortunately, offer the same "feel" characteristic of the lapped piston common to most British engines. They do, however, work most satisfactorily, and once employed, are soon appreciated for their design simplicity.

Another new feature of modern diesels is the reed valve as opposed to the rotary or piston port valve. This brings a new requirement for the diesel operator to understand, for there is a lengthy delay between time of needle adjustment and actual effect on all diesels using reed, clack or vibra-matic systems.

One must also allow for the diesel being a vintage type of the era when the contra piston was considered unnecessary and obviated by use of a blanked-off cylinder head. Such engines, with fixed comp. ratios demand a fuel with large ether content, and a quantity of mineral oil, RedEx being specially suitable. A typical formula would be 60 per cent Ether, 40 per cent RedEx. Arrange slight gravity feed for such engines if the fuel can drip away from the carburettor, and prime very carefully as there is no adjustment. Engine speed depends entirely on the heat of the cylinder and the amount of fuel supplied.

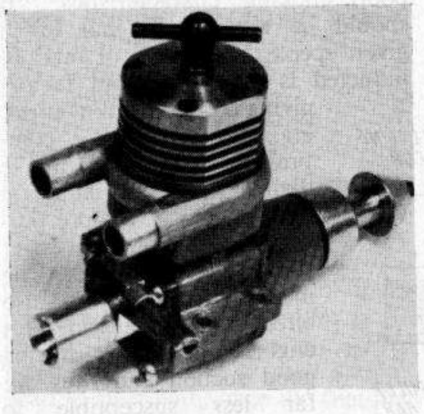
Above all, whatever the type of induction, cylinder head of manufacturer, your diesel will not run properly unless it is securely mounted on the bench or in a model, the former to be preferred if it is your first engine. Use strong hardwood measuring at least $\frac{3}{8}$ in. square, larger if possible, and screw the pair of beams on to the top of the workbench with long woodscrews. Nails will not do, and the



THE AM-10 AND AM-15 ARE IDENTICAL IN external appearance except for the colour of the fins. This is the radio control version with a rotary baffle above the needle valve control. These engines have a great reputation for simple single channel R/C.

space between bearers should be just enough to clear the crankcase. Use all four of the mounting holes in the cast lugs on the engine so that the running stresses are distributed around the casting as the manufacturer intended and make sure that all bolts are tight after each lengthy run.

DO follow the maker's instructions most carefully, and DON'T dismantle the engine unless you know that something has been damaged.



THE ETA 15 Mk. II FITTED WITH A REVISED Edmonds' carburettor by team race specialists who have also removed the fin area to crankcase width for easy assembly of the model. The needle valve is a Cox type.

CHAPTER TWELVE

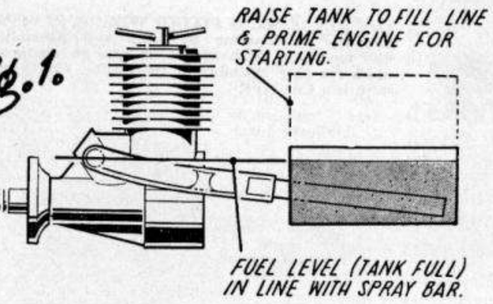
Fuel Tanks and Fuel Feeds

WITH the elementary form of carburettor used on all model aero engines (*i.e.*, a jet hole opening, normally controlled by a needle valve) anything other than suction feed will result in a continual flow of liquid fuel when the engine is stationary, unless the needle valve is closed right down. It is therefore more or less standard practice to arrange the tanks so that the standing fuel level is not above the level of the spraybar jet hole. For static running this is generally quite satisfactory for the suction head resulting (*i.e.*, the head or height through which the fuel has to be sucked to reach the jet hole) can be made quite small and there is little variation between this head with the tank full and empty—*Fig.* 1.

The actual suction available to lift the fuel varies a lot with different engine designs, although all may operate perfectly satisfactorily when running. By this we mean that some engines do not readily suck up fuel from the tank to fill the fuel line for starting, even when fully choked and with a minimum suction head. For bench running, in fact, with a set-up like *Fig.* 1 the easiest way to fill the fuel line may be to lift the tank bodily and so momentarily apply gravity feed.

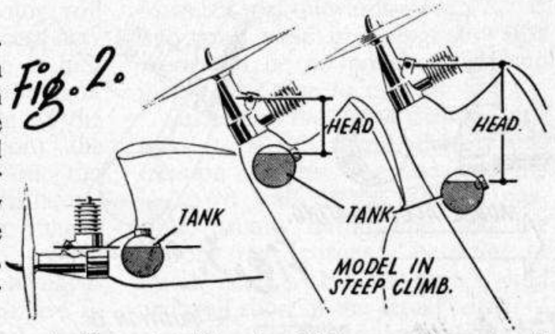
To a large extent the "suction" power is governed by the crankcase compression ratio. This can be defined as the ratios of the "under piston" volume with the piston at top dead centre and bottom dead centre. If the displacement of the engine is X c.c. and the actual crankcase volume at B.D.C. is Y c.c., then this ratio is X + Y/Y. The higher this ratio the more work has to be done on the piston travelling down to the B.D.C. position. Hence to minimise power losses a designer may deliberately use a low crankcase compression ratio, which may result in the engine having very poor choking characteristics. It should be appreciated that for a given speed the *volume* of mixture inducted is unaffected, but factors giving a rapid suction effect, *e.g.*, high crankcase compression ratio, small intake diameter, shorter induction timing, etc., give more pronounced suction and easier starting characteristics. The method of induction also affects the issue, side-port engines generally having good suction and thus are far less susceptible to

Fig. 1.



suction head. Most side-port engines will suck in and run satisfactorily with the tank well below the intake position. Most reed valve engines have similar characteristics, but rotary disc induction is at the opposite end of the scale, with rotary crankshaft induction intermediate.

Fig. 2.



The predominate pressure inside the crankcase of an engine is positive and suction pressure is realised only over a proportion of the cycle. Thus finger choking and turning the propeller over by hand to suck up fuel will apply both positive to blow-back and suction pressure to the fuel line. Particularly with racing engines or engines designed for high operating speeds the timing may be such that there is appreciable "blow-back" when turned over slowly. The correct technique for priming the fuel line in such cases is to move the propeller forwards and backwards across compression, apply finger choke *only* on the forward movement. In this way the intake is sealed only on the suction movement and blow-back can escape, through the intake on the return movement, the finger having been removed. This seems a small point but is one which gives trouble to many engine operators who are puzzled by the apparent lack of suction with a particular engine and perhaps have to resort to blowing through a fuel tank vent, holding a model up on its nose, etc., for priming the fuel line for starting. Provided there is only a small suction head to start with, once the fuel line is properly primed no further troubles should be experienced with it draining again, unless there is an air leak in the line connections.

Once running the engine can tolerate a far greater suction head without interrupting the fuel flow to the engine, although there are obvious limits. In the case of free flight power models, provided the tank is mounted on a level with the spraybar to start with, few troubles should be experienced due to changing fuel level in flight. It should be borne in mind, however, that to minimise the effect of changing attitude on suction head the tank should also be fitted as close behind the engine as possible—see *Fig.* 2 Almost any form or shape of tank is usually suitable, provided the fuel feed remains submerged for the duration of power run required. For example, mounting a circular tank with the feed to one side might result in the engine being starved in a steep climb or when the model is accelerating (when inertia will throw the fuel to the back of the tank)—see *Fig.* 3. Normally, however, on free flight models the fact that the engine will speed up due to the propeller becoming "unloaded" in forward flight will have more effect on needle valve setting than tank position.

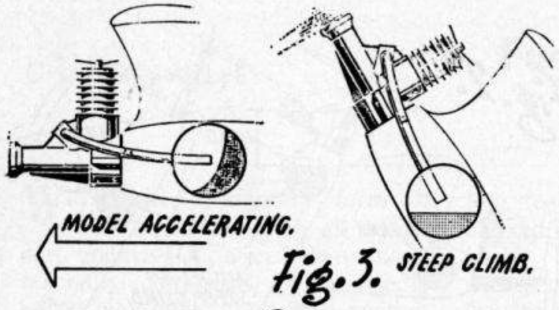


Fig. 3.

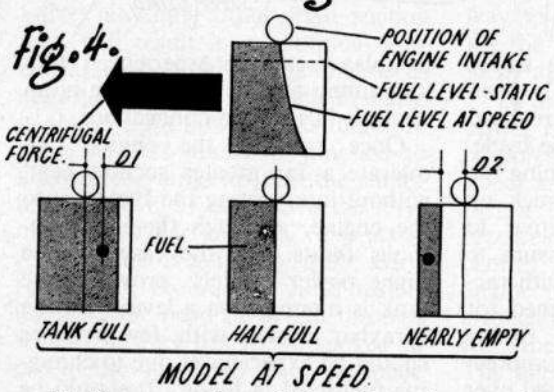


Fig. 4.

In the case of models subjected to considerable accelerations—*e.g.*, aerobatic radio control models and control line models—the question is far more important for inertia effects are exaggerated. On a control line model, for example, fuel will tend to be piled up with a near-vertical surface in the tank—*Fig.* 4.

Assuming that the engine intake is on a level with the top of the tank and the fuel pipe arranged suitably to pick up from the tank from full to empty, conditions for starting and adjusting the engine on the ground are zero suction lift. Now with the model flying and tank still nearly full there is effectively a *positive* pressure feed due to centrifugal force, which becomes a *negative* feed force after the tank is half empty.

This effect can be calculated quite easily. Centrifugal force is equal to $\frac{MV^2}{r}$, where M is the mass, V the speed and r the turning radius. For a given weight W of fuel,

$$\text{centrifugal force} = \frac{WV^2}{gr}$$

which for 136 m.p.h. (200 ft. per sec.) and a $52\frac{1}{2}$ ft. radius circle is centrifugal force = 23.7 W.

Thus in the case of the example sketched in *Fig.* 4, at the start of the run with the tank full conditions are equivalent to placing the tank some 23.7 × D_1 inches above the spraybar (*i.e.*, equivalent to nearly 12 inches gravity feed on a 2 inch wide tank). At the end of the run, conditions are equivalent to *suction* feed of a somewhat reduced order (due to the smaller weight of fuel left). But even so the total change of head from tank full to tank empty is considerable, and the greater the speed and the smaller the radius of the circle, the greater the effect.

Most tanks for speed models are, therefore, made tall and thin, a minimum practical width ensuring a minimum change of fuel head during flight. At the same time the lateral position of the tank with respect to the intake is significant. If too far towards the inboard side of the model (*i.e.*, large D_1 dimension) the motor may have a marked tendency to richen up, slowing the

model down when the motor will lean out again, and the process may be repeated until the tank is half empty. Conversely, with the tank too near the outboard side of the model (large D_2 dimension) the motor may tend to lean out too much—*Fig.* 5. Adjusting the lateral position of a speed tank is often a cure for such troubles.

Some of the more common shapes used for control line models are illustrated in *Fig.* 6. Actual shape will depend on the type of model. A rectangular shape is popular for team racers, and variations in vent positions are shown for two types. Where the tank has to be negotiated around an engine bearer or in a confined space, the piano shape is good as it allows for the wall of fuel to be constant when centrifugal force comes into effect.

Some advantage is claimed for utilising ram air pressure to provide a positive pressure in the tank, *e.g.*, most simply achieved by using forward facing vents. Straight (vertical) vents may actually have fuel sucked out of them at speed if normal to the airflow; or have fuel thrown out by centrifugal force (pressure build-up) if located on the *outboard* side of the tank. It is therefore the general rule on speed tanks to locate vent pipes on the inboard side, either facing forwards or outwards—*Fig.* 7. If only one vent is facing forwards there will be no pressure build-up in the tank due to ram air.

Since the fuel pipe requires always to be submerged, the obvious location for this is as near to the outboard wall of the tank as possible, usually terminating near the bottom rear corner. A point to watch here is that the pipe should end well short of the actual corner—*e.g.*, by at least $\frac{1}{4}$ in. and preferably more—as there is a tendency for

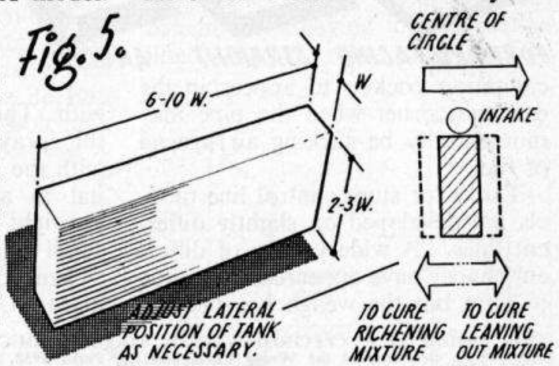


Fig. 5.

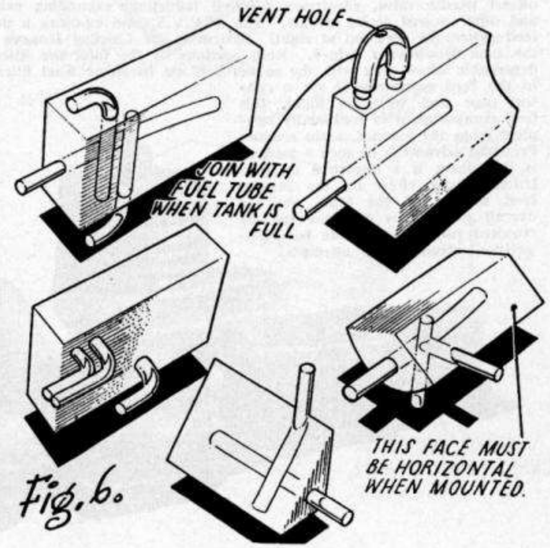
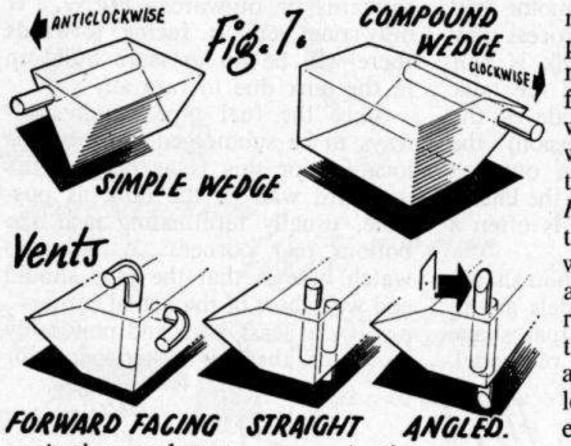


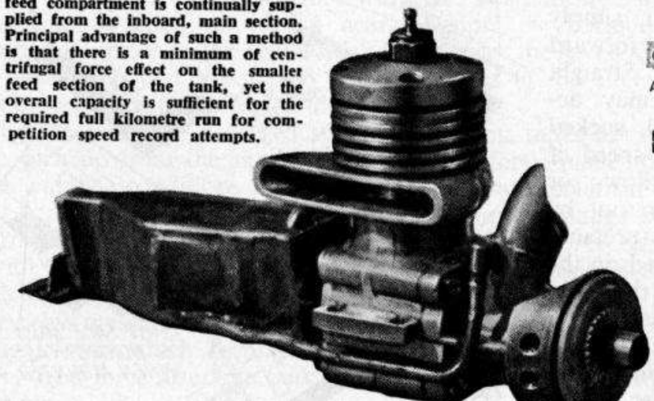
Fig. 6.

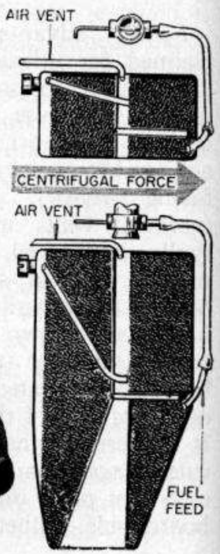


cavitation pockets to appear in the extreme corner when the pipe may momentarily be sucking air instead of fuel.

Tanks for stunt control line models are developed on slightly different lines. A wide variety of different shapes have appeared from time to time but the wedge has become more or less standard and perfectly satisfactory for most needs. This takes the form of either a triangular wedge of a compound wedge, with the fuel feed taken from the apex—*Fig.* 7—and the vents on the inboard side again. A wedge tank is symmetrical as regards feed both upright and inverted although, of course, there is a change in fuel head unless the spraybar of the engine is on the same level as the fuel pipe to start with. This will give gravity feed to the spraybar under static conditions with the tank full and it is more usual to arrange for zero head for "upright" running and accept the small change in head (tending to richen the mixture) in inverted flight. Usually this is not significant enough to cause trouble, except on a very "fussy" engine. Internal baffles are sometimes included in stunt and combat model tanks to minimise fuel surge during violent manoeuvres but this normally not necessary except on the larger sizes feeding the bigger engines. Baffles should be quite unnecessary on a wedge or "speed" tank used on any size of team racer.

THIS MOTOR IS A CZECHOSLOVAKIAN STATE PRODUCT, THE M.V.V.S.25, first seen at the World Championships, Paris, 1955, featuring off-set needle valve, slipstream directed induction, expanding exhaust port and other potent racing details. The M.V.V.S. also employs a unique fuel feed system as sketched at right. System is the Chicken Hopper principle, the tank is virtually 2-in-1. Both sections of the tank are filled through detachable screw plug with the model held on its side. Fuel filters through to the feed compartment via a central pipe and when in flight, this feed compartment is continually supplied from the inboard, main section. Principal advantage of such a method is that there is a minimum of centrifugal force effect on the smaller feed section of the tank, yet the overall capacity is sufficient for the required full kilometre run for competition speed record attempts.





Most wedge tanks are "handed" by arrangement of vents (*i.e.*, are usually designed for normal anticlockwise control line circuits). A non-handed type of stunt tank which has regained popularity with radio control models is the de Bolt —*Fig.* 8. The original de Bolt tank was rectangular in form with a weighted swivelling tube attached to the fuel pipe so that its end always tended to remain in the fuel, irrespective of the attitude of the tank.

It will be appreciated that this arrangement also compensates for displacement of the fuel sideways under centrifugal force, the same force that displaces the fuel tending to carry the tip of the swivel in the same direction. It cannot, however, compensate for fore and aft displacement of the fuel, so to minimise such changes the de Bolt tank is made tall and relatively short. A later de Bolt swivel tank (and a British counterpart, the EmDee, which appeared in the later 1940's) was cylindrical in shape and not so satisfactory in this respect.

For Radio Control

The de Bolt tank is well suited to the modern aerobatic radio control model since it can be made of ample capacity for the size of engine used and is generally fool-proof and trouble-free in operation. Some special forms of tanks developed specifically for radio control models are as sketched in *Fig.* 9. Normal stunt tanks, where used, generally benefit from having internal baffles fitted.

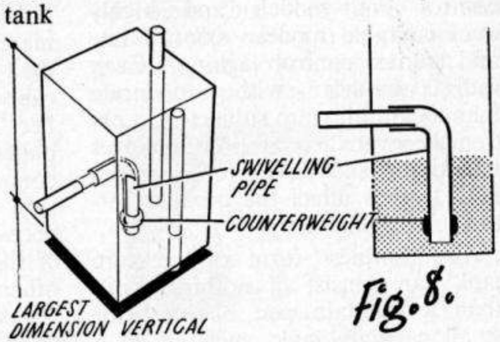


Fig. 8.

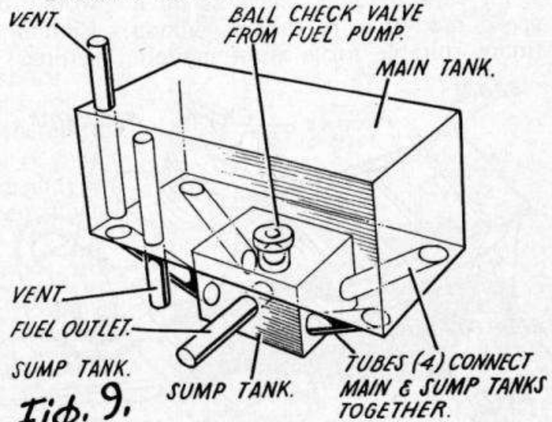


Fig. 9.

Pressure tanks have a definite value for supplying fuel under conditions where marked changes in head may occur, such as in speed control line models and highly manoeuvrable models—control line and radio control again. Even radio models with moderate manuoevrability are subjected to extremely severe accelerations, changes in attitude and inertia forces which may serious affect the mixture setting of an engine.

The simplest form of pressure tank can consist of nothing more than a fountain pen bladder (for small capacity tank, such as on a speed model) or a rubber balloon (more suitable for a stunt model).

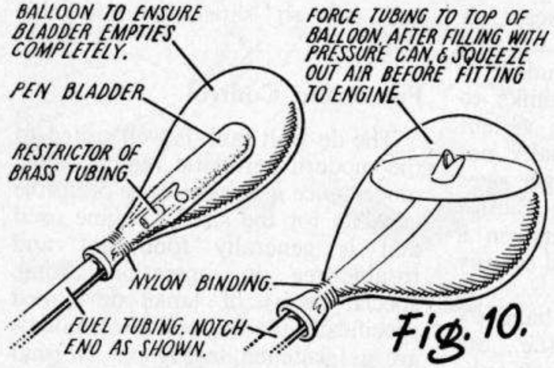


Fig. 10.

An ordinary rubber balloon is satisfactory for accommodating glow fuels but with diesel fuels a synthetic rubber variety must be used. A pen bladder can be filled *in situ* (*i.e.*, still attached to the engine) by means of a veterinary hypodermic to contain up to 30 c.c. of fuel. Alternatively, it can be filled with a pressure bulb. Balloon tanks are usually best filled by removing from the fuel line and pumped up with a pressure-type oilcan—*Fig.* 10.

Jim Walker introduced a commercial pressure tank on these lines, the tank material being synthetic rubber. Pressure is applied by rubber bands looped over the cover plates—*Fig.* 11—and the tank is connected to the engine via a pressure regulator. The regulator is necessary to equalise the pressure of the fuel as fed to the engine, as otherwise the change between "maximum" and "low" pressure would be too great for consistent running on a single needle valve setting.

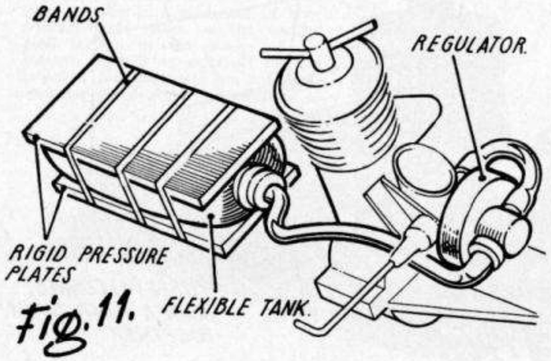
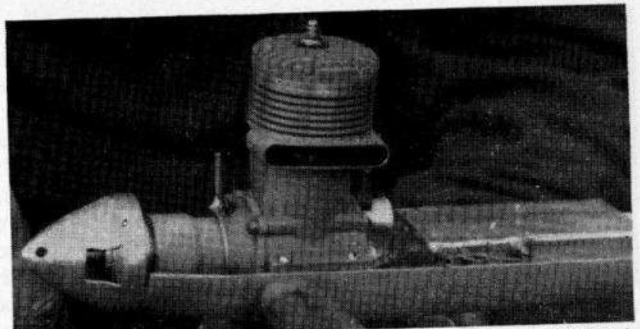


Fig. 11.

Another very successful approach to the effect of centrifugal force on fuel feed is the "Chicken-Hopper" tank, working virtually as *Fig.* 9, but on its side as sketched. This minimises the change in fuel head during flight and is sometimes taken to the extreme of using two separate tank components, one to act as reservoir for a small feed tank.



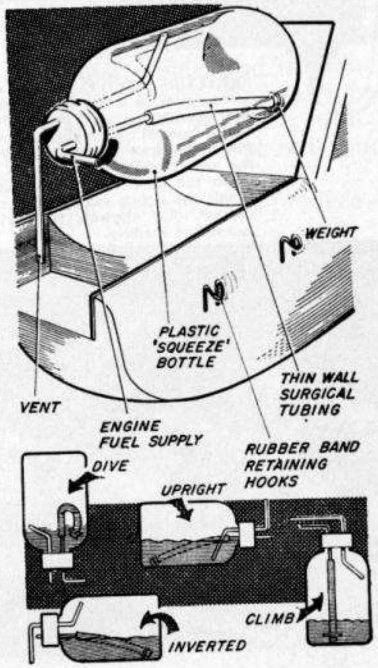
THIS IS THE KUZNETSOV 10 c.c. ENGINE WHICH ESTABLISHED A World Speed record of 196 m.p.h. at Leningrad on 30th Sept. '62. The tank is a metal type, using crankcase pressure for high rate of fuel consumption.

Invariably the "plumbing" in any fuel system is done with plastic tubing. A majority of the commercial tubing is manufactured in clear form and is to be preferred to opaque tubing since the state of the contents can be observed. It is of considerable advantage, for instance, with a cowled-in installation to take a length of the fuel line out through the cowling so that one can readily see when the line is full for starting.

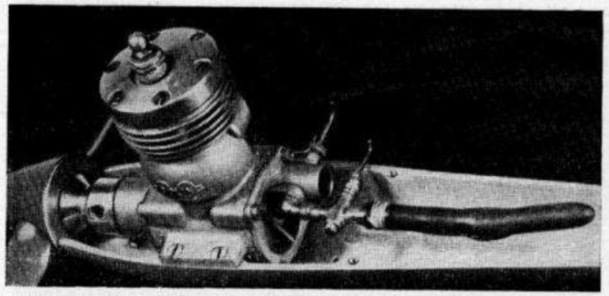
Tubing is normally made either from synthetic rubbers or P.V.C., both of which materials are fully resistant to fuel and oils. Neoprene tubing is the best from the point of view of remaining flexible in contact with fuels. Most of the plastic tubing age hardens to a rigid, brittle state in a matter of weeks after being in contact with fuel and the length usually requires renewing should it be disconnected for any purpose.

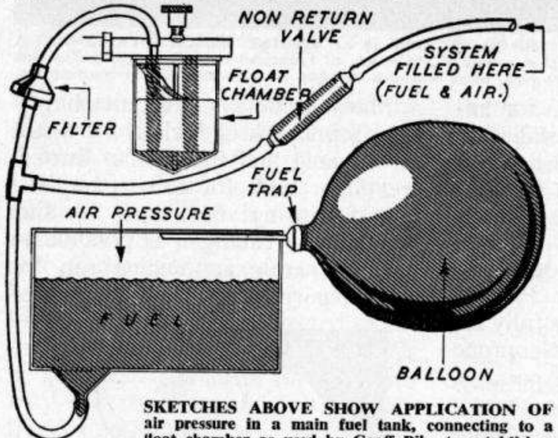
Most of the fuel tubing sold through British model shops is manufactured originally for surgical drain tubes, etc., and is therefore expected to remain flexible. A good tip for softening the hardened type is to warm it slowly then flush through with petrol.

Many modellers are quite happy to coil a length of fuel tubing about 18 in. long and use this to form a free-flight tank for contest models. By measuring the flow of fuel through the tubing it is possible to graduate an exact engine run for contest purposes with a tolerance of

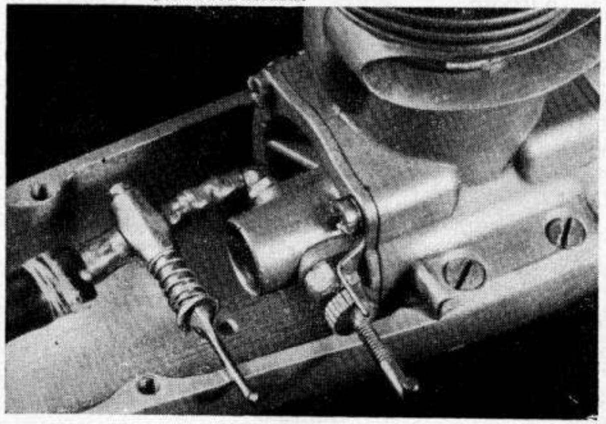


SIMPLEST AND PROBABLY ONE OF THE most effective stunt tanks for radio control is the use of an ordinary polythene bottle fitted with a metal screw cap and two brass tubes as in these sketches. Lower diagram illustrates how the flexible surgical tube—or cycle valve rubber, with wide end to fill the fuel to any position in the bottle and thus maintains a good flow to the engine. Tank also has the advantage of being quickly detachable and its translucent material offers an immediate gauge of fuel supply.





SKETCHES ABOVE SHOW APPLICATION OF air pressure in a main fuel tank, connecting to a float chamber as used by Geoff Pike to establish a radio control duration record at Nottingham. Photos illustrate speed expert Pete Wright's use of the pen bladder tank on his Dooling 29. Between the tank and needle valve (see below) is a simple tap which can be triggered as the engine is starting, thus allowing the fuel to flow from the pressurised bladder.



1 to 3 secs. allowance for precise judgment. Others, too, with an eye to simplicity employ the most useful transparent casings supplied with toothbrushes as fuel reservoirs. These can be cemented to plywood brackets and adhered to the side of the fuselage offering visible flow for a minimum of expenditure. They are not, however, robust enough to withstand rough treatment and for reliability one should always use the commercial products which have been in use for many seasons and have additional features, such as non-spill tops, and engraved graduations on the side as an indication for the fuel content. One last point concerns filling the tank. The arrival of the flexible polythene bottle now makes this a simple matter, and all that is necessary is to fit a suitable length of brass tube through a tight fitting hole in the screw top, then to fit to the tank and squeeze the contents out.

CHAPTER THIRTEEN

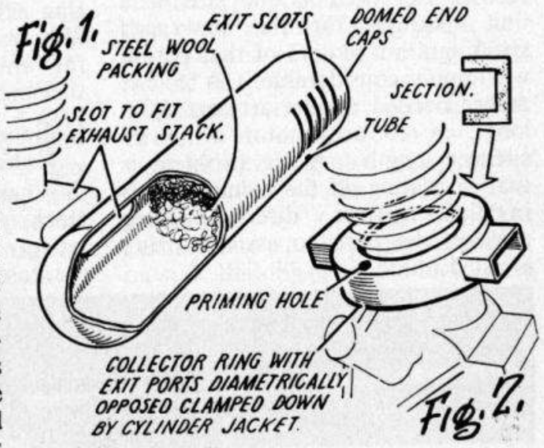
Silencers

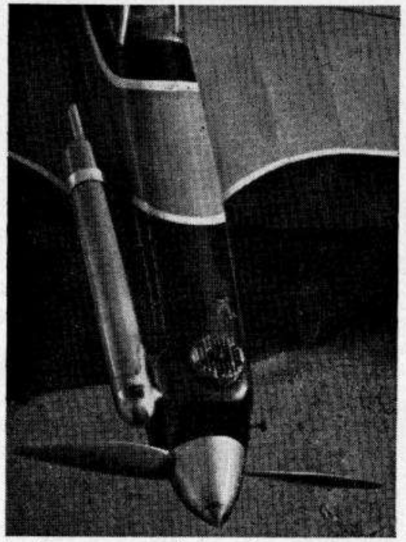
THE model engine is a remarkably noisy piece of machinery—a feature which cuts both ways. Noise implies power and so, to a large extent, the noisier and faster an engine the more potent it appears, and the greater its sales appeal. But to the outsider the nuisance value of a model engine being operated anywhere in his vicinity is considerable. It was noise more than the danger element which brought down drastic restrictions on the flying of power models in public parks—and, in fact, continues to get model flying banned in many areas. Even the test running of an engine in an average house or garage is apt to upset dozens of neighbours and although this problem has been with us for a number of years, very little attempt even had been made to find any sort of solution until 1965.

The apparent (complete) answer is an efficient silencer. Periodically one hears engine manufacturers condemned for not having thought to produce a fully-silenced engine for "urban", as opposed to "country" flying, but seldom have the critics given much thought to the implications involved. There have been commercial silencers produced for model engines (the American Mart-Lee unit appeared in 1947), and individual manufacturers have supplied silencer adaptations for their engines (Davies-Charlton and E.D., for example). But the silenced engine applied to a model aeroplane only became a reality when National rules required them in 1965.

The original Mart-Lee silencer consisted of an aluminium tube of roughly 1 in. diameter, blanked off at each end. A port was cut near one end of the tube to fit closely the exhaust stack of the engine and the other end of the tube cut with a number of slots for escape of the exhaust gases. This end of the tube was stuffed with steel wool—*Fig.* 1. One purchased the silencer as a complete unit, filed the slot to match the exhaust stack and held the contraption in place with a length of spring rod passed round the cylinder.

As an attempt to produce a





simple commercial unit the Mart-Lee silencer had many points in its favour. Provided the fit on the engine stack was reasonably close silencing was quite effective on the engines then current. It reduced the crackle of an Ohlsson to a "sewing machine" hum, with some rather peculiar side effects.

Silencing as such was quite effective, but even so the size of a unit required for the low-speed spark ignition motors of that period was quite considerable. A 5 c.c. motor needed a tube at least 5 in. long; a 10 c.c. motor a 10 in. silencer length. A big problem is that in sealing off the exhaust ports in this manner, direct priming through the ports to assist starting is ruled out.

SIMPLE TUBULAR SILENCER WITH ¼ in. I.D. outlet and partly filled with steel swarth, as applied to the Fox 35 giving excellent silencing to the level of a baby diesel in 1954.

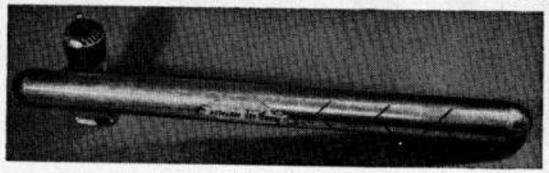
High Frequency Notes

The problem of silencing is not so much a question of reducing the noise level as one of filtering out and absorbing the objectionable high frequency notes. Size for size, the two-stroke engine is far the noisiest of the reciprocating internal combustion engines. For the same speed it has twice the explosion frequency of a four-stroke, and also a lower brake mean effective pressure. That means that the exhaust is open with the gases at a higher pressure, hence the more violent their escape of the gases, not the actual explosion or firing cycles.

The actual exhaust note varies considerably with different engines, and even with the same engine under different operating conditions. The "crackle" associated with high-performance engines is a welcome feature from the sales appeal angle and full size car and motor cycle manufacturers may go to considerable pains to achieve it (*e.g.*, in fitting exhausts of "resonant" length, although of course another reason for this is to improve cylinder scavenging).

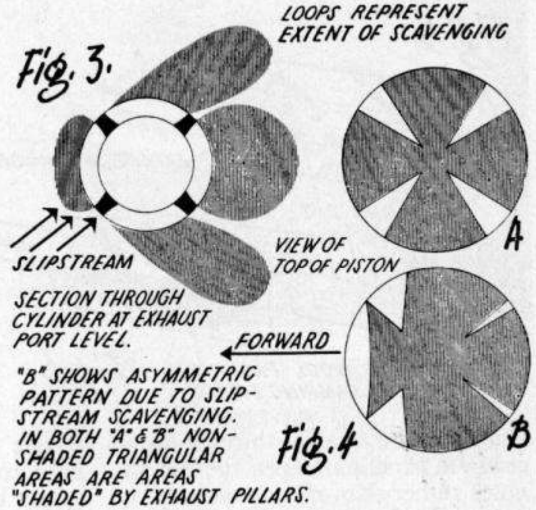
Collector Banjo for 360° Exhausts

Where an engine has an exhaust stack, *fitting* of a silencer is a relatively straightforward problem. In the case of circumferentially-ported



OHLSSON 23 IS DWARFED BY this American Mart-Lee tubular silencer with saw-cuts for outlets and partly filled with steel swarf.

engines a collector "banjo" is required, as sketched in *Fig.* 2. The groove should approximate to the depth of the port opening and the exit ports cut in the walls (for connection to the silencer) should be as large as possible. Preferably, there should be two such ports diametrically opposed.



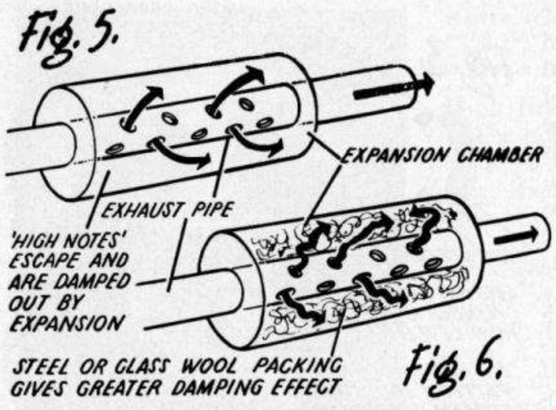
The fitting of such a banjo may affect the performance of the engine. If the design relies on sub-piston induction of air, this will no longer be effected. In fact, the engine will suck back exhaust gases instead of air (when the silencer is attached).

Another way in which running characteristics may be affected is that slipstream scavenging is now eliminated. With "open" porting, the slipstream playing back around the cylinder may materially improve scavenging—*Fig.* 3. On some engines the effect of removing such scavenging effect may be quite noticeable, on other negligible.

Incidentally, a fair idea of the normal "escape path" for the exhaust gases can be had by examining the top of a piston on a new engine after some twenty minutes running. Areas subjected to gas flow will be carboned up far more than "shaded" areas. A symmetrical pattern (with the light triangular patches indicating the "shading" effect of the exhaustpillars) as in 4(*a*) would indicate that there is no effective slipstream scavenging. A strong asymmetric pattern could mean strong slipstream effect—*Fig* 4(*b*).

As to the silencer units themselves, a "packed" silencer tube will provide most effective silencing, but the higher the operating speed of the engine the greater the adverse effect on performance through back pressure. The most satisfactory type of silencer is undoubtedly the straight-through layout with a surrounding expansion chamber. The length of pipe inside the expansion chamber is perforated, the expansion chamber itself being just a hollow cylinder—*Fig.* 5—or a cylinder packed with steel or glass wool.

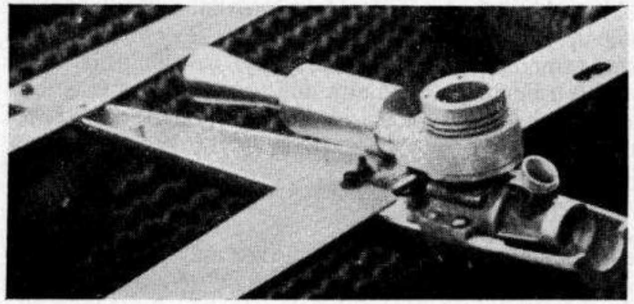
A straight-through silencer offers virtually no resistance to the passage of the exhaust gases (other than friction of the walls of the pipe) and by opening the flow radially into an expansion chamber, most of the objectionable high notes will be filtered off. In other words, a straight-through exhaust will only remove the high notes, whereas the packed silencer of *Fig.* 1 will remove both high and low notes. The effect of packing in the expansion



chamber of a straight-through silencer is to rapidly dampen the "high" notes rather than relying entirely on "expansive" damping and so should result in a lower overall noise level than the type of *Fig*. 5. The unpacked expansion chamber can, however, be quite effective if large enough.

The final note of such an exhaust (whichever type is used) will be affected by the total length of exhaust pipe. With a resonant length of pipe the final note can be quite loud (although not necessarily "objectionable" since it will be lacking the high notes). But it must be remembered that resonant effect will be achieved at only one speed. Thus if the normal operating speed corresponds to a resonant length of pipe, altering the length of pipe will cut down the overall exhaust note. Conversely, with a non-resonant length at operating speed, the exhaust may resonate at some lower or higher speed. Resonant length will also correspond to most efficient cylinder scavenging.

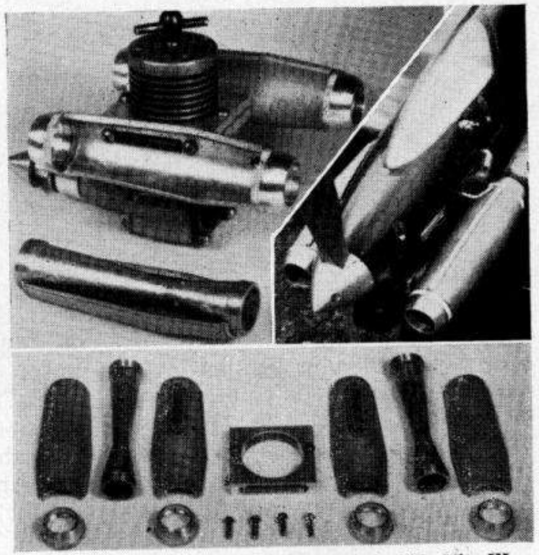
The size of expansion chamber required for effective silencing is quite large from an aeromodelling point of view. On power boat installations, where silencers are obligatory, a 5 c.c. engine commonly has twin exhausts and twin silencers, each with an expansion chamber some $1\frac{1}{8}$ in. to $1\frac{1}{4}$ in. diameter and 7 in. long, *i.e.*, a total expansion chamber of some 250 to 300 c.c.—more than 50 times the internal displacement of the engine! It is, therefore, difficult to think of an effective silencer for existing modern control line or free flight models fitted with an engine of more than 1.5 c.c. where the silencer would not be either too heavy or too large to accommodate on other than a specially designed model. A solution would be to design the fuselage around a silencer of the required size.



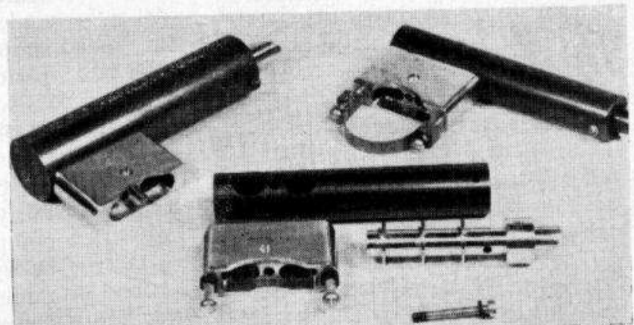
The extractor - venturi megaphone added to this exhaust aft of a sound absorption chamber shows one early approach to the noise problem by a speed flyer.

On small diesels of up to 1 c.c. or possibly slightly larger, a reasonable degree of silencing can be produced by fitting a collector ring and attaching a fairly long length of neoprene tubing for the "pipe". A length of at least 6 to 8 in. is usually required and the tubing diameter must be at least $\frac{3}{16}$ in. bore, preferably slightly larger. Some power loss will result but the noise level can be reduced substantially. Such an exhaust system is, necessarily, limited to short engine runs—a maximum of about 30 seconds—otherwise the tubing will melt. Also it cannot be used on glow motors.

Apart from the reduction in noise level, by *collecting* the exhaust objectionable oil waste need not be spattered all over the place and the exhaust fumes themselves can be led *out* of the test room (*e.g.*, through a window) by extending the length of tailpipe used. Some further suggested designs which should prove effective are sketched in *Fig*. 7.



THE ETA SILENCER SYSTEM ON A 15D Mk. III diesel employs a manifold and a pair of venturi shaped extractors. Exhaust feeds into the central venturi tube via small holes in a low pressure area. Display of parts indicates the complexity of such a system.



VARIOUS D.A.C. "Spinaflo" silencers with the core of one removed to show triple vanes or baffles. Later versions have round noses.

CHAPTER FOURTEEN

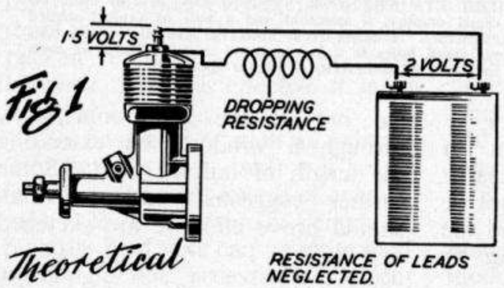
Operating Glow Plug Engines

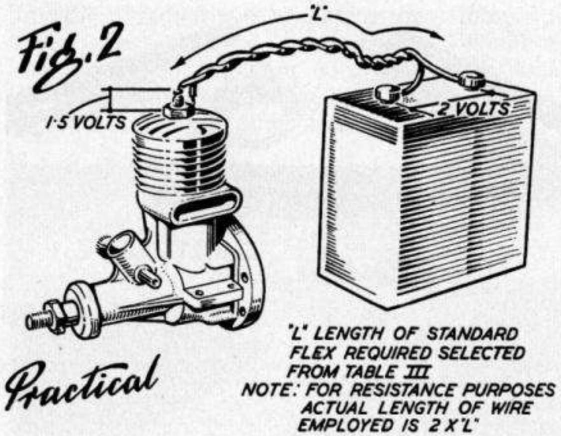
THE glowplug form of ignition for an internal combustion engine is one of the very original principles, used before either diesel or the spark ignition systems. The Wright Brothers flew with a motor of their own construction, using glowplugs, although of rather different character to those employed in the miniature two-strokes of today.

Our glow is provided by a wire element, usually platinum, which becomes incandescent when short circuited in a 1.5 volt (for U.S.A. plugs) or 2.2 volt (British plugs) circuit, and retains incandescence once the engine is firing, so that the short circuit can be removed and the engine runs as a self-contained unit.

The invention, or rather the application of glowplugs to our power units should be attributed to a brilliant American engineer, responsible for the greatest single advance in engine design and manufacture methods, by the name of Ray Arden. He introduced the plug as an accessory, for his own specially created engines. Overnight the idea swept the U.S.A., and came to Europe. The plugs enabled modellers to throw away deadweight in spark ignition systems and models were thus simplified, with higher performance.

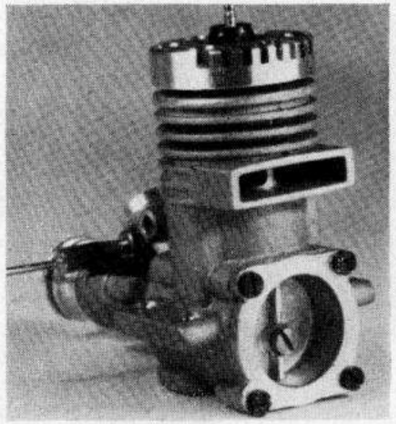
But conversion from spark to glow is not quite that automatic. Firstly, the piston seal has to be fairly good. Then the compression ratio has to be in the region of 8 : 1 minimum, and a good many ignition motors are less than that. Thirdly, the fuel has to be specially concocted, using alcohol base and castor lubricants. Additives to provide extra oxygen content have to be employed in racing glowplug engines, Nitro Methane being the greatest oxygen generator, and used in quantities up to 40 per cent of the whole. The fuel chapter provides a variety of formulae ; but for the first glow engine, you are recommended to adhere to a commercial fuel.





LARGER THAN LIFE PHOTO OF ASSORTED BRITISH AND AMERICAN glowplugs illustrates the diversity in design of the head connections. Nearest plug in holder is an Ohlson, extremely squat whilst those with spherical tops are KLG, the popular British make which has achieved great international reputation. Detached K & B plug in foreground shows the platinum-iridium wire coil and copper gasket.

Glowplugs are made in two thread fits, for $\frac{1}{4}$ in. or $\frac{3}{8}$ in. bodies, and in some cases, an adaptor can be employed to fit say, a $\frac{1}{4}$ in. plug in a $\frac{3}{8}$ in. hole. Plugs also vary in construction, even come integral with the cylinder head, and the most common means of fitting the wire element is to have it in a tightly wound coil of about 5 to 7 turns, Platinum-Iridium wire of about .015 diameter. Current drain on the booster battery is high, being around 2 to 5 amps, so it is better to remove the battery terminal connection as soon as the engine fires, or for any period when the engine is not actually being flick started. Dry batteries of the large bell-cell type, arranged in parallel pairs, are satisfactory: but for British plugs in particular the lead acid accumulator as used for Low Tension in a battery radio set, is far better. When fully charged, the wet accumulator



THE M.V.V.S. REAR EXHAUST 2.5 c.c. glowplug engines were made for the 1964 World Championships and since then imparted a great influence on model engine design. Screw in rear plate is for pressure tapping.

may be giving as high as 2.4 volts, so a resistance in one line, .25 ohm being average, is a safeguard against blowing the plug. See *Figs.* 1 and 2, Tables I, II, III for precise details.

To connect the glowplug to the battery, one can either employ stranded insulated cable, braided as a black and red pair for identification, or of the twin-flex type, about a yard long, and connect an alligator clip at each end. Have one lead shortened so that it cannot contact its opposite number and create a short circuit when the engine has started and the leads discarded hurriedly.

One more very important precaution arises before we can run the glow engine in a model for the first time. The model has to be "fuel-proofed" with a special varnish, as the fuel with castor, alcohol and nitro methane in it will soon act as a paint stripper and spoil the finish of any model decorated with cellulose dopes. The fuel proofer need only be a thin covering, not adding much weight, and certainly little trouble to apply. Two coats are advised around the areas in contact with the exhaust and filler positions on the fuselage.

To set the engine ready for starting, mount it securely in a test stand, arrange a fuel supply with the tank on a level with the crankcase, and fit a prop at the "quarter to three" position as on a clockface, securely tightening the prop-nut. Now blow through the fuel supply tube with it connected only to the needle valve body, and adjust the needle valve control until a steady hiss is heard at the jet in the centre of the intake tube. This will be the average setting for the carburettion when starting.

Next remove the plug, and connect with the battery. One lead to the top of the plug, the other to the plug body, or earth if the plug is left resting by its body on the engine cylinder head. There is no need for concern over polarity, just as long as one lead cannot touch the other, and that the alligator clips are safely spaced on the plug, then the element should glow bright orange. Dull red will indicate a poor contact, low battery or current leak in the circuit. If the plug glows rapidly to white heat, disconnect immediately as it is most probable that there is too much current going through the element and there is a danger of blowing out and fusing unless a resistance is fitted. American plugs on lead acid accumulators are specially susceptible. Once satisfied that the glow is present, we should inject a little of the special fuel through the plug hole, fit the plug and flick over a few times

without connecting the plug. A drop of fuel in the intake also helps this way to free up the engine ready for a quick start and the lubrication helps to seal the piston fit with a rich mixture.

When thoroughly prepared and ready to start, connect the fuel line to the tank which should be filled, do not choke the carburettor and then hook up the plug immediately prior to flicking over. A precaution is to take the prop tip and slowly rotate at one rev. per second, over the T.D.C. position of the piston. A "knock" should be felt at the fuel inside the combustion chamber is partially ignited especially if a trifle wet. Now flick hard: the engine should fire, continue to run, sucking fuel through the tube from the tank and giving the impression of running rich. Allow to pick up for five seconds or so, disconnect the lead off the top of the glowplug (the other can be more or less permanent for test purpose on a mounting lug or other convenient earthing point), then watch the characteristics.

The engine should be rather rich for the first run, not two-stroking but nevertheless operating smoothly with copious fumes from the exhaust and lots of noise. To get the engine to two-stroke, simply lean out the mixture by screwing the needle valve into its body, quarter turn at a time and waiting a moment between adjustments to check the effect in revolutions per minute. To stop the engine, simply screw the needle valve fully home and the effect will be that the engine speeds up to a peak speed, dies off and stops fairly abruptly through the fuel starvation. This will give the experience needed to identify too lean a fuel setting, which may in

TABLE I. LENGTH OF SOFT COPPER WIRE (SOLID) FOR .166 OHMS RESISTANCE

Size (s.w.g.) ...	14	16	18	20	22	24	26	28	30	32
Length (ins.) ...	1,250	800	450	250	150	95	63	42.5	30	23

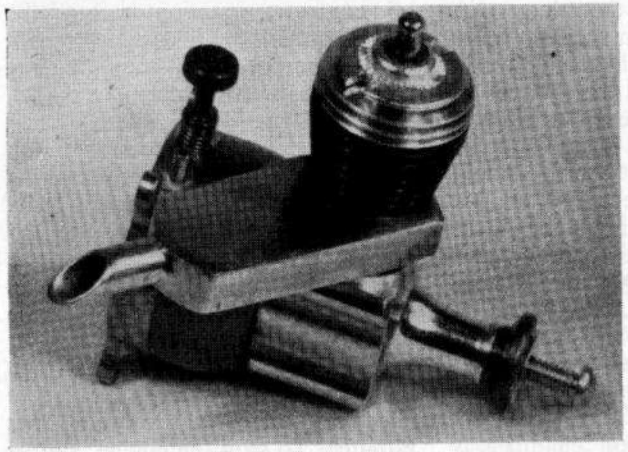
TABLE II. LENGTH OF COPPER-NICKEL RESISTANCE WIRE FOR .166 OHMS RESISTANCE

Size (s.w.g.) ...	20	21	22	23	24	25	26	27	28	30
Length (ins.) ...	8.75	7.0	5.3	3.9	3.3	2.7	2.2	1.85	1.5	1.05

TABLE III. TYPICAL PERFORMANCE DATA "STANDARD" (WOOLWORTH) FLEX LEADS

Price	Wire Size	Insulation	Voltage at Plug* 4 Yards Flex	Nominal Length for 1.5 Volts at Plug	Recommended Lead Length (yards)
2d.	7 Strands .0076	Plastic	0.5	1 yard	1½
4d.	14 Strands .0076	Rubber and Plastic	0.95	2 yards	2-2½
5d.	14 Strands .0076	Rubber and Braid	0.95	2 yards	2-2½
6d.	23 Strands .0076	Rubber and Braid	1.2	2½ yards	2½-3

* 1.95 Volts at Battery on Load



SILENCERS IN SILVER SOLDERED BRASS are made down to the smallest size shown here for the diminitive Cox "Pee-Wee" .32 c.c. engine. Known as the "308" silencer it is a combined expansion chamber and manifold.

fact have occurred instead of the pre-supposed rich start. If the engine bursts into life and dies out, firing only the prime given through the ports, then obviously it is not getting enough fuel and the needle valve should be opened. If the engine still does not get the fuel through with the needle valve wide open, then there is a blockage in the supply from the tank, either a restricted ventilator preventing air getting into the tank and causing a vacuum, or perhaps a blob of solder lying loose and getting sucked across the fuel supply tube. Both are common complaints, rarely diagnosed by the beginner.

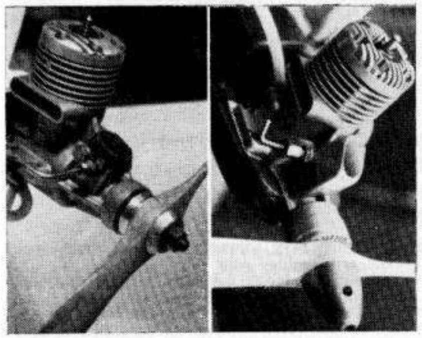
Run the engine rich for the first 30 minutes of its life, and after the first minute or so, stop it and tighten the head bolts while still warm.

If the engine just "plops" each time it is flicked, it needs an upper cylinder prime in most cases, but there is also the chance that the crankcase is flooded. The engine is then in a very rich state and will not clear until the plug is glowing bright, and the engine fires out some of the excess. Shut the needle off, connect the plug to battery and listen with the exhaust port open, to the plug sizzle. If silent, look at the wet reflection on the piston crown, and you should see a glow; if none, and no sizzle, then the plug is "out" and should be removed for examination. If it has burned out, a discouraging fact which is signified by a crumpled appearance of the coiled element, then the only answer is replacement. If it appears to be whole then the connections are at fault. Whichever has happened, examine the cause and effect so that you can profit by mistakes.

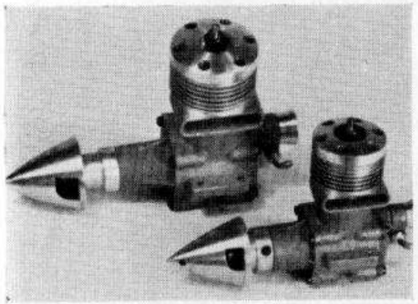
A fouled plug can give the same impression, as for example one of long-reach type touching the piston.

There are also hot plugs or cold plugs. Makers vary wire thickness and length according to the engine In America, many engines are supplied with special plugs and call for different fuels. These are usually the small .8 c.c. (.049) units known as the ½A class.

The differences between hot and cold plug requirement are decided by engine compression ratio. Low compression calls for a hotter glow, being provided by thinner gauge wire. High compression needs less glow to absorb less heat units and also to stand up to the compression itself. Such a cold plug may need to have the batteries connected for a longer period until there is no rev. drop on disconnection. This fault may persist, and will not therefore be entirely the cause of the trouble, so the fuel should be treated as suspect. A glow engine must have its special fuel, and cannot be expected to run on the mixture offered for diesel or spark ignition. If the correct fuel is being used, and the engine still will not run without the plug being boosted by the battery, then the compression ratio is not high enough and the cylinder head should be lowered if at all possible.



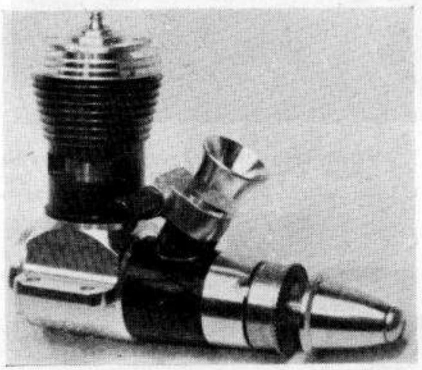
RIGHT IS THE COX SPECIAL 15 Mk. II WITH single exhaust port, thick cylinder and revised cylinder head contour.



Rough running indicates low compression and this is especially the case when a plug is well recessed in the head and the element shielded from the combustion chamber as designed. Manufacturers have found amazing gains in speed and power simply by moving the plug around the head experimentally. Most "special" engines use a solid unfinned head, with plugs at fancy angles and in all sorts of positions. One will find power with the plug over the transfer side, others on the exhaust side!

In the case of some mass produced miniature engines, the beginner's main problem is usually that of applying sufficient power in the

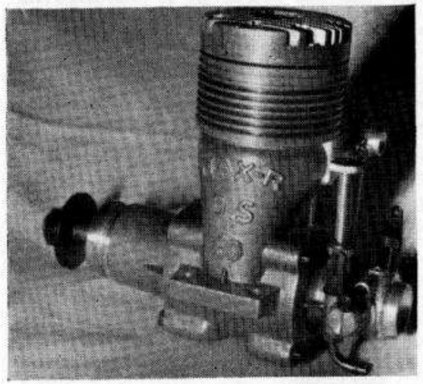
MOST EFFICIENT OF THE 2.5 c.c. glow engines is the Super Tigre G.15. At left are specials (note non-standard carbs) seen in the 1965 World F/F Championships. That at right was possibly most powerful at the contest in Finland, was prepared by Benno Schlosser.

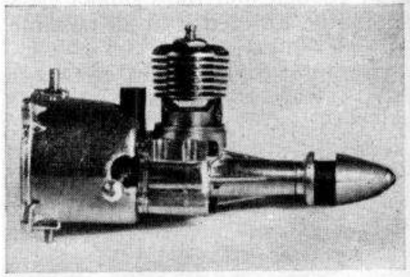


THE K & B ENGINES PRODUCED UNDER the supervision of World Speed Champion Bill Wisniewski have a family resemblance, obvious in the 15 and 29 here.

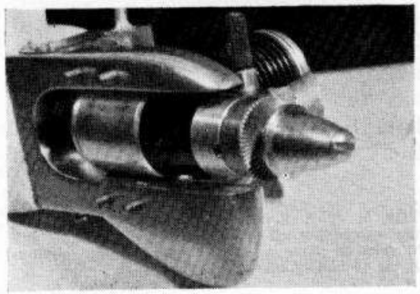
flick of the prop. to get the engine started. A tip we can advise is to inject a spot of machine oil on the piston crown when the exhaust is open, and so helping to seal compression. The engine should already be wet with a fuel choke, otherwise the plug may be fouled by the oil and does not glow enough to fire on those vital first few flicks after injection, but in most cases, machine oil will come to the rescue of any baulky engine.

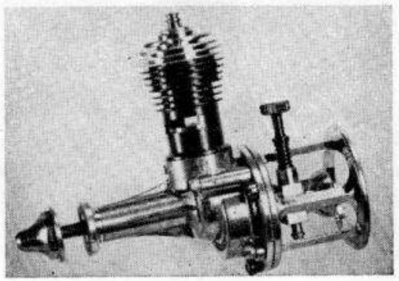
Until now, we have been considering only the starting side of glow. The most desirable feature of this form of ignition is that it is self-timing. There is no question of having to set the make and break for advanced or retarded position and we do not have to alter the compression as in a diesel. When the fuel compresses in the combustion chamber, it meets a glowing element and the firing is smoothly spread over the precise timing required according to the load. A big propeller brings the r.p.m. figure down, and the combustion takes place in a retarded position to suit. On a small prop., the revs. obtained from most glow engines are such





PRODUCED FOR EXTREME POWER IN SPITE of weight penalties with tough castings, the advanced OS 60 design has many unusual features. This is one of the 1965 prototypes for Radio Control. Above is the Ohlsson Midget .8 c.c. with "feather valve" and an incorporated tank, a very small engine. Bottom left is a Cox Special 15 mounted in a specially cast bearer which is also the landing skid for Frenchman Landeau's model. Below is the L. M. Cox "Thermal Hopper" a forerunner of today's successful small glow engines.

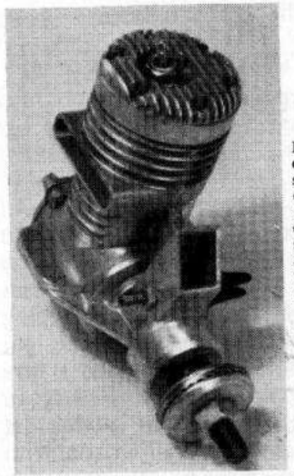




that combustion could only be advanced.

It is here that the glow motor has special applications. Its flexibility allows it to take a high speed model off the ground, gain up to 4,000 r.p.m. in the air and use the full power of the engine without the operator having to worry about re-setting the needle valve. A diesel will not do this and petrol/spark/ignition just does not have the power, unless in extra special magneto equipped engines.

This advantage has been explored by many manufacturers in turning out products which will provide their maximum performance high up in the r.p.m. scale. For this reason, they have intake or induction ports which are not favourably inclined to slow speed starting, and possess little suction. Such motors are best started with a finger partially blocking the intake, sensing the exact amount of choke needed until the power is built up to a point where it continues to gain speed after the needle valve is closed (there are few engines which only run with a pressurised fuel system.

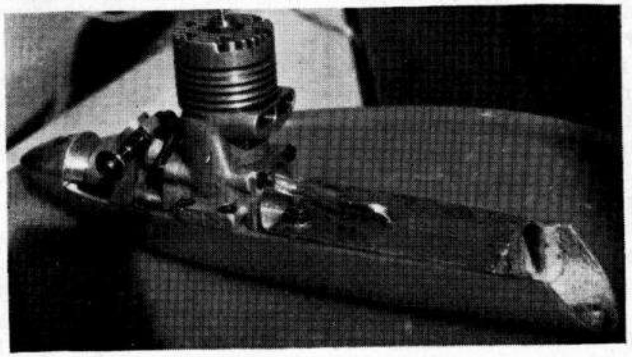


DUKE FOX'S characteristic square intakes, offset glow plug and crank-case finning are seen in the Fox 36xBB — a very powerful unit.

These are of course only for experts and are employed for speed work only).

If ever in difficulties, check the plug first. Then see if a crankcase leak is preventing the motor picking up from just an exhaust prime, followed by an examination of the fuel flow and jets in the needle assembly, that is all one needs to consult in 99 per cent cases of glowplug engine failure.

REARWARD FACING EXHAUST ON THE MVVS 2.5 RL (COMPARE WITH "production version" on page 96) as used by Czech National Speed team flier and designer J. Sladky. Metal tank and cast pan mount are units in a well engineered system.



CHAPTER FIFTEEN

Operating Spark Ignition Engines

TO newcomers the spark-ignition engine will be something of a mystery. Few are sold now that compression ignition and glowplug have simplified model engine design, but the spark motor remains a favourite for "heavy" duty, such as in a large radio controlled model. It is complicated by comparison with modern counterparts, though very satisfying in its operation which is by far easier, cleaner and more controllable than either diesel or glow.

The engine has an ignition system identical to that of many cars and motorcycles. It runs on a mixture of petrol (gasoline) and lubricating oil grade SAE 70 (medium to heavy grade) in the ratio of three parts petrol to one of oil. For ignition, there is a miniature sparking plug in the cylinder head which is electrically connected to a high voltage coil and interruptor or timing gear on the engine, plus batteries either 3 v. or 4.5 volt.

To start, one invariably employs a booster battery so that the flight batteries are not drained, and these are so arranged that the flight timer acts as a switch and transfers the power from booster to internal batteries when actuated ready for launching—*Fig.* 1. In the circuit we also have a condensor which consists of a series of tinfoil sheets separated from one another by an insulated wrapping of wax paper. The even numbered tinfoils are connected to one end and the odd numbers to another terminal so that we can position the condensor in circuit (average value .05 mfd.) between the earthed engine bearer and the insulated or non-moving point on the distributor or contact breaker. The purpose of the condensor is to absorb the self-induced current of the primary circuit, thus allowing the magnetic field to collapse as quickly as possible and eliminating to a great extent, sparking at the contact points. Without a satisfactory condensor, we get very intermittent running, hard starting and arcing of the contacts to the extent of burning the faces badly.

The coil is a more important component, for it boosts 3 or 4.5 volts one thousandfold so that a high tension spark can jump the plug gap in the cylinder under compression. A weak coil, or weak batteries can only result in good sparks out of the cylinder, none whatsoever in the place where needed!

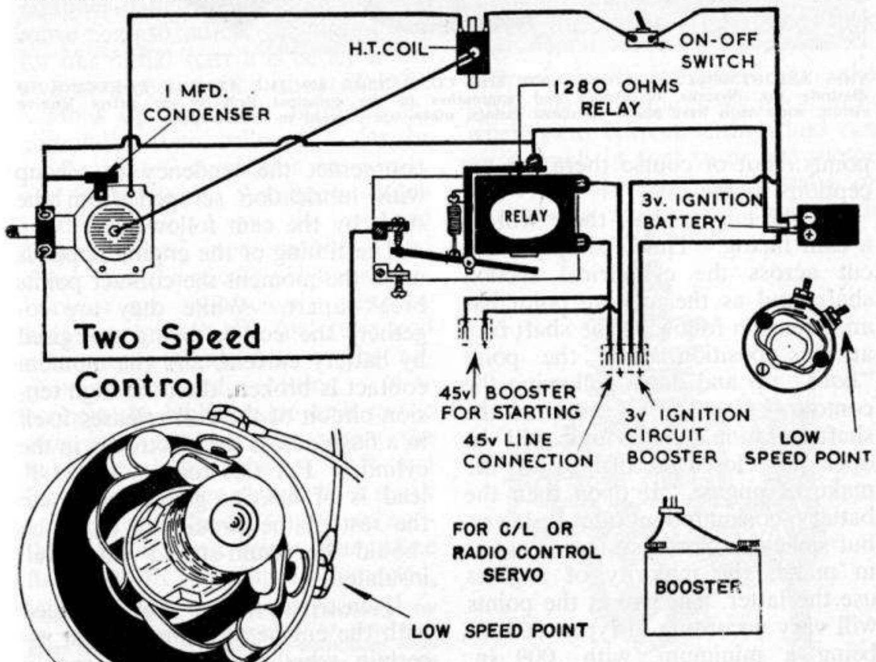
The coil consists of a core made up of laminations to prevent them becoming permanent magnets, and multiple windings of insulated copper wire built up to a diameter of about 1 in. and length about $1\frac{1}{4}$ in., or even longer. The coil is a heavy item, and in order to conserve weight in duration models, the majority of spark ignition operators were obliged to use lightest possible batteries of the pen-cell type. For

radio control, weight is less critical and batteries with larger reserve can be used.

There remains the plug as the completing accessory in the spark circuit and this consists of an electrode, conducting high tension from the coil, and insulated from the plug body by a porcelain body. The gap between the pointed end of the electrode and the body is adjustable by force, just as with the full size engine, and this setting can be critical. A setting of .008 in. is average but the actual gap varies according to the purpose of the engine, and indeed according to the efficiency of the coil.

On the engine, one very important component that determines the timing of the whole spark ignition system is the distributor, also referred to as the contact breaker, make and break, timer or interruptor gear. It contains a set of points which are usually made with tungsten facings, one of them following a cam on the engine shaft and known as the moving point, while the other is fixed and insulated. The insulated point is normally the one which has screw adjustment for the gap between

FIG. 1.—THIS CIRCUIT INCORPORATES ALL THE POSSIBILITIES LIKELY TO BE INVOLVED in a coil ignition system. For the basic circuit, eliminate the two-speed control side and relay, complete with 45 volt booster lines. For free-flight, eliminate the on/off switch which is replaced by a flight timer. The 3 volt ignition circuit booster should always be retained to save flight battery drain during the initial starting period; 45 volts are applied for control line only, being the necessary allowance for voltage drop along 50 ft. of insulated lines from the pilot's handle to the model.



THIS ASSORTMENT OF COILS AND THE CONDENSOR IN THE CENTRE FOREGROUND illustrate the diversity of shapes and approaches to the principal item in the petrol ignition circuit, some coils have plastic insulated casings, others are encased in a fibre tube.

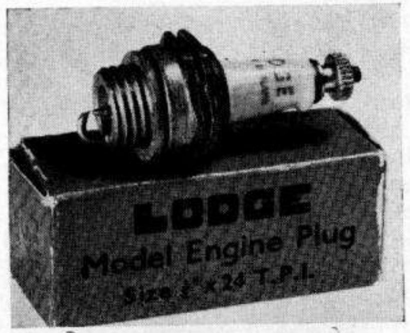
points: but of course there are exceptions to the rule.

On the engine shaft there will be a cam facing. This is simply a flat cut across the cylindrical section shaft, and as the moving point has an extension following the shaft face at this position, then the point "bobs" up and down following the contour. For 320° or more of the shaft rotation, the points will be open or closed according to the make of engine. If open then the battery consumption rate is lower, but since the closed system is easier to make, the majority of engines use the latter. The gap at the points will vary according to type, .005 in. being a minimum, with .009 in more suitable on most makes to counteract the tendency to oil up with lubrication scraped from the shaft by the cam follower.

The timing of the engine depends upon the moment the contact points break apart. While they are together, the coil is being energised by battery current, and the moment contact is broken, then the high tension circuit of the coil releases itself in a flash across the electrodes in the cylinder. For this reason, the H.T. lead is of heavier gauge wire than the rest of the circuit, all of which should be multi-strand and well insulated.

If instructions are not provided with the engine, one should first ascertain whether the points are in fact breaking correctly, and at the

TYPICAL SPARK IGNITION PLUG IS THIS approximately actual size Lodge ⅜, a true miniature of the full size plug.

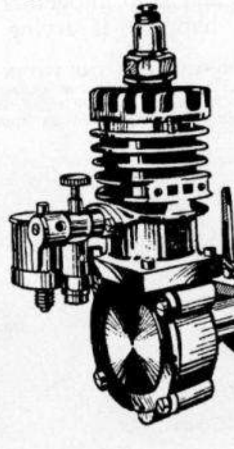


right moment relative to the piston movement. This can be decided by removing the plug so that the engine can be turned over easily, with the points visible. Rotate the shaft so that the piston is on the up stroke and place a piece of balsa strip through the plug hole, resting one end on the piston top so it forms an indicator. Mark a line on it flush with the outer cylinder head when at T.D.C., and turn over slowly in the normal anti-clockwise direction, watching the points. They should break when the piston is just before the top dead centre position. If at exact T.D.C., then the points are retarded, if long before T.D.C., then they are well advanced. Point assemblies usually possess an arm for movement to adjust the timing, so for our initial start it is better to arrive at a setting just about T.D.C.

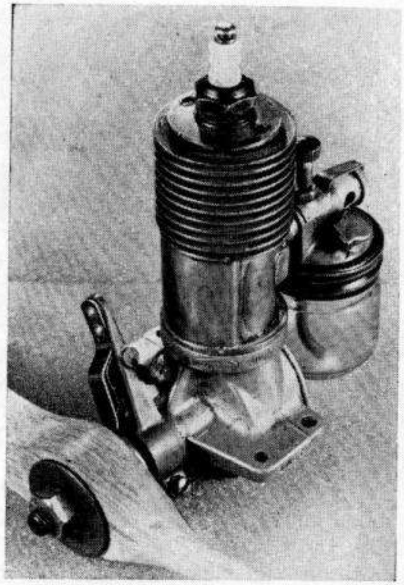
Hook up the circuit, using an accumulator (two cells of a car or motor-cycle accumulator are ideal), and fill the tank. Arrange a switch in the circuit to save the coil and battery when not running, and check that the needle valve and carburettor are satisfactorily clear by blowing through the fuel tube and obtaining a gentle hiss at the approximated needle setting. Now connect the high tension lead to the plug, and lay the plug across the head. Switch contact "on" and flick over. There should be a spark at the plug sufficient to give an audible "crack" and bright blue/white flash. With the plug disconnected, the spark should jump a ⅜ in. gap to earth on the engine, and will give a shock through the arm of anyone holding both engine and the H.T. lead.

Fit the plug, hook up leads, and as a last check, one can oscillate the prop. around T.D.C. piston position and hear an echo of the spark in the dry cylinder. Now prime the exhaust port and flick over.

Spark ignition engines have the advantage of firing first time if anywhere near correct settings and run on a smell of fuel so as the motor picks up, open the needle valve or just give a momentary choke with



EARLY PETROL IGNITION ENGINES OF pre-war years were characterised by their large plugs and small ports. Note the particularly small exhausts on this version which could be built from kits of parts.



AN ENGINE WHICH REALLY SET POWER models on the road to success was the Brown Junior which appeared in various forms, this one having the later type carburettor with true needle valve and added facility, a carb. choke. It has a long shaft, long con. rod and its light weight made it an easy to start favourite with many modellers.

a forefinger on the carburettor, and the motor will roar away.

If retarded, there may be a tendency for slow running with smoky exhaust, so push the advance lever over and speed up things with a better timing. If one takes the ignition point too far advanced, the motor will backfire and stop altogether.

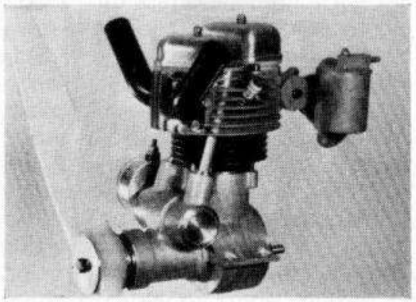
This also happens if trying to start with the engine too far advanced and the backfire often loosens the prop. nut which can be the source of starting difficulty. Always ensure that the prop. is bolted on really tight. Symptom of a loose fit is a "crack" in the backfire note.

A petrol engine will run with fairly poor compression and piston seal but must have a good spark. Most of the starting problems will be centred around the electrics and it is as well to familiarise oneself with the circuit to be able to check back on any possible fault. Misfiring at high speeds can be due to oily points and insufficient gap allowing "bounce". A loose H.T. lead on the plug gives the same symptom, while an oiled up plug (which happens often due to the 3 : 1 Petroil mixture) can be detected by visible sparks outside the plug down the porcelain insulator.

The easiest way in which one can check the condensor if it is suspected, is to carry a new spare all the time (they are cheap accessories) and double up on a test run with the new one in circuit, the other temporarily de-earthed or disconnected from the insulated points. The difference in running will soon show if the condensor is working satisfactorily.

To check the spark points, arrange to have them open and with the plug lead fitted, ignition "on", short circuit with a screwdriver between the moving and insulated points. Touching the cylinder with

THIS FINE FOUR-STROKE IS THE WORK OF D. H. Chaddock. A 5 c.c. push-rod design, it drives a 10 x 6 in. propeller at 9,600 r.p.m. Twin exhaust stacks and valve covers add an appearance of power.



one end while the 'driver rests on the insulated point will suffice. Inside the cylinder, one will hear the spark jumping the plug gap if the exhaust is just open, and this indicates that any faulty running must have been due to the points being burned or dirty.

Clean with a fine wetstone as used to sharpen knife blades. In emergencies a piece of fine sandpaper would do, and a lady's manicure file consisting of sandpaper fixed each side of a card finger, should be part of all spark-ignition operators' tool kits.

Battery check by torch bulb is useless. One can use discarded batteries for the torch, and the only safe means of a battery check is the ammeter, which most radio control enthusiasts happen to possess, but which falls outside the scope of other modellers. When batteries were scarce, all sorts of means were taken to rejuvenate the dry cells Heating in the oven for an hour works wonders, but the cheap dry cell cannot compete with the Venner type dry accumulator for amp-hours and these are thoroughly advised for anyone intending to use spark ignition extensively.

One more point arises. Most spark engines are older designs, created before minor mechanical problems were overcome in engine construction. One of these is the carburettor system. Earlier carbs were made for attached tanks and unless a special leather gasket made to suit was fitted on Ohlsson engines in particular, the air leak between the carb and the spraybar assembly spoiled the carburettion. This is overcome if a separated tank is used and the spraybar made a tightly sealed fit where it passes through the tubular intake pipe.

To eliminate the oil splash on engines with exposed contact breakers such as O.K. engines, Anderson, Atwood and Cyclone, it is possible to solder a small tinplate oilguard on the moving point arm.

Despite the intricacies of electrics, the spark engine is more easy to handle than many of its counterparts and will always find a place for itself in aeromodelling.

MOST RENOWNED OF ALL petrol ignition engines, are the Ohlsson Marque, here a "29" (5 c.c.) is being started with slight finger choke in a pylon model. Note the ignition circuit lead to the make and break behind the spinner. Ohlsson engines were unique in having the cylinder liner spot welded to the alloy crankcase and this later version had an alloy head also attached to the cylinder making it virtually impossible to dismantle the liner, apart from removing the front rotary housing and extracting the piston by angling the con. rod.