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MODEL AERO ENGINE ENCYCLOPÆDIA

COMPILED BY
R G MOULTON

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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 spray-bar 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

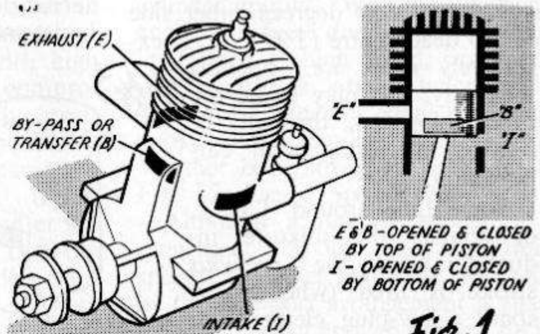
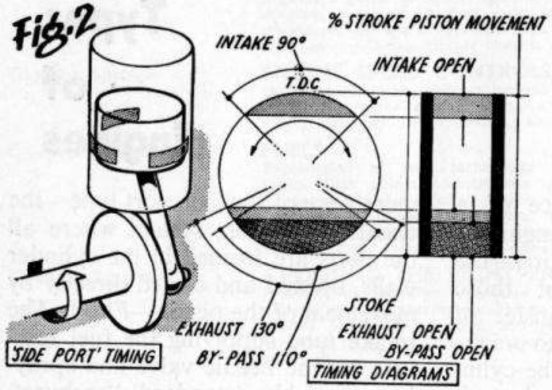


Fig. 1

Fig. 2



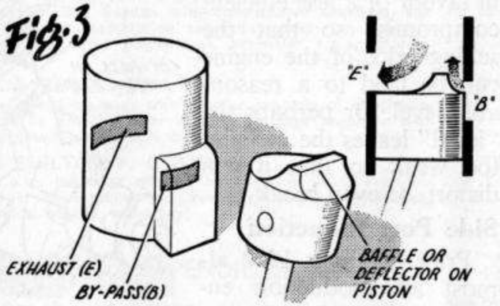
How It Works

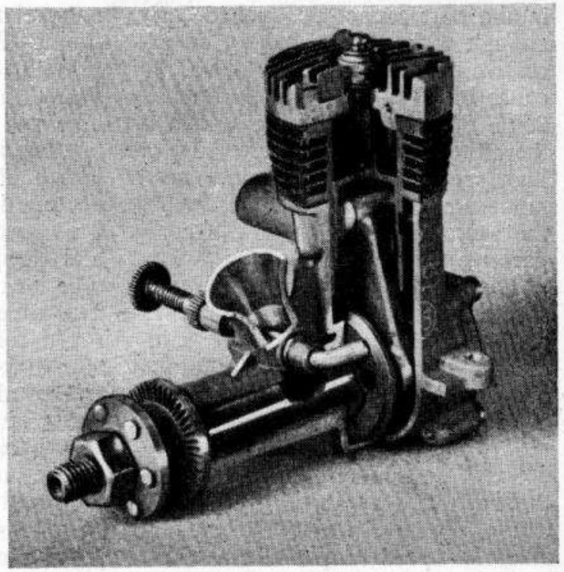
Approaching the top of its stroke the piston must uncover the intake opening or port so that the lowering of pressure produced in the crank-case can draw in the fuel mixture, *i.e.*, the intake port is so positioned that it is uncovered so many degrees of rotation, or a specific fraction of the stroke, 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

Fig. 3





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.

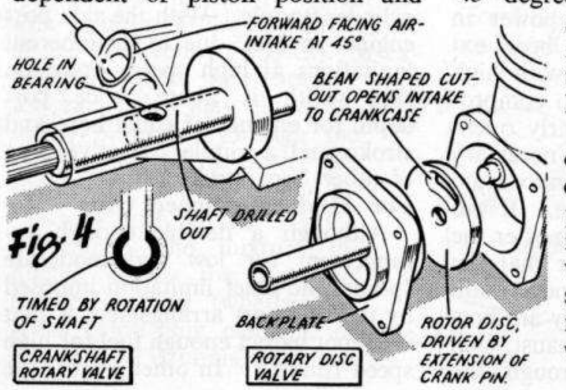
Rotary Valve Induction

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.

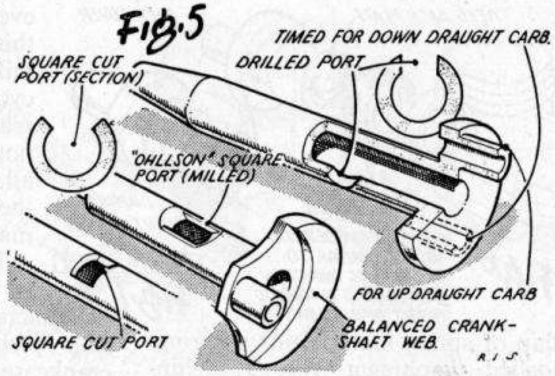
With the CRANKSHAFT ROTARY VALVE the port is a round or square hole cut in the crankshaft itself, opening into a hole drilled along the length of the shaft (and thus connecting directly with the crankcase). This port is timed by its appearance and disappearance past the intake tube let into the crankshaft 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.

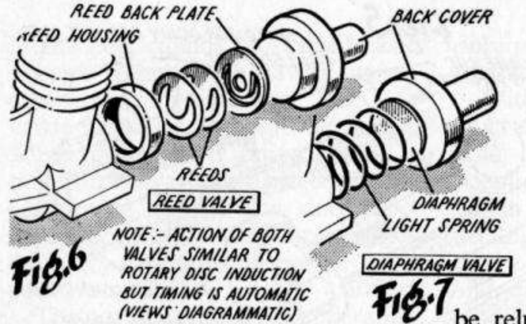
ROTARY DISC type of induction is virtually unlimited as regards timing at the expense of being a more critical proposition mechanically. It is quite obvious that to increase the opening it is only necessary to increase the length of the slot in the rotor disc, without any resulting weakening of stressed 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

A*



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.

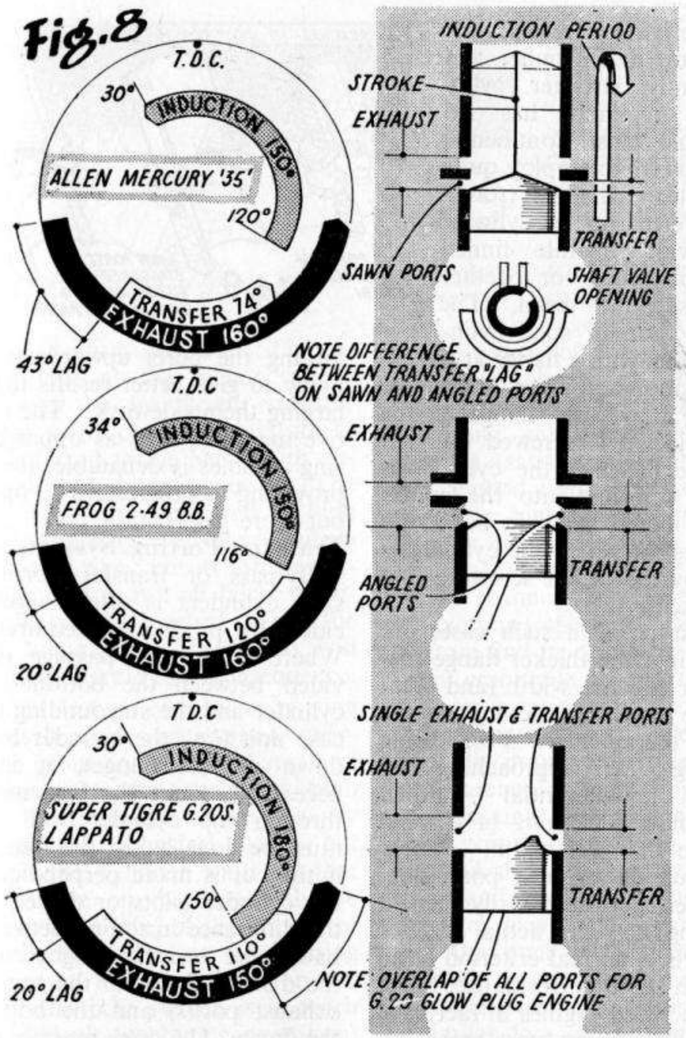
Limitations are the fatigue life of the spring material under operation stresses and the inertia of the system. The conventional reed valve consists of a flap of beryllium copper or phosphor-bronze of about .002-.004 in. thick (depending on size). How greatly this is stressed can only be guess estimated. 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-



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 by-pass 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-

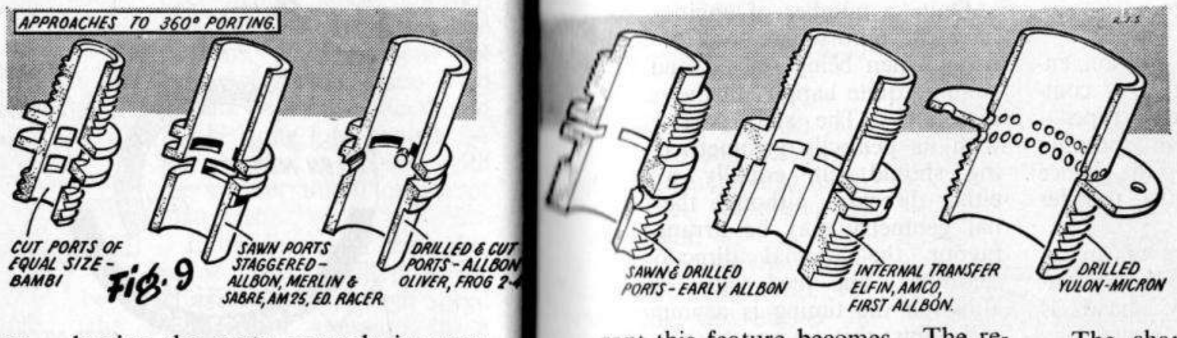


Fig. 9

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 signifi-

cant 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.

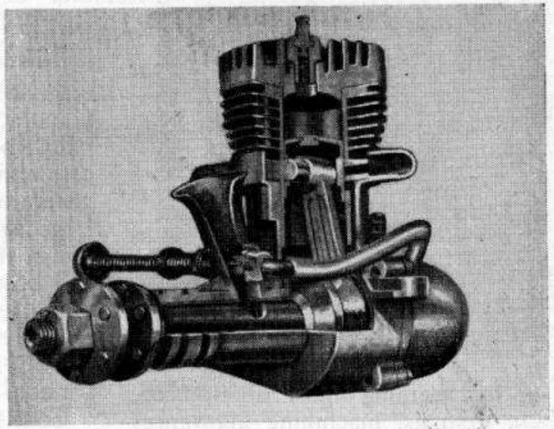
Starting Technique

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.

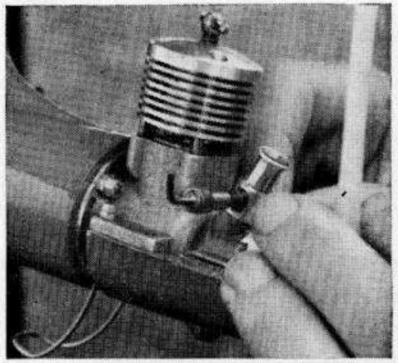
But for the man with the necessary skill, reworking an engine in search of that little extra in performance can often be an attractive and interesting, proposition. The mass-produced commercial engine, is, after all, a compromise between design for performance and design for production, with particular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

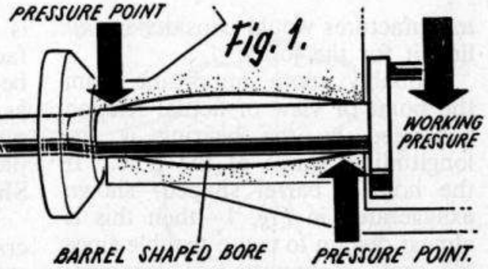
The mating fits we are most closely concerned with in model engine manufacture are the crankshaft-main bearing and piston-cylinder assemblies. These are the main generators of friction which to a large extent govern the power output of the engine. The big and little end bearings on the connecting rod (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.

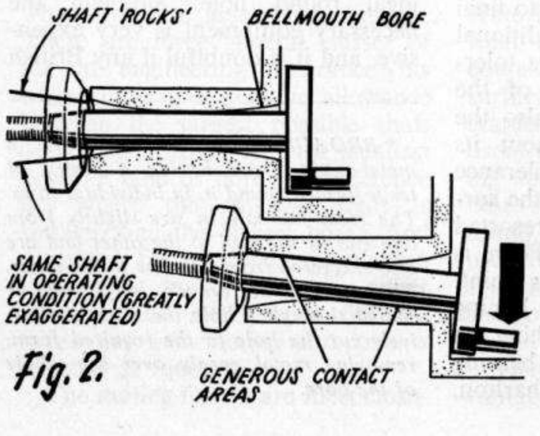


Fig. 2.

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.

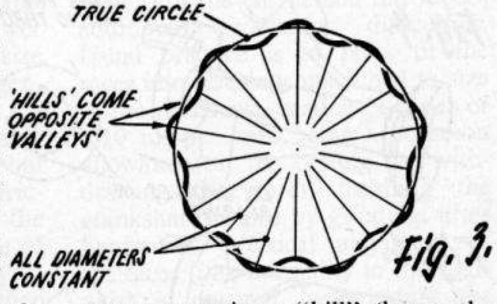


Fig. 3.

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.

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.

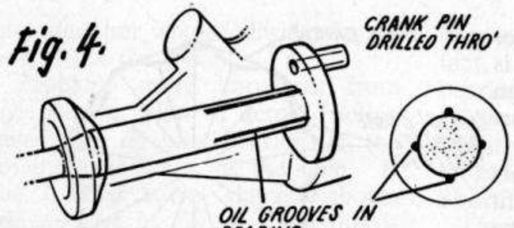
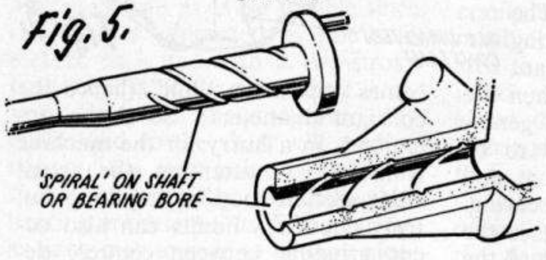


Fig. 5.



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

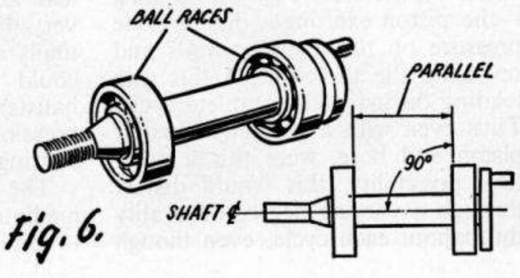
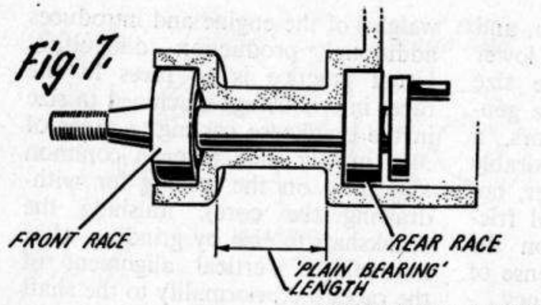
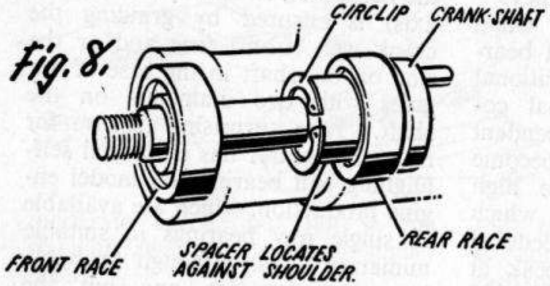


Fig. 6.





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.

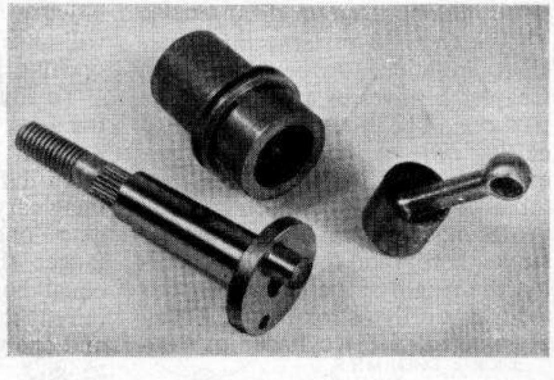
The ideal is obviously an intermediate fit. A piston loose enough to minimise friction at the operating

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 indlcation 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.

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



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.

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.

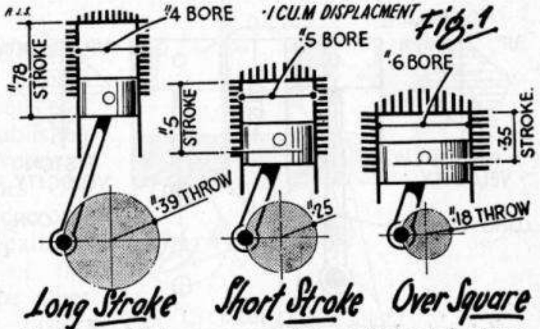
From the previous comments regarding centreless grinding, it will also be seen that a ground piston is not necessarily truly circular, although would indicate so with micrometer measurement (the jaws spanning the minute "hollows"). Pistons ground by other methods should be circular as finished (provided 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.

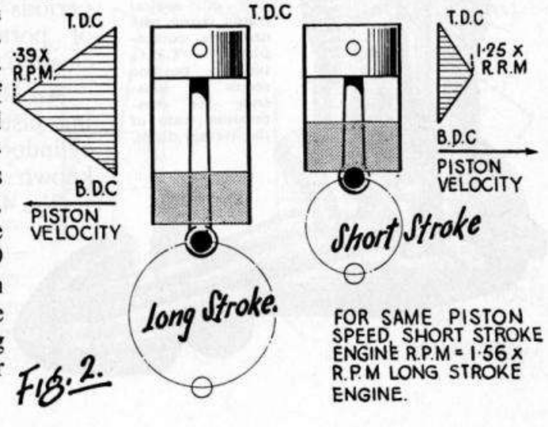
To illustrate the effect of varying the sizes of the bore and stroke for a given capacity we can take the three different arrangements for an imaginary 0.1 cu. in. (1.6 c.c.) engine—one with a stroke appreciably longer than the bore; one with equal bore and stroke (usually referred to as a "square" layout);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



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

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-



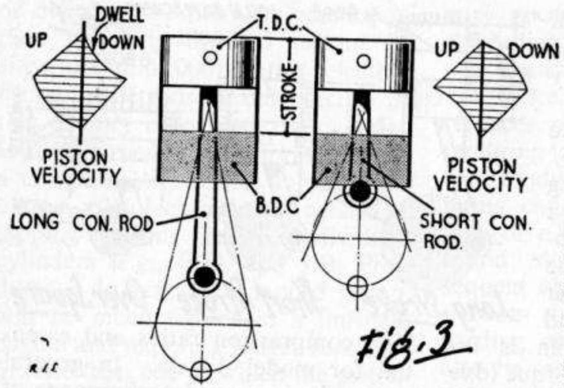
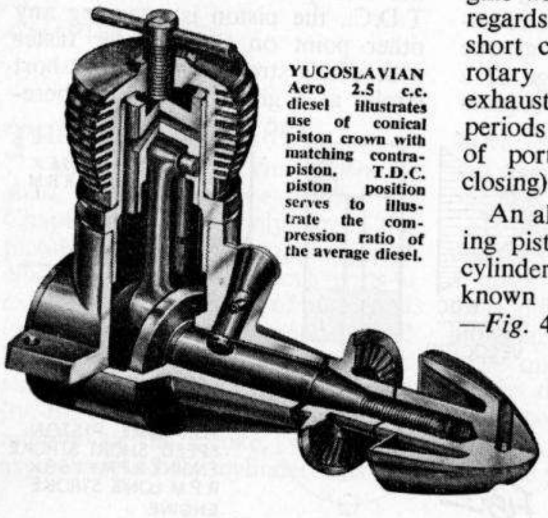


Fig. 3

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.

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).


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.

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).

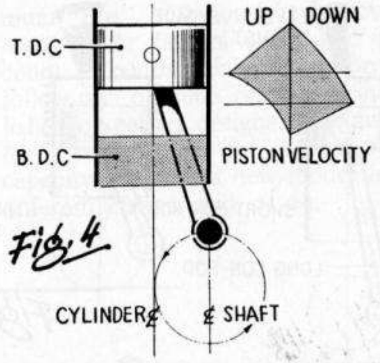


Fig. 4

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

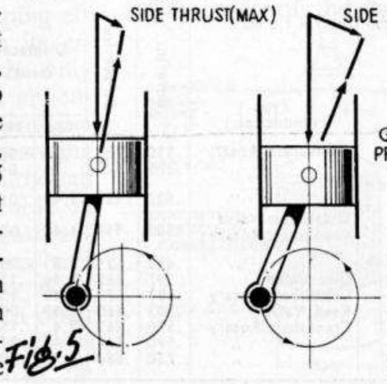
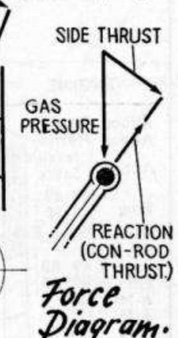


Fig. 5



Force Diagram.

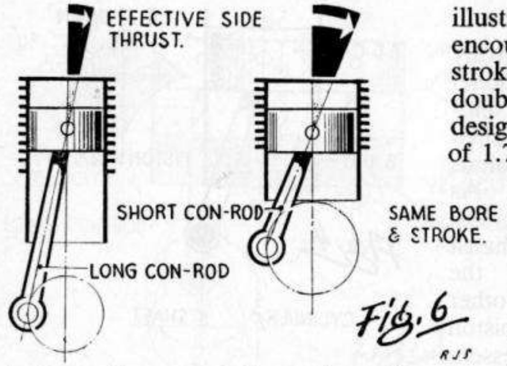


Fig. 6

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 $\frac{3}{4}$ x diameter.

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

TECHNICAL DATA ON BORE/STROKE OF BRITISH ENGINES

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

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

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.

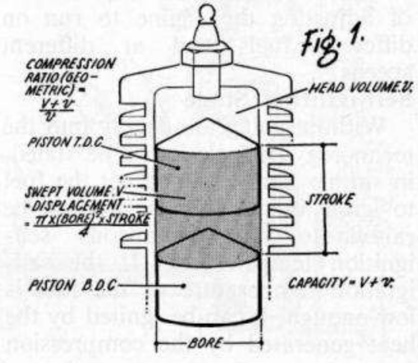
CHAPTER FOUR

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

Fig. 1.



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.

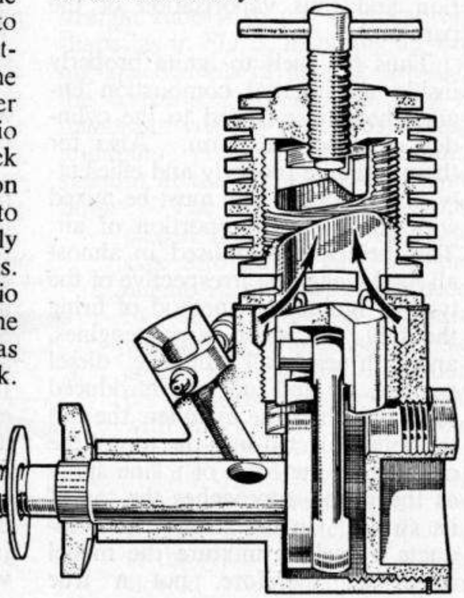
In the case of glow motors, the position is rather different. The base fuel is methanol, which has a self-ignition temperature nearly twice that of diesel fuel and nearly three times that of ether. Hence a heated element is required to ignite it. (It is an interesting fact that an ether-methanol mixture *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

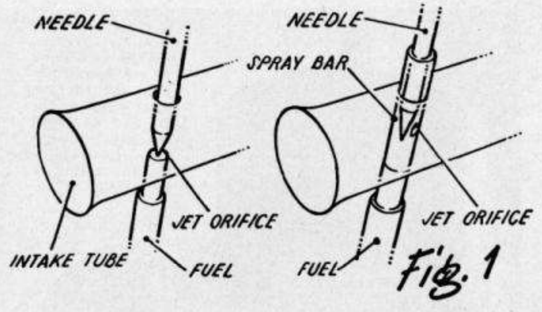


Fig. 1

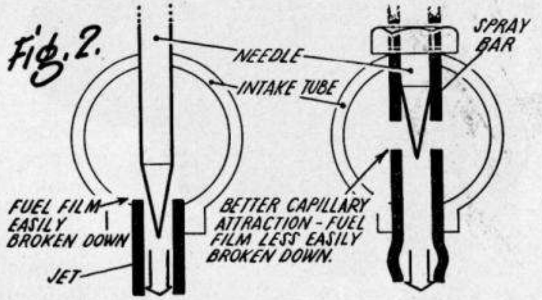


Fig. 2.

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.

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.

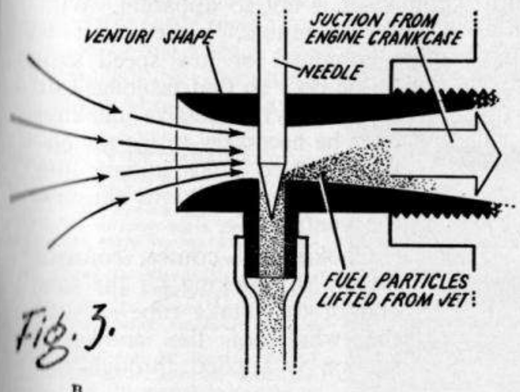
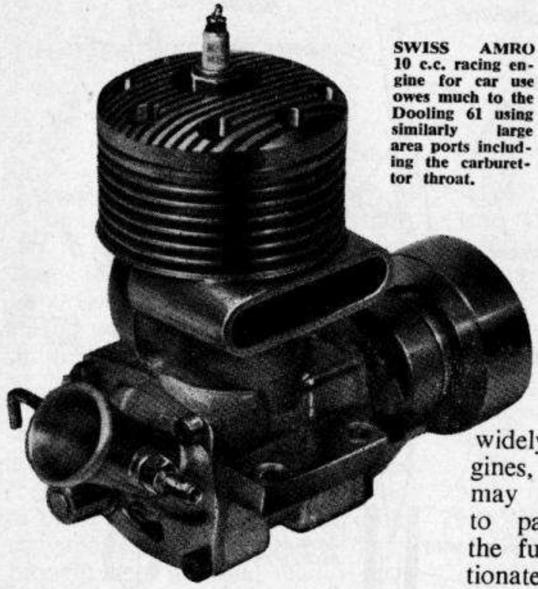


Fig. 3.

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.

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.

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.

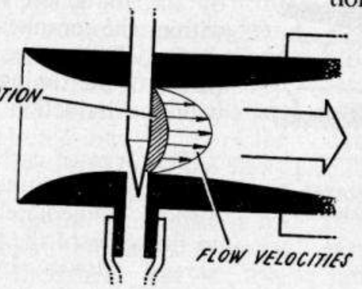
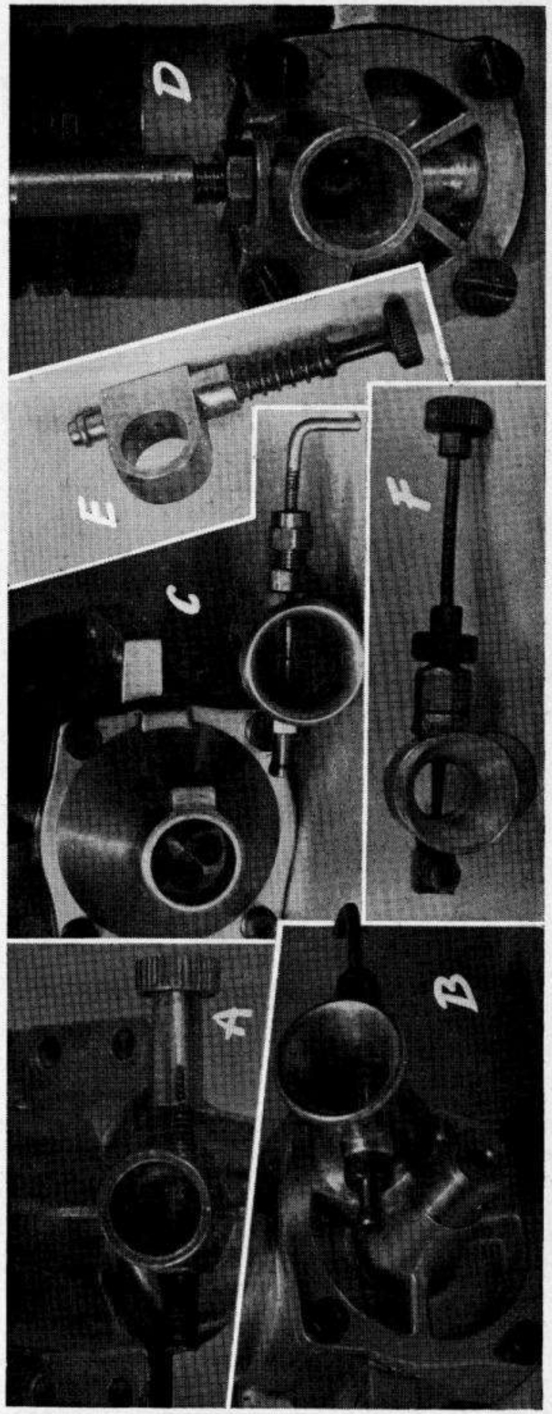


Fig. 4.

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.

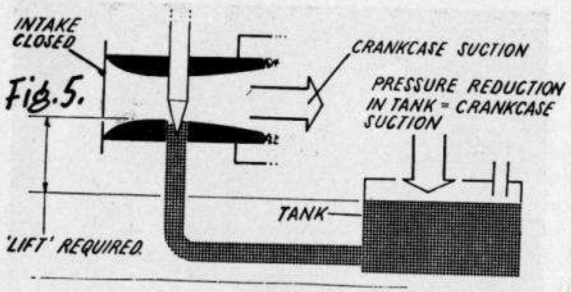


Fig. 5.

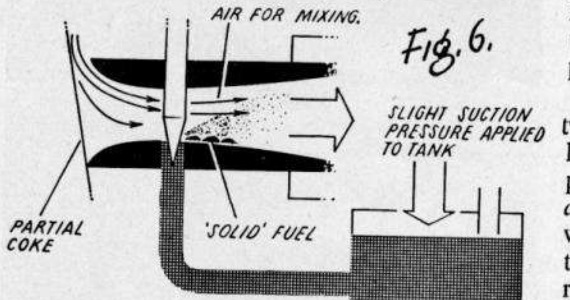


Fig. 6.

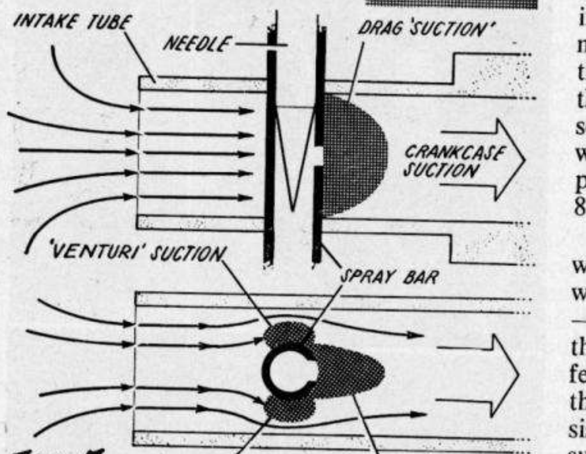


Fig. 7.

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).

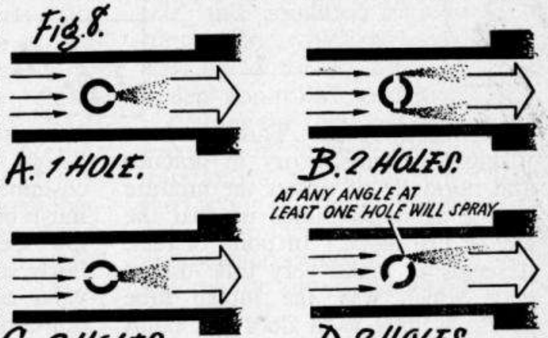
Locating the spraybar with one hole facing forward and the other back—*Fig.* 8 (c)—will mean that only one hole is effectively spraying, but this action will be assisted by a positive pressure build-up through the other hole. In any other position—*Fig.* 8 (d)—either one or both holes may spray, but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



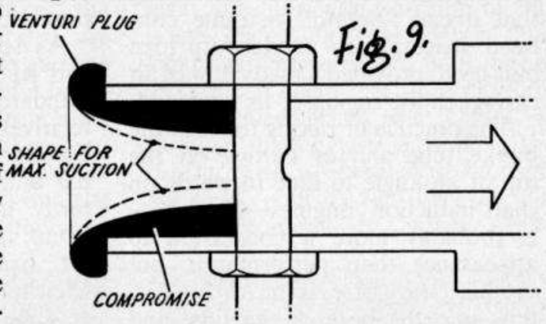


Fig. 9.

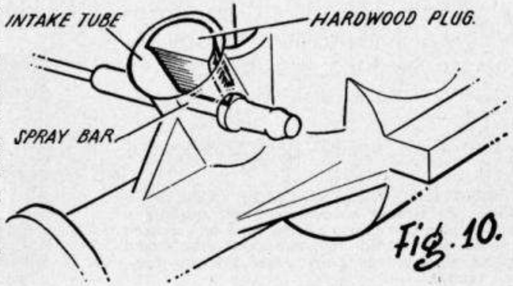
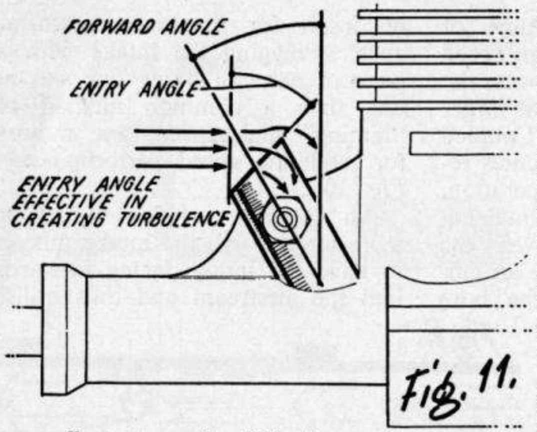


Fig. 10.



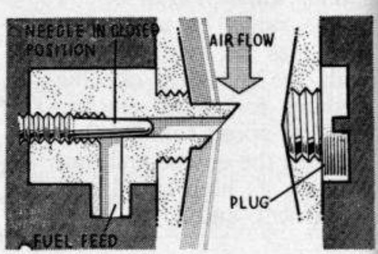
ram effect at speed. This, however, is highly unsatisfactory in practice and normally so upsets the mixture setting as speed builds up that the engine just stops. In point of fact, it seems to make very little difference which way the intake tube faces, as long as it does not point straight ahead. It 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.

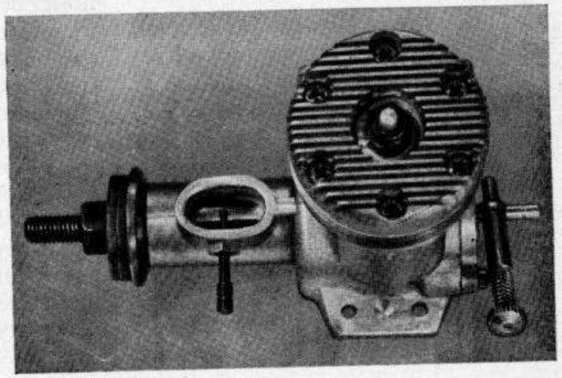
CHAPTER SIX

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.

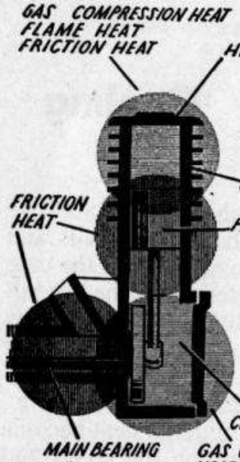


Fig. 1.

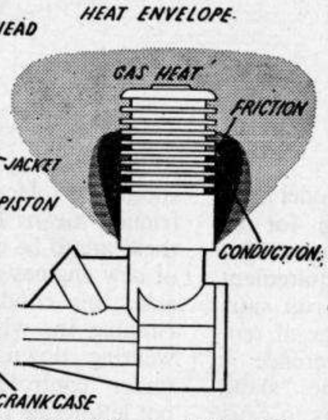


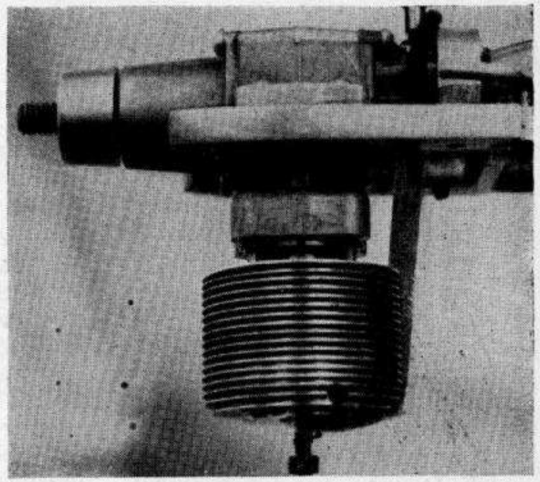
Fig. 2.

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.

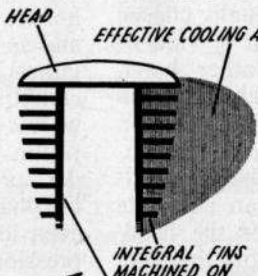


Fig. 3.

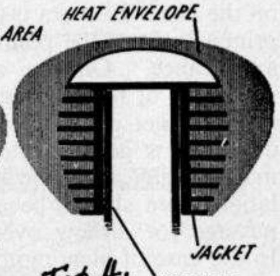


Fig. 4.

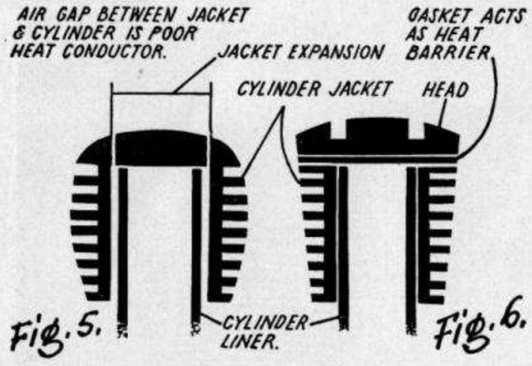


Fig. 5. Fig. 6.

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.

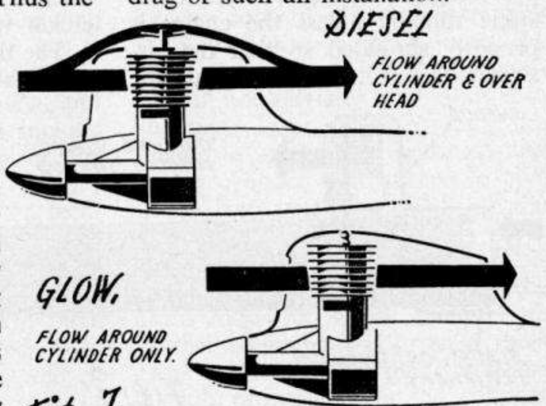


Fig. 7.

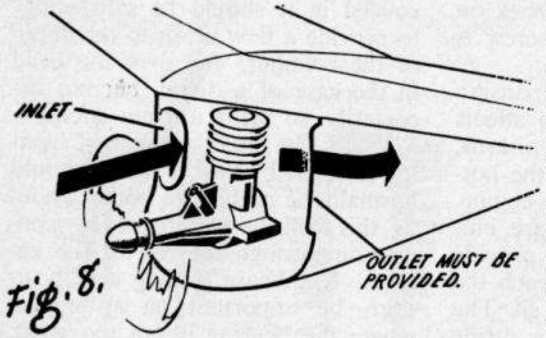


Fig. 8.

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 $\frac{1}{4}$ 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.

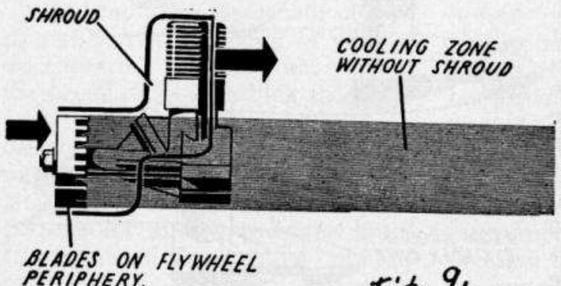


Fig. 9.

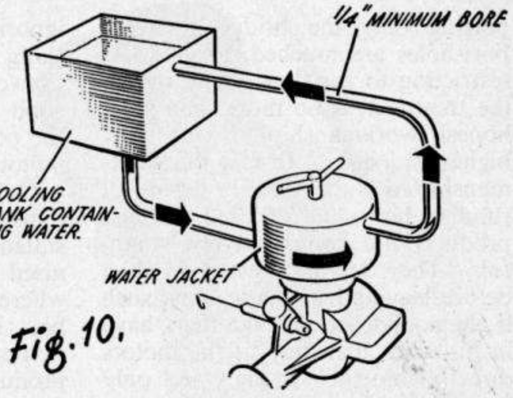


Fig. 10.

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.

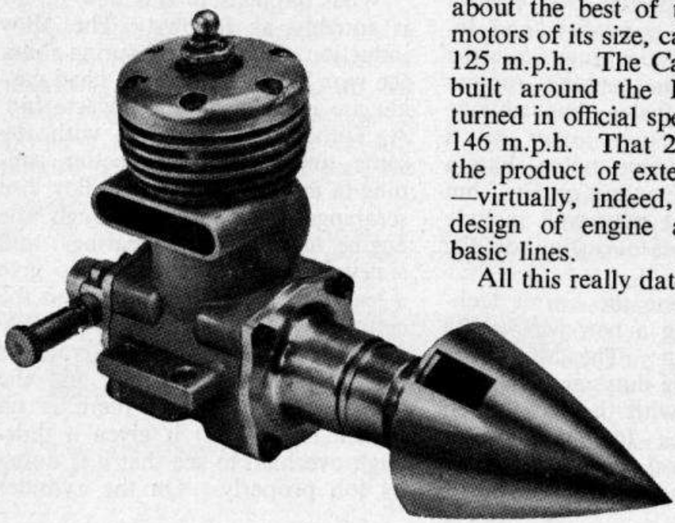
The short-life rear ball race is exchanged for a Hoffman and the front race (for which there is no British equivalent) is given a thorough overhaul to see that it is doing its job properly. On the cylinder porting only the bridges between port-holes are touched, to give less restriction to gas flow. 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.

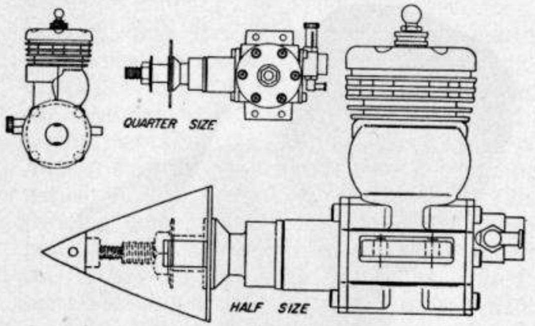
All this really dates back to about 1950 when Fred Carter was flying control line stunt, was being tempted by speed and had the feeling in the back of his mind that having found out a little about commercial engines and their limitations that he could do a bit better. Perhaps the contrast between a precision 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.



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

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.

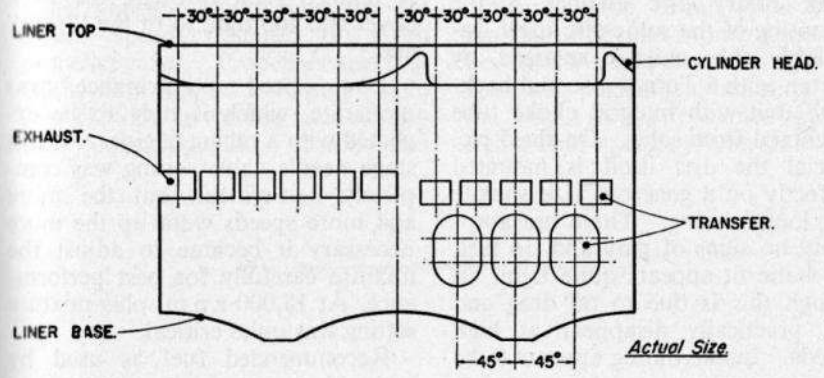
The original shrunk-in liner was removed and replaced with one made from Meehanite bar stock carried down to the full depth of the casting. Ports were filed by hand, duplicating the layout of the original but with a slightly later exhaust opening and slightly later transfer. (Shallower ports providing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

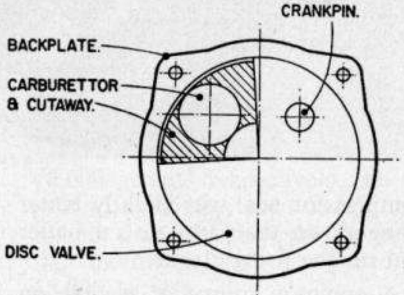
"UNWRAPPED" CYLINDER FOR CARTER Special shows how the Dooling ports are actually reduced in area to a degree, while transfer is opened up. This is a characteristic of most Carter engines.

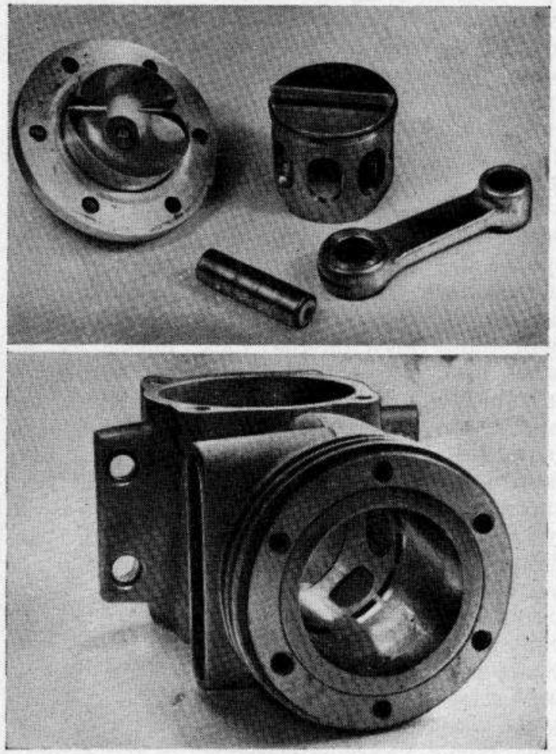


If performance does not come up to scratch, a new head and piston would be an obvious "second try" We get the impression, however, that with Fred Carter, second tries are a bit of a rarity.

"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.

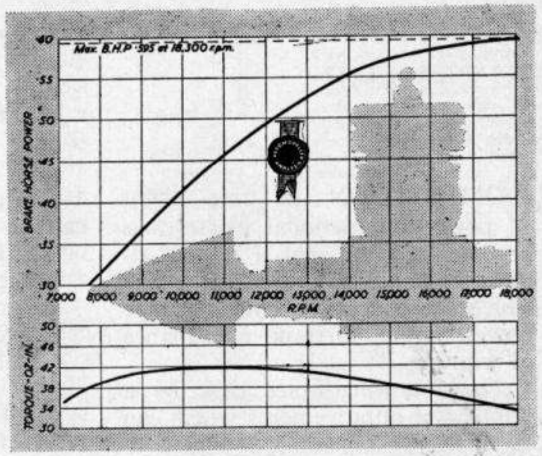
Low speed performance was moderate, which is only to be expected with a racing engine. At this stage needle valve setting was completely non-critical, but the more and more speeds went up the more necessary it became to adjust the mixture carefully for best performance. At 18,000 r.p.m. plus mixture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

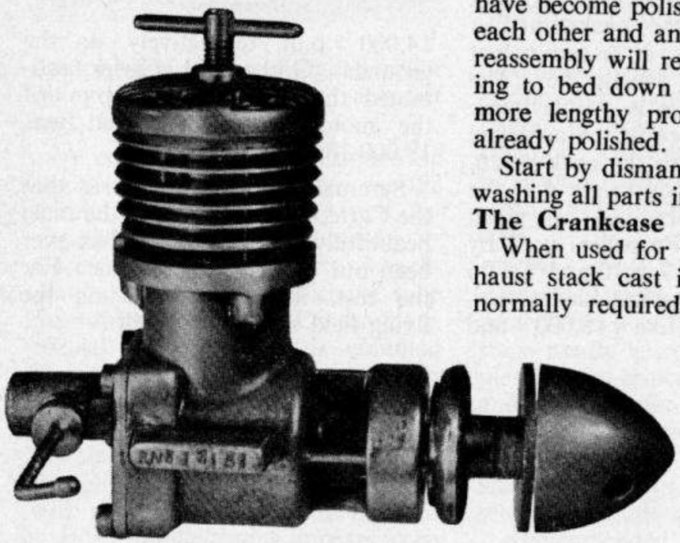
(5) To decrease vibration by improving the balance of reciprocating parts.

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

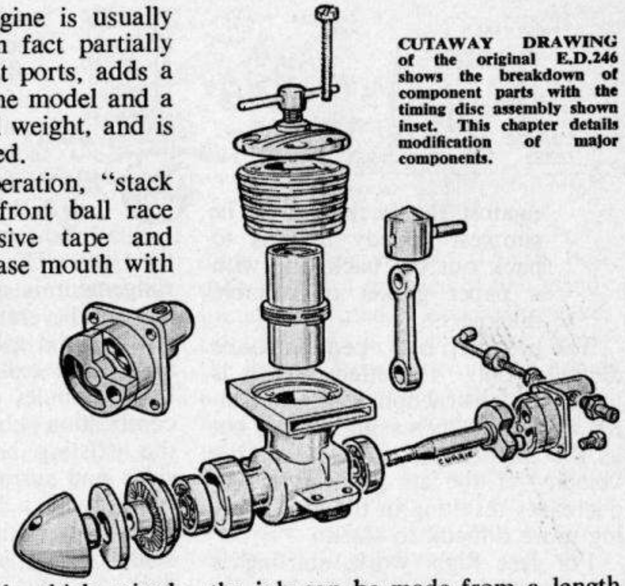


THE E.D.246 AS modified by P. Buskell. The only external difference is complete lack of exhaust stack—compare with drawing opposite.

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

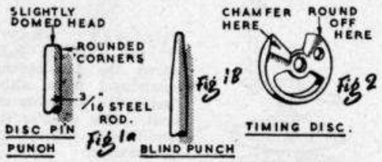
Begin by checking the end play on the timing disc. On a new component the ideal figure is about 1 to $1\frac{1}{2}$ thou. inch, *i.e.*, so that the disc appears to rub on the back plate slightly; this will wear off with running-in to a free close fit. A maximum of about 2 thou. is permissible in a new 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



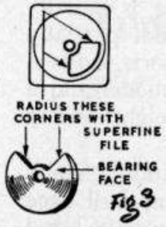
against the backplate. The simplest remedy here is to pack out the backplate with a paper gasket of suitable thickness.

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



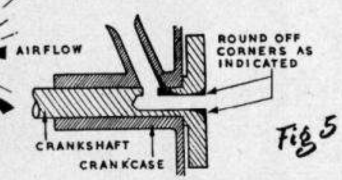
obstruction in the choke to a minimum Do not carry this too far otherwise there is a danger of it breaking in two when the fixing nut is tightened.

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.

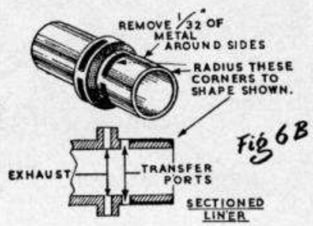
If the open period is less than this it is usually possible to file out the crankcase port by the required amount. Finally radius all port edges with an oil stone slip.

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



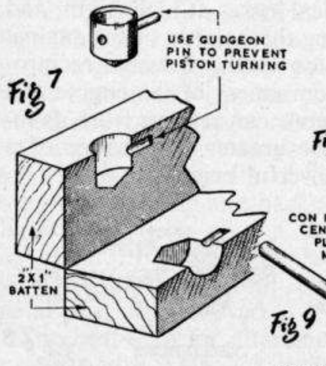
done on the cylinder. This is best done with a 4 in. bastard file, grinding can be used if done very carefully but has a tendency to distort the liner. Take your time to avoid any chance of distortion. The bottom end of the liner may be found to be hardened, this is of small depth and can be removed 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

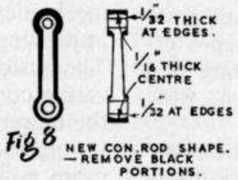
This requires the use of a lathe and $\frac{1}{4}$ in. power drill or a flexible shaft set. If you cannot get the use of these items do not attempt the work. The idea is to remove as much metal from the inside of the piston as possible, so decreasing mechanical losses at high r.p.m. and improving the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

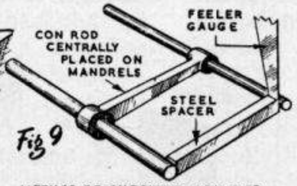
PHOTO SHOWS THE METHOD OF RELIEVING the rotor load as detailed on page 53. The back-plate is positioned so that the rotor pin is mounted over a punch and is now being rivetted securely.

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.



PISTON JIG (OPENED UP)





METHOD OF CHECKING CON ROD BEARING ALIGNMENT

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.

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.

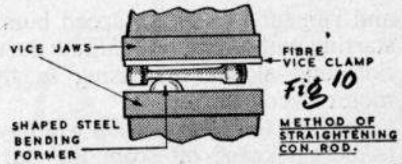
The alignment of big and little ends should also be checked. Purchase two lengths of ground silver steel rod (obtainable from most tool shops) to the big and little end diameters; these should be a close fit. *Fig.* 9 shows how to check using a length of steel bar and feeler gauges. If slight errors 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

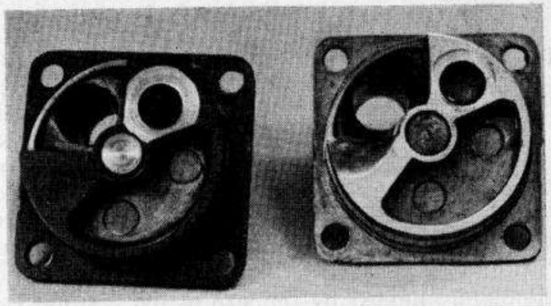
Clean off components in petrol and assemble the disc and big and little end bearings with graphite. Smear cylinder and piston with light machine oil.

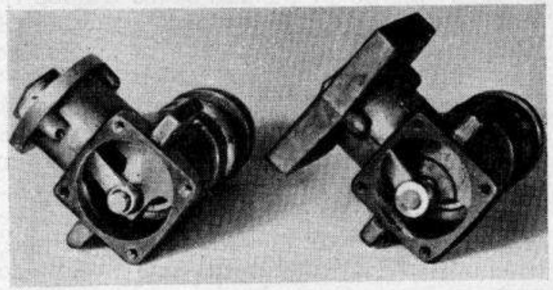


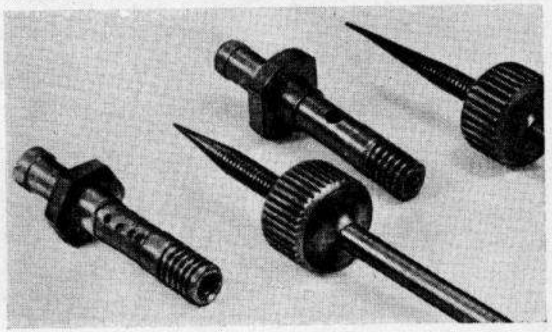
METHOD OF STRAIGHTENING CON. ROD.

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.

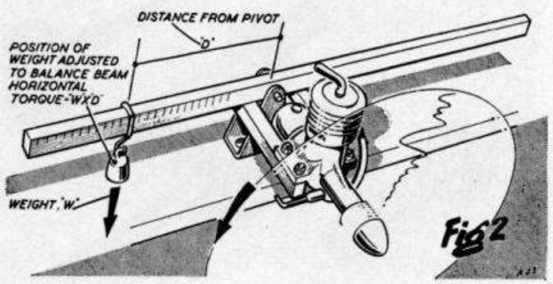


Fig 2

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.

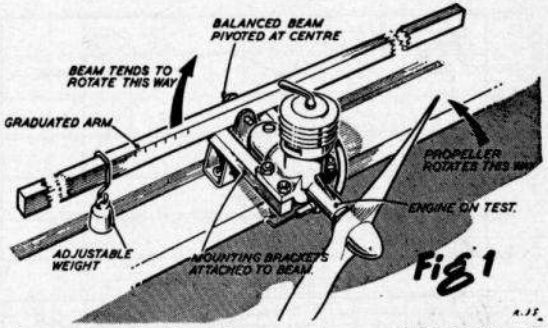


Fig 1

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

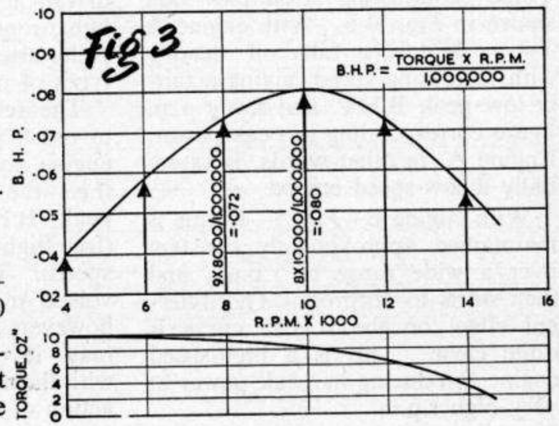
If this equation is simplified, we find that:—

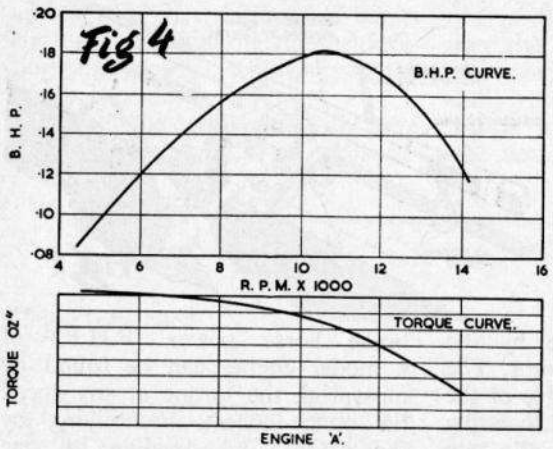
$$\text{Horse power} = \frac{NQ}{1{,}008{,}000}$$

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., Torque x 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.

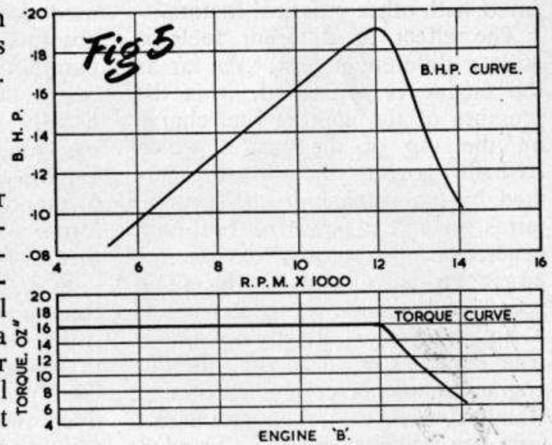
Engine C—*Fig*. 6—is relatively inefficient at low speeds and, again, essentially a high-speed type, peaking at a high r.p.m. value.

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.

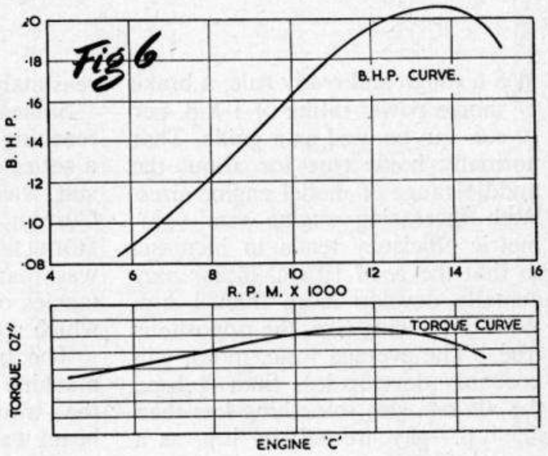


Important Points to Consider

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.

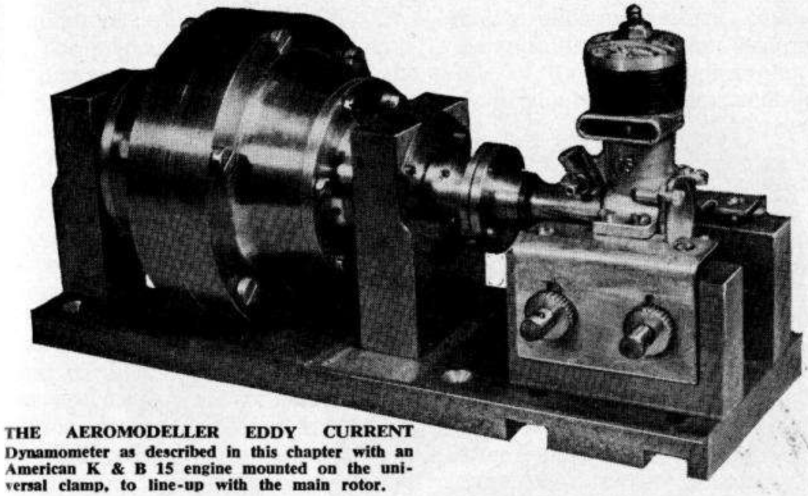
CHAPTER TEN

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

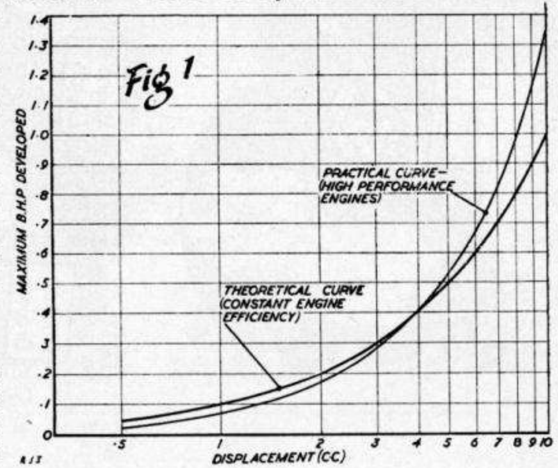


THE AEROMODELLER EDDY CURRENT Dynamometer as described in this chapter with an American K & B 15 engine mounted on the universal clamp, to line-up with the main rotor.

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



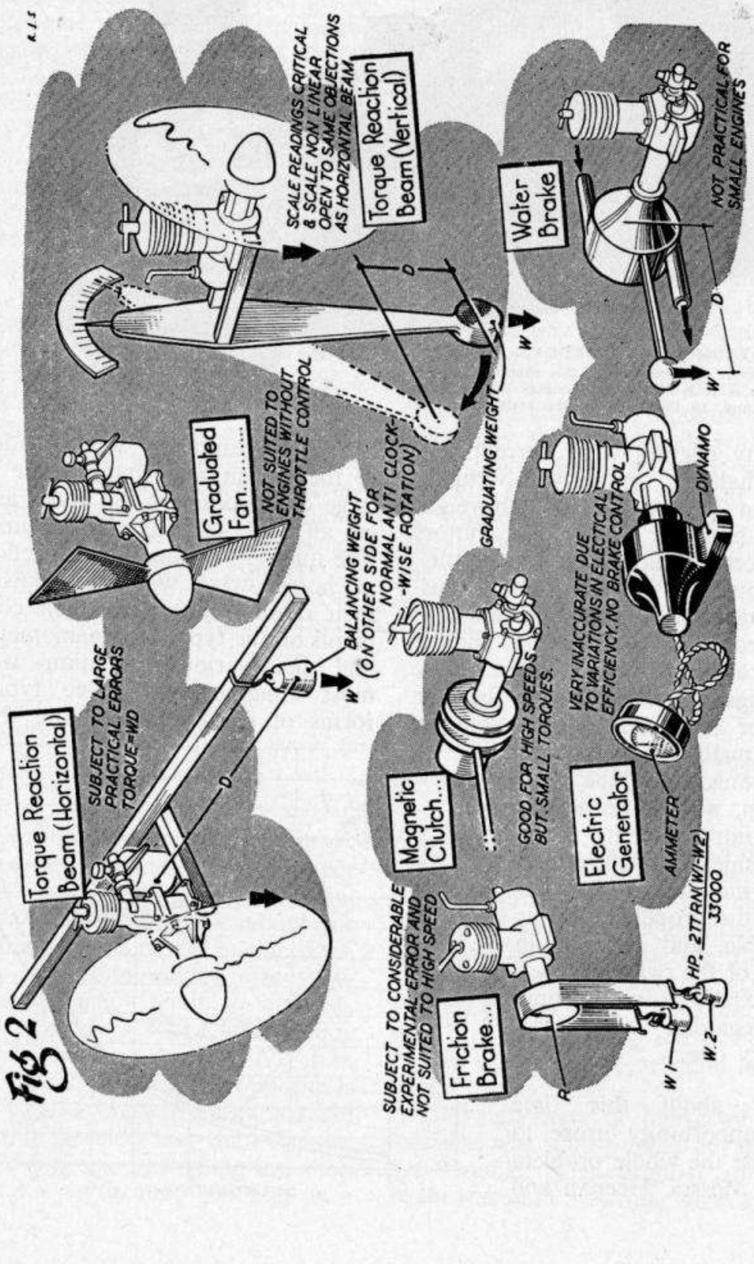


Fig 2

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, air-brake 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

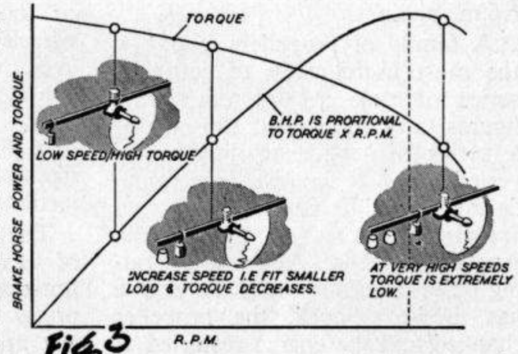
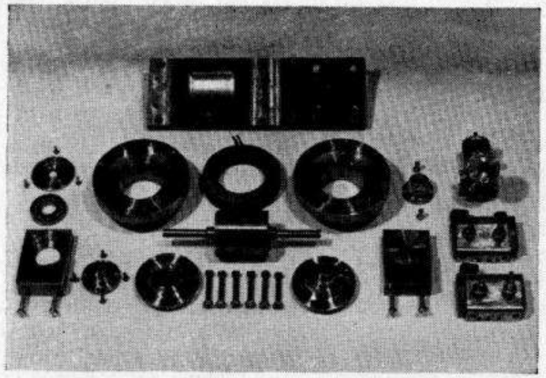
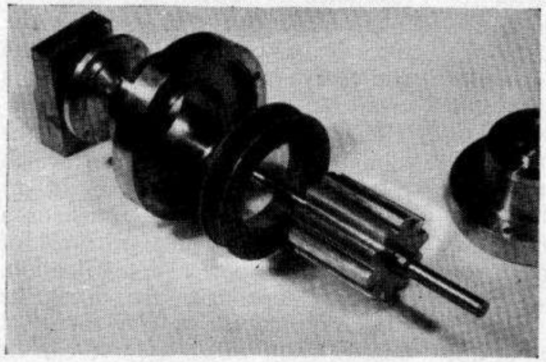


Fig 3





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 air-brake 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

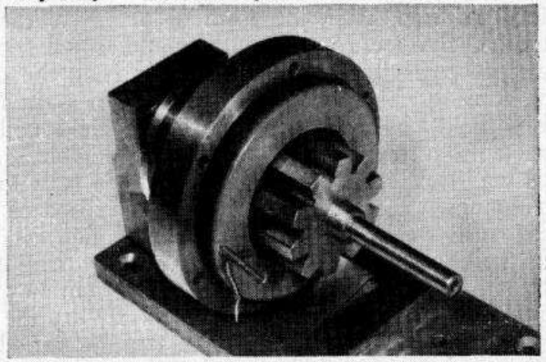
A possibility which occurred was to work strictly on a family of propellers of similar geometric proportions but different pitches and diameters, basing b.h.p. figures on power absorption figures to the r.p.m. at which individual propellers were driven. One major difficulty was how to calibrate the family 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.

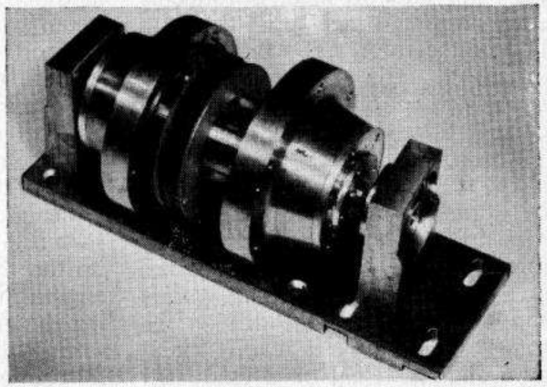
The first alternative investigated was a motor-generator or, virtually, a dynamo coupled to the engine shaft. Shaft torque driving the dynamo would then be measurable in terms of current produced. This method, however, did not lend itself too readily to accurate speed control and overall accuracy was 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.

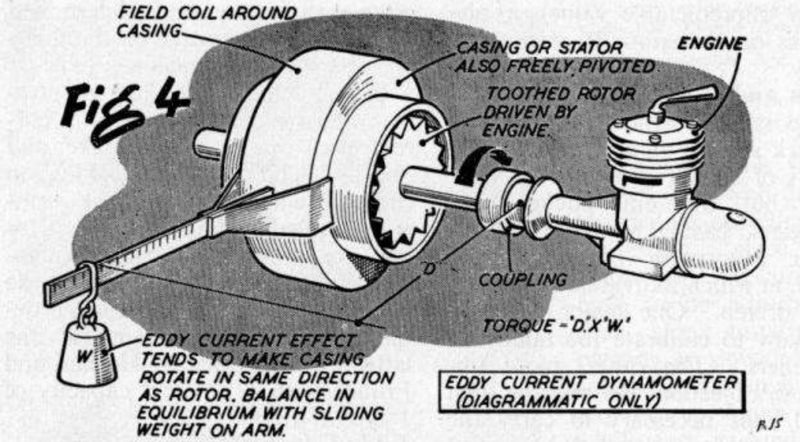
Eddy-Current Dynamometer

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

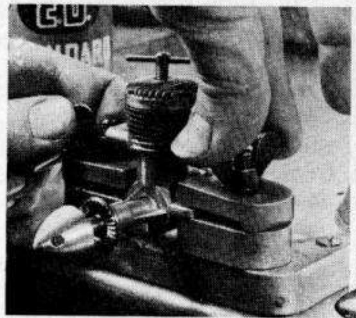
Operating a First Diesel

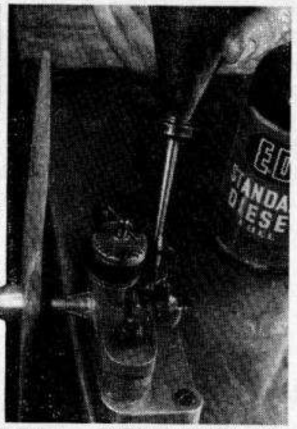
IT must be every young aeromodeller's aim and ambition to have a model engine to call his own—and these days it is inevitable that his first power unit will be a diesel of 1 c.c. or less. With this in mind, we intend to run through the process of purchasing and learning how to operate a small diesel. 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.

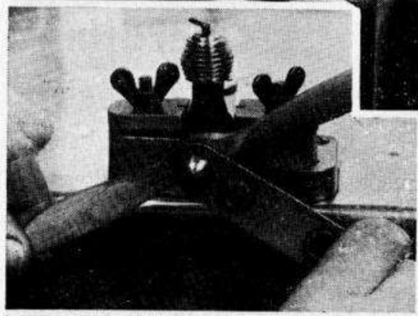
Mention of "two-stroke" recalls the method by which our engine will work. It will fire, or explode a combustible mixture every second stroke. (Unlike a four-stroke, where the firing is every fourth stroke and mechanical valves are employed to control intake and exhaust.) The fuel is drawn from the tank, 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.

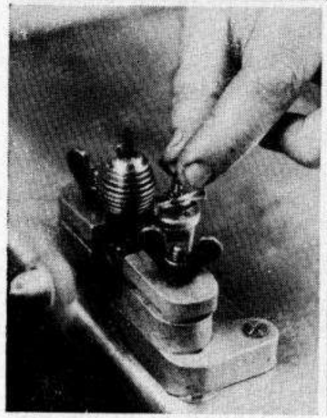
Now all this effort is transmitted through the crankshaft to the propeller, and according to the mixture we give the engine, and the compression ratio we can adjust, so can we vary the power and r.p.m. (revolutions per minute) of our engine. All we have to do is to learn the right way to go about the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



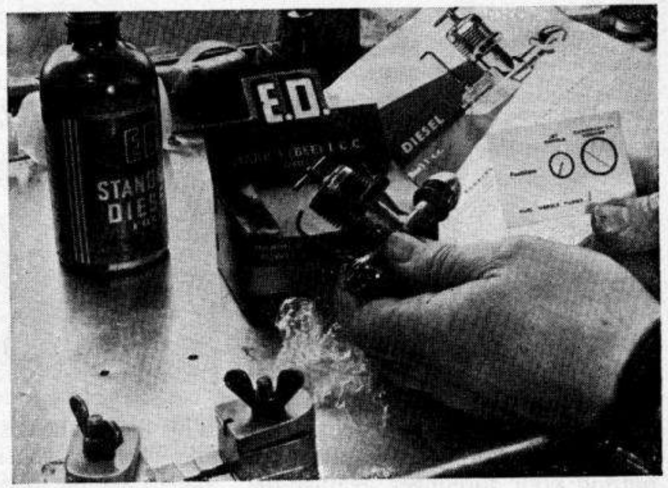






THESE PHOTOS show how one should follow the manufacturer's instructions. Mount the engine rigidly, fill up with the right fuel and set the propeller correctly for starting, using the proper spanner. Below right, how to choke carburettor; left, altering the needle valve.

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.

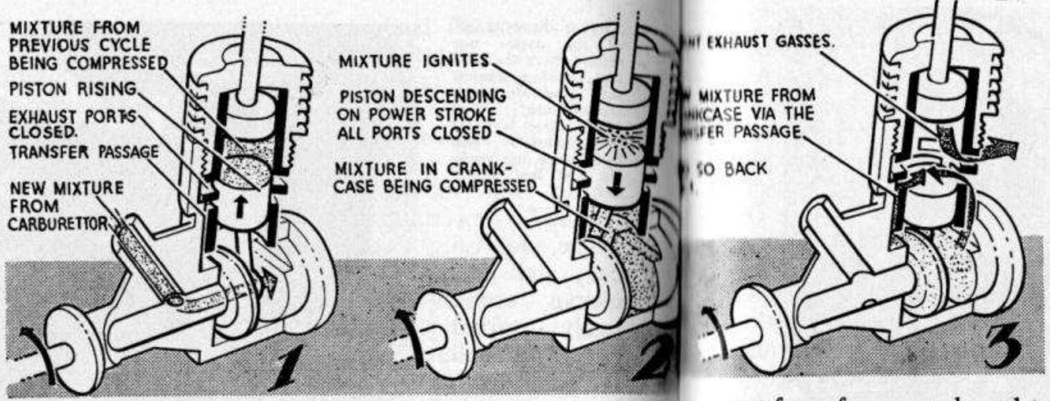
1 c.c. to 1.5 c.c., and purchase some ready - mixed fuel. We shall not dabble with home - made mixtures here, for the beginner can do no better than use a branded fuel with all its easy starting additives. Most engines have their own special brews, and the model shop will see that you are well equipped.

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.

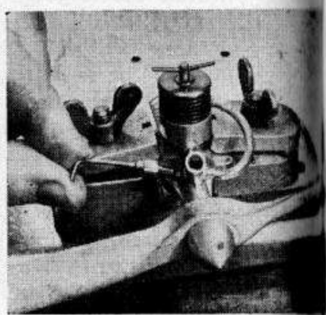
If the engine refuses to show any inclination to work, look through the exhaust ports and see if the top of the piston is at all wet with fuel. If it is, then use the compression screw as though you have your hand on the pulse of the engine; treat it as you would a human, and raise the compression by 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

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.

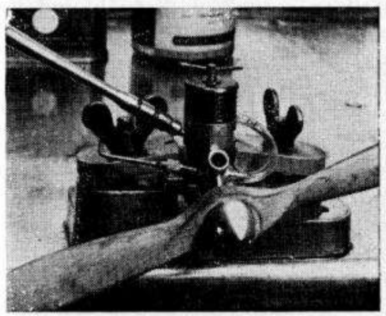


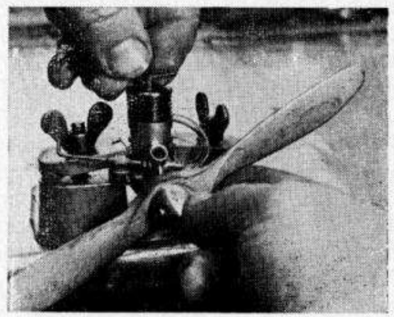
set for a few seconds and take stock by watching the exhaust and listening to the note. Smoky, rich exhaust is cured by screwing in the needle valve, and a staccato misfire indicats the need for more compression. Most engines start for the first time in this condition, and will not harm themselves if 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.

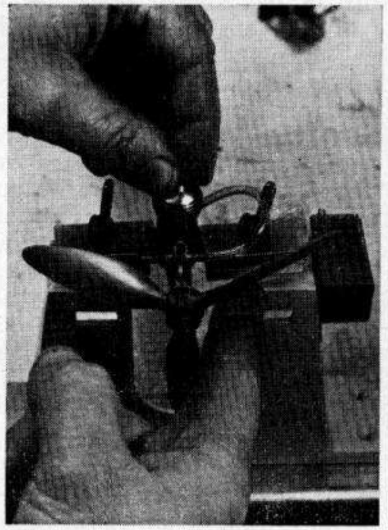
Now all of the foregoing has made one very critical assumption. That you have purchased a trouble-free engine (and it should certainly be so if brand new), and with it you have obtained some ready-mixed fuel of the advised brand for the particular engine. It may be that you do not have a local source 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.

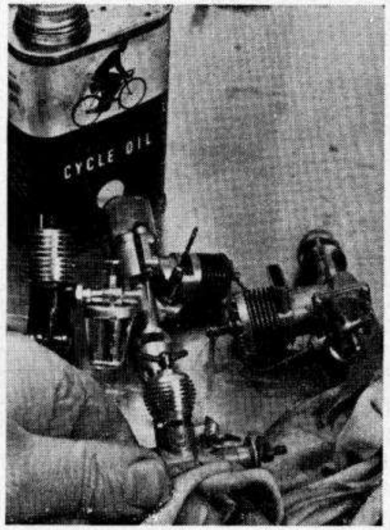
The procedure for mounting the engine, checking the fuel flow and getting oneself accustomed to the "flick" start as described in the first part of this chapter are simple enough to follow and will not need further enlargement: but let us presume that before we can start, we have to obtain the fuel.

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









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.

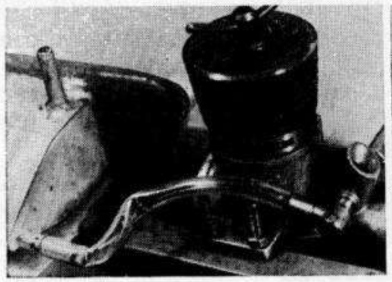
Quantities of the fuel components can fluctuate by wide margins. Simplest mixture, and one advised by Messrs. Electronic Developments Ltd. for their engines over a ten year period is equal parts (33 per cent) of each, Paraffin, Ether, Castor Oil plus additive. If you want to run faster then reduce the 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.

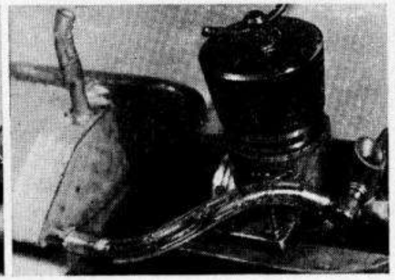
More on fuels will be found in chapter 17, but now we know the basic principles, we can assume that the first diesel is now ready to run on our own concoction.

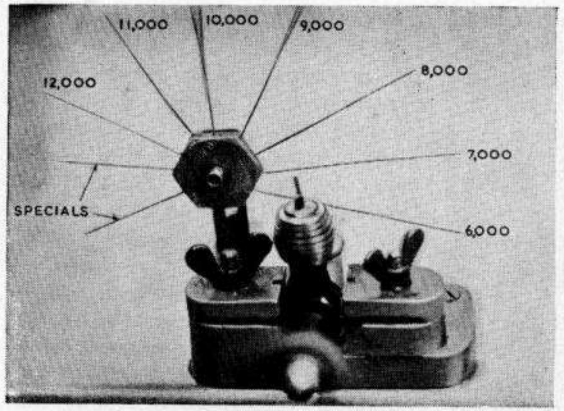
Let us make one more presumption for the "Outback" modeller—the engine arrives without any guide as to its working settings, or it does have such a guide, but the Customs & Excise people have been playing with the engine so much that factory settings cannot be identified on the controls.

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

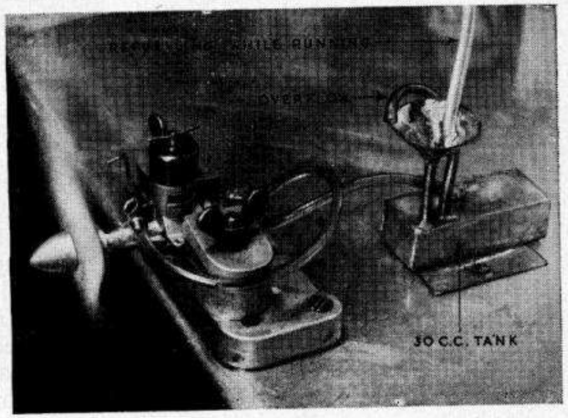
though against an obstruction, then you have primed too generously. The condition now created is known technically as a hydraulic lock, and the cure is to swing the engine backwards 180° so that the exhaust port is opened, and to blow into the port to distribute some of the excess mixture. One must also 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 oscilating, in some cases runs correctly and mostly stops, ready for an immediate start if flicked correctly.

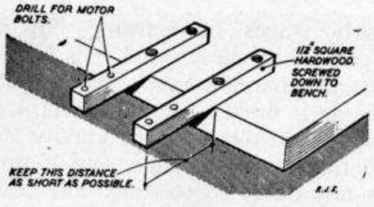
The tank position should not be any higher than the jet during these early stages of learning how to operate the diesel, and it is best to use the manufacturer's integral tank if one is fitted. Should the tank be high, then there will be gravity feed to the jet all the time, and if there is any delay 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.

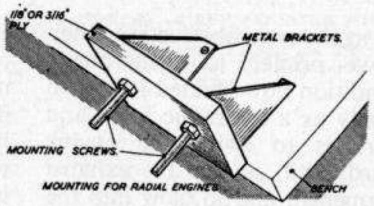
We have avoided touching the needle valve through the initial stages because, if near enough to average running setting by the "blow through" test, then the diesel can be fully adjusted to run continuously by means of the compression alone. For maximum performance the needle valve should be used to adjust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

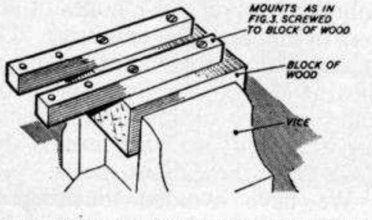
As the carburettor is "leaned-out" or made to run with a restricted fuel supply, then the engine will speed up at normal compression settings,



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







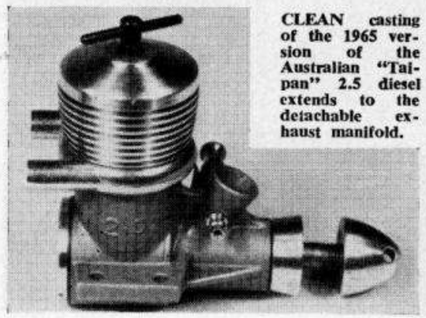
THIS SKETCH ILLUSTRATES THE USE OF hardwood engine bearers if a cast test mould is not available or desired.

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!)

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.

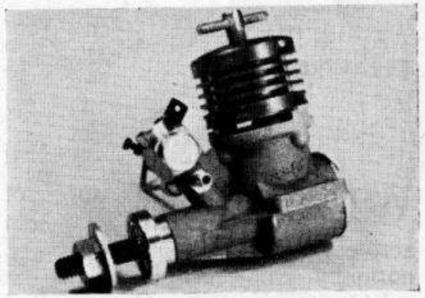


CLEAN casting of the 1965 version of the Australian "Taipan" 2.5 diesel extends to the detachable exhaust manifold.

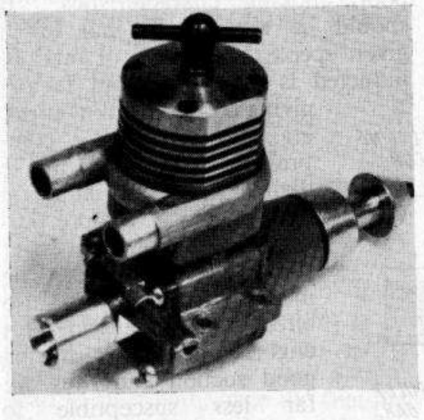
One must also allow for the diesel being a vintage type of the era when the contra piston was considered unnecessary and obviated by use of a blanked-off cylinder head. Such engines, with fixed comp. ratios demand a fuel with large ether content, and a quantity of mineral oil, RedEx being specially suitable. 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 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 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.



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

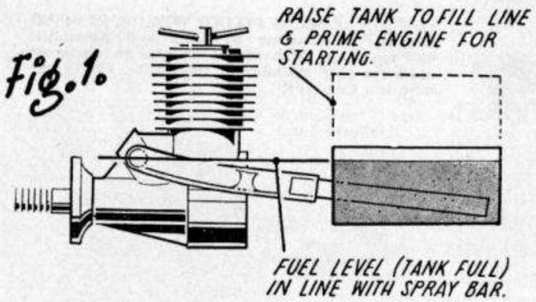
Fuel Tanks and Fuel Feeds

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.

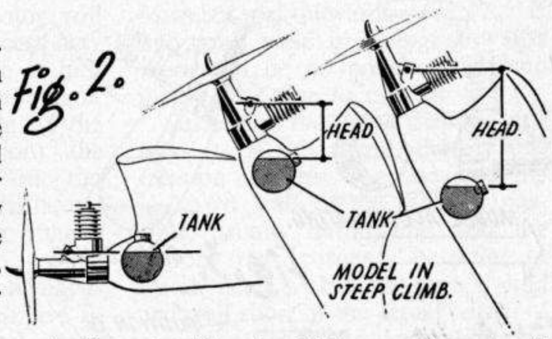
To a large extent the "suction" power is governed by the crankcase compression ratio. This can be defined as the ratios of the "under piston" volume with the piston at top dead centre and bottom dead centre. If the displacement of the engine is X c.c. and the actual crankcase volume at B.D.C. is Y c.c., 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.

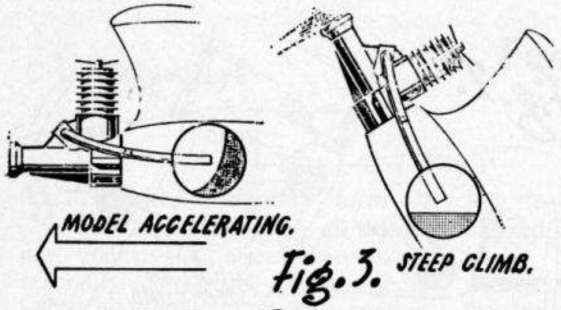


Fig. 3.

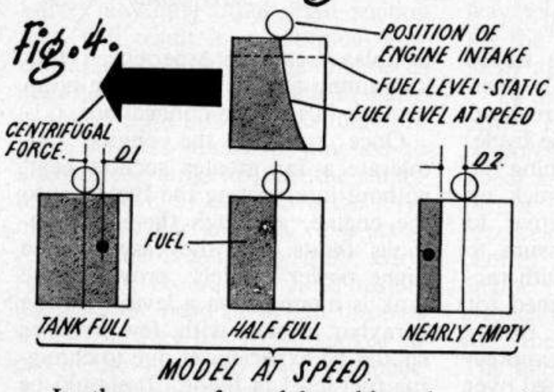


Fig. 4.

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 $23.7 \times D_1$ 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

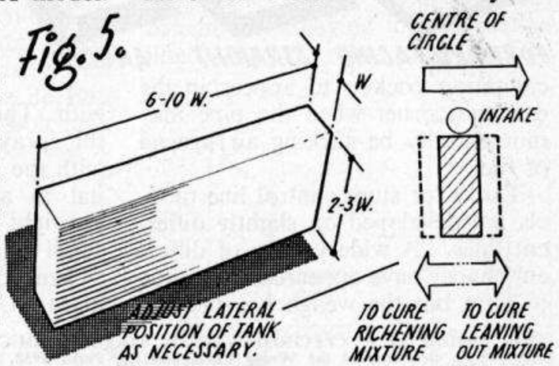
model down when the motor will lean out again, and the process may be repeated until the tank is half empty. Conversely, with the tank too near the outboard side of the model (large D_2 dimension) the motor may tend to lean out too much—*Fig.* 5. Adjusting the lateral position of a speed tank is often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



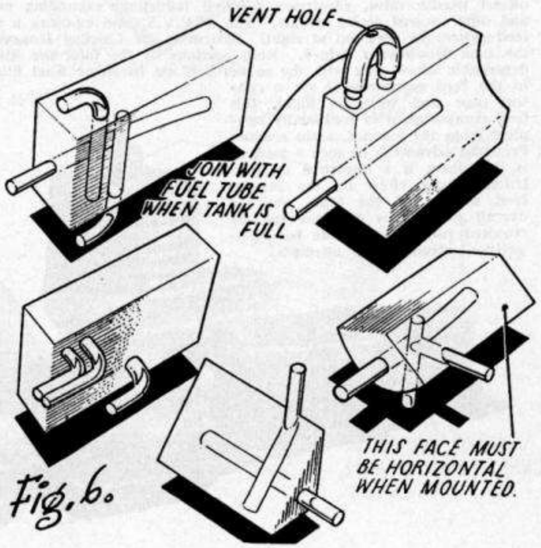
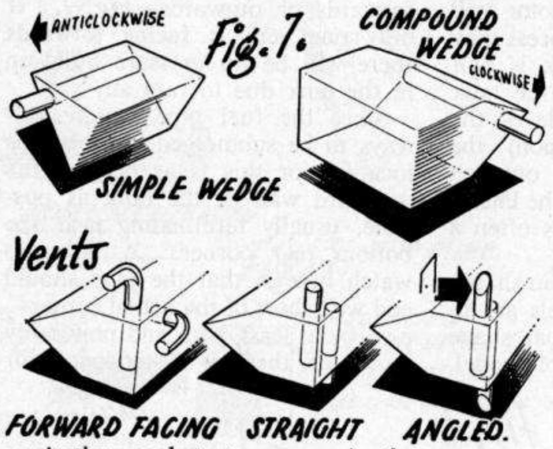


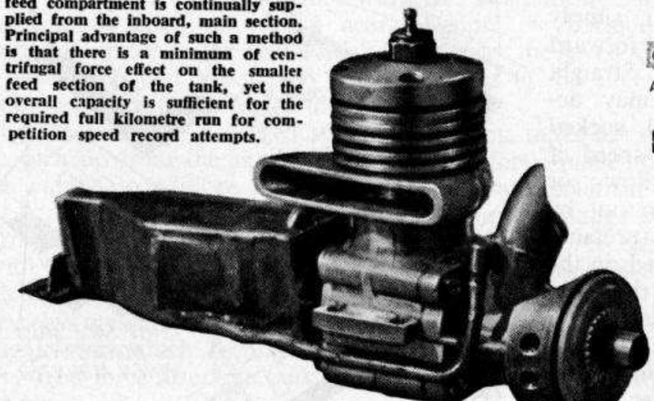
Fig. 6.

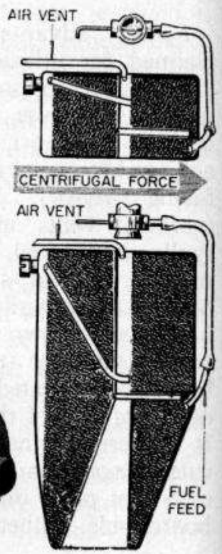


cavitation pockets to appear in the extreme corner when the pipe may momentarily be sucking air instead of fuel.

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

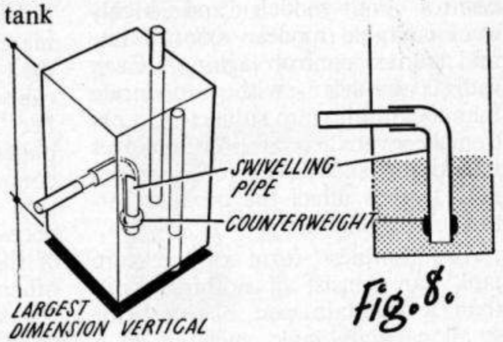
THIS MOTOR IS A CZECHOSLOVAKIAN STATE PRODUCT, THE M.V.V.S.25, first seen at the World Championships, Paris, 1955, featuring off-set needle valve, slipstream directed induction, expanding exhaust port and other potent racing details. The M.V.V.S. also employs a unique fuel feed system as sketched at 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.





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.

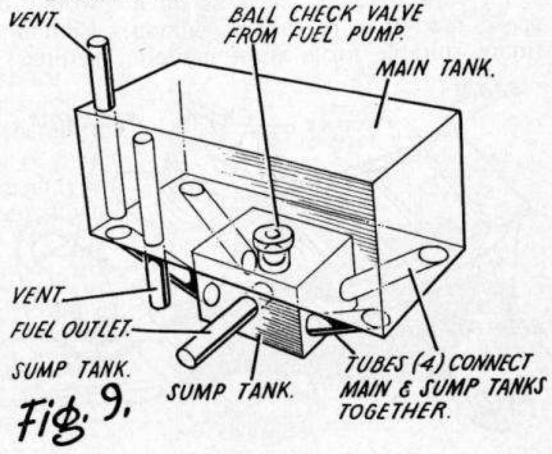
Most wedge tanks are "handed" by arrangement of vents (*i.e.*, are usually designed for normal anti-clockwise control line circuits). A non-handed type of stunt tank which has regained popularity with radio control models is the de Bolt —*Fig.* 8. The original de Bolt tank was rectangular in form with 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.



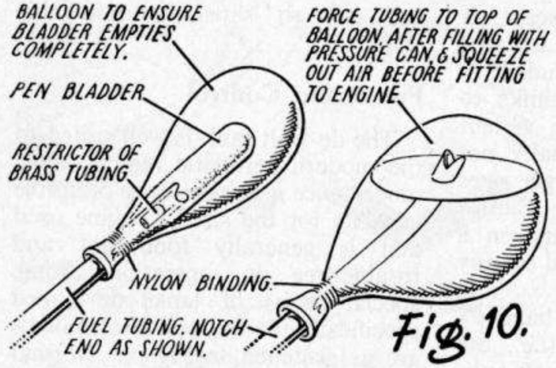


Fig. 10.

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.

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).

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.

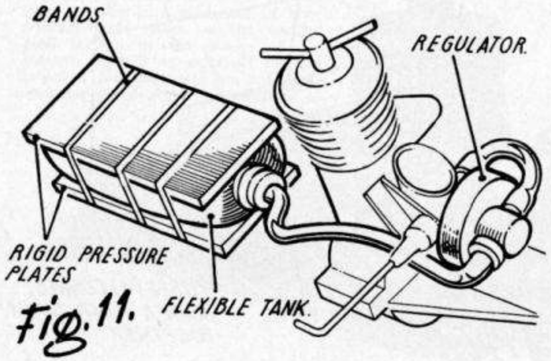
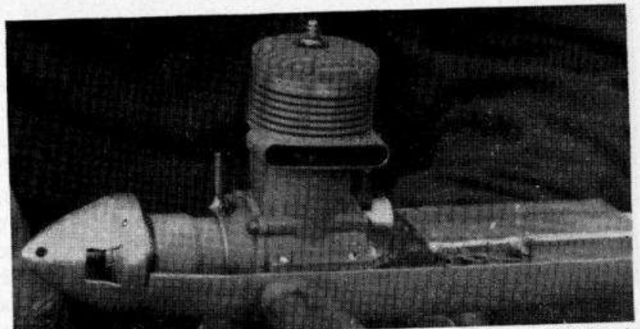


Fig. 11.

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.

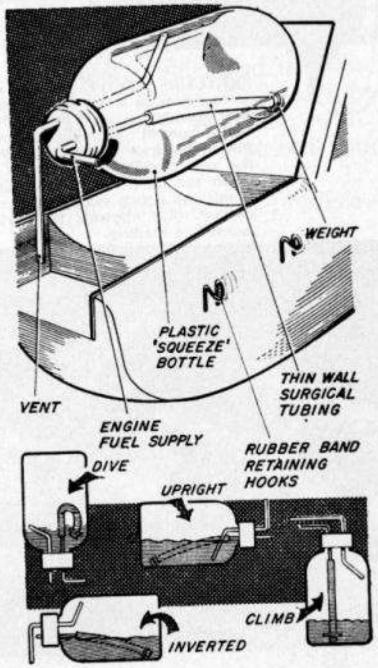
Invariably the "plumbing" in any fuel system is done with plastic tubing. A majority of the commercial tubing is manufactured in clear form and is to be preferred to opaque tubing since the state of the contents can be observed. It is of considerable advantage, for instance, with a cowled-in installation 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.

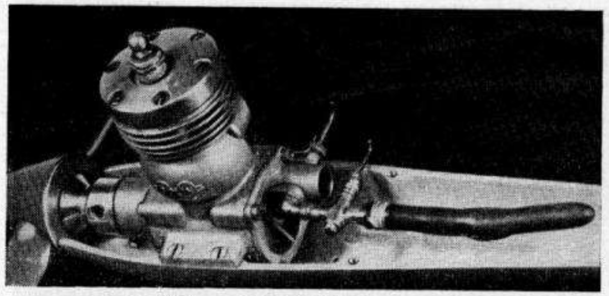
Tubing is normally made either from synthetic rubbers or P.V.C., both of which materials are fully resistant to fuel and oils. Neoprene tubing is the best from the point of view of remaining flexible in contact with fuels. Most of the plastic tubing age hardens to a rigid, brittle state in a matter of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

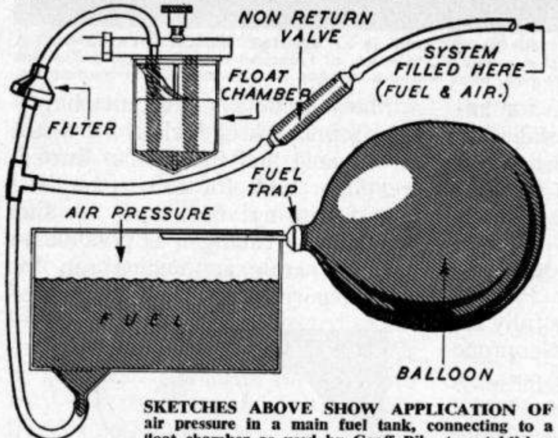
Most of the fuel tubing sold through British model shops is manufactured originally for surgical drain tubes, etc., and is therefore expected to remain flexible. A good tip for softening the hardened type is to warm it slowly then flush through with petrol.

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.

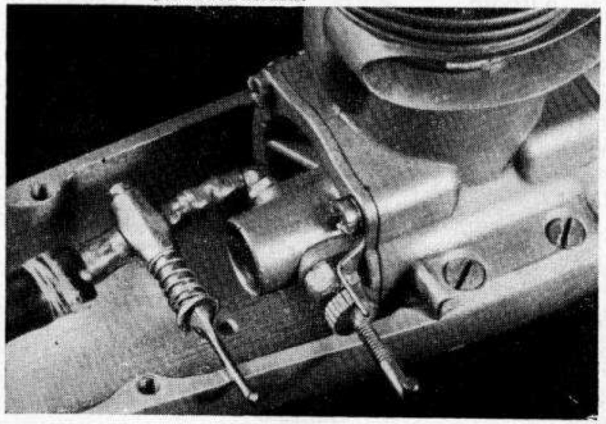
Many modellers are quite happy to coil a length of fuel tubing about 18 in. long and use this to form a free-flight tank for contest models. By measuring the flow of fuel through the tubing it is possible to graduate an exact engine run for contest purposes with a tolerance of







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

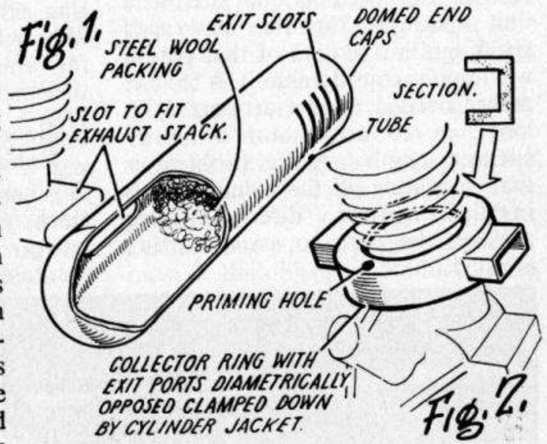
Silencers

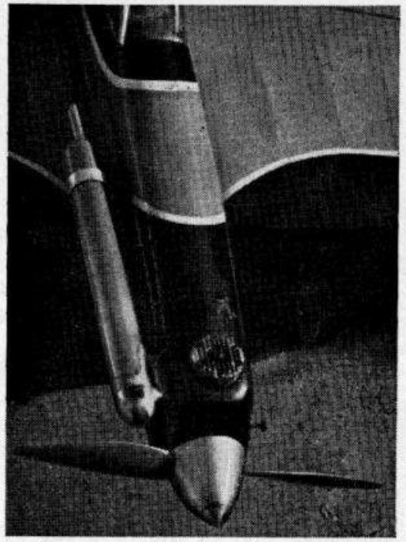
THE model engine is a remarkably noisy piece of machinery—a feature which cuts both ways. Noise implies power and so, to a large extent, the noisier and faster an engine the more potent it appears, and the greater its sales appeal. But to the outsider the nuisance value of a model engine being operated 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.

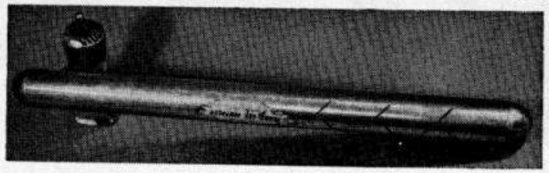
High Frequency Notes

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).

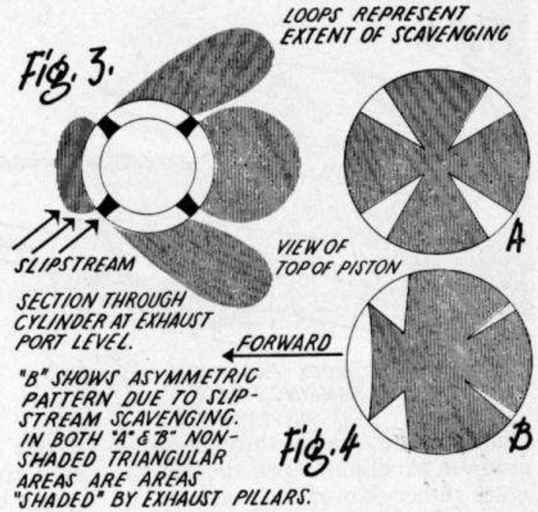
Collector Banjo for 360° Exhausts

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.



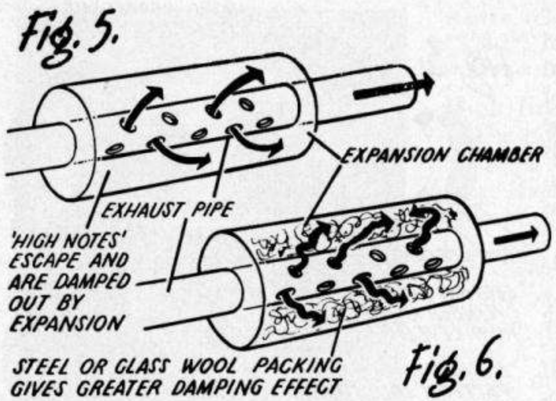
The fitting of such a banjo may affect the performance of the engine. If the design relies on sub-piston induction of air, this will no longer be effected. In fact, the engine will suck back exhaust gases instead of air (when the silencer is attached).

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.

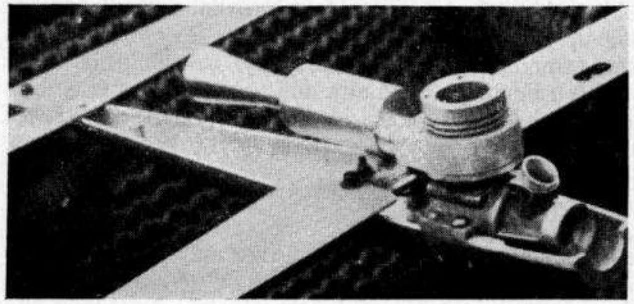
A straight-through silencer offers virtually no resistance to the passage of the exhaust gases (other than friction of the walls of the pipe) and by opening the flow radially into an expansion chamber, most of the objectionable high notes will be filtered off. In other words, a straight-through exhaust 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.

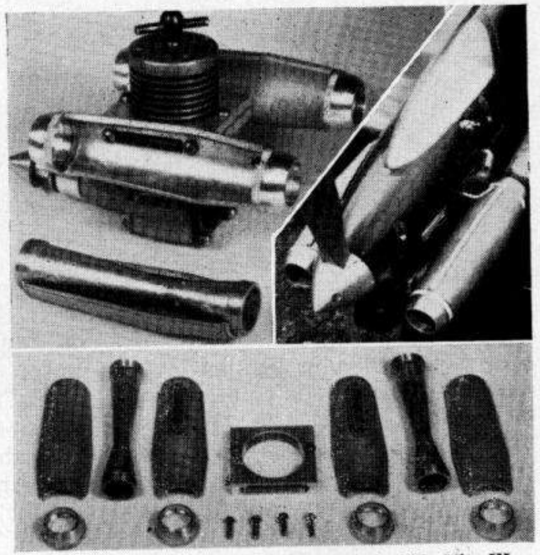
The size of expansion chamber required for effective silencing is quite large from an aeromodelling point of view. On power boat installations, where silencers are obligatory, a 5 c.c. engine commonly has twin exhausts and twin silencers, each with an expansion chamber some $1\frac{1}{8}$ in. to $1\frac{1}{4}$ 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.



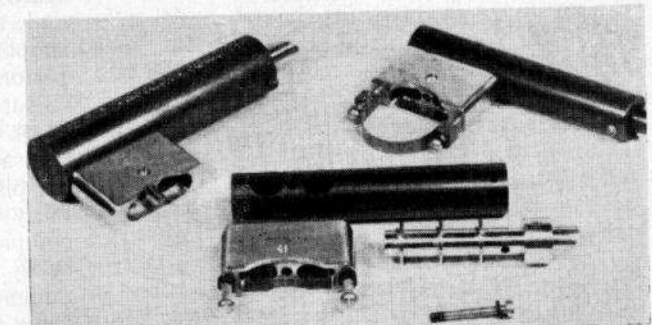
The extractor - venturi megaphone added to this exhaust aft of a sound absorption chamber shows one early approach to the noise problem by a speed flyer.

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.

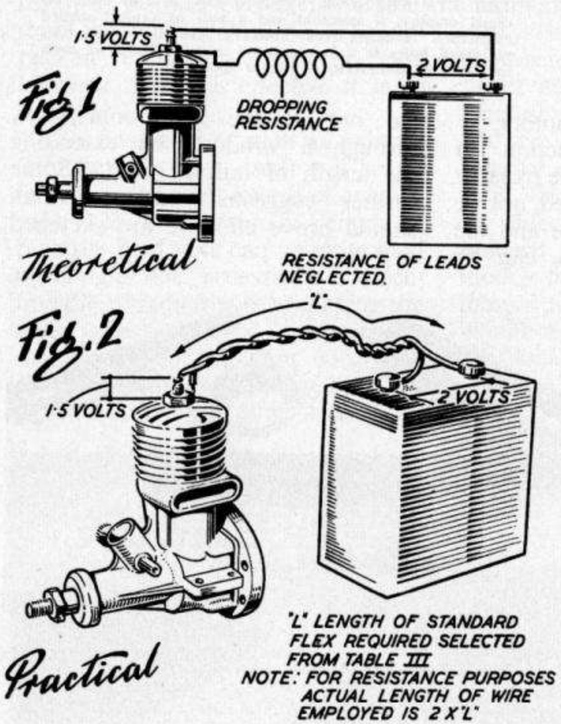
CHAPTER FOURTEEN

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.

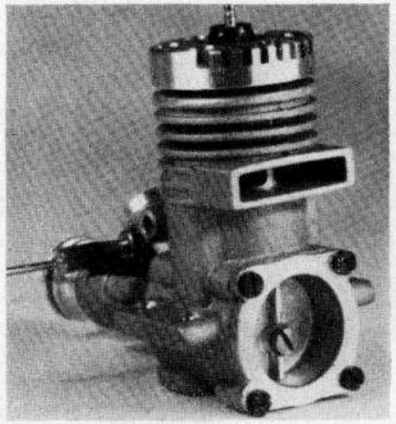


But conversion from spark to glow is not quite that automatic. Firstly, the piston seal has to be fairly good. Then the compression ratio has to be in the region of 8 : 1 minimum, and a good many ignition motors are less than that. Thirdly, the fuel has to be specially concocted, using alcohol base and cas-

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 Ohlsson, 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.

tor 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.

Glowplugs are made in two thread fits, for ¼ in. or ⅜ in. bodies, and in some cases, an adaptor can be employed to fit say, a ¼ in. plug in a ⅜ 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

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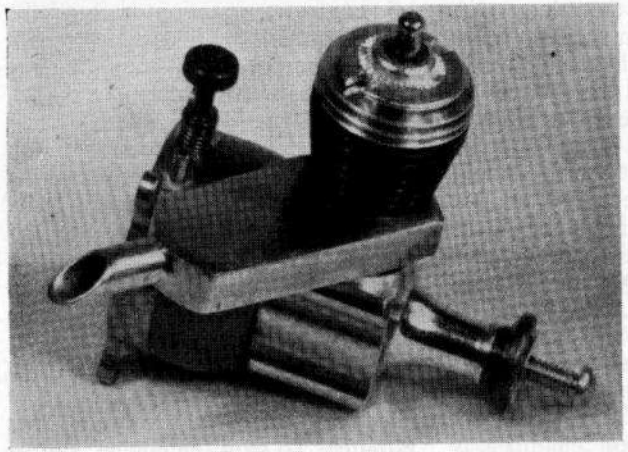


SILENCERS IN SILVER SOLDERED BRASS are made down to the smallest size shown here for the diminitive Cox "Pee-Wee" .32 c.c. engine. Known as the "308" silencer it is a combined expansion chamber and manifold.

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.

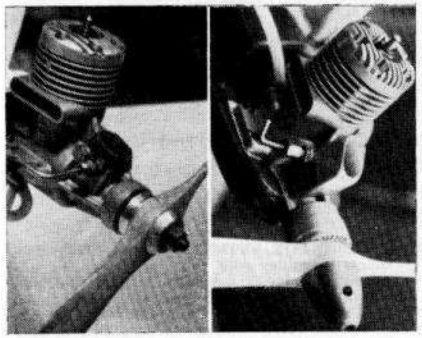
If the engine just "plops" each time it is flicked, it needs an upper cylinder prime in most cases, but there is also the chance that the crankcase is flooded. The engine is then in a very rich state and will not clear until the plug is glowing bright, and the engine fires out some of the excess. Shut 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.

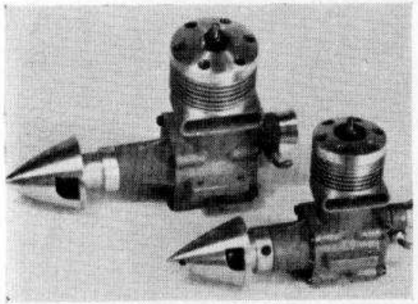
There are also hot plugs or cold plugs. Makers vary wire thickness and length according to the engine. In America, many engines are supplied with special plugs and call for different fuels. These are usually the small .8 c.c. (.049) units known as the ½A class.

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.



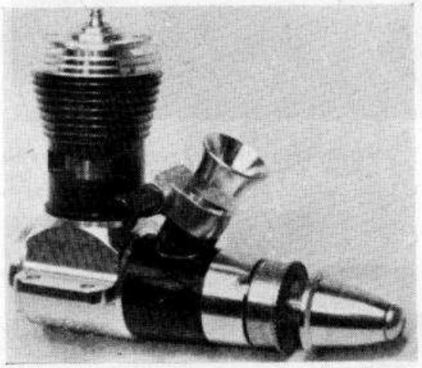
RIGHT IS THE COX SPECIAL 15 Mk. II WITH single exhaust port, thick cylinder and revised cylinder head contour.



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

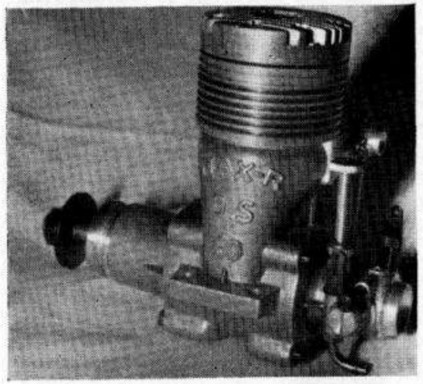
MOST EFFICIENT OF THE 2.5 c.c. glow engines is the Super Tigre G.15. At left are specials (note non-standard carbs) seen in the 1965 World F/F Championships. That at right was possibly most powerful at the contest in Finland, was prepared by Benno Schlosser.

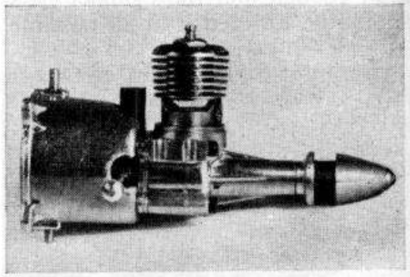


THE K & B ENGINES PRODUCED UNDER the supervision of World Speed Champion Bill Wisniewski have a family resemblance, obvious in the 15 and 29 here.

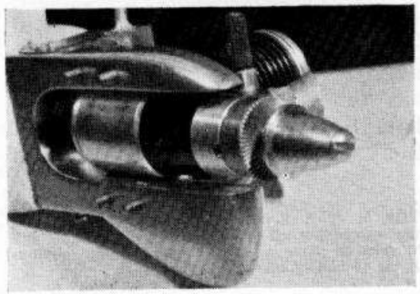
flick of the prop. to get the engine started. A tip we can advise is to inject a spot of machine oil on the piston crown when the exhaust is open, and so helping to seal compression. The engine should already be wet with a fuel choke, otherwise the plug may be fouled by the oil and does not glow enough 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.

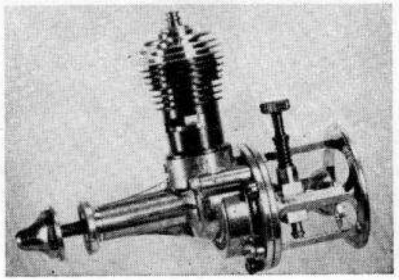
Until now, we have been considering only the starting side of glow. The most desirable feature of this form of ignition is that it is self-timing. There is no question of having to set the make and break for advanced or retarded position and we do not have to alter the compression as in a diesel. When 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 Ohlson 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.

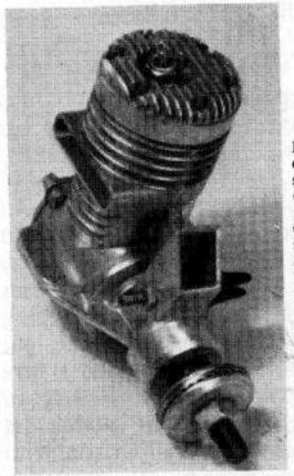




that combustion could only be advanced.

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.

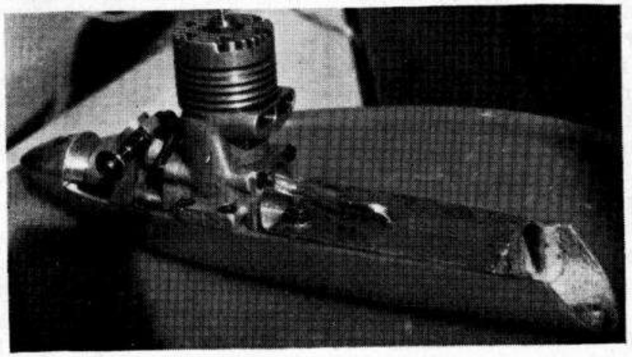


DUKE FOX'S characteristic square intakes, offset glow plug and crankcase finning are seen in the Fox 36xBB — a very powerful unit.

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!

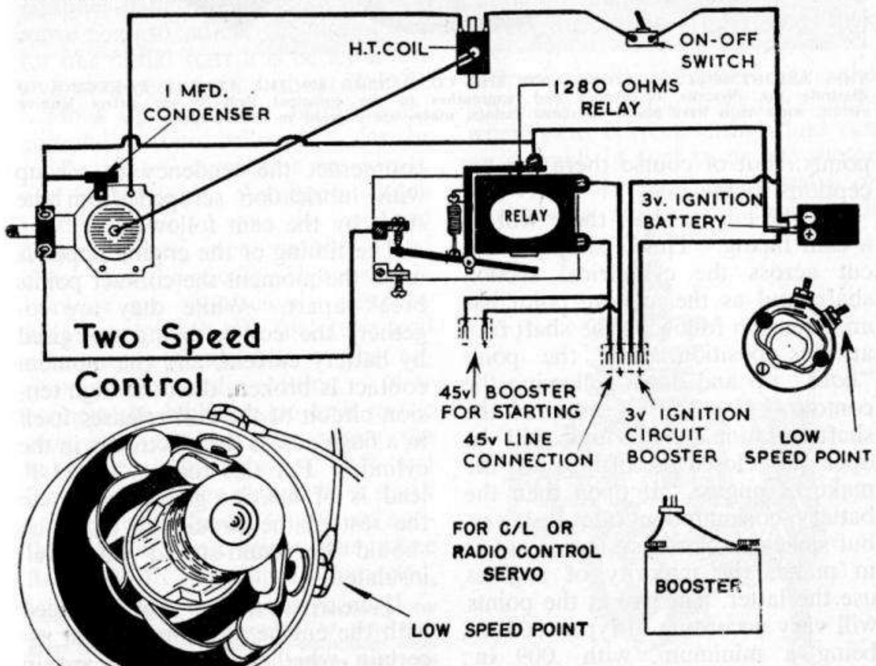
The coil consists of a core made up of laminations to prevent them becoming permanent magnets, and multiple windings of insulated copper wire built up to a diameter of about 1 in. and length about 1¼ in., or even longer. The coil is a heavy item, and in order to conserve weight in duration models, the 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.

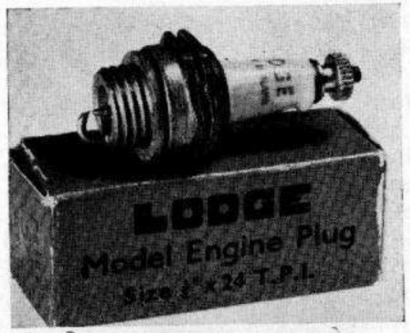
points: but of course there are exceptions to the rule.

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 $\frac{3}{8}$, a true miniature of the full size plug.

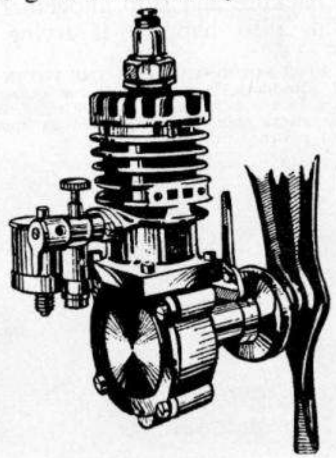


right moment relative to the piston movement. This can be decided by removing the plug so that the engine can be turned over easily, with the points visible. Rotate the shaft so that the piston is on the up stroke and place a piece of balsa strip through the plug hole, resting one end on the piston top 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.

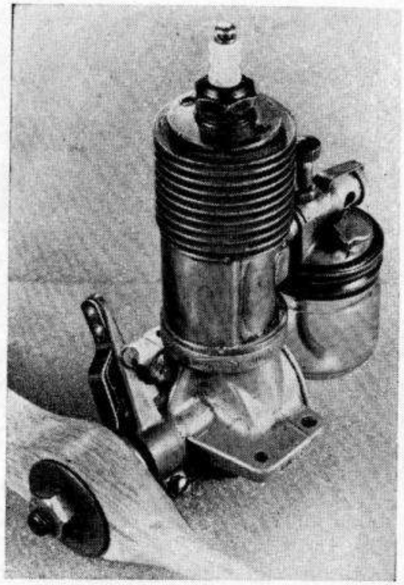
Hook up the circuit, using an accumulator (two cells of a car or motor-cycle accumulator are ideal), and fill the tank. Arrange a switch in the circuit to save the coil and battery when not running, and check that the needle valve and carburettor are satisfactorily clear by blowing through the fuel tube 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 $\frac{3}{8}$ 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.

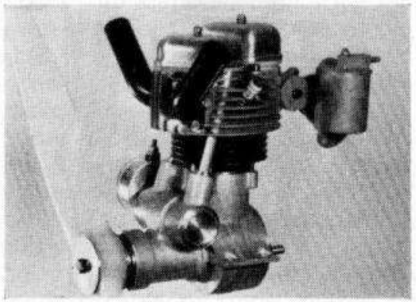
This also happens if trying to start with the engine too far advanced and the backfire often loosens the prop. nut which can be the source of starting difficulty. Always ensure that the prop. is bolted on really tight. Symptom of a loose fit is a "crack" in the backfire note.

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.

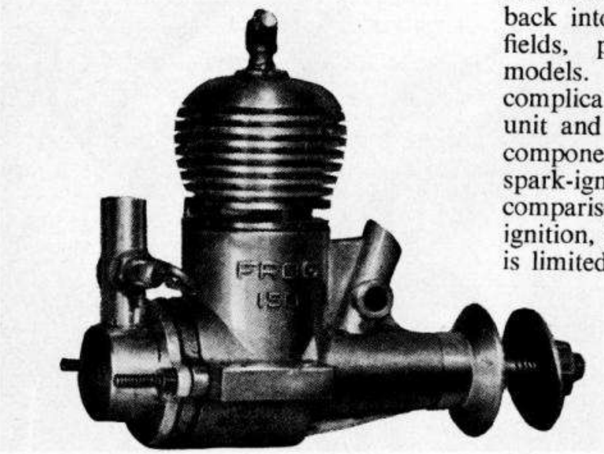
CHAPTER SIXTEEN

Engine Speed Controls

INHERENTLY the small two-stroke engine is less amenable to throttling by varying the amount of fuel fed to the cylinder than its full size counterparts, due both to an apparent "scale" effect with small volumetric sizes and, particularly, the method of carburettion employed. Thus throttles, as such, are absent from standard aero-engine designs. Running speed is then largely dependent on the load (*i.e.*, the size of propeller driven) with the mixture control (needle valve) adjusted for optimum performance, except in the case of spark-ignition engines. In the latter case, speed and any load can effectively be varied by adjusting the timing or the instant at which the plug sparks, *e.g.*, by rotating the contact breaker in the same direction as the direction of rotation to retard the spark and slow the engine; and against the direction of rotation to advance the spark to make the engine run faster.

This is a particularly positive form of speed control for with the spark retarded, for instance, it is not possible for the engine to speed up unless the contact breaker unit is rotated. Also the engine will continue to run steadily at this setting. Throttling a diesel or glow motor, on the other hand, *e.g.*, by running on a very rich mixture setting or reduced compression (diesel) *can* lead to the engine speeding up, or stopping during flight without any change in adjustment.

Although no spark-ignition areo-motors are now produced in this country or the United States (the last was the spark-ignition version of the Frog "500" which was withdrawn in 1956) the type is coming back into favour in certain limited fields, particularly radio control models. Apart from the additional complication of the contact breaker unit and the weight of the ignition components to be carried, the spark-ignition motor suffers, by comparison with diesel, and glow ignition, in that its maximum speed is limited, and hence its power out-

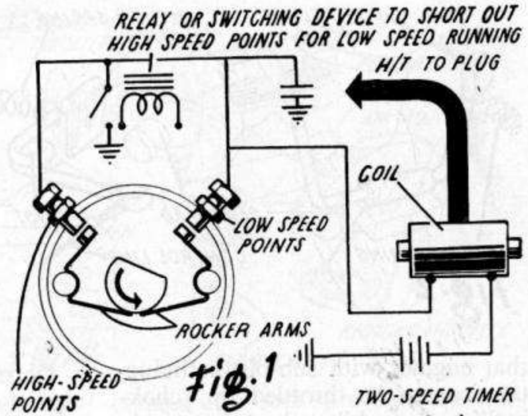


NOT A CASE OF USING TWO carbs. for engine speed control, this experimental Frog 150 is actually a prototype of the latest 149 version using the vibra-matic induction system. A standard engine was employed with blanked main shaft.

put per c.c. inferior. But where this can be tolerated its ready adaptability to speed control and its comparative cleanliness are still in its favour.

Dual Points for Spark-Ignition

To avoid the necessity of mechanical movement of the contact breaker assembly, electrical switching can be used for speed control by fitting a duplicate set of contact points—*Fig.* 1. The contact breaker unit is now fixed, the two set of contacts corresponding to "retard" and "advance" positions. (Fixed, here, is a relative term: the whole unit may still be movable for initial adjustment.) It is then possible to switch the ignition circuit from one set of contacts to the other, to change from "low" to "high" speed, or vice versa. Alternatively, the switching can be arranged to short out the high speed contacts for the low speed running, this being the more general arrangement.



Both diesel and glow motors can, however, be throttled quite effectively, although not always to the same positive degree. In both cases this is nearly always accomplished by supplying the engine with an excessively rich mixture for low speed running, although some diesels can equally well be throttled back by reducing compression. All diesels will reduce speed when compression is backed off from the running position, accompanied by "missing". It is very much an individual characteristic of the design, however, whether the engine will continue to run with reasonable consistency when this is done, or whether it will tend to stop. On a limited number of engines, the non-critical response to reducing compression may be such that the engine cannot be stopped by this means, *i.e.*, with the range of upward movement possible of the contra-piston, the compression ratio still remains high enough to continue to fire the mixture.

Altering the compression, however, is not a practical means of speed control, allied to light servo mechanisms which have to be contained on the model. Thus the rich mixture method is preferred as this can be accomplished with a simple clapper valve or similar device. There is one marked difference between diesels and glow motors when slowed by running on over-rich mixtures. Nearly all diesels have a tendency to die if run too slowly, whereas glow motors will generally keep going. The main objection to "throttling" by the use of a very rich mixture is the extremely messy running, a large proportion of solid fuel being ejected through the exhausts.

It must be remembered, however,

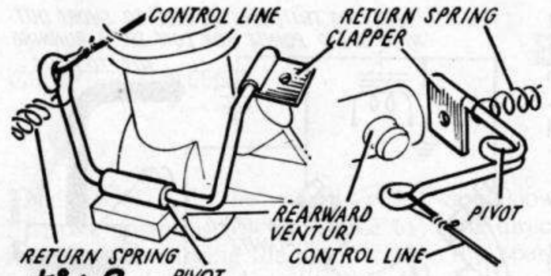
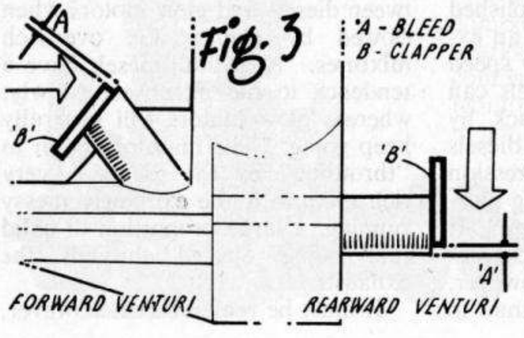


Fig. 2

that engines with sub-piston induction cannot be throttled by "choking" since whilst the air supply through the intake pipe is restricted by this method, additional air is drawn into the crankcase through the exhaust at the top of the stroke and hence the final mixture remains on the weak side. Some measure of speed control may be produced by choking, but seldom can a marked degree of speed difference be obtained for the slow running mixture can never be made rich enough if the needle is set for optimum lean mixture for normal running.

Simple Clappers

A simple clapper valve merely consists of a flat disc which can be lowered over the end of the intake tube—*Fig.* 2. The valve must seat reasonably well, but not completely



seal off the air supply. Rather than rely on an indifferent seating to give the necessary air bleed it is better to make the clapper seat quite well and pierce it with a small bleed hole, *e.g.*, about $\frac{1}{32}$ in. diameter. The size of this hole can then be adjusted to produce consistent slow running with the clapper over the intake.

Depending on the individual design of engine again, it is possible to use a clapper valve to give a range of engine speeds, by varying the final position of the clapper offered up to the intake. In other words, the amount of air induction is modified by the proximity of the clapper. The clapper itself need not be pierced, the air bleed in the fully shut position being given by a slight clearance between the rim of the clapper and the induction pipe. Quite small movement of the clapper may then produce a marked variation in engine speed but the system is rather difficult to adapt to variable speed control by servo mechanisms. It would be more easily worked on the principle of sliding the clapper over the end of the intake, rather than lowering it in position—*Fig.* 3.

Throttles

Two proprietary throttles produced some years ago in America operated on the "choke" principle—the Drone throttle—*Fig.* 4*a*—designed specifically for the Drone diesel, and the Redwing speed control. The latter was designed for linkage to the

bellcrank on control line models so that a violent manoeuvre threw the control quadrant over to the "choked" position when it could be restored to normal running position by a sharp pull on the line—*Fig.* 4*b*. Operation is self-explanatory from the illustrations. The Drone control, however, was more useful as a motor cut-out since throttling could only be effected by screw adjustment of the plunger and thus required several turns either way to change from normal running to rich and back again.

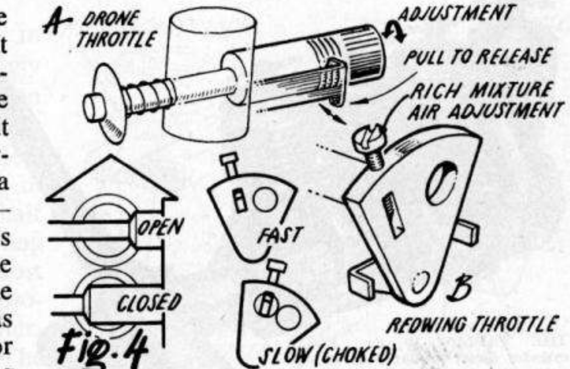


Fig. 4

A favourite American method of achieving two-speed engine control is to employ twin spraybars and needle valves, one, usually the top, being adjusted for a rich mixture and the other for optimum lean mixture. The fuel supply is then switched from one to the other spraybar, as required, to select the required speed. Provision can also be made to incorporate an air bleed in the switching system so that two engine speeds and "stop" are available as controls. A typical method of achieving this is shown in *Fig.* 6 which utilises a Bonner (American) escapement type valving switch, designed for radio control work.

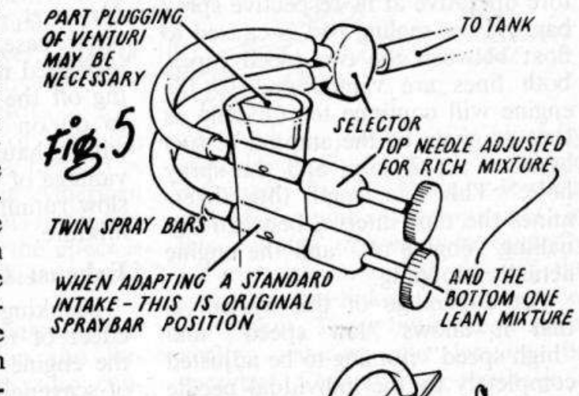


Fig. 5

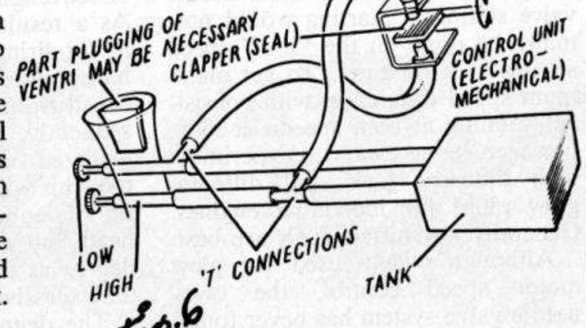
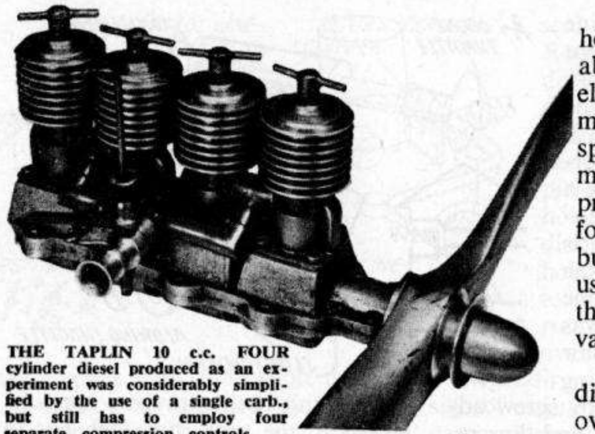


Fig. 6

Escapement Controls

In the Bonner motor control unit both spraybars are connected to the tank, but each has in it a "T" joint connecting to open-ended pipes on the contro unit. The escapement controls the position of



THE TAPLIN 10 c.c. FOUR cylinder diesel produced as an experiment was considerably simplified by the use of a single carb., but still has to employ four separate compression controls.

a sealing pad which blanks off one or other of the bleed tubes. Whichever line is thus sealing off is therefore operative at its respective spray bar. If the sealing pad is caused to float between the two bleed pipes, both lines are vented and so the engine will continue to run only as long as given by the amount of fuel between the T joint and the spray hole. This line length thus determines the time interval between signalling "engine off" and the engine actually stopping.

An advantage of this system is that it allows "low speed" and "high speed" running to be adjusted completely by the individual needle valve settings. Starting would normally be done on the "low" speed setting (rich mixture). To get maximum speed difference, with consistent running at both speeds, it may, however, be necessary to experiment with different fuels and different glow plugs for individual engines. Generally low nitrate fuels are best.

Although widely used for glow motor speed control, the twin-needle valve system has never found great favour with diesels. It does, however, work reasonably well on most diesels. Similarly, glow motors will normally respond to the "clapper" method of choking to produce a rich mixture for slow speed running, but this method is less used with glow engines than the twin needle valve set-up.

An objection with diesels to running on an over-rich mixture is the danger of getting too much solid fuel in the cylinder head leading to over-compression. Some diesels can tolerate running with the head very wet. Others cannot. In the latter case, speed control may be produced more effectively by blanking off the exhaust. This is easiest to do on engines which are fitted with exhaust stacks and has the advantage of being far less messy than slow running on an excessively rich mixture.

Exhaust Choke

Blanking off the exhaust has the effect of reducing the efficiency of the engine by reducing the amount of scavenging which can take place. As a result the fresh charge drawn in for firing is adulterated with exhaust gases which have not escaped from the previous cycle. Consequently, there will be less power gathered on the fire stroke. Again, too, this will lead to a similar build-up of unburnt (solid) fuel in the head, but not normally to the same degree as that given by running on an excessively rich mixture.

The degree to which the exhaust area must be blanked off to achieve

any appreciable drop in speed may be surprisingly high. Most exhaust areas on model engines are larger than necessary for complete scavenging to start with, in any case. Blanking off half the port area (*e.g.*, sealing off one of a pair of exhaust stacks) will normally produce no appreciable difference in running speed. The exhaust area may have to be cut almost completely off before a marked loss of speed is produced—*Fig.* 7. Also a good seal may be necessary on such a device. Depending on individual engine designs, however, this method can be effective for variable speed control, variable on the degree of blanking off. The exhaust choke method is mainly applicable to the American "35" size engines; but the noise can be deceptive, often the effect is more of silencing than speed control.

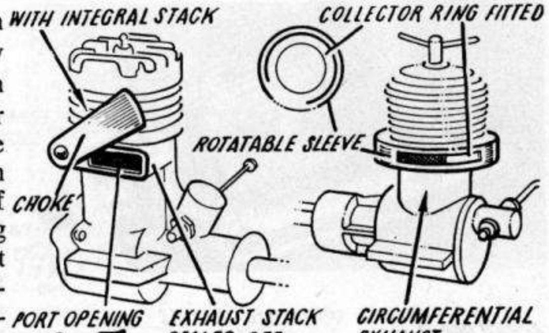


Fig. 7

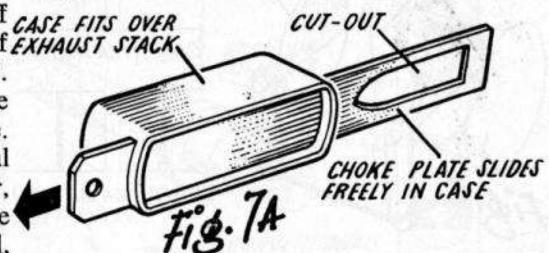
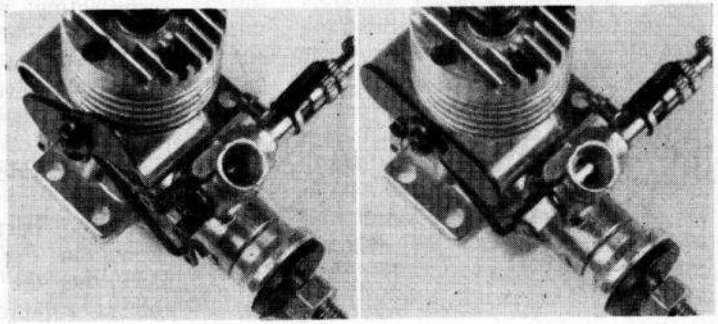
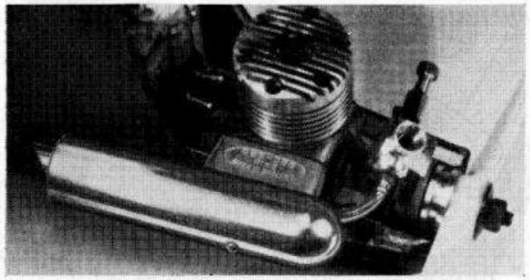


Fig. 7A

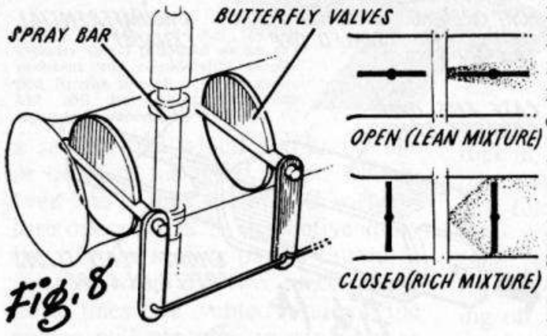
A method of speed control which has become popular on the larger diesels (mainly for radio control work) is the use of a butterfly valve in the intake tube. A scheme developed by G. Honnest-Redlich and Electronic Developments in this country utilises twin butterfly valves, one on either side of the spraybar, and linked to have parallel movement. These butterflies control airflow entering and leaving the spraybar. Closing them results in a

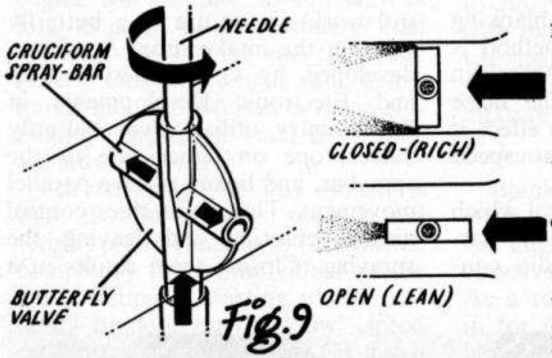


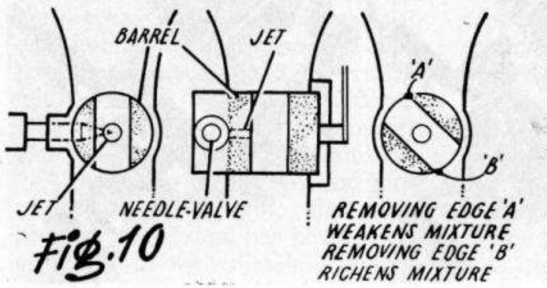
Open and shut positions of a Japanese O.S. coupled exhaust baffle and rotary intake barrel on an O.S. 10 R/C engine. This gives an excellent range of control.



SOME SILENCERS GIVE AN improved fuel economy and better idling control. One of them is the Gee Dee Pike unit seen on a Merco 61 here (refer to page 182).





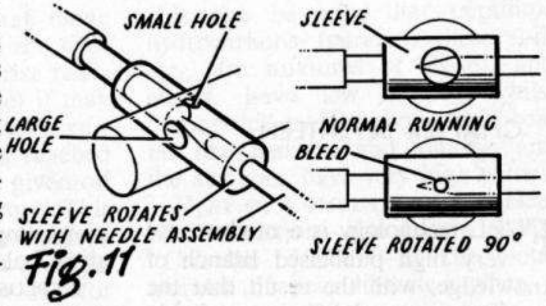


richening of the mixture, and vice versa. The butterflies must be quite snug fitting for maximum effect in the "closed" position and represent a certain difficulty in assemblage within the intake tube, but the external linkage necessary to operate them is simple, and power to move them low.

Much the same effect can be achieved with a single butterfly valve with a single "T"-shaped spraybar mounted on it—*Fig.* 9. Here the spraybar is rotated with the butterfly from fully closed (maximum rich) to fully open (maximum lean), the actual mixture being determined by the setting of the needle valve proper as controlling the fuel supply to the bar of the "T".

A valve of this type with the greatest possibilities is the barrel valve, although representing more of an "engineering" job in the matter of manufacture. In general, however, it is rather easier to make and fit a barrel-type valve than single-or double-butterfly units.

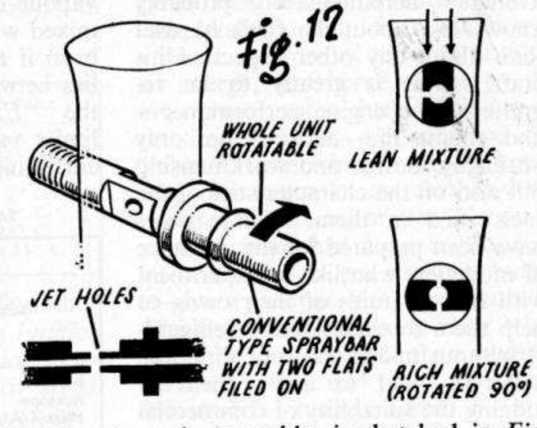
A true barrel valve is shown in *Fig.* 10. The barrel fits the choke tube and is rotatable, the relatively large hole drilled through it normal to its axis of mounting thus providing an adjustable air opening. The fuel jet opening terminates in the throat of the barrel opening, the mixture setting being controlled by a normal needle valve adjustment.



Quite accurate adjustment of the mixture can then be obtained by trimming the edges of the barrel. In the closed (slow running) position, removing metal from edge "A" weakens the mixture, whilst removing metal from edge "B" richens it. It is thus possible to arrive at an optimum mixture for slow running for a required needle valve setting (*i.e.*, for optimum "high speed" running mixture).

The proprietary Mills throttle—*Fig.* 11—works on a similar principle, the needle valve being surrounded with a rotatable sleeve. Actually, needle valve assembly and sleeve rotate as one unit, for convenience. The sleeve is then bored through with a large diameter hole (the same as the diameter of the venturi throat) to line up with the intake for normal running; but when rotated 90 degrees blanks off the intake except for a small bleed hole. In this position the air supply is considerably restricted and so the mixture is much richer for the same needle valve setting.

An adaption of the barrel valve principle, using in effect on ordinary spraybar, slightly modified and



made rotatable, is sketched in *Fig.* 12. Here the centre portion of the spraybar is formed into two flats to serve the same function as the hole through the barrel valve (except that the spraybar may not completely fill the intake tube and so some air passage is left on each side of it). Rotating the spraybar both "throttles" the air supply and also slightly affects the mixture due to the displacement of the jet holes relative to the intake axis giving, in effect, a variable mixture control between "open" and "shut" from fast (optimum lean mixture) to "slow" (maximum rich mixture). Like the barrel valve the final mixture for slow running can be finely adjusted by trimming the edges of the flats.

CHAPTER SEVENTEEN

Fuels

FUEL technology is a modest, not very high publicised branch of knowledge, with the result that the average aeromodeller probably knows less about the fuels he uses than about any other aspect of his craft. This is greatly to be regretted since engine performance—and engine life—depends not only on engine design and workmanship but also on the characteristics of the fuels used in them. These notes have been prepared for the guidance of modellers who like to experiment with fuel mixtures of their own—to help them to experiment intelligently without undue waste of time and materials—and to assist them in judging the suitability of commercial brands of fuel for whatever purpose they may have in mind. No attempt has been made to write a "Formulary" or to review existing commercial fuels. What has been attempted is a concise and simplified account of the properties and functions of the major fuel ingredients, and an outline of the basic scientific principles to be followed in working out the design of a fuel for any particular purpose.

Before it is possible to proceed to the formulation of a satisfactory "Diesel" or "Glo" fuel it is necessary to be familiar with certain fundamental properties of fuel components such as *"Flash Point"*, *"Heat of Combustion"*, *"S.I.T."*, etc., and a short explanation of the more important of these terms is given below.

EXPLOSIVE LIMITS. When the vapour of an inflammable liquid is mixed with air the mixture will only burn if the concentration of vapour lies between certain limits known as the *"Explosive Limits"*. These limits vary considerably for different liquids, as shown in Table 1.

TABLE I.—EXPLOSIVE LIMITS.

Substance	Explosive Limits per cent of vapour in the air	
	Lower	Upper
Benzene	1.35	8
Acetone	3	13
Methyl Alcohol (Methanol) ...	5.5	21
Ethyl Alcohol (ordinary alcohol)	2.8	9.5
Ethyl Ether	1.7	48
Paraffin Hydrocarbons	about 1	about 3.5

Taking Methanol as an example it can be seen that if the concentration of methanol vapour in the air is less than 5½ per cent the mixture will be too weak to fire, whilst if it exceeds 21 per cent the mixture will be too rich.

FLASH POINT. *"Flash Point"* is a measure of the inflammability of a liquid. If a little inflammable liquid is placed in the bottom of a small metal cup it will give off vapour into the air space above it. If this concentration reaches the lower explosive limit the mixture of air and

vapour will "flash" if a small flame or spark is brought above the cup. If the liquid does not vaporise readily (paraffin oil for example) it may be necessary to warm it until a certain critical temperature is reached at which enough vapour is given off to form the explosive mixture. This temperature, below which ignition will not take place, is known as the *"Flash Point"*, and varies widely for different liquids, as shown in Table 2.

TABLE II.—FLASH POINTS.

Liquid	Flash Point
Ethyl Ether	—41° C.
Benzene	—21° C.
Acetone	—17° C.
Toluene	— 2° C.
Methanol	0° C.
Butyl Acetate ...	25° C.
Paraffin and Diesel Oil ...	about 65° C.

SPONTANEOUS IGNITION TEMPERATURE. Also known as *"Self Ignition Temperature"*, *"Auto-Ignition Temperature"*, and *"S.I.T."* for short. This is the temperature at which a mixture of inflammable vapour and air will ignite *without* the application of a flame or spark. S.I.T. is totally unrelated to the Flash Point, and should not be confused with it. Table 3 gives some typical values.

TABLE III.—SPONTANEOUS IGNITION TEMPERATURES.

Substance	Self-Ignition Temperature
Acetone	630° C.
Benzene	580° C.
Toluene	553° C.
Ethyl Acetate	484° C.
Methanol	475° C.
Ethyl Alcohol	421° C.
Amyl Acetate	379° C.
*"Petrol"	280° C.
Coml. Diesel Oil ...	240° to 260° C.
Paraffin	about 250° C.
High Cetane Val. Gas Oil	220° to 240° C.
Ethyl Ether	188° C.

It can be seen that paraffinic hydrocarbons (paraffin, diesel oil, etc., are mixtures of these), and ethers, have low S.I.T.'s whilst "aromatic" hydrocarbons from coal tar like benzene and toluene, and the alcohols, have very high values.

HEAT OF COMBUSTION. The Heat of Combustion—also known as the *"Calorific Value"*—is the total amount of heat liberated when a given quantity of a substance is completely burned. It is, therefore, a direct measure of the total intrinsic energy, and hence of the available power, of a fuel. Some approximate values are recorded in Table 4, from which it can be seen why, for example, an alcohol fuel requires larger carburettor jets than petrol; more fuel must be flooded into the cylinders per stroke in order to give a comparable power output. The figures also make it clear why alcohols run "cooler" than hydrocarbon fuels, and are therefore favoured for racing engines.

OCTANE VALUE. Pure Iso-Octane is a very good anti-knock fuel for spark ignition engines, since it has a high S.I.T., whilst Pentane, with a very low S.I.T. is a bad fuel. Other fuels are compared as regards performance with mixtures of iso-octane and pentane and thereby given an "Octane" rating. If the fuel is as good as iso-octane its Octane Value is 100, whilst if it is only as good as a mixture of equal parts iso-octane and pentane its Octane Value is 50.

* This refers to a straight-run petroleum fraction of low Octane Value, before "leading" or admixture with benzene, etc.

A good commercial petrol will be higher than 280°, and an aviation spirit higher still.

CETANE VALUE. This is a method of assessing the values of diesel fuels by comparing their performance in a test engine with mixtures of different proportions of the excellent diesel fuel cetane and the very poor diesel fuel methylnaphthalene. Cetane and Cetene Values may also be calculated indirectly from the specific gravity and Aniline Point of the fuel, but this method is not applicable if "dopes" are present. A high Cetane Value means a low Octane Value, and vice versa.

IGNITION LAG. When a mixture of a diesel fuel vapour and air is raised to the Self Ignition Temperature, there may be a considerable delay before the explosion actually takes place. This time interval is known as the "Ignition Lag" and for smooth running should be small. The running characteristics of a poor fuel may be enormously improved by reducing the ignition lag by making small additions of certain "dopes". This must not be overdone since *too* short an ignition lag causes detonation, etc.

LIQUID FUELS for internal combustion engines are of two fundamentally different types, namely those to be fired by spark or hot-wire ignition and those designed to ignite under the heat of compression alone, without the application of a spark or other local hot-spot. The former fuels, of which petrol is the commonest example, should contain a low-boiling fraction (the "light ends") of low Flash Point to ensure starting from cold, but must have a *high* S.I.T. to prevent firing taking place under compression alone before the spark passes. The second type of fuel, for use in Diesel engines, need not possess a low Flash Point but *must* have a low S.I.T. It follows that a good petrol will be a bad diesel fuel—and vice versa.

Miniature Diesel Fuels

The diesel fuels used in road transport vehicles are fairly high-boiling fractions from natural petroleum consisting mainly of certain types of "paraffinic" hydrocarbons. Such a "gas oil" has a Spontaneous Ignition Temperature around 250° C. and when forced into the cylinders in finely atomised form will fire satisfactorily under the high-temperature conditions prevailing in these very high-compression full-scale engines. But they will not ignite in a model "Diesel" unless it is hot, and to enable miniature compression-ignition two-stroke engines to be started it is customary to add a proportion of Ethyl Ether, which combines the phenomenally low S.I.T. of 188° C. with very wide Explosive Limits. Since the miniature "Diesel" is a two-stroke engine, lubricant must also be incorporated in the fuel. Finally, to ensure smooth even running it is often advantageous to include a small proportion of a further component, the "dope". It is worth while to study in some detail the functions and properties of these *four vital components*.

(1) *The Paraffinic Base-Fuel*.

This is the main ingredient of the fuel. Its function is to provide most of the energy of the fuel, and it should therefore possess high Calorific Value and low S.I.T. Reference to Table 3 will show that, with the exception of certain ethers, the only readily available substances with relatively low S.I.T.'s are the

paraffin hydrocarbons—which fortunately also possess very high Calorific Values. Ruling out individual pure hydrocarbons like pentane, hexane, heptane, etc., on the grounds of expense, this virtually narrows down our choice of base fuel to paraffin oil, commercial diesel oil and special high cetane gas oil fractions, if available. There is little to choose between paraffin and diesel oil, the latter having its higher viscosity and greater "oiliness" to recommend it. It can be seen, partly by reference to Table 3, that the addition of petrol, benzene, toluene, naphthalene, turpentine, white spirit, or in fact any of the fantastic materials that have from time to time been recommended, must of necessity make the fuel worse, because of the high S.I.T.'s of these substances.

TABLE IV.—CALORIFIC VALUES.

Substance		Heat of Combustion (calories)
HYDRO-CARBONS:	Paraffin Oil ...	11,000
	Diesel Oil ...	10,900
	Petrol ...	10,000
	Benzene ...	9,960
ETHERS:	Ethyl Ether ...	8,800
	Methylal ...	7,900
KETONES:	Acetone ...	7,300
ESTERS:	Ethyl Acetate	6,100
ALCOHOLS:	Ethyl Alcohol	7,080
	Methanol ...	5,330
NITRO HYDRO-CARBONS:	Nitrobenzene	6,030
	Nitromethane	5,370
	Nitroethane	4,300
	Nitropropane	2,790
ETHYL NITRITE		4,450
ETHYL NITRATE		3,560

Their use to "deaden down" the detonation of the ether is a case of two wrongs failing to make a right: a fuel that needs deadening down has got far too much ether in it.

(2) *The Lubricant*.

The lubricating component of the fuel may be any good quality lubricating oil, either mineral or vegetable. The only limitation imposed by vegetable oils like Castor Oil is that alone, they will not blend with paraffin base fuels; castor oil can be used only in a fuel ready-mixed with ether, which will keep all the components in solution. There is scope for experimenting with different grades and qualities of oil.

With regard to the *quantity* of oil to incorporate in the fuel, this again is a matter for experiment. Many miniature engine fuels are grossly over-lubricated, with the result that they are unnecessarily messy in use, and also require more ether than they otherwise would. In designing a diesel fuel it should be borne in mind that the oil has one function only—to provide adequate lubrication—and that it should not be expected to burn, to moderate the explosive tendencies of excess ether, or to do anything else. A two-stroke motor-cycle engine runs on the road for long periods at a time under much greater (and varying) loads than any model engine, and with considerably greater bearing and piston speeds, yet seldom does the percentage of lubricant in the fuel exceed 7½ per cent. It is desirable in formulating a model diesel fuel to increase this proportion for the following reasons: (1) a new engine may have tight spots and require excessive lubrication till it is run-in; (2) in a very old, or badly made engine, the piston may be a poor fit in the bore, so that a fairly thick viscuous fuel is needed in order to seal the compression, and (3) the manufacturer must allow a reasonable safety factor. Point 2 normally affects only the ease of starting: once the engine has been started it will usually continue to run perfectly satisfactorily even on a very thin fuel. With old engines starting can usually be facilitated by injecting a drop or two of lubricating oil through the ports.

For a normal fuel for use in a run-in engine in good condition, oil percentages in the region 30 per cent to 50 per cent are unnecessarily high. If the aeromodeller experiments with proportions of oil in the range 12 per cent to 20 per cent for racing blends and 20 per cent to 30 per cent for general-purpose and running-in fuels, he will not go far wrong. Diesel oil based fuels tend to require rather less than those blended with paraffin.

(3) *Ether*.

Apart from its low S.I.T., which enables it to start easily, and its wide Explosive Limits which ensure that throttle settings are not critical, *ether is a bad diesel fuel*. It has a considerably lower Calorific Value than the paraffinic base-fuel and it detonates or "knocks" badly. Excess of ether means correspondingly less base-fuel in the formulation, and hence a fuel of lower calorific value than need be, whilst its detonating propensities when present in excess cause diesel knock and impose undue strains on the con-rod. Ether should, therefore, be added to a diesel fuel for one purpose only, namely to make the engine start. Just enough for this purpose should be added—*and no more*. Thirty per cent to 35 per cent is excessive, and modellers are recommended to experiment in the range 20 per cent to 30 per cent. It cannot be overstressed that the function of the ether is solely to bring about easy starting; it should not be expected to usurp the function of the base-fuel.

There seems to be some confusion regarding the grades of ether suitable for use in fuels. Ether is manufactured from ordinary ethyl alcohol, two molecules of which join together, with the elimination of water, thus:—

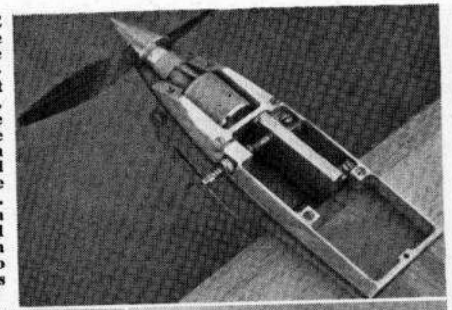
$$\underbrace{\overbrace{C_2H_5\text{–}O\text{–}H + H\text{–}O\text{–}C_2H_5}}_{\text{2 Ethyl Alcohol}}$$

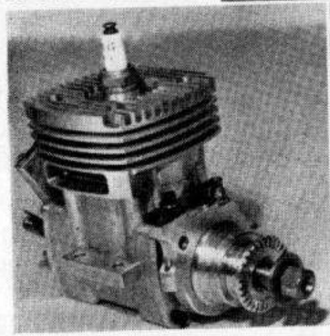
$$\longrightarrow \quad C_2H_5\text{–}O\text{–}C\text{–}_2H_5 + H_2O$$

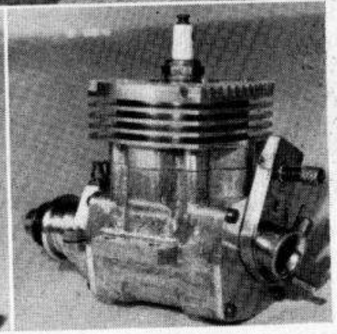
1 Ethyl Ether Water

The process is usually carried out by heating the alcohol with concentrated sulphuric acid, which absorbs the water formed—which is why the product is sometimes called "sulphuric ether". The ether which distils over is washed free from acid, purified, dried and re-distilled. It therefore contains no acid whether it is sold as "Anaesthetic Ether" "Ether .720", "Ether B.S.S. 759". "Sulphuric Ether", or "Ether Meth.". All these materials are, effectively, the same thing; and if properly manufactured are all harmless to model engines. The .720 refers to the specific gravity of the product and shows the substantial absence of water; B.S.S. 579 refers to the appropriate British Standards Specification laying down the standard of purity; "Ether Meths." indicates that the ether was not manufactured from pure ethyl alcohol but from methylated spirits, which contain a few per cent of

ENGINES WHICH ARE specially fuel sensitive are the high performance types used for team racing and speed. At right is Drazek's M.V.V.S. diesel in "Orion". Note the neat mount and the slim 10 c.c. tank with fuel filler valve supported in the mount. Below are two views of the R. Kinnersly 10 c.c. ultra short stroke engine tested on petrol/ignition with slide throttle control to establish a true basis for further experiment.







methanol—this will give traces of methyl-ethyl and di-methyl ethers in the product, which are not harmful. Anaesthetic ether is made from pure alcohol and usually contains a proportion of deliberately added alcohol, and sometimes other additives, to prevent peroxide formation on storage. It is more expensive than other grades and, if anything, is slightly less suitable for fuel work.

The di-ether, Methylal, with the chemical formula $CH_3\text{–}O\text{–}CH_2\text{–}O\text{–}CH_3$, may be used partly or wholly to replace ethyl ether in certain specialised fuel formulations. The higher ethers Amyl Ether and Butyl Ether are too high boiling to be valuable alone, but may be used mixed with ethyl ether. Isopropyl Ether, unlike the straight-chain ethers above, has a very high S.I.T. and is not suitable for use in diesel fuels. It is a possible ingredient of glo-fuels.

(4) *Dopes*.

There are a number of well recognised "dopes" which may be added to diesel fuels, best known of which are

- Ethyl and Amyl Nitrites
- Ethyl and Amyl Nitrates
- 3–Chloro-ethyl Nitrate
- Paraldehyde
- Various organic peroxides like Tertiary Butyl Hydro-Peroxide, Di-Tertiary Butyl Peroxide, etc.

The choice of dope is usually determined by price and availability.

The function of the dope is to reduce "Ignition Lag" and thereby give smooth, powerful running. Very little dope is needed for this purpose, the precise amount depending on the particular fuel formulation, and is a matter for experiment in each case. Seldom is more than 3 per cent required, and modellers would be well advised to start with about 1 per cent of dope and gradually increase, by not more than ½ per cent at a time up to a maximum of about 2½ per cent, until smooth even running is obtained—and then to *stop*. This is a case of "a little of what you fancy does you good" —but a little bit more can play hell. Dopes should be used *solely* for the purpose described above and should under no circumstances be used in excess to assist starting. They do, indeed, lower S.I.T. somewhat, but their effect in this direction is most marked with the first few per cent and then falls off very rapidly. It should be remembered that nitrate dopes are, in effect, high explosives and that when they burn they generate nitrous fumes. An overdoped fuel requires the compression setting of the engine to be drastically reduced as the engine warms up, it sets up unnecessary strains in the engine, and it is corrosive.

A proprietary brand of fuel will be a carefully balanced blend of ingredients with the correct amount of dope; no attempt should be made to "improve" it by further dope additions.

Following the basic principles discussed above, and bearing in mind that each component of the mixture has its own specialised part to play in the performance of the final fuel, it is now possible to set about designing a good diesel fuel for a particular engine or for a specific purpose. A good running-in fuel for new engines and for general purpose flying would look something like this:

Paraffinic Base Fuel	45-60%
Lubricant	20-30%
Dope	1-2½%
Ether	20-25%

whilst a Racing or Competition fuel might well be:

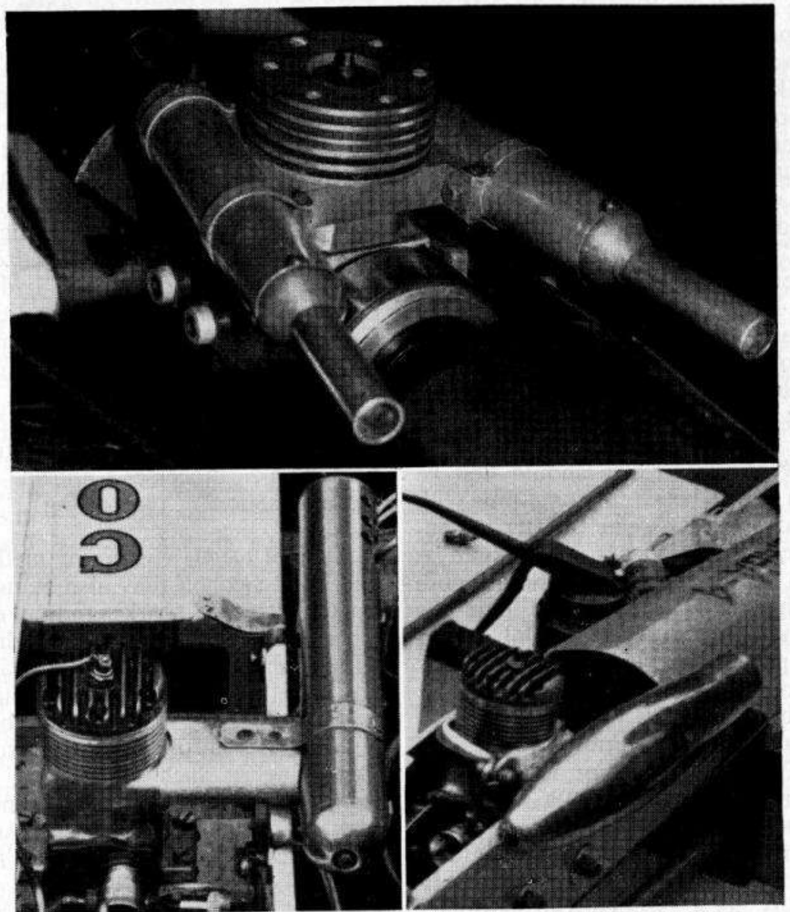
Paraffinic Base Fuel	55-65%
Lubricant	12½-20%
Dope	1-3%
Ether	20%

If the fuel is of the ready-mixed variety all the ingredients are mixed together, and the lubricant may be castor oil. But if the fuel is to have its ether added immediately before use, only the first three components are mixed in each case; in which event mineral lubricant must be employed.

Starting with either of the above basic formulations as a guide, the ideal fuel for a particular purpose and individual engine can readily be worked out on the test bench by modifying the components of the appropriate formula a very few per cent at a time until optimum performance is obtained. It should be borne in mind that the perfect fuel for one engine may not be ideal for another with totally different design characteristics, and the really scientific flying enthusiast will study the individual fuel requirements of all the more important engines in his "armoury". It should also, of course, be appreciated that different fuels may require different starting and running settings—and the careful experimenter has to develop a considerable amount of patience.

Running-in: Engine Temperature

It follows from the increased proportion of base-fuel and the reduced proportion of ether that a "racing" fuel will run hotter than a running-



HYDROPLANE ENTHUSIASTS WERE THE FIRST TO HAVE TO USE the silencer. These three units are typical of those engineered to fit special, and McCoy engines.

in or general-purpose fuel, because of its higher Calorific Value. This relatively high-temperature running has been known to worry some modellers, who sometimes attribute it to frictional heat arising from under-lubrication. Any well-formulated racing fuel is, by its very nature, bound to run hot—and it is advantageous that it should do. The efficiency of operation of the internal combustion engine increases, within reasonable limits, with increase in temperature of running, hence the modern practice of cooling full-scale aero engines with ethylene glycol (b.p. 198° C.) instead of water (b.p. 100° C.).

If the modeller is anxious, it is suggested that he feel, not the cylinder head where combustion of powerful fuel is taking place, but the crankcase main-bearing. If this remains moderately cool he need have no fear of a seizure.

WARNING. In fairness to the manufacturer, as well as in his own interests, the modeller should, of course, be careful only to use a fuel for the purpose for which it is intended. A "Competition" or "Racing" mixture is, as its name implies, intended for high-speed work, and the manufacturer assumes his customer will not be expecting to develop maximum power and revs. with a new engine straight out of its box. A "Standard" or "Running-in" fuel should always be used with new engines, which should first be run on the bench for some time with an oversize propeller. After the engine has loosened up it should be run for another half hour or more with a standard prop., still on the same type of fuel. Only after proper running-in, and after a fair amount of work should peak output with racing fuels be attempted.

Spark Ignition Fuels

The usual fuel for an ordinary spark plug engine is "Petrol". A motor spirit which is a simple "cut" from the distillation of natural petroleum—what is known in U.S.A. as a "straight-run gasoline" —consists mainly of paraffinic and naphthenic hydrocarbons boiling over the range 40°–190° C. Because of its high paraffinic content it has a fairly low S.I.T. and tends to "knock" or "pink" badly in a modern high-compression automobile engine. Its low Octane Value is raised by either of two methods. The first is to incorporate a small amount of a dope having precisely the opposite effect of a diesel dope, in order to *suppress* pre-ignition, *i.e.,* to *raise* the S.I.T. Lead Tetra-Ethyl is pre-eminent for this purpose, although when used alone it has the disadvantage of giving hard deposits of lead oxide inside the engine. Modern "Ethyl Fluid" contains ethylene dibromide to minimise this trouble. The second method is to enrich the straight-run fuel by additions of benzene (benzole), toluene, other hydrocarbons of high Octane Value (high S.I.T.), or alcohols. The high octane hydrocarbons may be obtained from coal-tar distillation or from the gasoline itself by various high - pressure, high - temperature "cracking" processes known as "aromatisation", "preforming", "alkylation", etc.

Alcohol blends containing methyl and ethyl alcohols may also be used satisfactorily in spark ignition engines. They perform best in engines with high compression ratios and are therefore most suited to motorcycles and racing cars, where their high S.I.T.'s ensure immunity from "knocking". Such blends are, of course, eminently suited to miniature spark-ignition engines, castor oil lubricant being incorporated for two-stroke engines. The relatively low calorific values of alcohol blends, and their higher price, makes their use for ordinary purposes uneconomic if hydrocarbon fuels are available. But the increased volume of fuel that has to be flooded into the cylinders in order to obtain comparable power output tends to keep the engine moderately cool at high speeds, an important consideration with racing engines. The calorific value of methanol blends may be increased by replacement of part of the methanol by Methylal, the di-ether already referred to above, which is not prohibitive in cost for specialised fuels. Methylal can be used alone as a motor fuel.

The higher the compression ratio of an engine the higher must be the Octane Value of its fuel. But the spark-ignition engine possesses a certain measure of tolerance for poor fuels resulting from the ability to vary the ignition timing—retarding for starting and with fuels of low rating, and advancing for high speeds and with high octane fuels. This flexibility is lacking with glo-plug motors.

Glo-Plug Motor Fuels

The glo-plug engine is without ignition control, and fuel formulation might therefore be expected to be more critical than for spark ignition engines. For maximum racing performance this is undoubtedly true, yet it is surprising on how many weird and wonderful concoctions the average glo-motor will run passably well. A good general-purpose fuel on which any glo-plug engine will run is a simple mixture of

Castor Oil	... ...	30%
Methanol	... ...	70%

but performance may not be outstanding. The castor oil proportion may with advantage be increased for some engines for the preliminary running-in ; it should seldom be reduced below 20 per cent even with well seasoned engines. Methanol does not have the natural inherent oiliness of diesel oil, and glo-fuels must have a higher oil content than diesel fuels. In order to develop the high revs. of which it is capable the glo-engine must be fairly "sloppy", and to ensure adequate compression for starting a fairly oily viscous fuel is needed. Castor oil, and not a blended lubricant like Castrol "R", is to be preferred since it does not contain additives insoluble in methanol and therefore yields a clear fuel without sediment.

A very large number of substances have been suggested from time to time as useful additives to simple Castor Oil/Methanol blends in order to give increased performance. This list includes Amyl Acetate, Ethyl and Amyl Nitrates, Acetone, various cellulose solvents, Nitrobenzene, and many more. Extensive experiments with a host of other materials have led to the conclusion that whilst one or two may have a slight effect in glo-plug engines of early type, most of them are valueless in a modern glo-engine. In work with, for example, the latest type McCoy, replacement of part of the methanol in a methanol/castor oil blend by

Ethyl Nitrate
Amyl Nitrate and Nitrite
Amyl, Butyl, Ethyl and Isopropyl Ethers
Ethyl and Amyl Acetates
Paraldehyde
Acetaldehyde
Nitrobenzene

and many other solvents was found to have *little or no useful effect*, even when added in quite substantial quantities. It is true that in some instances the engine developed a very satisfying staccato note suggestive of increased revs., a very potent exhaust flavour, or both, but in no case was any significant speed improvement recorded by the instruments.

An approach to the problem of improving simple methanol blends can be made by replacing part of the methanol by a fuel of higher calorific value such as benzene, toluene, acetone, ethyl alcohol or methylal. In some cases these materials effect a slight improvement, but usually more in the direction of improved fuel consumption than in increased speed. In any case there is a limit to the proportion of such substances that can be added since, without exception, they have narrower Explosive Limits than methanol; after quite a small percentage has been added the throttle setting may become too critical for reasonably easy control. Furthermore, excess of some of these compounds of high calorific value can cause an engine to run very hot indeed and to eject showers of red sparks, so that risk of seizure becomes very real. Acetone was invariably found to give erratic running, which is surprising.

METHANOL. Some straight methanol castor oil blends have been found to run more smoothly than others. Modellers would be well advised to purchase only the purest methanol. Methyl and Ethyl alcohols come on the market in various "Proof" strengths, *i.e.*, containing varying proportions of water and for best results only 74° over-proof methanol should be used (this contains over 99 per cent of methanol).

METHANOL/CASTOR OIL RATIO. Unlike diesel fuels, the speed is not greatly influenced by variation in the base-fuel/oil/ratio. If a particular engine is adequately lubricated by, say, 20 per cent of oil and 80 per cent of methanol, there is no significant loss of speed when the ratio is altered to 30 : 70. On the other hand, if the former mixture is somewhat under-lubricating the engine, there may be a substantial increase in r.p.m. when the oil ratio is raised.

Nitroparaffins

Whilst most of the substances so far discussed are without any profound effect on the speed of a glo-engine, this is certainly not true of the nitroparaffins. Replacement of part of the methanol in a Methanol/castor oil blend by Nitromethane, Nitroethane or Nitropropane may increase engine speed by between 1,000 and 2,000 r.p.m. In this respect the nitroparaffins appear to be unique—and are indispensable for really high speed work. Unfortunately, they have not been readily available and they have been fantastically expensive, and except in carefully balanced fuel formulation they involve high fuel consumption. Supplies of Nitromethane are not stocked by the model traders and should be ordered through a dispensing chemist or direct from an organisation such as British Drug Houses.

Just why nitroparaffins are so effective is not clear. They have very low energy contents, as reference to their Calorific Values in Table 4 will show. Nitromethane, for example, has only half the Calorific Value of Methanol and a nitromethane fuel might more logically, in fact, be described as a "cool" fuel than a "hot" fuel. Their effectiveness would seem to lie not in their intrinsic energy contents (which are very low) but rather in the extreme rapidity with which this energy and oxygen content can be liberated. In effectiveness, Nitromethane and the Nitropropanes are closely similar on the test-bench, nitropropane possibly giving a slightly more stable mixture under flight conditions. They would appear to be interchangeable in glo-fuel formulations, the choice depending mainly on price and availability. Nitroparaffin blends require a slightly wider throttle setting than non-nitrated blends, and are hence a little less economical in use.

With regard to the *amount* of nitromethane or nitropropane to include in a glo-fuel formulation, it is the considered opinion of many that the proportions sometimes advocated are excessive. A fuel with 25 per cent to 40 per cent of nitromethane, apart from its exhorbitant cost, usually seems to kill off glo-plugs with fair rapidity. Secondly, careful speed tests on a number of engines have shown that at first there is a considerable speed increase when nitromethane is added, but that the effect gets progressively less with each further addition until it becomes insignificant. The experimenter is recommended to start off with a fairly small percentage of nitroparaffin in his fuel mixture and to carry out several speed determinations on his engine. Another mixture should then be prepared with the same base-fuel/oil ratio, but with a few per cent more nitromethane, and further speed readings taken. This process should be repeated with small nitroparaffin increases until there is no further speed increase measurable. In this way the most effective, and at the same time most economical, fuel will be worked out with the minimum waste of expensive materials. It will often be found by trials of this sort that 20 per cent of nitromethane is just as useful as 30 per cent.

The response of an engine to changes in fuel composition depends to a very considerable extent on the design of the engine, particularly as regards timing, porting and compression ratio. One engine may be found on test to be very much faster on a nitroparaffin blend than on a straight castor oil/methanol, whilst the performance of another engine may be found to be almost identical on either fuel. The moral is, clearly, do not run on expensive nitroparaffine blends if a non-nitrated racing methanol blend will give as good results. And equally, a commercial fuel should not be condemned because it does not improve the performance of *your* engine; it may be giving your friend another 1,000 revs. on his engine of identical make. Engine manufacturers are constantly experimenting and incorporating minor design changes so that two apparently similar engines may, in fact, differ noticeably in compression ratio, timing, or both.

Finally, there is ample scope for studying the effect of combining nitroparaffins with other additives like amyl acetate, etc., which are ineffective by themselves. The guiding principle in all such work always being to make only one change at a time, and to make the changes small gradual ones.

Fuel Testing

Smoothness of running, the absence of "missing", etc., can be tested fairly well with a critical ear —although an electronic stroboscope is better if you can borrow one. Adequacy of lubrication can be checked by feeling the crankshaft bearing (not the head!) by holding a plate behind the engine when it is running and noting how much oil is

TABLE V.—ADVISED FUELS FOR AMERICAN RACING GLOW ENGINES.

Engine	Prop.	U.S. Commercial Fuel	Standard Blends (per cent)	Glow Plugs
Thermal Hopper	4 x 6 4½ x 5	Thimble-Drome Racing	30 N.M. 20 C.O. 50 A.	Thimble-Drome Racing Plug
McCoy 19	6 x 9	Supersonic 1,000 This Is It	35 N.M. 30 C.O. 10 N.B. 25 A.	McCoy Hotpoint O & R Racing
Torpedo 19	6 x 10	Supersonic 1,000 This Is It	50 N.M. 25 C.O. 25 A.	O & R Racing McCoy Hotpoint O.K. Long
Fox 19	6 x 10½ 6 x ½ x 10	This Is It	50 N.M. 25-30 C.O. 20-25 A.	O & R Racing McCoy Hotpoint O.K. Long
McCoy 29	7 x 10	O & P No. 4 This Is It	35 N.M. 30 C.O. 10 N.B. 25 A.	McCoy Hotpoint Champion VG-2
Dooling 29*	7 x 9*	This Is It*	40 N.M. 25 C.O. 35 A.	O & R Racing McCoy Hotpoint O.K. Long
McCoy 60	9 x 12	OBR No. 4 This Is It Stardust H.	30-40 N.M. 25-30 C.O. 20-35 A. 10 N.B.	O.K. Long Champion VG-2
Dooling 61	8 x 11 9 x 11	This Is It (with added Nitro)	50 N.M. 25 C.O. 10 N.B. 15 A.	O & R Racing McCoy Hotpoint

* As used by winner Bob Lutker, Texas, U.S.A., at 1954 World Speed Championships, The Hague.
Standard Blends Code : N.M.=Nitro Methane N.B.=Nitro Benzine C.O.=Castor Oil A=Alcohol.

TABLE VI.—ADVISED FUELS FOR BRITISH DIESEL ENGINES.

Manufacturer	Ether per cent	Paraffin per cent	DERV per cent	Tvo per cent	Castor per cent	Redex per cent	Mineral Oil per cent	Amyl Nitrate per cent	A. Nitrite per cent
Oliver ...	30	50			20			3	
E.D. ...	33	33			33				
Allen-Mercury	32.5	40			25			2.5	
Davies-Charlton (Allbon)	30		45		5		25		2.5
Sugden ...	27		55		15				3
Buskell ...	35		30		35				3
Redex Racing Fuel	30			40		25	5 Esso/ Racer TFD 46		
Fixed Head Fuel	70					30			1

ejected, by noting whether the engine slows of its own accord when hot even with correct throttle and compression settings, and by seeing whether the engine runs any better when a few per cent more oil is added to the fuel.

But *speed* cannot be checked by ear—*use instruments*. An electronic stroboscope, if available, is the ideal instrument since it puts no load on the engine and since it shows *variations* in speed from second to second as well as overall average speed. Failing this, use a good Revolution Counter and watch, or Tachometer. The vibrating reed type of Revolution Indicator, if properly calibrated and carefully used, is capable of detecting reasonable variations in r.p.m. at the slower speeds, but is not capable of showing up small speed differences. It is suitable, therefore, for the preliminary experiments with diesel fuels, but is too insensitive at the higher revs. to be of much value in glo-fuel development. In all cases the engine should be reasonably solidly mounted ; a well balanced engine fitted with a properly balanced prop., if firmly clamped in a vice, seldom gives an early reading on a reed indicator: but vibration can kill r.p.m.

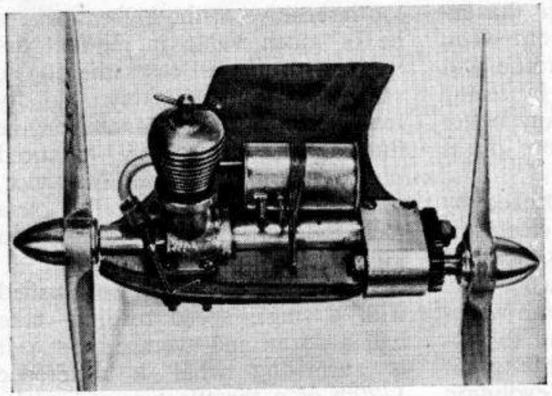
In conclusion, do not be satisfied with a single speed reading—take half a dozen and average them. It is surprising what a difference 1/20th of a throttle turn can make to a precision engine running near its flat-out maximum speed. And check back from time to time the values of your earlier fuel mixtures—the apparent increases in speed you have been getting with the later mixtures may be due to the engine loosening up with prolonged running. Elementary, but it happens every day.

CHAPTER EIGHTEEN

Choosing the Right Propeller

SELECTING the best propeller for a particular engine to go with a particular model can be a very complicated business, if you take into account all the theoretical aspects involved—or just a simple case of "cut and try", using a number of different propellers until you find one which gives the best results. Neither method on its own, however, guarantees that you will get the best possible results. Whatever propeller size may be worked out as being the best possible for a particular design case still has to be proved in practice, and there are so many factors which can affect the result that the finer details can only be worked out by practical tests.

Fortunately, this does not apply to the majority of power models. For everyday or "sports" flying, in fact, the modern engine is far more powerful than it need to be for the job, so if the propeller efficiency is low it does not matter. It may



TWIN PROPS ON THE GERMAN WAF-1 experiment with an extension rear shaft. Rear prop. has higher pitch, working in the slipstream, and contra-rotation eliminates torque troubles.

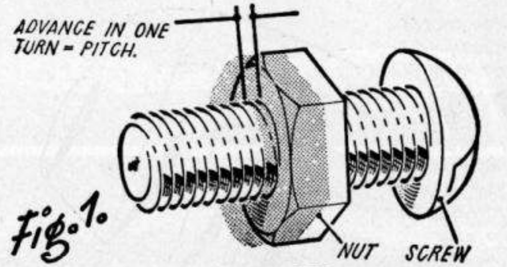
even be a good thing in that it makes the model that much more docile to handle. But for power duration flying and control line work, the right propeller for the job can make all the difference. In control line speed, ultimately it is the best engine-propeller combination which counts (with the propeller size limited to a certain extent by the model characteristics) since performance here is truly an expression of the amount of power the engine is capable of delivering and the efficiency with which it is turned into useful work.

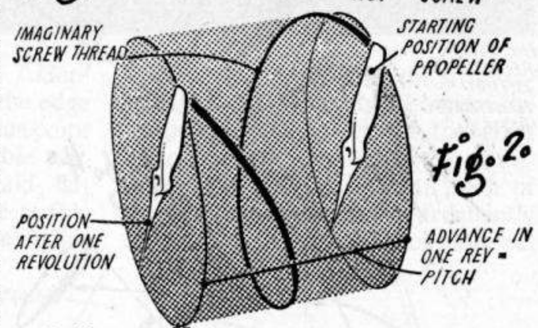
Since practical results are what we are after we do not propose to go into the theory of propeller design and performance but only touch on those aspects, important for an understanding of how to compare and select different propeller sizes. For those interested in the theoretical side we recommend a study of the books and more technical articles previously published on the subject. We would emphasise, however, that full size propeller theory does not hold good in model sizes, particularly where small diameter propellers are concerned. Propeller efficiency appears to drop alarmingly once propeller diameter is reduced to about 6 in. and below. It appears, in practice, that much of this loss of efficiency can be recovered by increasing the pitch of a small diameter propeller. This would appear to indicate that, in the smaller propellers at least much of the useful work in producing thrust is done by the *back* of the blade rather than the front or upper surface of the blade (aerofoil) section. Hence blade section seems to become less important as propeller size diminishes, except for the general rule that the greater the thickness of the blade section the more drag it has and thus the more power it requires to rotate it.

The pitch of a propeller, as quoted, is more often than not an arbitary figure. Pitch is measured on the assumption that a propeller acts like a screw. Rotate a screw in a matching nut or screw thread and it will advance into (or out of) the thread a distance equal to the *pitch* of the thread in one revolution. In this case there can be no slipping between the threads, unless they are stripped, and so the pitch is clearly defined. It is equally clearly seen that pitch can be measured as the distance between two similar points on adjacent threads on the screw—*Fig*. 1.

The pitch of a propeller is defined on similar lines, assuming that

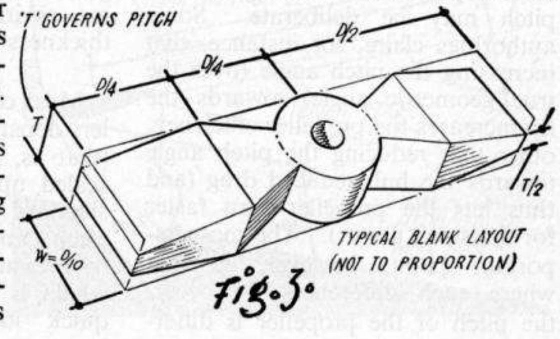
it is screwing itself through some medium where it cannot slip. If easier to visualise, you can consider it as an element of a large imaginary screw thread screwing into a matching thread tapped in "solid" air—*Fig*. 2.





Since the propeller is only an element of the complete screw thread, however, its pitch can only be measured as the theoretical geometric advance per revolution. Mathematically this can be calculated as 44/7 *times* the radius *times* the tangent of the angle of the blade at the radius (*i.e.*, Pitch $= 2\pi R \tan \emptyset$). The possible radii dimensions vary from zero (at the hub centre) to half the diameter (at the tip), with a corresponding different value of the blade angle (Ø) for each. For the pitch to be the same all along the blade from centre to tip, radius *times* the tangent of the blade angle must be the same.

This accounts for the characteristic twist of propeller blades, the greatest blade angle being near the root (where the radius is the smallest) and the least blade angle at the tip. In a carved propeller this necessary "twist" is arranged for in the shaping of the blank—see *Fig*. 3. In machine cut propellers, the necessary twist is incorporated in the movement of the cutters along the blade. But most practical propellers are compromises between a propeller with true pitch angles all along the blade and one which is easy to produce, so that effectively the propeller, as finished, departs from a *true geometric pitch design*. *Fig*. 4, for example, shows the theoretical blade angles required for a 10 x 4 propeller at 1 in. blade stations compared with the blade angles actually produced by carving from a typical blank of the type shown in *Fig*. 3.



Normally these differences are small, and can be ignored. In some

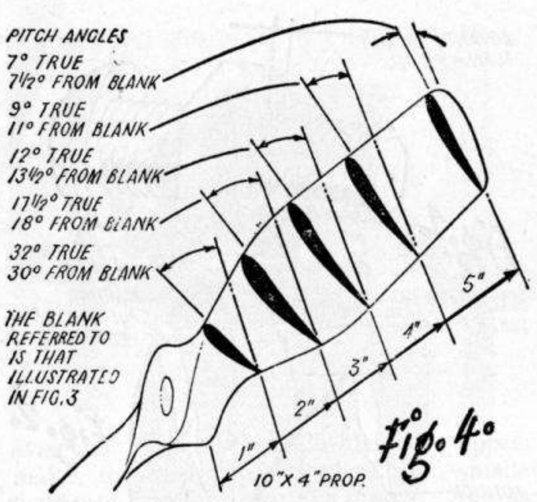


Fig. 4.

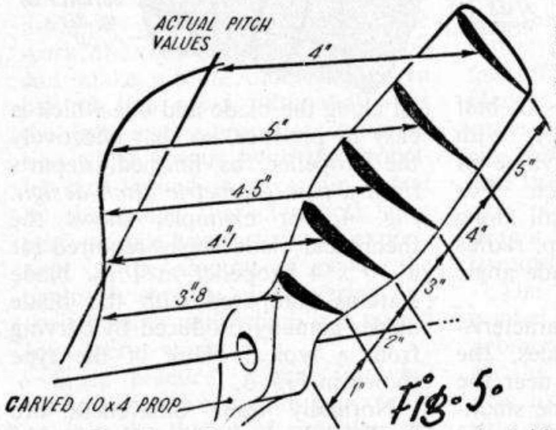


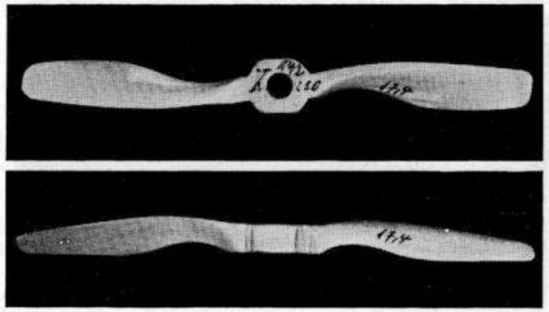
Fig. 5.

cases departure from true geometric pitch may be deliberate. Some authorities claim, for instance, that increasing the pitch angle (over the true geometric angle) towards the tip increases the propeller efficiency; other that reducing the pitch angle towards the hub reduced drag (and thus lets the propeller turn faster for the same power). The most important point, however, is that where such differences do occur, the pitch of the propeller is different along the blade length and so which pitch figure is quoted as being *the* pitch of the propeller, for selection of comparison purposes, will depend on which point this pitch is measured or calculated from. *Fig.* 5, for instance, shows actual pitch of carved propeller of the previous example at different blade stations.

Various "standard" methods of measuring pitch quote .5, .6 and .75 of the full radius (half diameter) as the station at which the propeller pitch should be measured. Measurement at half radius is generally best for model work, since most propeller blanks are laid out on this basis. On power model propellers, too, the depth of the blank, and thus the thickness of the propeller, is left constant from the centre to the half radius and so the thickness of the hub becomes a suitable measure of pitch for any given family of propellers. If the width of the original blank is measured at half radius, as in *Fig.* 6 and also the hub thickness

$$\text{Pitch} = \frac{11 \times \text{dia.} \times T}{7 \times W}$$

Most commercial wooden propellers depart from a true family layout (that is, the blank layout is not scaled up or down exactly from a standard shape) and so nominal pitch cannot be determined simply by measuring the hub thickness (T) which is a more useful method of quick identification with a true



SPEED PROPS., MARKED FOR dia., pitch and r.p.m., used by Czech expert J. Sladky. Note how the root area is trimmed away for speed.

family of propellers, *e.g.*, laid out like *Fig.* 3.

Commercial propellers can be checked for pitch by the method shown in *Fig.* 6 or by the direct method of measuring the blade angle as shown in *Fig.* 7 Here the propeller is held against the edge of a table so that the half radius point comes just level with the table top. A straight edge is then laid flat against the back of the blade at this station. Measuring from a point $\frac{11 \times \text{prop. dia.}}{7}$ from the centre of the blade along the straight edge, and then at right angles out to the table edge gives a measurement of the pitch of the propeller. To save calculation each time the straight edge, *e.g.*, a strip of 1 in. x $\frac{1}{8}$ in. balsa, can be "calibrated" ready for use by marking out as in Table 2 for different propeller diameters, leaving only the pitch dimension to measure direct with a ruler.

One limitation of this method is that the backs of commercial propeller blades are seldom flat and judging the true tangent position of the straight edge can sometimes be difficult, especially with moulded plastic props.

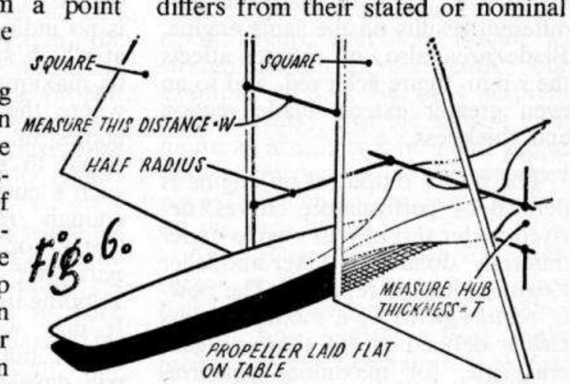


Fig. 6.

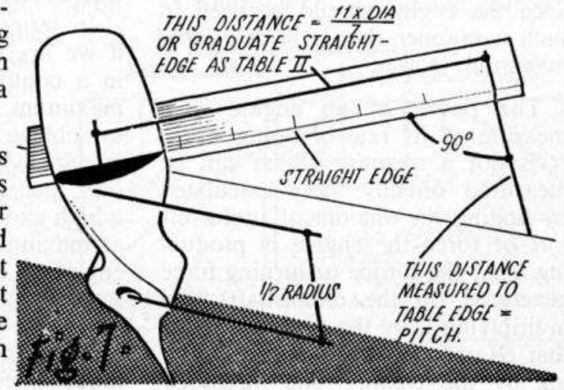
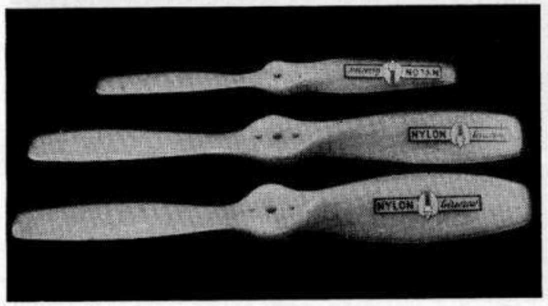


Fig. 7.

We have dealt with pitch determination in some detail since the propeller pitch is one of the chief factors affecting propeller performance, and since the actual pitch of commercial propellers frequently differs from their stated or nominal



FROG FAMILY OF NYLON moulded airscrews, good for general purpose flying and frequently the subject of thinning experiments by free-flight exponents.

pitch, a method of checking for comparative purposes is strictly necessary. It is this difference between nominal and actual pitch which often accounts for why two commercial propellers of the same stated size (pitch and diameter) give different results on the same engine. Blade area also, of course, affects the r.p.m. figure achieved, and to an even greater extent, blade section and thickness.

The power output of an engine is defined by performance curves, derived under test of that engine under different loads—of Aeromodeller Engine Analysis reports. The power *requirements* of a model are less clearly defined except, that as a general rule, for maximum performance the engine should be used in such a manner that it is delivering maximum power.

The power of an engine is a measure of its rate of doing work It is not a quantity which can be measured directly, only calculated by finding the amount of useful effort or force the engine is producing (*e.g.*, the torque or turning force generated on the crankshaft) and multiplying it by the speed at which that effort is applied, *i.e.*, multiplying by the r.p.m. The *measured* performance of torque curves of an engine is thus in many ways more useful data than the power (B.H.P.) curve.

The torque output of an engine varies with speed. As the r.p.m. increase the torque tends to fall off (largely because more torque is being absorbed *inside* the engine in overcoming the increased friction of the faster moving parts). But there is no indication on the torque curve at which speed the engine is giving its maximum performance. That is where the power or B.H.P. curve comes in. Horsepower is proportional to torque *times* r.p.m., and such a curve, if plotted over a wide enough range, always shows a "peak" or maximum value. At this particular speed, the engine is developing its *maximum power output*. It may well be capable of going faster, but at any higher speed it will develop less power.

In terms of a practical example, if we are using a particular engine in a control line speed model, the maximum speed which we can hope to achieve is that particular propeller size which both suits the model (*i.e.*, generates enough thrust with a high enough pitch to fly the model at maximum speed) and allows the engine to operate at the r.p.m. corresponding to peak power. Trimming that propeller to make the *engine* go faster will then *reduce* the model speed. On the other hand.

a propeller which does not let the motor reach peak r.p.m. in flight will again result in loss of speed.

That is the chief significance of the power curve, for any particular engine. We can use it to determine at what *speed* we should operate the engine for maximum performance. For instance, taking the top half or power curve of a typical Aeromodeller Engine Analysis graph—*Fig.* 8—we can see the operating speed of that engine for maximum performance.

Actually we cannot use this figure directly. It refers to the required engine r.p.m. *in flight*. If we used a propeller of the right size to give this speed on the ground, *e.g.*, static running, where we can conveniently measure the r.p.m., the engine speed in the air would be higher because in flight the propeller is "unloaded" to a certain extent, therefore, offers less resistance to being rotated and so the engine r.p.m. increases.

We can only guesstimate what this increase in speed is between static running and "in flight" running. A fair estimate, which seems to work out well in practice, is that r.p.m. goes up by some 10 per cent. To arrive at the required *static r.p.m.* figure for peak r.p.m. in the air we shall not be far wrong if we take this as 10 per cent *less* than the graph figure—see *Fig.* 8 again. This is a general rule for models aiming at maximum performance with a particular engine. For sports models it is generally better to work at a lower r.p.m. figure—say taking 20 per cent off the peak r.p.m. as given on the graph.

Either way we arrive at what is the required operating r.p.m of the engine, and the next job is to select the propeller size to suit.

It can be pointed out that the peak r.p.m. figure arrived at by test is usually typical for other engines of the same type, although there may well be some variations. The peak may well be altered by using different fuels, however, particularly "doped" fuels which enhance performance. But for average selection purposes, test figures are usually an accurate enough guide.

The obvious way of finding the right propeller size is then to try different propellers until we find one which gives the required static r.p.m. figure, but this is not as straightforward as it appears. For example, an 8 in. dia. by 4 in. pitch propeller may give the same r.p.m. figure as a 6 in. x 8 in. pitch prop., both corresponding to the required r.p.m. Which to use is dependent on the type of model.

As a general rule, propeller *diameter* is most important on free flight models and propeller *pitch* on control line models. Free flight models do not normally fly as fast as control line models and perform

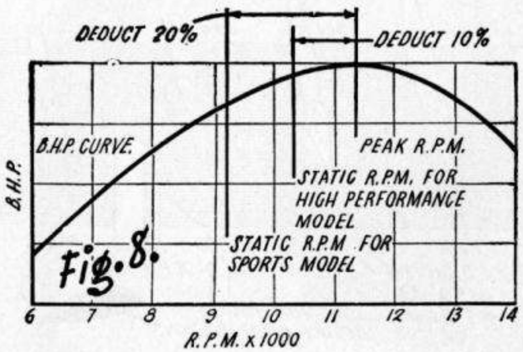
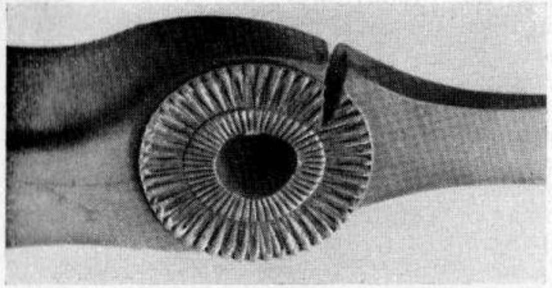


Fig. 8.



WARNING.—THIS IS WHAT happens to a plastic acetate prop. if over-revved. The blade would shear on the next run. Nylon mouldings rarely exhibit this fault.

better with large diameter small pitch propellers. Control line models, on the other hand—particularly speed models—will not be able to attain the necessary speed unless the propeller pitch is high, which means trimming down the diameter to enable the engine to drive the propeller at the required r.p.m.

Suppose we want to match the engine whose power curve is given in *Fig.* 8 for maximum free flight performance. The required static speed is 10,400 r.p.m. so we try it out on a series of commercial propellers, 10 x 4, 9 x 4, 8 x 4, and 7 x 4, with the results shown in *Fig.* 9. The required propeller size in this case is obviously something between an 8 x 4 and a 7 x 4. Try an 8 x 3½, for instance, if available, or an 8 x 4 of another make (the difference in nominal pitch may make it "faster" or "slower" for the same size).

The other method is to trim the propeller which is the next size too large, *i.e.*, the 8 x 4. Nearly all commercial propellers can be increased in efficiency by thinning down the blades, making sure the underside is flat (not convex) and generally working to a lower drag aerofoil section with sharper edges. Such modifications will increase the r.p.m. achieved with the same propeller quite considerably—perhaps even to the extent where it is overdone. In that case start again with the next largest propeller (9 x 4 in this case, and treat similarly). Then any final trimming for speed, if necessary, can be done by reducing the diameter a little at a time.

This method is tedious, but it will give the best results. It means, however, that if you break a propeller you have to re-work another one to replace it. Most people—except the contest experts—prefer to work with the nearest available stock size of propeller and leave it at that. Contest propellers are commonly reworked from standard commercial varieties, or carved individually (when the best material, for a long-lasting propeller, is probably red fibre). An approximate propeller size may be arrived at using stock commercial sizes, the pitch measured and checked and the final propeller laid out from these days for trimming down to final size by practical tests.

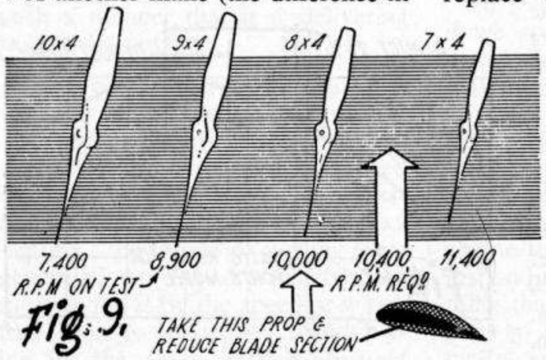


Fig. 9.

Usually a 3 to 4 in. pitch gives best results on free flight models with propellers from 5 to 12 in. diameter. There are exceptions. Some contest modellers prefer to work with higher pitches and reasonably large diameters and trim blade *area* down to arrive at the required r.p.m. figure, giving the "toothpick" type of propeller more usually seen on a control line model.

Lack of pitch on a control line model propeller means that the model cannot fly fast enough—perhaps not even fast enough to keep the lines taut. To fly at 60 m.p.h for instance, with an engine giving 12,000 r.p.m., requires a propeller pitch of at least 6 in. (allowing for

TABLE I.—PITCH AS A FUNCTION OF PROPELLER. Hub thickness for blade layout as Fig. 3.

Pitch	Hub Thickness
4	.25
5	.32
6	.38
7	.45
8	.51
9	.57
10	.64
11	.70
12	.76
14	.89

All dimensions in inches.

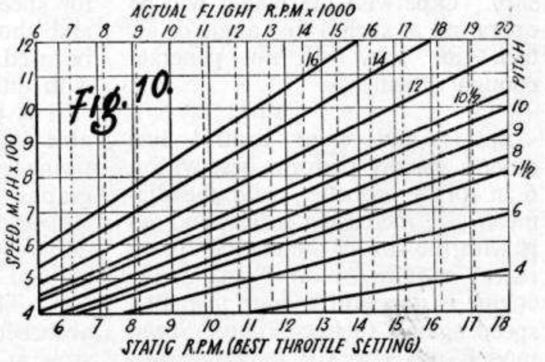
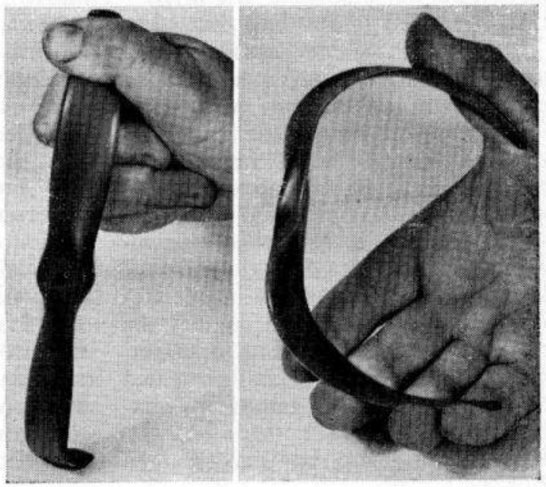


Fig. 10.

TABLE II.—LAYOUT DIMENSIONS FOR STRAIGHTEDGE. (For direct propeller pitch measurement —See Fig. 7.)

Propeller Diameter (inches)	5	6	7	8	9
Scale Dimension (inches)	7.85	9.42	11.0	12.57	14.14
Propeller Diameter (inches)	10	11	12	13	14
Scale Dimension (inches)	15.71	17.28	18.85	20.42	22.00



COMPARISON OF FLEXIBILITY. At left, a Frog Acetate type, and at right, an American coloured Nylon Perm-a-prop by Windsor Eng., both 10 x 6 in. size.

slip). Otherwise the blades will be operating at such a fine angle of attack that they will not generate enough thrust.

Sports and stunt control line models usually perform best with a 6 in. pitch propeller, with the diameter size increasing correspondingly with the size of the engine. Team racers require up to 9 in. if the engine is powerful enough to match speed against fuel economy. Speed models use 9 to $10\frac{1}{2}$ in. pitch propellers, as a general rule. Your cut and try selections can therefore be made with available propeller sizes within these recommended ranges. The graph reproduced in *Fig.* 10, incidentally, gives quite good results for speed model propeller selection and shows the pitches which must be used to achieve different speeds with different engine r.p.m.

It is possible to save a considerable amount of time and trouble in trying out propellers by using the graphs published in the AERO-MODELLER of torque absorption curves for families of different commercial propellers (see Appendix VI). These graphs show the torque absorbed by individual propeller sizes at different speeds. Superimposed over a motor torque curve, speeds at which that particular engine will drive each propeller are given by the intersection of the engine torque curve with the propeller torque absorption curves.

CHAPTER NINETEEN

Equipment and Its Influence on Design

Making Your Own Diesel Engine

HOW many of you have longed to make the perfect model diesel engine, designed to meet exactly your own requirements and incorporating your own ideas, but have cast the thought aside, thinking that the construction of model engines can only be accomplished by skilled experts ? This is not so, for although Dave Sugden, designer of the engine which follows in this section, has made several engines, he is not a skilled machine operator. The information to be presented has simply been gained by experience.

For those who have access to a reasonable lathe may I say that this work is no more difficult than aeromodelling and only demands the same qualities of ingenuity and patience. The less fortunate ones who have no machining facilities will probably find interest in the processes involved, whilst some of the information derives from, and is applicable to, aeromodelling. Earlier chapters have told us the basic design features of production engines, now we can try to fabricate our own miniature two-stroke.

Equipment

The most important machine tool required is a reasonable centre lathe. Without one, making an engine would be almost impossible and the project would become a nightmare. The use of other tools, whilst making certain operations easier is not vital; for a lathe suitably set up is capable of handling all the operations.

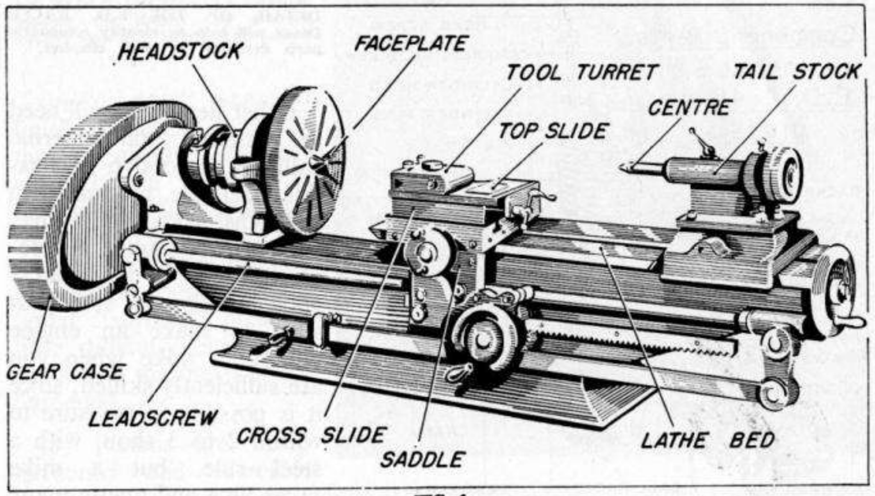


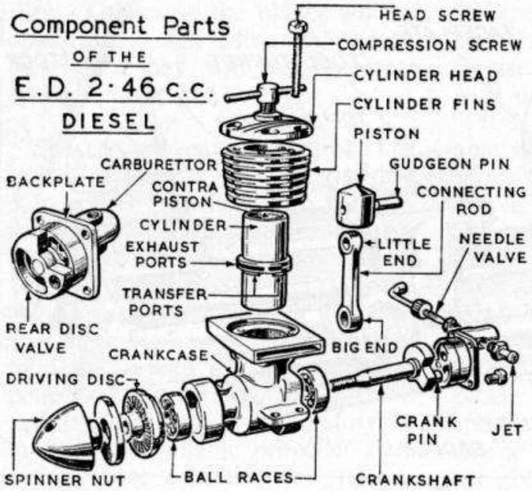
FIG. 1.

The most convenient size is a $3\frac{1}{2}$ in. centre lathe, having the centres set $3\frac{1}{2}$ in. above the bed. Anything much smaller will present serious clearance problems when turning such parts as an engine crankcase of the $2\frac{1}{2}$ c.c. size. The $3\frac{1}{2}$ in. lathe possesses the advantages of being capable of removing excess metal at a higher rate and are less prone to chatter with work of our size. *Fig.* 1 shows the various parts. Lathes of the watchmaker pattern and those not having feed along and across the bed are of little use for our sort of work.

Although a lathe is capable of doing all the jobs required certain other equipment is also necessary. A grinding wheel is almost essential and one with a hard wheel should be obtained. The cheap soft wheels will be found to be useless on high quality high speed steel tools.

A milling machine is also a useful piece of equipment as it can do jobs easily which take a great deal of setting up on a lathe, but a vertical slide as supplied by Myfords for use with their lathes is more useful still, since angular motion in two directions as well as vertical and transverse motion is available.

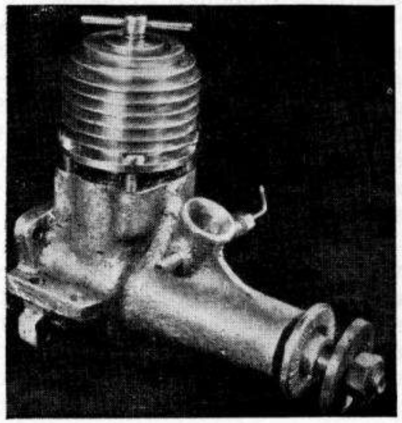
Grinding equipment is a luxury not essential for making engines provided that your ability to produce a respectable turned finish is reasonable. Finishing of cylinder bores, pistons, etc., is facilitated by a small internal grinding and attachment which can be easily made. The largest size of grinding wheel which will fit into the bore is procured from a good engineering shop and mounted either directly on to an electric motor, or a spindle geared to run at about 20,000 r.p.m. The grinding attachment is mounted in



DETAIL OF THE E.D. RACER Diesel will help to identify component parts described in these chapters.

the tool holder and used to remove turning marks prior to lapping.

Certain minor jobs such as drilling carburettor spraybar holes, jets, and exhaust parts are facilitated with a drilling machine or Wolf type drill stand. The drill holder which comes as part of the latter equipment is useful as a means of providing a mounting for the drill on the lathe saddle or vertical slide, for milling operations.



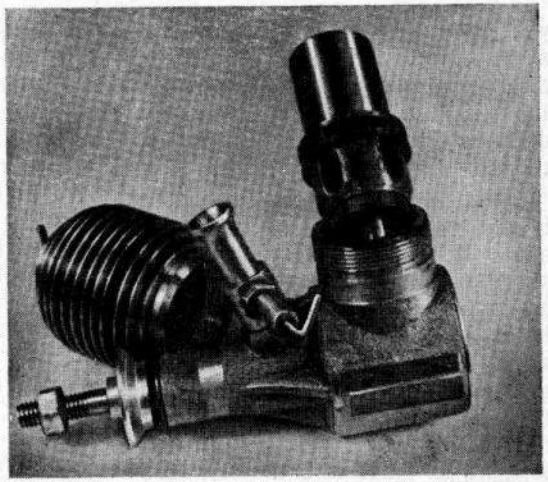
THE SUGDEN 2.5 SPECIAL DIESEL IS THE feature item of these chapters on "making your own" engine. It can be made by anyone with lathe facilities.

Other items you will need are internal and external callipers, a good 6 in. flexible steel rule, graduated in 1/100ths. and 1/64ths., some sort of square, a scriber, centre pop, files and a good micrometer. It is possible to make an engine without a mike when you are sufficiently skilled, since it is possible to measure to within 2 to 3 thou. with a steel rule, but a mike saves time and averts many mistakes. A depth gauge is useful but not vital.

Influence of Available Equipment Upon Motor Design

You may find that because of lack of equipment or skill, certain operations such as milling cannot be carried out. These need not be major stumbling blocks. We shall consider the various machines or operations, the jobs they accomplish and the means of achieving the same object, or even avoiding it, by modification of design. GRINDING is a means of producing a good finish after ordinary turning and is often intermediary between turning and lapping or honing. Its advantage is felt on parts which are case hardened, when applying a good finish by other means becomes tedious. Parts usually ground are (*a*) Cylinder bores; (*b*) pistons; (*c*) crankshaft and (*d*) crankpins.

AN EARLY SUGDEN 2.5 DIESEL showing the drilled and grooved transfer passages referred to below.



(*a*) A good reamered finish is nearly equal to that of grinding so that whether the cylinder is hardened or not it is not too difficult to go from the reamered finish directly to the lapped one. A good bored surface presents little difficulty either, especially if the metal is not hardened.

(*b*) Pistons are never hardened and therefore are easily lapped to fit from plain turning, in fact probably more easily lapped because the volume of metal being removed may be less due to the rough surface.

(*c*) An equivalent finish is easily obtained by careful use of fine files and emery cloth, it being not advisable to case harden crankshafts.

(*d*) Even if grinding were possible it would hardly be convenient to grind this item and a finish is obtained as in (*c*).
does not affect design.

Thus lack of grinding facilities

MILLING is usually carried out in transfer passages, or ports, pistons for lightening, glo-motor cylinder head finning, etc. Lack of a milling machine is no handicap; a lathe will be found to be capable of doing most jobs (to be covered later).

The chief point to watch is that the design is such that the milling cutter can get in to do the operation. On a side port design an Eta type transfer passage cannot be milled out since a suitable cutter will not reach in, and it must be shaped in the casting, a rather tricky process for the amateur. It is best to design this sort of crankcase *a la* K & B engines, making a joint at the top of the transfer passage so that the miller can enter. Alternatively a joint can be made at the base of the cylinder as on the Hornet. Transfer passages affect performance considerably and offer much scope for detail design and experiments. Milling is completely eliminated in a design like the E.D. Racer, and it can be avoided in the Oliver Tiger type of porting by making grooves, using a rotary cutter in an electric drill or by hand in the cylinder liner, instead of the crankcase as on the Javelin. There is also the soldered up tinplate E.D. Comp., Special type and the Elfin layout only requires very careful drilling, aided by using a jig, before the cylinder is drilled and bored.

If you think that milling out ports is going to be too difficult they may be drilled (360° Elfin type) or sawn (K & B or E.D. racer type), and filed out, in which case the cylinder wall will be designed to be as thin as possible to reduce the labour. A drilling jig will have to be made for the Yulon type ports to assist the drill to start on the correct spot on the curved surface.

Milling out between gudgeon pin bosses for lightness is not essential, and if it is felt to be too tedious to set upon the lathe it can either be omitted or effected by drilling small holes, which is not worth the effort anyway. Another system is simply to drill out the inside of the piston to a larger diameter than normal, thus allowing the gudgeon pin end to float, as on K & B motors. Do however, see that the gudgeon pin is strong enough to take the increased bending moment.

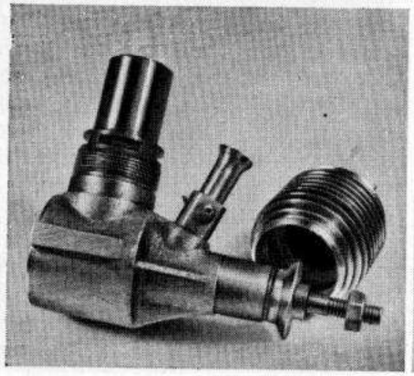
Cylinder head finning as on some glo-motors is easily accomplished in the lathe by setting up the head on the cross slide and mounting the cutter in the chuck.

SCREW CUTTING is usually possible since most self-respecting lathes possess a lead screw, but if yours does not, the parts are merely designed to be held together with screws as in the E.D. Racer or K & B 15. Parts which screw together as in the Elfin are usually lighter and are quicker and easier to manufacture and probably more reliable. In certain cases separate screws just have to be employed.

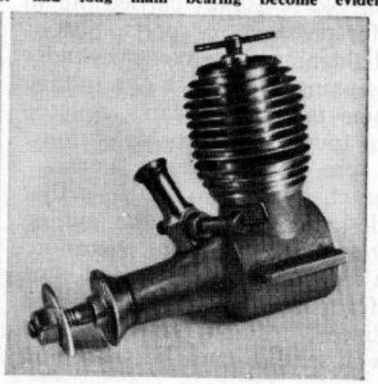
BORING is a normal operation on a lathe but some designs are impossible by direct methods. In the case where crankshaft housing cannot be fully machined directly but must be parted off, turned round and bored from the other side, the difficulty is to get both bores concentric.

Do not design built up crankshafts; these are easily turned from solid bar.

When designing your engine, constantly keep in mind the operations involved in producing each detail. The ensuing chapters, in which the various machining operations will be described, should be of assistance in design. Although you will have your own pet theories on design for high performance some indication of the merits of the various materials required as found by experience is given in the following chapter.



FIRST HOME-BUILT SUGDEN DIESELS had the cylinder retained by the screwed jacket bearing down on to the crankcase.



HERE, THE CYLINDER JACKET IS fitted, and the deep finning, long carburettor, and long main bearing become evident.

CHAPTER TWENTY

Materials, Pattern-making and Casting

Materials

THE efficiency with respect to life, power and weight of an engine depends to a large extent upon the choice of materials. All engines depend upon bearing surfaces of one sort or another and the better the bearings the greater the performance. Good plain bearings consist of one very hard surface bearing against another which is tough and malleable. The softer surface then runs in and work hardens to mate perfectly with the opposing part, usually the shaft. This principle, together with other requirements of the parts determines the choice of metal.

CYLINDERS when run in must have a glass-like surface, so that if they cannot be hard chrome plated or case hardened, a work hardening steel, *i.e.*, one containing chromium or nickel, or molybdenum iron must be used. A high tensile steel S82, S96, or such as that used for car half-shafts is very good. With surface or heat treatment a mild steel is the best choice, *i.e.*, S.1, S.15.

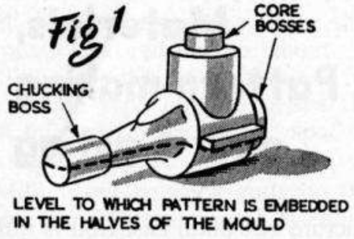
PISTONS are best made from cast iron because its porosity results in it being very difficult to seize and having long-wearing properties, due to the oil and graphite which its surface retains. Meehanite, having a fine grain structure and globular graphite inclusion, is best. Centrifugally cast iron rod is next best since it has a fine uniform crystal structure but plain cast iron is quite good enough.

CRANKSHAFTS must withstand high stresses due to the piston and crash loading and require to be strong and tough. They must be capable of being bent without cracking and must work harden. A high tensile steel is called for, *i.e.*, S.96 or a piece of car half-shaft Case hardening is not recommended because of the uncertainty of the depth of the brittle surface. Hard chrome plating would be advantageous but remember to allow for the thickness of the plate, about .0005 in.

CONNECTING RODS have to be very strong and light and must possess good bearings. Super dural of about 38 tons/in.^{2} is ideal, *i.e.*, DTD 363 or DTD 683. Pure aluminium is useless but ordinary alloy good enough.

CRANKCASES are usually cast from DTD 424, a general purpose casting alloy containing silicon used in foundries. It is rather soft and superior metals are Y alloy or RR 56, *i.e.*, car or aero pistons.

General parts such as the cylinder head, carburettor and driving disc are catered for by ordinary alum. alloy rod but of course the stronger this is the better. The spraybar can be turned from alum. or brass, but since the needle cap is soldered to the needle, brass is used.



Phosphor bronze is a good bearing metal for crankshaft journals and con.-rod big ends, but cast iron is just as good for the former. Ground silver steel is the gudgeon pin material. It is also useful for making special tools when hardened and tempered for work on the softer metals. Magnesium is beautiful to machine but is structurally weak. It requires chromate treatment to render its surface inert to the various corrosive chemicals in fuel. Tufnol is light and strong and is highly resistant to wear, especially where no lubrication can be permitted. It may be used for disc valves.

Pattern Making

This should be relatively straightforward for aeromodellers though there are a few points to note. Any part of the pattern which has to be drawn out of the sand in the moulding operation should possess a slight taper to facilitate this, although on castings of our size this is hardly necessary. It would be appreciated at a foundry where the casting was being made and would indicate which way you wished the pattern to be set in the mould.

Balsa is suitable for patterns but because of its absorbent nature must be given several coats of pigmented dope to harden the surface. Patterns must be capable of withstanding rough treatment as they are liable to be hit during the ramming process of moulding. All lugs and projections should therefore be notched in. Machining is often simplified by the addition of an extra boss which can be gripped in the chuck whilst machining proceeds and which is parted off on completion of the part—see *Fig.* 1.

Coring Out

A pattern made for a cored out casting has bosses attached to the faces into which the cores will enter —see *Fig.* 1. The mould is made in the usual way with the cores arranged to lie on the dividing or parting lines. The cores are made from a special sand mixed with a binding agent such as linseed oil and are baked hard before being placed into the mould in the core prints left by the special bosses. This rather tricky process is best done at a foundry.

One difficulty which comes with using cores is that there is no metal on which to scribe the centre through while the boring and other machining takes place. It may be possible to set the casting up to the outside surface but this will most probably not be true enough for the accuracy of 2 to 3 thou., which is required. Cores are most useful, however, for cutting down machining time and for ensuring a sounder casting, when setting up for machining can be accomplished without difficulty.

Casting

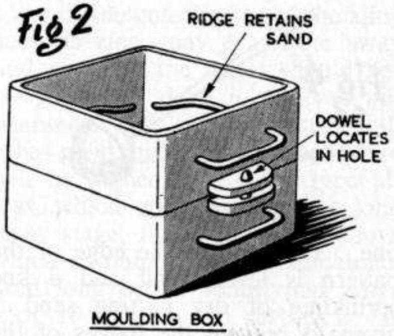
The crankcase and possibly the back cover of a rotary disc induction motor are the only parts usually cast. It is most convenient to take the patterns to the local foundry where the castings will be done cheaply, but making the castings yourself can be interesting.

CAUSES OF DEFECTIVE CASTINGS. By understanding some of the causes of blow holes and other defects of castings, certain points of moulding and melting will be made readily understood. Uneven cooling of the metal, caused by non-uniform volumes distribution of the casting, results in one portion solidifying before the rest. When this happens the large contraction of aluminium alloy causes cracks or draw-holes at the junction of the regions of unequal volume. Putting cores in a casting brings it to more uniform proportions eliminating the trouble, which is most likely to occur at the junction of the crankshaft housing with the main body of the crankcase.

Blowholes are caused by the inclusion of air that cannot escape due to poor ventilation. The positioning of risers and air vents, made with a knitting needle, is of utmost importance in the production of sound castings and only comes with experience. However, with a bit of imagination the requirements of a small crankcase can easily be catered for.

Porosity may be found in castings made at a foundry due to a cleansing pellet, added to the molten metal for purification purposes, not being allowed to complete its action before pouring.

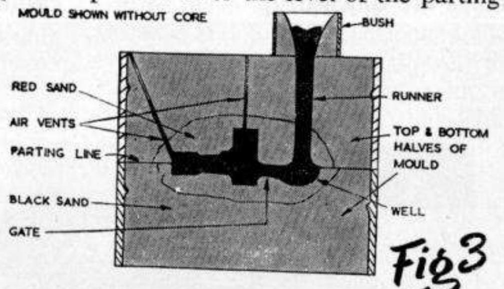
ARRANGEMENT OF THE MOULD. The pattern may be arranged in the mould either with the parting line in the plane of the lugs, as shown in *Figs.* 1 and 3, or along the line of the shaft and up the cylinder, or on an ETA type crankcase across the cylinder. Whichever way is



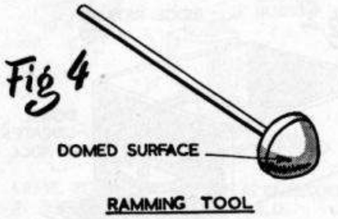
chosen the pattern must be capable of withdrawal from both sides of the mould. Porosity usually occurs in the uppermost regions of aluminium castings which should be arranged to be the part which will be machined away, *i.e.*, where the cylinder fits. The method shown has been found to give the best results.

Moulds may be made from black sand (sand and coal dust moistened with sufficient water to make it bind and feel cool), plaster of paris and steel. Steel dies are only used for mass production of accurate castings and will not be dealt with.

SAND MOULDS. One half of the moulding box—*Fig.* 2— is packed level with sand and the pattern is pressed in to the level of the parting



CROSS SECTION THROUGH MOULD WHEN METAL HAS BEEN POURED

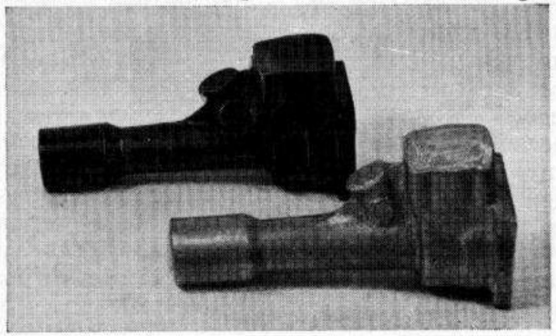


line. The sand at the edge of the pattern is levelled off and a fine sprinkling of dry parting sand is given to prevent the halves of the mould from sticking. This is the odd side which is not used for casting.

The other part of the box is fitted and twisted clockwise to take up any play. Fine red sand containing a little more moisture than the black sand is riddled into the box to cover the pattern and the remaining space is filled with black sand. This is rammed down fairly firmly with a tool of the type shown in *Fig.* 4, a process requiring skill to accomplish correctly. More sand is added and rammed down until the box is levelled off. This part of the mould is lifted off steadily, care being taken to see that it comes away squarely and does not rotate, and inverted. The edges round the pattern are trimmed and any slight faults are touched up.

The pattern is replaced, parting sand applied, the box containing the odd side knocked out, fitted, and the process repeated to produce the other half of the mould. A runner, down which the metal is poured, is made in the top side by withdrawing the sand in a thin metal tube about 1 in. dia. pushed through the sand. In the other side of the mould a well is made to receive the metal from the runner and a passage, known as a gate, is made to connect this to the mould. All corners in this region are rounded to prevent pieces from being washed away and carried into the mould with the rushing metal. The length of the runner is governed to a certain extent by the depth of the box but should be as long as possible to create a good pressure head, to enable the molten metal to flow into all corners, driving out all occluded air.

A bush into which the metal is poured is made by packing sand into a metal ring which is placed on the entrance of the runner. Before the box is finally assembled, not forgetting the clockwise twist, the mould is dusted with graphite to impart a good surface finish to the casting, and all loose particles of sand are blown out. With large castings a heavy weight is rested across the top of the box to prevent the internal gas pressure from separating the mould.



PATTERN FOR THE SUGDEN Special crankcase is carved in balsa, doped and polished. Sand casting obtained from it is in foreground. Similar gravity die-cast crankcases can be bought for 8/- through Aeromodeller Plans Service.

PLASTER OF PARIS MOULDS. A similar procedure to that used with sand is carried out with the exception that an odd side is not required. Runners and venting are similar. The important point is that the mould must be allowed to dry for two or three days and it is best to heat it in the oven to drive out all moisture, otherwise when the metal is poured in, steam is formed which cracks the mould and ruins the casting. These moulds can be used several times and give a good finish.

Melting

The melting point of aluminium alloy is about 550° C. whilst the temperature of red heat is almost 650° C. Red heat should be avoided during melting, because some of the constituents of the alloy such as zinc, may evaporate away and certainly the metal should not be poured if there is any red appearance. The metal is heated until it has good fluidity. The temperature will be higher for forging type alloys, which on cooling have a long pasty stage, than for casting alloys which contain silicon so remaining fluid long before solidifying. After 5 or 10 minutes the casting is extracted. A smooth shiny surface denotes that the metal was too cool, a rough one indicates that it was on the hot side. The latter is preferred since the casting should be sounder.

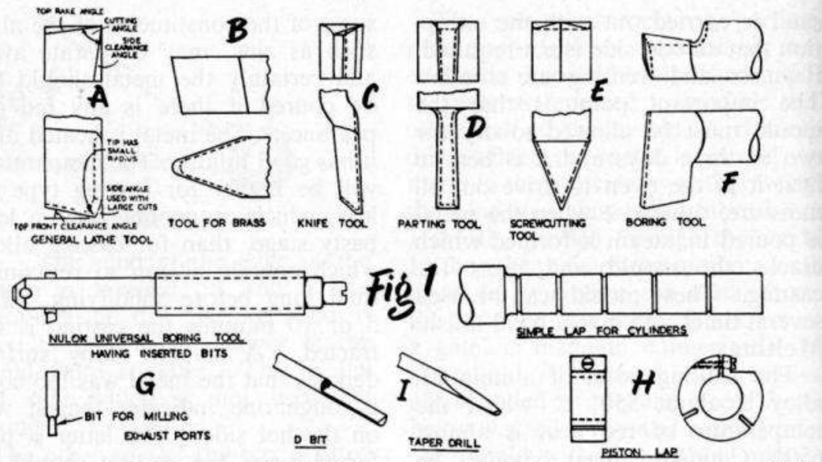
CHAPTER TWENTY-ONE

Tools and Processes in the Lathe

SINCE most of the work is turning, lathe tools will be dealt with first. *Fig.* 1 shows the various types for general work. Each turner has his own pet way of sharpening tools and those shown will merely serve as a guide. A few general rules apply to all lathe tools. Overhang from the tool post must be reduced to a minimum to prevent chatter. Top side and front clearance angles of no less than 3 degrees should be allowed between the finished surface and the tool, so that swarf cannot jam between the tool and the work, thus spoiling the finish. It has been found best to set up a tool at centre height despite what some people may say to the contrary. It is advisable to touch up the tool tip prior to taking a final cut, especially on ferrous metals.

TURNING HIGH TENSILE STEEL (A) The cutting angle should be made fairly large to strengthen the cutting edge and reduce its wear. Because H.T.S. work hardens rapidly the tool must not be allowed to rub and is best operated with a coarse feed at 200 to 400 r.p.m. with as big a depth of cut as allowed by the motor power. Soluble oil and water is a suitable coolant if the work and tool overheat.

CAST IRON. Although C.I. crumbles off when machined, it requires similar treatment to H.T.S. On no account should a cutting fluid



be used as this will cause the tool to rub.

ALUMINIUM. This is easy to machine and any combination of feed, cut and r.p.m. can be used, though for a good finish high r.p.m. is best. Larger rake and clearance angles may be used and, indeed, are essential for some of the softer alloys which tend to build up on the tool tip. Paraffin used as a cutting oil cures this trouble.

BRASS. Being such a soft metal, brass is so easily cut that a tool as shown at (B) with no rake angle plus side angle to prevent digging-in must be used. No lubricant is required. Any combination of feed, r.p.m., and depth of cut is permissible.

PHOSPHOR BRONZE. Although a fairly soft metal, it is very tough and quickly work hardens. It should at all times be treated with respect. A sharp ordinarily-shaped tool will be satisfactory. If difficulty is experienced, cutting fluid may be used to good effect. Any speed with moderate feed and cut is suitable.

Special Tools

KNIFE TOOL. This tool (C) is for cleaning out square corners. It is made either left or right-handed with more rake angle than usual. The point is not robust and will not stand heavy wear. It may be used on any of the various metals above with cutting fluid if necessary, and in general the r.p.m. should be somewhat lower than that used with the ordinary tool.

PARTING TOOL (D). Cutting takes place on the front edge and corners which should therefore be ground true and square to prevent the tool from wandering. A small clearance angle is given to the sides, but adequate metal must be left at the root to take the cutting loads which can be heavy. Even on a good lathe a parting tool tends to chatter and a low speed is often used together with a coarse feel. To stop chatter the feed must be increased. If this does not do the trick, the speed has to be lowered; 200 r.p.m. is easily possible on dural and also on H.T.S if the tool is good. Always use cutting fluid to prevent chips from jamming.

SCREW CUTTING. A screw cutting tool is ground to the profile of the

thread as shown (E). It may be fed in either perpendicularly or at an angle of $27\frac{1}{2}$ degrees and is set up with the aid of a special template. A 5-thou. depth of cut is suitable and r.p.m. is governed by chatter and the skill of the operator; bottom speed is best for a start. Choose a thread pitch which divides evenly into the pitch of the lead screw so that the "nut" can be engaged at any position. A screw-cutting dial eases this problem. Having set up the gears, and with a suitable cut, make a run, disengage the "nut" when the tool has run into the groove which should be provided at the end of the thread. Wind out the tool, return it to the beginning, and reset to a new cut. Should anything go wrong, don't panic. Stop the lathe and wind out the tool instantly. Cutting fluid is often useful, as is also a touch of emery cloth to ease the tops of tight threads.

BORING (F). A boring tool should possess properties similar to an ordinary turning tool. The overhang which tends to make the tool chatter should be kept as small as possible. This reduces the whip which makes boring to an accurate parallel diameter a little difficult. Provided that a good finish is obtained the tool may be mounted above centre height so that it does not foul the hole. In general the r.p.m. will be slightly lower than that used for plain turning.

MILLING. The chief difficulty here comes in putting up the job. It is fairly easy to grip it in a machine vice bolted on to a vertical slide which permits 3D motion, but it is considerably more tedious to clamp it on to the cross slide. A vee block with lots of packing including paper is most useful here. The cutter mounted in the chuck will be run at maximum speed and often completes the operation in a couple of minutes. Cutting fluid is useful in preventing clogging.

An end milling cutter will be found to be the most useful for surfacing, cutting transfer passages, lightening pistons, etc. Its size will probably be governed by the radius of the curves. For milling exhaust ports a fly cutter is most convenient This is similar to a boring tool, with the tip ground like a parting tool, mounted in the chuck. The bar part of a Nulok tool with a bit as shown in (G), is admirable.

GRINDING. Means of avoiding grinding and the construction of a small internal grinder were described in Part I. Should you be lucky enough to have a friend who can do grinding for you the following hints may be helpful. Grinding is often done between centres and if at all possible the part should be made with centres for this reason. It will be necessary to leave about 5 thou. on the diameter for grinding. If centres cannot be made and the grinding has to be done in a chuck, a boss suitable for gripping in the jaws must be arranged, and from 10 to 20 thou. left on the diameter to allow for eccentricity of the chuck and setting up. An extra 5 thou should be allowed for distortion if heat treatment is being carried out prior to grinding on parts which are not robust.

LAPPING. This is the process by which the accurate finish and fit of the piston and cylinder is obtained The principle is that of impregnating the surface of a piece of metal with rubbing compound which is then used to "wear" the part down

AN EXPANDING LAP, SHOWING THE GRUB screws used for adjustment.

to the required dimensions. The rate of cutting is dependent on the amount of compound charged into the lap, the coarseness of the grit, and the fit of the lap to the part. A softer material than that being worked upon is used for the lap, so that it will absorb the compound. C.I., copper, aluminium, and brass are the usual materials. Because the lap is made of a soft metal it tends to wear rather rapidly, and if the rate of cut and accuracy of finish are to be maintained the lap must be expandable. For 1-off jobs where little lapping is needed the extra complication of expanding laps is not justified, but on parts which are at all distorted, probably due to heat treatment, they are essential. (H) and the accompanying photograph (above) show the usual types.

A corkscrew type of motion is applied to the part held by hand with the r.p.m. at about 600. Medium grade valve grinding paste has been found to be the most suitable; it then only takes a few minutes to lap out a cylinder from the reamered finish. The surface obtained is fairly smooth, but is rough enough to enable it to run in easily. A dry lap with a little paste gives the best finish. As always, to remove metal quickly, power must be used, and on one occasion when lapping out a case-hardened cylinder which had distorted 5-thou. out of round, a cast iron lap tightly expanded with a liberal amount of paste, employing paraffin for cooling and lubrication, trued the bore track in half an hour. The part was not held by hand as is usual as the torque and temperature were too great. A hone as marketed by Delpena is far superior if your pocket will stand it.

TAPS AND DIES. Taps are made usually in three forms: taper, second, and plug taps which are used to make the initial through to the final cuts. After each half turn the tap should be rotated backwards far enough to free the chips, which on soft metals tends to clog. Cutting oil should always be used except on brass and C.I. It is a good idea to withdraw the tap completely several times to clear the swarf. Large taps are manipulated with a wrench and frequently have a centre hole in the shank which when located by a centre greatly assists a true perpendicular feed. Small taps are best gripped in a drill chuck.

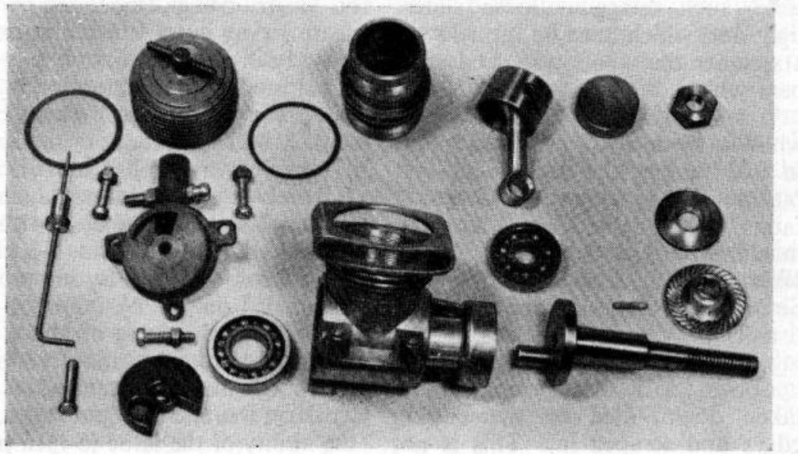
A died thread should always be made after the tapped one since the die is adjustable. The swarf is freed and cutting oil applied as for taps.

DRILLS. Modellers should need little introduction here, though a few hints may be of use. It will be found easiest to drill most holes with at least two drills. The last drill then has a better chance of producing the hole to size since only a small portion of the cutting edges is being employed. It follows the hole already made by the first pilot drill which should therefore be in very good order, or new if possible, to ensure that the drill does not run off centre. Always start the hole

with a centre drill or a good centre pop. Should the drill not be starting on centre before the lands enter the hole, it is possible to pull it back on centre by cutting a groove with a centre pop on the side of the conical depression to which you wish the drill to return. Use cutting fluid to assist swarf removal and cooling and don't use too high r.p.m. with large drills (about 300 r.p.m. for a $\frac{1}{2}$ in. drill in steel) otherwise they may burn out. For small sizes steel wire sharpened to a suitable point is often useful with soft metals.

REAMERS. Reamers are made either to size or expandable. They possess a small taper for the first one-third of their length and so should be able to pass through the hole being finished. Lots of cutting oil and a fastish feed are combined with low r.p.m. to remove only 3 to 4 thou. of metal. A reamer will not produce a good finish if called upon to remove more than this and for sizes where drills increase by $\frac{1}{64}$ in. the hole must be bored out to bring it to a size suitable for the reamer. If the hole cannot be bored and an expanding reamer is not to hand a badly ground drill might be used after a pilot drill to produce a sloppy hole which with luck might to acceptable to the reamer. An accurately ground drill having the corners rounded with an oil stone may be used with a fast feed in soft metal to replace the reamer for finishing the hole. In the smaller sizes, below $\frac{1}{4}$ in., reamers can be replaced by D bits or taper drills (I) made from ground silver steel, hardened and tempered if necessary.

ONE OF THE MOST COMPLEX DIESELS EVER PRODUCED 'en masse' was the Amco 3.5 BB. The alloy crankcase illustrates fine detail involving costly pressure die-casting. Disc valve and crankcase backplate are also castings, otherwise the Amco calls for exactly the same technique of engine manufacture described in these chapters.



CHAPTER TWENTY-TWO

Looking after the Lathe

LIFE is made much more enjoyable if the lathe to be used is in good condition as there is nothing more exasperating than, after having spent half of the evening in turning a part, to have it ruined through no fault of your own as a result of some defect of the machine. Human error is too frequently responsible for the spoiling of a part and it is really worth while to see that the equipment is in the best possible condition. The major difference between working with wood and metal is that if a mistake is made, wood can be stuck back. Metal cannot. Thus every step must be taken with complete certainty that it will be correct. To do this a settled state of mind must prevail which can only be assured if there are as few irritants as possible, such things as loose slides, tight nuts which must be turned with a spanner the whole way, and spanners which don't fit anyway.

It is, therefore, well worth your while to spend two to three evenings in overhauling the equipment and putting it into good order. Common faults with lathes, in fact, all machines, are loose bearings, sloppy slides, end float in spindles, etc. Since this is the case, manufacturers usually provide some means of taking up these slacknesses.

Loose plain bearings must be taken down, filed on the butting edges and scraped in. This is not easy and the aid of a skilled man should be sought. Some lathes like the Myford merely require special 2 thou. packing shims to be removed from beneath the bearing cap to take up the play. Spindles mounted on roller races will have some means of taking up wear which will also remove end play. Although end float has no effect in most ordinary turning work it can be responsible for poor facing or parting off, to say nothing of the havoc it can play with fins or screw-cutting. It should be removed if at all possible.

Slides are easily tightened by means of the screws set into one side which bear on to the adjusting gib plate. It is as well to take slides to pieces to give them a thorough clean out should there be any signs of swarf being embedded underneath. Any burrs which are present must be filed away before the slide is adjusted so that its motion is even along the whole travel whilst being slightly stiff. Slackness in the saddle is taken up by similar means.

Rarely is a lathe found which will turn a constant diameter with the work mounted either in the chuck or between centres. By mounting a piece of 1 in. to $1\frac{1}{2}$ in. mild steel bar with an overhang of about 5 in. in the chuck and without centre, a check on the effectiveness of your work on the bearings and slides, and the ability of the lathe to turn paral-

DESIGNER OF THE HOME-BUILT ENGINE DESCRIBED IN THESE CHAPTERS, Dave Sugden works on the topslide adjuster of his Myford ML7, a favourite lathe with model-makers, amateur and professional.

lel may be made. With a correctly sharpened tool (see Chapter 21) a good $3\frac{1}{2}$ in. lathe should take a $\frac{3}{16}$ in. cut without chattering along the whole length using automatic feed. This is governed by a combination of r.p.m., feed, and shape of tool, and requires much experiment or skill to achieve. It is possible for an unskilled person to turn the last 3 in. without having to drop the r.p.m. below 200. By taking a final cut of a few thou. the amount of taper present can be checked with a good micrometer. One to 2 thou. taper on the 6 in. length can be tolerated for model engines and for anything much above this figure, resulting from further checks, the headstock should be adjusted with the help of your skilled friend. The tailstock is mounted on slides which are perpendicular to the bed to provide adjustment for turning parallel. To check for adjustment for turning parallel the free end of the previous test-piece is centre drilled and the centre inserted. A small cut is taken and the taper measured. To correct this the tailstock is loosened and tapped in a direction across the bed away from the tool if the diameter is larger at the chuck end, and vice versa.

A bent spindle can only be corrected by turning up new back plates for the chucks and facing off the face plate. The former is a rather long job for which you may not have time, but eccentric chucks need not cause trouble provided that in certain cases care in setting up is exercised.

Slackness in belt drives is a common fault which should be rectified to obtain best efficiency. There is nothing so exasperating as to be constantly stopping the machine because the belt drives will not transmit the power.

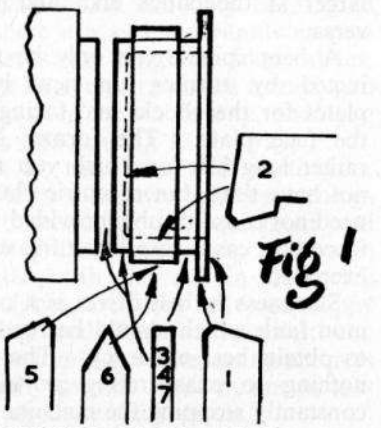
CHAPTER TWENTY-THREE

Work on the Components

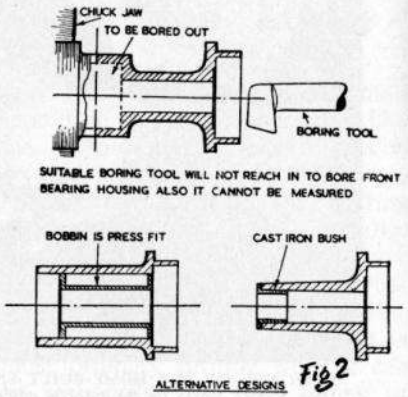
THE components will be dealt with in the order which has been found best in obtaining the most satisfactory mating of parts. A knowledge of the metals and tools to be used will be assumed.

BACK COVER. *Fig.* 1. In order to machine this part without resorting to a jig it must be set up as shown. This means that the threads cannot be tried for fit and so must be made prior to those of the crankcase. The order of machining is not really important, that indicated is as good as any. See that the sealing face is true and has a good finish and that the threads are satisfactory before parting off.

CRANKSHAFT HOUSING. Where a ball race is required, it cannot be machined directly but must be parted off, turned and bored from the other side—*Fig.* 2. The part

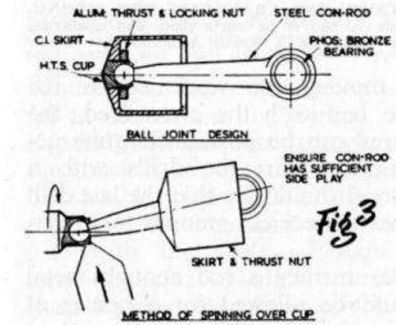


MACHINING BACK COVER



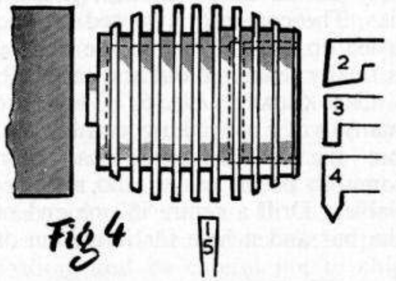
could be shrunk on to a mandrel turned up in the chuck, but if it slipped with subsequent machining the results might be disastrous. It might be possible to use a steady on the outside of the bearing housing so that the boring is concentric with the outside which is true with the rest of the part. A special jig could also be used for the same purpose. These methods are involved and a modification of design as shown surmounts the problem easily. The front ball race could be dispensed with and a plain bearing substituted. Front ball races are hardly worth the extra effort anyway.

BALL JOINT con.-rod-piston assemblies require special spinning equipment for the cup fitting used up inside the piston. Alternative is to use a composite piston as shown in *Fig.* 3, where the piston skirt is secured and soldered to the cup after spinning. The ball end is



turned, filed, and lapped spherical with a piece of copper tubing and grinding paste. A specially ground drill will form the cup contour which is spun round the ball end with a piece of steel, held in the tool post, and well lubricated.

CYLINDER HEAD. *Fig.* 4. This consists usually of simple turning and should present little difficulty. It is easiest to turn the fins before applying the contour. A centre will reduce chatter if this is troublesome. Bore out the inside after completing the fins and do not forget to drill and tap the hole for the compression adjuster before parting off. The order shown is best.

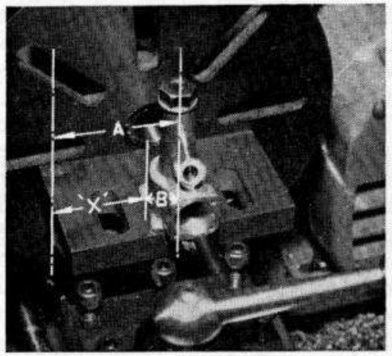


MACHINING CYLINDER HEAD

CRANKCASE, Having gained some experience on easy parts we now turn to something demanding a little more care. By means of the special cast on boss, or at some alternative suitable position, the casting is gripped very firmly, since castings tend to work loose in the chuck. The inside is drilled and bored, the threads are cut and fitted to the back cover and the crankshaft hole and bearing is carefully drilled and reamered. This must have a true finish to permit the crankshaft to run freely, an essential for high power output. A lapped finish on plain bearings where the shaft runs directly in the casting is not recommended due to the difficulty of removing all the grit. Turning out the housing for a ball race on a crankcase of a motor as illustrated has been found to be the most difficult job in the whole engine. It requires a very sharp tool and the utmost skill and patience to obtain the correct fit (see table). It has been found best to use the actual bearing as a plug gauge but be very careful to see that no swarf enters it.

Work on the upper region of the crankcase, where the cylinder fits, can be done next but it is best left until the crankshaft, cylinder, piston and con. rod are finished, so that it can be fitted to the cylinder and ports and be turned down to the height which will give the correct part opening. In this way accumulated errors on the various parts which can completely upset port timing are eliminated.

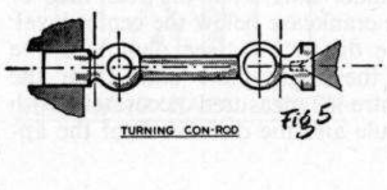
The angle plate is bolted on to the face plate at the distance of the cylinder axis from the rear face of the crankcase below the centre level. The distance A (see photo) of one of the angle plate ends from the centre is measured accurately with a rule and the distance B of the ap-



propriate side of the crankcase from the crankshaft axis is similarly measured. The crankcase is then mounted squarely on to the angle plate at distance X = A-B from the end which puts the cylinder axis true for the subsequent machining which is straightforward.

Do not take heavy cuts since the part may easily be put off centre. It might be convenient to mill out transfer passages or drill the carburettor hole by mounting an electric drill on the cross slide while the job is still set up.

CON ROD. *Fig.* 5. The most difficult part in making this item is in drilling the holes at the correct spacing and ensuring that they are true. Whether the rod is being filed from solid or turned from bar it is best to finish the holes first. With care they can be drilled from centre pops reasonably accurately under a pillar drill, but the most reliable method is to mount the work on a vertical slide, putting the drills in the chuck.



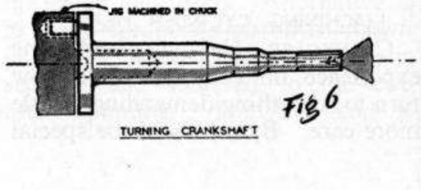
CENTRING THE CRANKCASE FOR BORING, measure A and B on angle plate and crankcase, setting dimension X equals A minus B.

By moving the work across the lathe bed with the cross feed, the centres can be positioned quite accurately. Start the drills with a centre drill and see that the last drill leaves the correct amount for reaming.

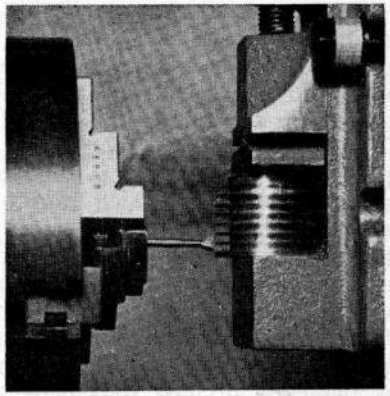
For turning a rod enough metal should be allowed for a centre at one end and a boss at the other. The centres for these must be marked out accurately so that the holes will be on the centre line of the rod and be square to it. The boss is nipped to run true in the chuck whilst the other end is supported with the centre. A form tool will assist in shaping the ball ends but a file will produce the required shape without effort. Part off when as much as possible has been turned and finish by hand.

BIG END BUSHES. These and any other bushes should be of fairly stout proportions, *i.e.*, a wall thickness for one suitable for a 2½ c.c. motor should not be less than 20 thou. When turned and drilled they should be the tightest possible fit. The con. rod is forced on and trued up squarely before the bearing is finally reamered out and sawn off.

CRANKSHAFT. *Fig.* 6. There are many ways of making crankshafts, but the one described has been found to be the easiest and most reliable. Drill a centre in one end of the bar and a hole for induction or

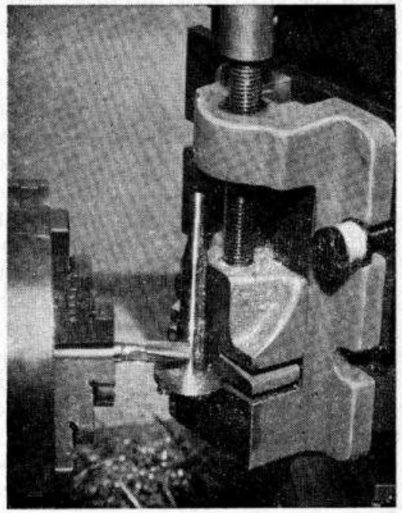


USING A DENTAL BURR TO MILL AWAY fin for holding down nuts on engines such as the early Sugden and Denis Allen 25 (Mk. I version).



lightening in the other, being careful to see that a line joining them is parallel to the outside surface. If the chuck is out of true the bar must be suitably packed to ensure this, otherwise the crankpin will not be true with the shaft. Plough off some of the excess metal round the shaft leaving enough to chuck for turning the crankpin.

By mounting either in a 4-jaw chuck, or in a 3-jaw chuck in which No. 3 jaw has been removed whilst the other two jaws have been wound in, with two or three revolutions of the scroll, before it has been replaced, and using packing if necessary, the bar is set up with the required amount of eccentricity to give the correct stroke. When rotated the throw of the bar is the stroke of crankshaft. This can be measured with a steel rule to 2 or 3 thou. or by winding the tool from a setting with the bar on one side to one when the bar is diametrically opposite, taking readings of the movement from the dial collar. The accuracy of this method depends upon the quality of the screw thread but should be good enough for our purposes. Having mounted the work piece solidly and with as little overhang as possible on it and the tool, turning of the crankpin can proceed at as fast a rate as the lathe and tool will permit. Do not allow the tool to deflect under the impact loading and be careful not to chip it. A sharpened tool will probably be needed to make the finishing cut which should leave 1 thou. on crankpin diameter. A Swiss file will easily bring the pin nearly to size, leaving a good finish which is quickly polished to an accurate fit with fine emery cloth backed up with a rule. Be careful not to round the corners and use long strokes with the file to reduce the chances of the pin being put out of round. The big end should be a smooth tightish fit and is tried for fit, when thoroughly cleaned.



MILLING THE CRANKSHAFT INDUCTION hole to marks scribed through the throat in the crankcase. Note use of a Myford vertical slide.

To turn the shaft a simple jig is made as illustrated. A hole is drilled off centre and a tapered boss is turned which fits into the hole drilled up the centre of the shaft. To prevent frettage corrosion a piece of soft metal shim is wrapped round the crankpin, which transmits the drive, before it is fitted into the hole in the jig, with the boss acting as a centre at the crankpin end, and the tailstock centre steadying the front end. Use tallow to lubricate this centre if possible. The centre is tightened up just sufficiently to take up any end play but be on the lookout for any chatter, indicating looseness or squeaking, a warning that the centre is burning out. Check the centre occasionally and always before making an important cut. The turning is straightforward and bearings are finished in a similar manner to the crankpin.

The induction hole is milled, sawn and/or drilled and filed to marks scribed on at the internal corners of the carburettor intake, when the shaft is set up in its housing in the correct opening and closing positions.

CYLINDER. This is simple turning work and should present no difficulties. Complete the turning before finishing the bore. A good reamer provides the easiest means of obtaining a suitable finish but failing this—boring will produce the desired result with care. See that the tool is well sharpened and makes the last cut without spring, *i.e.*, continue to make passes along the bore at the same tool setting until it ceases to cut. The bore should then be parallel and true.

The ports come next. It may be found convenient to use the piece nipped in the chuck to grip in the vice whilst filing or milling out the parts, so mark the position of No. 1 jaw before taking the work out of the chuck so that it can be re-chucked accurately for parting off. Milling out of radial ports is easily done with a fly cutter, see page 149. on milling, set to the correct radius. The cylinder is held in a machine vice bolted on to a vertical slide or angle plate mounted on the cross slide. A small drilling jig will be necessary to prevent the drill from running if ports are to be drilled. Much patience is required if they are to be sawn and filed out.

Having finished the ports and parted off the cylinder it may now be heat treated but make it as hefty as possible to minimise distortion when quenching. The last job of all is lapping. To reduce friction it is a good plan to lap cylinders bell-mouthed so that the piston is a loose fit at the bottom of the stroke, where a good compression seal is not necessary, and is a tight fit up the bore where good compression is essential. When mating the piston it is then easy to judge the fit by the distance it will pass up the bore.

GUDGEON PIN. Ground silver steel rod is used here. It can be drilled out for lightness (hole $\frac{1}{2}$ to $\frac{1}{3}$ of the outside diameter depending on the general proportions), and dural or brass end pads inserted. However, use a good drill otherwise it may run off centre.

PISTON. The operations here are fairly straightforward except for drilling the gudgeon pin hole. The inside is drilled and bored and as the outside is turned to within 10 thou. when the job is marked and taken out of the chuck. It is then mounted on the vertical slide or clamped by some means on to the cross slide so that drills held in the chuck pass through perpendicularly across the centre line. Alternatively the piston may be mounted on a vee block on a drilling machine or held, on a vee block, on to an angle plate bolted to a lathe face plate. The operation cannot be set up very accurately on a drilling machine which should only be used as a last resort. A vertical slide makes jobs of this sort child's play. The gudgeon pin hole is finally reamered out such that the pin will not quite pass through and requires a light tap to make it fit.

Milling out the inside for lightening demands a similar setting up technique. The slight increase in performance that this gives is not worth the effort involved if you have no vertical slide attachment.

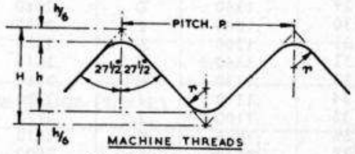
The piston is now returned to the chuck where it is turned down to within $1\frac{1}{2}$ thou. of size, half parted off and lapped to fit the cylinder. This being successfully accomplished it is completely parted off. When fitting pistons to cylinders both parts must be cleaned well to remove grit, enabling an accurate estimate of the interference to be made. Tallow is very useful in preventing the parts from sticking when close fits are being obtained. The correct fit is such that, with tallow the parts will pass fairly freely. They will then feel a little stiff when lubricated with fuel.

CONTRA PISTON. This is easily catered for by allowing some excess length when turning the piston, that is only partially lapped when fitting the latter component. The contra piston is merely parted off at the

position which gives the best fit.

JET ASSEMBLY. The screw cap for the needle is made first. The needle is best made from 18 s.w.g. wire finely tapered by careful filing and finished with emery cloth. It is very easy to distort the spraybar when threading and drilling. The hole is finished with a piece of 18 s.w.g. wire sharpened to a point like a screwdriver. Solder the needle very securely to the screw cap. To make the needle a firm fit in the spraybar it can either be bent slightly so that it binds in the spraybar hole, or it can be soldered out of line so that the threads rub a little. There is then no need to go to the trouble of making a saw cut in the screw cap for tightening purposes. 2 s.w.g. wire is suitable for drilling the jet hole on $2\frac{1}{2}$ c.c. engines.

DRIVING DISC. This is simple turning. The taper which fits on to the crankshaft should be done if possible with the same setting on the top slide as was used for turning the crankshaft taper. Serrations are easily applied by winding a boring tool across the driving face.



MACHINE THREADS

TPI	Pitch in.	Standard thread depth. in.	Outside dia. min	Outside dia. max.
40	.025	.0160	$\frac{1}{8}$	$1\frac{1}{4}$
36	.02778	.0178	$\frac{1}{8}$	$1\frac{1}{4}$
32	.03125	.0200	$\frac{1}{8}$	$1\frac{1}{4}$
28	.03571	.0229	$\frac{1}{8}$	$1\frac{1}{4}$
26	.03846	.0246	$\frac{1}{8}$	3 in.
24	.04167	.0267	$\frac{1}{8}$	3 in.
20	.050	.0320	$\frac{1}{4}$	3 in.

DATA TABLES

of use to those making their own engines, whether from the Sugden Special design, or their own designs.

DRILL SIZES

Number and Letter Drill dia.			
No.	**Size in.**	**No.**	**Size in.**
1	.2280	52	.0635
2	.2210	53	.0595
3	.2130	54	.0550
4	.2090	55	.0520
5	.2055	56	.0465
6	.2040	57	.0430
7	.2010	58	.0420
8	.1990	59	.0410
9	.1960	60	.0400
10	.1935	61	.0390
11	.1910	62	.0380
12	.1890	63	.0370
13	.1850	64	.0360
14	.1820	65	.0350
15	.1800	66	.0330
16	.1770	67	.0320
17	.1730	68	.0310
18	.1695	69	.0292
19	.1660	70	.0280
20	.1610	71	.0260
21	.1590	72	.0250
22	.1570	73	.0240
23	.1540	74	.0225
24	.1520	75	.0210
		Letter Drills	
25	.1495		
26	.1470	A	.2340
27	.1440	B	.2380
28	.1405	C	.2420
29	.1360	D	.2460
30	.1285	E	.2500
31	.1200	F	.2570
32	.1160	G	.2610
33	.1130	H	.2660
34	.1110	I	.2720
35	.1100	J	.2770
36	.1065	K	.2810
37	.1040	L	.2900
38	.1015	M	.2950
39	.0995	N	.3020
40	.0980	O	.3160
41	.0960	P	.3230
42	.0935	Q	.3320
43	.0890	R	.3390
44	.0860	S	.3480
45	.0820	T	.3580
46	.0810	U	.3680
47	.0785	V	.3770
48	.0760	W	.3860
49	.0730	X	.3970
50	.0700	Y	.4040
51	.0670	Z	.4130

HEAT TREATMENT

Tempering Temperatures		Heat Colours	
Colour	Temp. C.	Colour	Temp. C.
Pale Yellow	222	Dull Red	650- 750
Straw Yellow	238	Cherry Red	780- 800
Brown	254	Bright Red	830- 880
Light Purple	277	Dull Yellow	1050-1150
Dark Blue	306	White	1250-1300

SCREW THREADS

BRITISH ASSOCIATION STANDARD

No.	Diameter	Approx. T.P.I.	Root Dia.	Tapping drill
0	.236	25.4	.189	12
1	.209	28.2	.166	19
2	.185	31.4	.147	25
3	.161	34.8	.127	30
4	.142	38.5	.111	34
5	.126	43.0	.098	40
6	.110	47.9	.085	44
7	.098	53.0	.076	48
8	.087	59.1	.064	51
9	.075	65.1	.056	53

WHITWORTH STANDARD

Size	T.P.I.	Root dia. ins.	Thread depth ins.	Tapping drill
1/16	60	.0412	.0107	58
3/32	48	.0670	.0133	50
1/8	40	.0930	.0160	41
5/32	32	.1162	.0200	31
3/16	24	.1341	.0267	9/64"
7/32	24	.1653	.0267	18
1/4	20	.1860	.0320	11
5/16	18	.2414	.0355	D
3/8	16	.2950	.0400	N
7/16	14	.3460	.0457	S
1/2"	12	.3933	.0534	13/32"
BRITISH STANDARD FINE THREAD				
7/32	28	.1730	.0229	16
1/4	26	.2007	.0246	13/64"
9/32	26	.2320	.0246	15/64"
5/16	22	.2543	.0291	G
3/8	20	.3110	.0320	O
7/16	18	.3664	.0356	3/8"
1/2"	16	.4200	.0400	27/64"

BALL BEARINGS

(Dimensions in inches unless stated otherwise)

Type bore.	Width	Outside dia.	Weight oz.	Limits of shaft	Limits of housing	Abutment dia. shaft	Abutment dia. housing	Manufacturers' Code No.: F.B.C.	Hoffman	R & M	S.K.F.
4 m/m	5 m/m	13 m/m	.166	.1569 .1564	.6305 .6297	.281	.5	R4	104	LJ4	R4
6 m/m	6 m/m	19 m/m	.288	.2356 .2351	.7486 .7478	.375	.625	R6	106	LJ6	R5/6
8 m/m	7 m/m	22 m/m	.432	.3144 .3139	.8667 .8659	.437	.75	R8	108	LJ8	R7/8
¼	7/32	¾	.256	.2495 .2490	.7502 .7495	.40	.62	EE2	S.1	KLNJ¼	EE2
⅜	7/32	⅞	.40	.3745 .3740	.8752 .8745	.53	.75	EE3	S.3	KLNJ⅜	EE3
½	¼	1⅛	.482	.4995 .4990	1.1252 1.1245	.68	1.0	EE4	S.5	KLNJ½	EE4

METALS FOR YOUR ENGINE

Material	Specification	Use	Ultimate Tensile strength tons/sq. in.	Colour Code Identification
Mild steel ...	S.1.	Cylinders when case hardened. General lightly loaded parts.	35	yellow.
Case hardening steel ...	S.15 ... S.82 ...	... Cylinders ... ... Crankshafts	35 75	yellow, brown, yellow. green, red, yellow.
High tensile steel	S.96 ...	... Cylinders and crank-shafts ...	55	black, red, blue.
Aluminium alloy .	DTD 363 ... DTD 364 ...	Con Rods ... ... Con rods or general	38 30	brown, green, brown. green, brown, green.
Aluminium forging alloy ...	DTD 683 RR 77 DTD 130 RR 56	Con rods or general General	31 26	blue, yellow, red. red, black, yellow.
Aluminium casting alloy	DTD 424	Crankcase	10	
Phosphor Bronze .	B.8 ...	... Bearings	11	brown.

IMPERIAL STANDARD WIRE GAUGE (S.W.G.)

No.	Ins.	No.	Ins.	No.	Ins.	No.	Ins.
0	.3240	13	.0920	26	.0181	39	.0052
1	.3000	14	.0800	27	.0164	40	.0048
2	.2760	15	.0720	28	.0148	41	.0044
3	.2520	16	.0640	29	.0136	42	.0040
4	.2320	17	.0560	30	.0124	43	.0036
5	.2120	18	.0480	31	.0116	44	.0032
6	.1920	19	.0400	32	.0108	45	.0028
7	.1760	20	.0360	33	.0100	46	.0024
8	.1600	21	.0320	34	.0092	47	.0020
9	.1440	22	.0280	35	.0084	48	.0016
10	.1280	23	.0240	36	.0076	49	.0012
11	.1160	24	.0220	37	.0068	50	.0010
12	.1040	25	.0200	38	.0060		

CHAPTER TWENTY-FOUR

Assembly and Test

HAVING at last reached the stage where all constructional obstacles have been overcome, the assembly and test of a new engine must not be rushed, but be carried out with as much care as was put into the construction. Cleanliness is of supreme importance in ensuring that the motor shall have a long working life.

CLEANING. Wash the parts thoroughly in petrol, paying special attention to bearings. Assemble all the parts and check that everything fits together properly. Usually something does not. The con. rod may foul the sides of the crankcase and the piston may touch the top of the back cover Whatever the trouble, attend to it now. This does, of course, assume that you have designed an engine which *can* be assembled !

Dismantle and select all the ferrous parts (shaft, piston, cylinder, etc.), excepting ball races, which should still be perfectly clean ; put them into a tin containing washing soda, and if this is not available, soap powder and water, and boil for several minutes to remove ingrained grit. The parts are super-clean after boiling, and if not oiled will rust rapidly. Lay them out on clean paper and handle as little as possible.

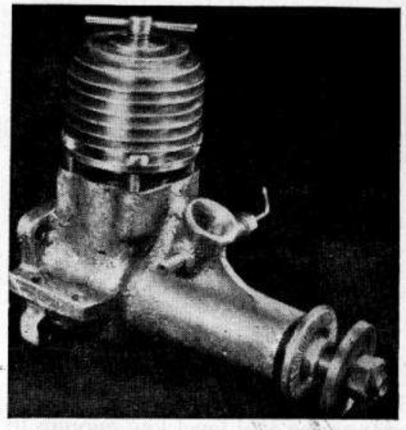
If a ball race has become dirty, it can be cleaned out by carefully spinning it over in a bath of clean petrol. Aluminium parts are cleaned with petrol and a smooth cloth, bearing surfaces being rubbed as vigorously as possible to remove grit. Give them a final dip in petrol to remove fluff.

CRANKSHAFT ASSEMBLY. Assembling an engine having a plain bearing crankshaft is not difficult. Inserting a plain journal bearing is effected by warming the crankcase and pressing home the bush in a vice, using suitable blocks of wood, being careful to avoid any transverse loading which might distort the bearing.

Fitting a ball bearing crankshaft can be decidedly tricky. Using a piece of tube to bear upon the inner race, press the rear ball race on to the crankshaft, using a vice. Warm the front bearing housing over a clean gas flame and insert the race. Tight force with a vice might be needed. There should be a face against which the outer race may seat squarely. It is easy to distort the outer face and excessive force must not be used. If the bearing will not enter or has distorted out of round or does not run freely, it must be removed and the offending part of the housing, usually detected by the score marks, scraped down. A balsa knife makes a good tool for this job. Be careful to remove metal evenly all the way round if the bearing is too tight. To remove the bearing, re-warm and tap out with a drift or with a piece of ground silver steel made to a very tight fit in the inner race.

Having fitted the front bearing

THE PROTOTYPE SUGDEN SPECIAL AS detailed in Chapter 25 was running within moments of its first assembly, due to careful work as prescribed here.

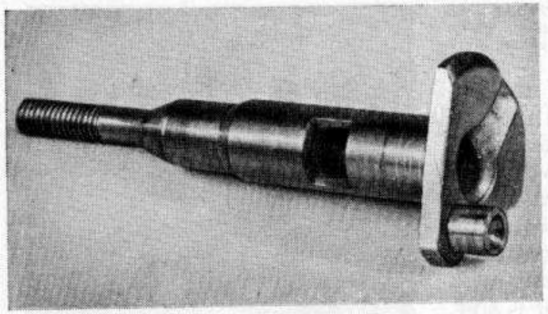


into the crankcase part of the engine and with the rear ball race on the crankshaft the next step is to mate the two assemblies. Warm the rear housing, or the crankcase, and with the front bearing supported on both its outer and inner races press the crankshaft into position with the vice. If it will not fit or is stiff, remove and attend to the trouble. When fitted, a slight tap in the reverse direction relieves stresses set up between the bearings, and the shaft should then spin freely. Slight "lumpiness" may be tolerated on an ordinary engine and it may be found to disappear when the engine warms up.

GENERAL ASSEMBLY. The remainder of the assembly is not difficult. As parts are put together they should be marked so that on re-assembly they can be fitted in the same position. *E.g.*, mark the front of the con. rod and insert the gudgeon pin from the front of the piston. Mark the cylinder. Always tighten opposite screws progressively to ensure that the part—a cylinder head or back cover—is evenly seated.

Gaskets should be used on all joints for the purpose of making absolutely certain that there are no leaks. Many a good engine fails to perform properly due to a leak. A leaky crankcase can make starting tricky which on a new engine can be quite difficult enough with unknown settings. Tough paper is ideal for gaskets.

With everything assembled the engine should turn over rather stiffly. It is a good thing, having mounted the piston and cylinder, to turn over the motor whilst it is immersed in petrol to flush any remaining dirt from the bore. If on tightening up, the piston has become stiff in the bore, either the cylinder is out of line or the con. rod holes are not true. Try the effect of turning the con. rod back to front. If the cylinder is giving the trouble, turn the motor over several times, remove the cylinder and look for rubbing marks which will indicate which way it is out of line, and where the seating should be adjusted. Tightness due to malalignment will loosen up with running as the con. rod bearings wear, but if it is the cylinder that is out of line the motor will never deliver peak power and the con. rod will rapidly wear and may even bend or break. Every effort should be made to assemble the motor free from binding of any sort, as this is the best means of ensuring a long life and high power output. If any parts have to be worked upon they should be re-cleaned in the manner prior to the commencement of assembly. Lubricate with castor oil.



SCOOPED CRANKWEB AND square cut induction port are aids to high speed running. Shaft is from earlier motor seen opposite.

TESTING AND RUNNING. The motor is at last ready to be run. Bolt it to a suitable mounting and arrange the fuel level low enough to prevent the motor from flooding, but high enough to keep the fuel at the jet. The weight of a plastic prop. eases starting and one should be selected which will not allow the r.p.m. to exceed 8,000 on a plain bearing engine or 11,000 on one fitted with ball races.

A suitable fuel is equal parts of castor oil, Derv or paraffin, with either a 2 per cent addition of amyl nitrite or nitrate. As the motor becomes more free this may be modified to a final mixture as follows: 15 per cent castor oil, 55 per cent Derv, 27 per cent ether and 3 per cent amyl nitrite.

Open the needle so that when choked the fuel is drawn through the tubing at a normal rate for the size of engine. Choke a couple of times and screw down the compression until it feels reasonable. Flick several times and if without success try and prime through the ports. Whilst flicking, turn down the compression until the engine "pops" Further priming should result in a burst and if the engine fails to roar into life, open up the fuel setting. If the motor still does not start and shows no signs of excess fuel, drill out the jet to a slightly larger size. A reasonable design, made with moderate skill, must run, and perseverance will end in success. My first motor took $1\frac{1}{2}$ hours to start; the latest one went on the third flick.

Glowplugs

Glowplugs present no appreciable starting difficulties. Equip with a long reach or warm plug, wind the needle well open, prime through the exhaust port with fuel, say Mercury 5, and with a good glow the motor will run. A reduction of fuel brings it to the best running setting. It is easy as that! The compression ratio is difficult to assess on glow motors because of their deflector head pistons and shaped cylinder heads, but it is easily judged by performance on various fuels and plugs. If on changing to a short reach plug the engine runs as though it is starved of fuel when the correct running setting is approached, compression ratio is too low. On the other hand, if it runs well on all plugs but sounds rough on certain fuels the compression ratio is too high especially for the fuel involved. The best ratio is determined only by checking performances carefully, on the best fuel with the best plug.

When the motor has lost all stiffness in the piston bearing it is virtually run in and small props. can be used. If after a run the piston

TURNING 11,000 R.P.M. ON AN 8 x 6 AND being checked with a vibrating spring steel rev.-reader, this is a plain bearing Sugden engine with screw-over jacket assembly.

feels on the stiff side or looks dry the oil content of the fuel should be raised or a thicker oil used. To prime a new engine with oil before a run is a good thing, making for a long life.

The state of the oil thrown out of the exhaust ports gives a good indication of the conditions within the engine. More nitrite is needed if it contains carbon. If after two or three runs however it has a "polychromatic" look, rapid wear on some part is indicated and the motor should be stripped, examined, and the trouble rectified, before the part —usually the con. rod—is too badly worn. Engines fitted with ringed aluminium pistons are more prone to produce this phenomenon, but it should not be allowed to persist.

A spot of jeweller's rouge or Brasso in the fuel assists a stiff engine to run in more rapidly, where without it the process might take many hours; but it is only recommended where peak power is wanted quickly. It can knock hours off the life of the engine.

The most important instrument for engine testing is a good tachometer. Wire reed indicators are not sufficiently sensitive or reliable. The most convenient method is to run the motor whilst someone compares its note on a piano. If the instrument is in tune you have a fairly accurate check on r.p.m., and anyway it provides a sensitive means of checking small variations in r.p.m. The table gives the r.p.m. indicated by the various notes. R.p.m. are halved if the note is an octave lower and doubled if it is an octave higher!

NOTES OF ENGINE R.P.M.

Note	...	c	d	e	f
R.P.M.	...	7,680	8,640	9,600	10,200
Note	...	g	a	b	Middle c
R.P.M.	...	11,500	12,800	14,400	15,350

CHAPTER TWENTY-FIVE

Making the Sugden Special

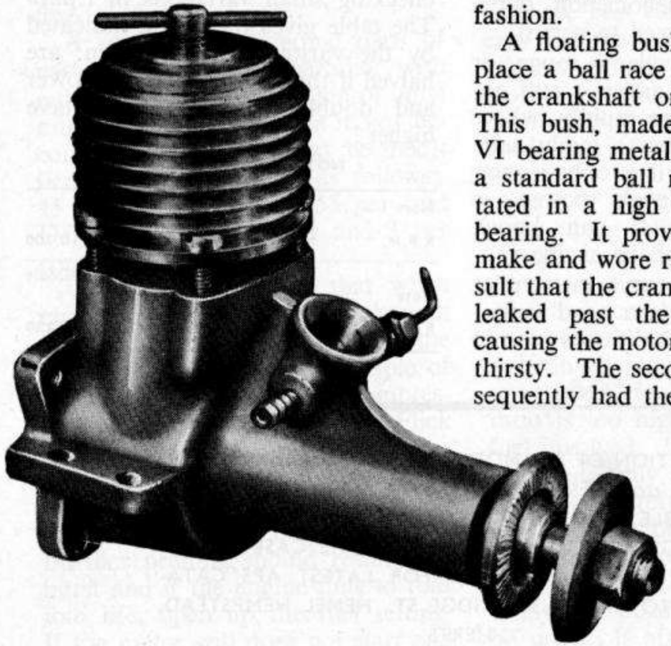
IN designing this engine the requirements were: high power output, low weight and easy construction. The first implies Oliver type porting and low friction crankshaft bearings whilst the others rule out ball races. The effectiveness of the compromises made is indicated by the test results, *i.e.*, the internal shape of the engine is satisfactory, but the output could be raised with ball races for racing purposes.

Design and Development

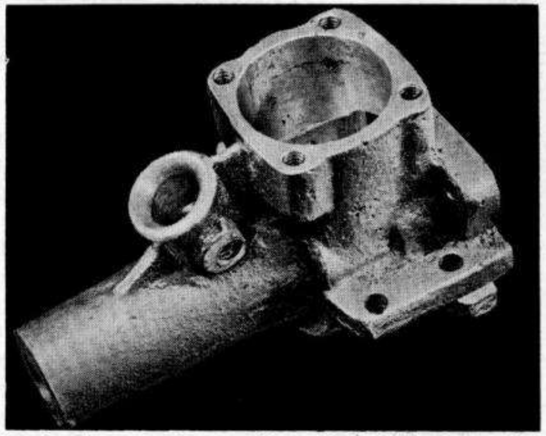
The stroke/bore ratio was chosen small enough to produce a light compact design with docile starting characteristics, but large enough to prevent the internal stresses from being excessive. A value of 1.06 was obtained when a stroke of .6 in. was chosen.

With Oliver transfer ports it is not very practicable to screw in the cylinder, a feature making for lightness, and on the first prototype motor a system of studs was devised which has been superseded by the method shown here. The cylinder is prevented from rotating by the holding-down screws which locate in the grooves between the exhaust ports. The cylinder head then screws on in the normal fashion.

A floating bush was used to replace a ball race at the web end of the crankshaft on the first engine. This bush, made from Immadium VI bearing metal, was the width of a standard ball race and itself rotated in a high tensile steel outer bearing. It proved to be tricky to make and wore rapidly with the result that the crankcase compression leaked past the crankshaft, thus causing the motor to become rather thirsty. The second prototype consequently had the short plain bearing, shown on the drawing, which has proved to be quite satisfactory. A separate inserted front bush used on the first engine was made integral with the crankcase on the later designs since this bearing is lightly loaded.



THE SUGDEN SPECIAL, a remarkable design, already very successful with many home-constructors and capable of high performance. Gravity die cast crankcases ready for machining and full-size drawings complete with instructions are available price 12/6d. the set from Aeromodeller Plans Service.



A GRAVITY DIE-CAST CRANKCASE after all machining operations are completed.

The exhaust ports were reduced from the 80 thou. depth of the first to 60 thou. depth on the second engine and the slightly lower power output of the latter unit could be attributed to this modification. Eighty thou. ports were used on the third and final engine.

The second and third motors had the carburettor intake drilled $\frac{1}{4}$ in. diameter for the simple reason that a suitable smaller size drill was not available. This did not appear to affect starting which is easy on all three engines. Alternative beam mount lugs were provided on the third motor.

Construction

Tolerances are not indicated on the drawing since the design is not intended for mass production. As a matter of principle all dimensions should be produced as accurately as possible and if the order of working as followed below is used, all parts will fit accurately and errors will be eliminated.

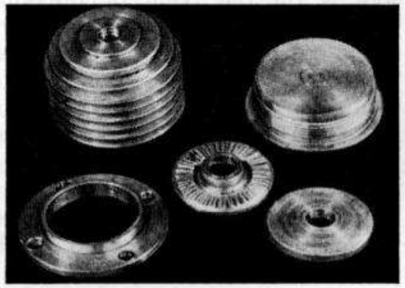
The *pattern* required much care to make and as many parts as possible were turned on a Wolf drill lathe. An additional $\frac{1}{16}$ in. was allowed on the faces to be machined. The local foundry did the casting. Aeromodeller Plans Service can supply quality die castings ready for machining, price 8/- each.

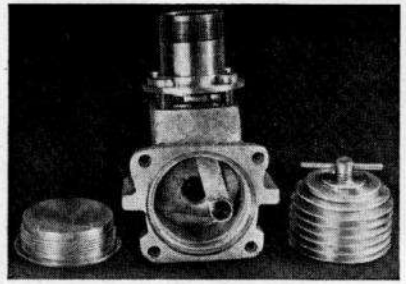
The *back cover* was turned from dural as described earlier.

The *con. rod* was milled from a piece of DTD 610 $\frac{1}{4}$ in. dural plate which despite its rather low strength has proved adequately strong and wear resistant. The holes were drilled and reamered, and the milling on one side completed at one setting up on the vertical slide to ensure good alignment of the rod, which is important. The remainder was easily filed.

The *cylinder head* and *holding down ring* were machined, the finning being completed before the boring and screw cutting were commenced.

SIMPLE TURNED PARTS FOR THE "Special" from Dural.





DISMANTLED, FROM THE REAR, SHOWS cylinder retaining method of flanges for radial mounting, etc.

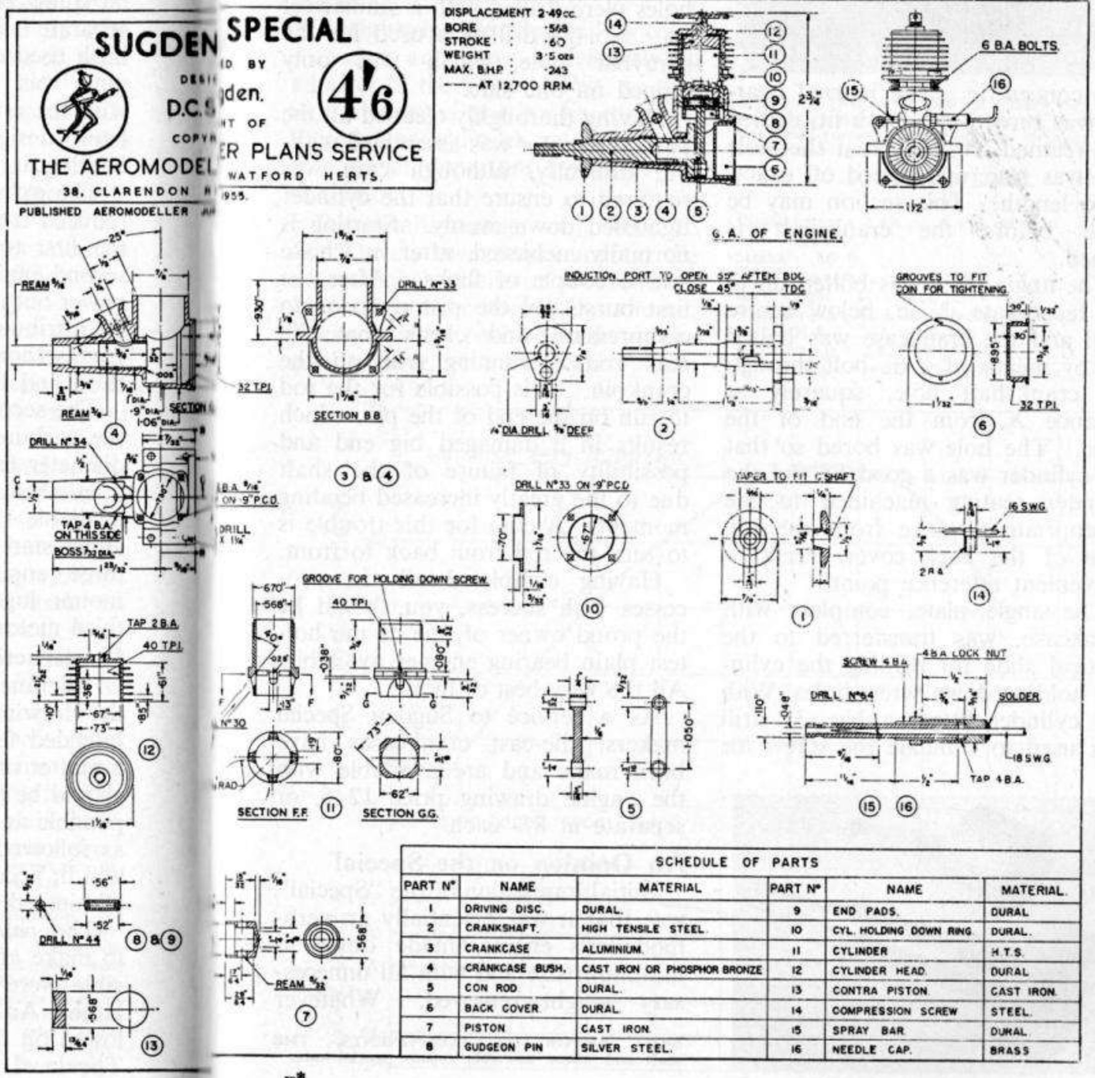
From a piece of S11 stock, sufficiently long to allow 1 in. for chucking, the *cylinder* was machined and drilled $\frac{1}{2}$ in. dia.

Marks are made at 90 degrees intervals round the exhaust ring. The work was marked at No. 1 jaw and transferred to the vice on the vertical slide for milling the transfer grooves. These could be filed out if necessary. A $\frac{1}{4}$ in. end mill was used although an old drill carefully ground for the purpose would do, and the grooves were located by the 90 degree marks as also was the drilling for the transfer ports. It was not found necessary to use a jig here and with the cylinder set across the lathe bed at 40 degrees, a centre drill and drills No. 40 and 30 completed the hole. This was squared off between the exhaust ports after they had been milled as described in Part V. The milling has been taken to the correct depth when the tool is on the point of breaking through the cylinder bore ($\frac{1}{2}$ in. dia.). The work was then returned to the chuck, trued up and bored to $1\frac{1}{2}$ thou. undersize. A half hour of filing remained to complete the porting, after which the liner was parted off. The lap was machined to a tight fit and the cylinder was tapped on for screw cutting the 40 T.P.I. thread. A few minutes lapping with coarse and then fine grinding paste completed the cylinder. It is not recommended that this cylinder design be subjected to the stresses of case hardening.

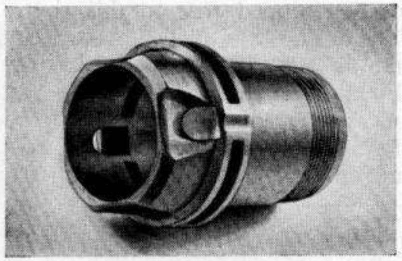
The *piston* and *contra piston* were made as previously described.

Next the *crankshaft* was machined from a piece of S.11 55 tons H.T. steel. The machined bar was mounted with the correct amount of eccentricity in a 3-jaw chuck which had had No. 3 jaw offset by two or three revolutions of the scroll, using shim packing for the final adjustment. My chuck could not "swallow" the full length of the bar and after checking for alignment with the lathe bed, a $\frac{1}{8}$ in. centre drill hole was made. The overhanging end was stabilised with the tailstock centre during subsequent machining of the crankpin. The remainder of the crankshaft machining was carried out as described earlier. The induction port was milled out to port opening and closing lines scribed through the carburettor intake hole as shown in the photograph.

The rear face of the *crankcase* casting was centre popped and the chucking boss was filed until the casting nipped true to this centre. The inside was drilled, bored and the threads cut to fit the back cover. Finally the hole was drilled and reamed to take the crankshaft before parting off. Care was needed to see that these holes were all true



PART No	NAME	MATERIAL	PART No	NAME	MATERIAL
1	DRIVING DISC.	DURAL.	9	END PADS.	DURAL
2	CRANKSHAFT.	HIGH TENSILE STEEL.	10	CYL. HOLDING DOWN RING.	DURAL.
3	CRANKCASE	ALUMINIUM.	11	CYLINDER	H.T.S.
4	CRANKCASE BUSH.	CAST IRON OR PHOSPHOR BRONZE	12	CYLINDER HEAD.	DURAL
5	CON ROD	DURAL.	13	CONTRA PISTON	CAST IRON.
6	BACK COVER	DURAL.	14	COMPRESSION SCREW	STEEL.
7	PISTON	CAST IRON.	15	SPRAY BAR	DURAL.
8	GUDGEON PIN	SILVER STEEL.	16	NEEDLE CAP.	BRASS



FULLY MACHINED CYLINDER ILLUSTRATING the angled transfer ports, and in particular their overlap of the exhaust timing.

and concentric. The journal bearing was turned to a push fit, drilled and reamed, ensuring that this hole also was true, and parted off exactly to length. This section may be done before the crankshaft is turned.

The angle plate was bolted on to the face plate $\frac{9}{16}$ in. below centre level and the crankcase was bolted on, by means of a tie bolt through the crankshaft hole, squarely at distance X from the end of the plate. The hole was bored so that the cylinder was a good fit and the cylinder seating machined to the appropriate distance from the top edge of the back cover recess, a convenient reference point.

The angle plate, complete with crankcase, was transferred to the vertical slide for drilling the cylinder holding down screw holes. With the cylinder fitted, a No. 35 drill was used to simulate the screw for obtaining the correct centres. Holes were drilled in the holding down ring at the same setting by slipping it on to the cylinder in the appropriate position. The carburettoı holes were drilled with a similar set up. No. 34 drill was used for the spraybar hole which was only tapped on one side.

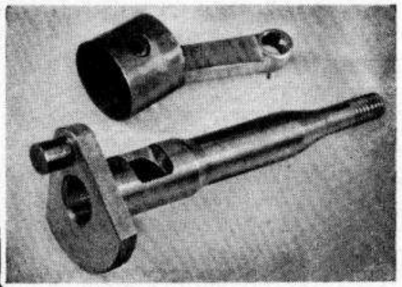
Having thoroughly cleaned all the parts the motor was assembled without difficulty, although care was required to ensure that the cylinder tightened down evenly. Starting is normally achieved after a choke and a couple of flicks. After the first burst, put the piston up on to compression and check that the con. rod is running true on the crankpin. It is possible for the rod to run on the end of the pin, which results in a damaged big end and possibility of failure of the shaft due to the greatly increased bending moment. A cure for this trouble is to turn the rod from back to front.

Having completed all the processes with success, you should be the proud owner of one of the hottest plain bearing engines available. All the very best of luck.

As a service to Sugden Special makers, die-cast crankcases have been made and are available with the engine drawing price 12/6, or separate at 8/- each.

An Opinion on the Special

Initial impression of the "Special" was that it was essentially an aeromodeller's engine, made down to minimum size and with all unnecessary weight removed. Whatever

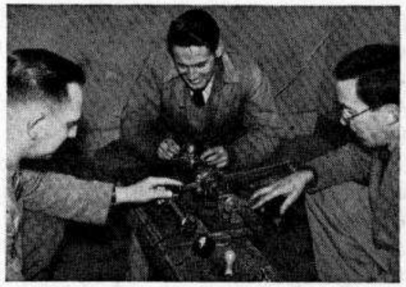


MOST IMPORTANT COMPONENTS, THE shaft and piston assembly require special care.

the claims that a little extra weight may not be important—and perhaps such claims are quite justified—when one thinks of "optimum" aircraft design one automatically links with it minimum weight. In this respect the "Special" rates top marks.

Design-wise the "Special" is quite orthodox and Dave is evidently an admirer of the Oliver stable, as seen in the form of porting he employs, though the cylinder retaining ring is a novel departure from the orthodox.

Being tailor-made, as it were, to suit Dave's own requirements, the test engine had one or two features which the writer would have altered. There was no positive lock on the needle valve, also the contra-piston

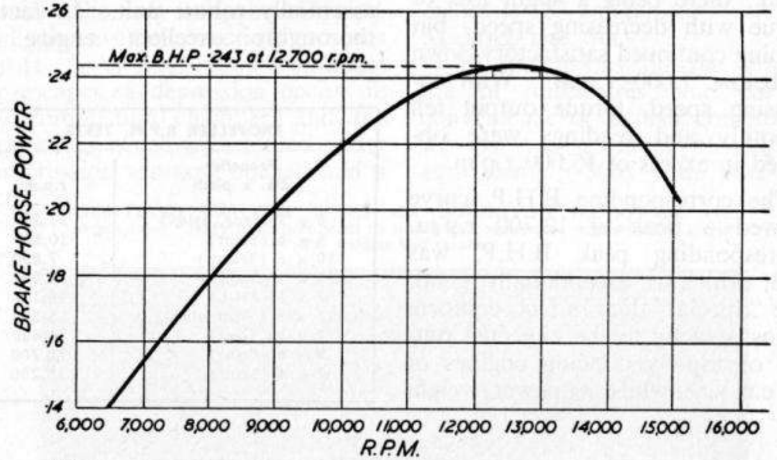


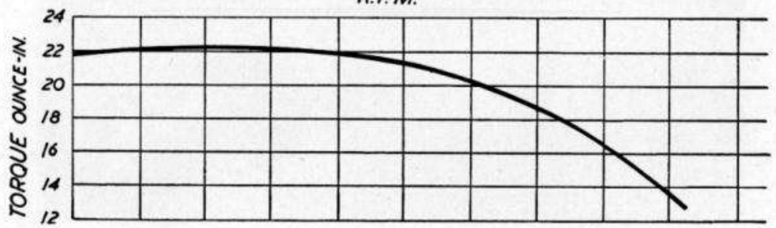
TESTING ONE OF THE SUGDEN SPECIALS are Eric Hook, whose father built the Aeromodeller Eddy-current Dynamometer, designer Dave Sugden and Ron Warring.

SUGDEN SPECIAL

DATA

Displacement : 2.49 c.c. (.152 cu. in.).
Bore : .568 in.
Stroke : .60 in.
Bore/stroke ratio : .93.
Bare weight : $3\frac{1}{2}$ ounces.
Max. torque : 22.2 ounce-inches at 8.500 r.p.m.
Bax. B.H.P. : .243 at 12.700 r.p.m.
Power rating : .0975 B.H.P. per c.c.
Power/weight ratio : .07 B.H.P. per oz.





fit was tighter than most would consider comfortable to handle. But apart from these minor points, everything else had the appearance of being "right".

Starting characteristics appeared quite satisfactory, without being outstanding. Hand-starting with small propellers was typical of a racing engine and the "Special" did appear to like a really generous prime and show a definite preference for the fuel tank being located on an approximate level with the needle valve.

Tests were started with large propeller loads, when it was quickly evident that the torque output was going to be at least as high as any 2.5 c.c. engine yet tested. Maximum torque was generated at 8,500 r.p.m., there being a slight loss of torque with decreasing speed, but running continued satisfactory down to below 7,000 r.p.m. With increasing speed, torque output fell smoothly and readings were obtained in excess of 15,000 r.p.m.

The corresponding B.H.P. curve showed a peak at 12,700 r.p.m. Corresponding peak B.H.P. was .243, which is exceptionally good. The "Special" does in fact, conform almost exactly to the expected output of top-class racing engines of 2.5 c.c. size, whilst its power/weight ratio is appreciably better than average.

Fuel used was quite heavily nitrated, the exact formulation being a matter of conjecture. Basically it started off as 25 per cent Ether, 50 per cent Derv, 22 per cent Castrol XXL, 3 per cent Amyl Nitrate with a *further* proportion of amyl nitrate added to Dave's own ideas. Some separate r.p.m. checks were made with other fuels, all of which showed varying degrees of inferior performance.

Summarising, the "Special" undoubtedly lives up to the requirements of a high-performance 2.5 c.c. engine with no apparent vices, consistent in running with a relatively low vibration level. Particularly commendable is the power/weight ratio achieved whilst retaining an essentially robust unit. In fact, a thoroughly excellent engine all round.

PROPELLER R.P.M. TESTS

Propeller dia. x pitch	r.p.m.
8 x 5 (Frog. nylon)	13,500
9 x 6 (Stant)	10,300
10 x 6 (Trucut)	7,800
11 x 5 (Stant)	7,000
7 x 10 (H-L)	10,100
11 x 8 (Whirlwind)	5,500
9 x 3 (Tiger)	11,400
9 x 4 (Stant)	10,700
7 x 4 (Stant)	15,200

CHAPTER TWENTY-SIX

Making a Pulse Jet

EMIL Brauner of Kladno in Czechoslovakia is a model maker who was forced by circumstances to make his own jet engine. It powers a scale model MiG 15 fighter at 85 m.p.h. We have, therefore, a powerful jet unit and one which can be made by anyone with access to lathe and welding facilities.

First, how it works: Petrol or White Spirit (cigarette lighter fuel) is induced to spray through a metering jet by a fast airflow into the nose cone. The fuel/air mixture passes through flap valves and into the combustion chamber, where it is ignited. Immediately after combustion the burning gases pass through the only exit, the tail pipe, and the resultant reaction provides thrust. As this column of burnt air escapes, a depression occurs in the combustion chamber, and the flap valves which were closed under compression are now opened and a further supply of fuel/air drawn in. So the cycle repeats itself in a series of pulses, each one igniting itself with the heat of the tail pipe which rapidly achieves the state of red heat, as the frequency of explosions is in the region of 200-300 cycles per second.

Because of the fire risk, and the possibility of personal danger, pulse jets are neither to be advised for free-flying nor are they tolerated for such a purpose in Great Britain. They are, however, insurable under a special scheme by the S.M.A.E. for control-line flying and a class exists for Jet Speed, usually flown at the National contests. Current record is 133.3 m.p.h.

Making the Jet

All dimensions on the drawing are in millimetres, and for the convenience of British constructors we provide a table of required equivalents. Start with part 1, a

COMPLETED BRAUNER PULSE JET SERVES TO ILLUSTRATE ITS simple lines. Many have been built as school and apprentice workshop test pieces for welding and turning.

METRIC DIMENSIONS AND DECIMAL EQUIVALENTS

mm.	in.	mm.	in.	mm.	in.
.1	.004	8	.31496	40	1.5748
.15	.006	9	.35433	43	1.69291
.5	.020	10	.3937	44	1.73228
.8	.0314	13	.51181	54	2.12598
.9	.036	14	.55118	55	2.16535
1.0	.03937	15	.59055	56	2.20472
1.1	.04337	16	.62992	60	2.3622
2.0	.07874	17	.66929	64	2.51968
2.5	.0984	18	.70866	65	2.55905
3.0	.11811	19	.74803	86	3.38582
3.2	.12611	22	.86614	129	5.07873
4.0	.15748	24	.94488	134	5.27558
5.0	.19685	30	1.1811	370	14.5669
6.0	.23622	32	1.25984	490	19.2913
7.0	.27559	38	1.49606		

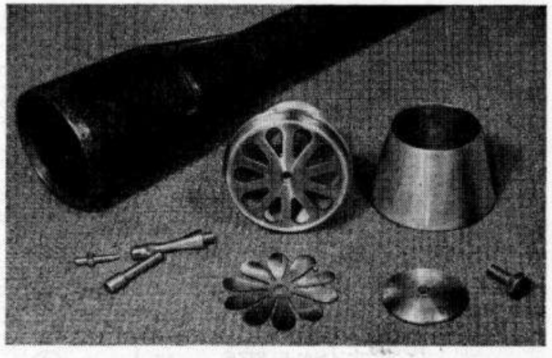
brass turning which serves as an adaptor for the compressed air or car tyre pump air supply during starting. It is brazed at 37° to part 3, the carburettor, which is another brass turning tapped to receive the pilot jet and threaded at the rear end to fit part 6. The pilot jet, part 2, has a 1 mm. orifice. It is advisable to make a set with .9 mm. and .95 mm. alternative jet sizes to determine best diameter for performance. Fuel flows directly from the tank to the pilot jet, thence into the carburettor; and out at 70°-80° through the two .8 mm. oblique holes.

The head—or cone, part 4, is a light metal turning threaded at the rear to fit the collar in the combustion chamber. Care to adhere to the aerodynamic curve, and external relieving to give a wall thickness of 2 mm., will improve performance and save weight. Note that a 3 mm. recess is needed to take part 6 at a later stage. Part 5 is a simple light alloy fairing to blend the carburettor to the valve plate, part 6, and this latter item is turned from the solid in mild steel. There are ten valve holes, each 9 mm. diameter and tapering down to the centre for maximum opening. The valve itself, part 7, is the heart of the jet, and as such is a most critical component. .15 mm. spring steel sheet was used in the original jet; while an alternative, cold drawn sheet, is easier to stamp out and will last for up to 30 starts. Mass production by means of a steel die and hard rubber blanking plate would be one answer to the valve replacement problem. To limit the opening of the valve, part 8 is a backing plate from dural, and here again it is advisable to make alternatives with different curvature to test for optimum performance. Part 9 is merely a standard metric thread bolt to hold the valve assembly together.

Part 6 is peened in place in the head, see detail at 12, and a light alloy nose fairing, 10, riveted as a cone before being "clicked" in place between shoulders. All that remains is the tail pipe, of welded heat-resisting or stainless steel, thickness is not critical between .015 in. and .025 in. made up in three stages to the dimensions in 11. Weld a steel collar in the combustion chamber, and thread to suit the head. Now mount the unit by means of metal collars to a stout board and prepare for first tests.

With fuel in the tank, and a car pump connected to the adaptor, part 1, start pumping with alternate long and short strokes, checking that fuel is drawn through to the carburettor. This done, use the Continental method of ignition by playing a blow-lamp across the jet orifice (not on the tail pipe) and providing a fuel/air mixture is passing through into the combustion chamber, a start is soon effected. There is no such thing as

ALL THERE IS TO THE WORKING parts of a pulse jet! Only really critical component is the flutter valve made from spring steel sheet.



a "misfire" in a pulse jet, either it is going or it is stopped. If the jet appears to show no inclination to keep going, then one should try variations with (*a*) the pilot jet and (*b*) valve backing plate. A low tone indicates a rich mixture and a high note, or short, barking tone, a weak mixture. Hot weather calls for a larger pilot jet, extreme cold a small jet. Having made your own unit, you will soon appreciate these symptoms and their cures.

DIMENSIONED DRAWING APPEARS ON PAGES 176-177

CHAPTER TWENTY-SEVEN

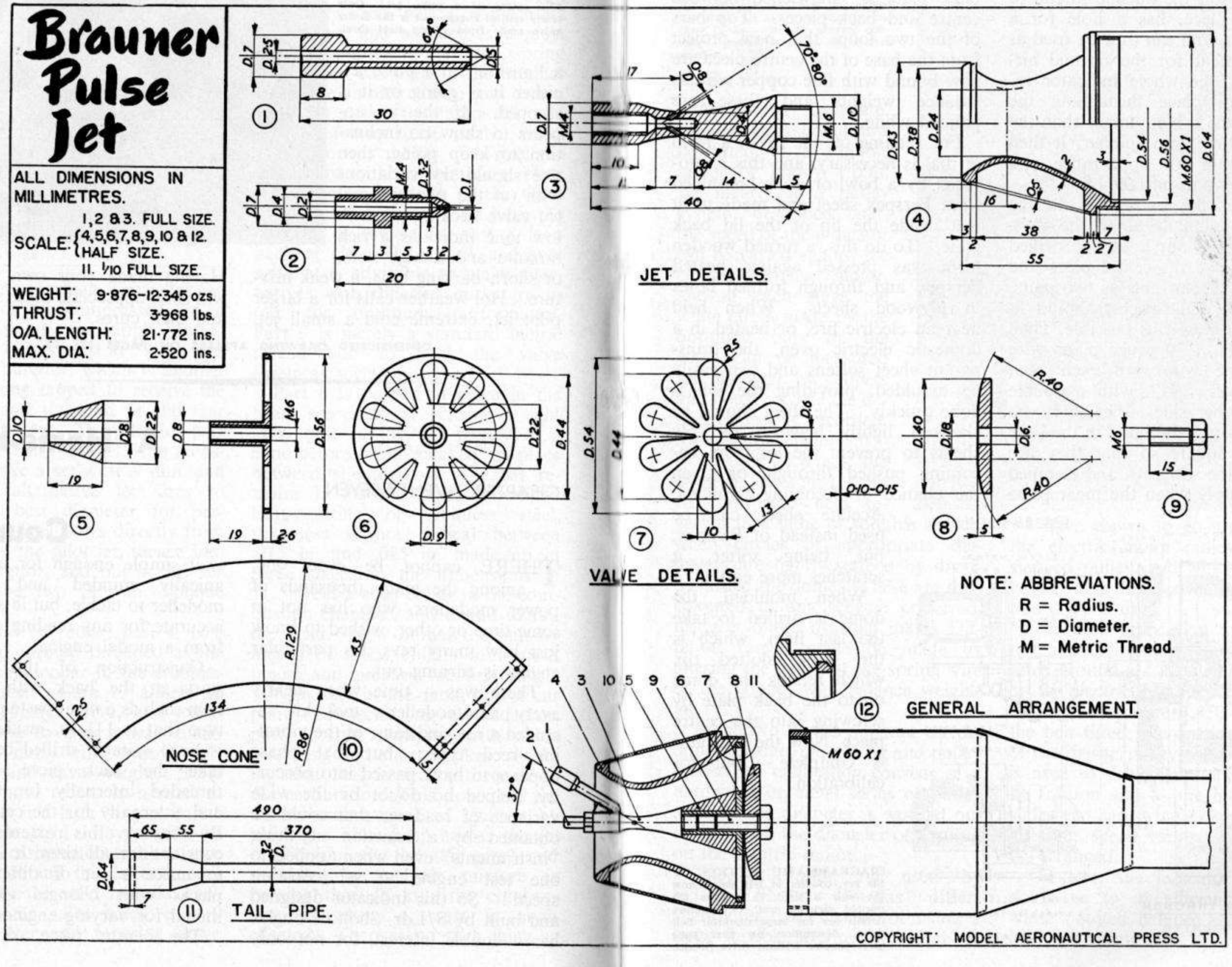
A Home-built Rev Counter

THERE cannot be even one, among the many thousands of power modellers, who has not at some time or other, wished to know just how many revs. his particular engine is turning out.

There was a time when nearly every aeromodeller's tool kit included a rev. indicator of the vibrating reed type; but that phase appears to have passed into obscurity, helped no doubt by the wide variance of readings that could be obtained by a selection of those "instruments" even when applied to one test engine at set constant speed! So this indicator designed and built by S/Ldr. Sholto Douglas has a double interest, for not only is it simple enough for any mechanically minded and equipped modeller to tackle, but it is also very accurate for any reading obtainable from a model engine.

Construction of this indicator starts at the back with a simple item such as a meat paste pot lid (the type that is 1$\frac{1}{16}$ in. inside diameter is best) suitably drilled centrally to take the back piece, which is threaded internally for the shaft, and externally for the centre piece. By having this external thread constant for all sizes, it is possible to make a set of different back pieces with changes of internal thread for varying engines.

The centre piece, drilled and

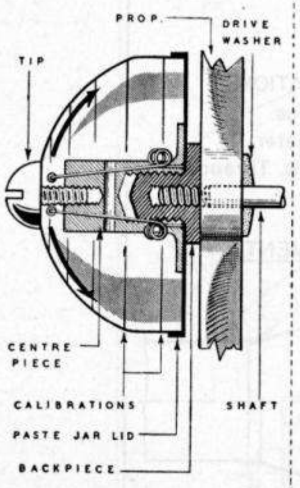

Brauner Pulse Jet
ALL DIMENSIONS IN MILLIMETRES.
SCALE: 1,2 & 3. FULL SIZE. 4,5,6,7,8,9,10 & 12. HALF SIZE. 11. 1/10 FULL SIZE.
WEIGHT: 9·876-12·345 ozs.
THRUST: 3·968 lbs.
O/A. LENGTH: 21·772 ins.
MAX. DIA: 2·520 ins.
JET DETAILS.
VALVE DETAILS.
NOTE: ABBREVIATIONS.
R = Radius.
D = Diameter.
M = Metric Thread.
NOSE CONE
TAIL PIPE.
GENERAL ARRANGEMENT.
M60 X1
COPYRIGHT: MODEL AERONAUTICAL PRESS LTD.

tapped to suit the outside thread of the back piece, has a hole for a tommy bar, and can thus be used as a replacement for the normal airscrew nut, the whole indicator remaining in place throughout the flight. Since it is no larger than the average aluminium spinner, it then doubles both for appearance and indicating r.p.m. at take-off.

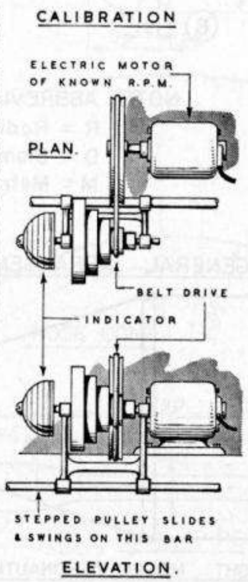
Now for the parts that do the work and actually show the revs. per minute. Four holes are drilled in the base of the centre piece and connected by saw cuts as two pairs. Both holes and sawcuts should fit the wire as closely as possible. Then two lengths of 30 gauge piano wire (control-line wire) are each bent into a square "U", with a double coil on either side. The ends are passed through the holes in the base and bent square so that they are locked in the saw cuts, and clamped there securely when the meat paste back plate is sandwiched between centre and back pieces. Top bars of the two loops that now project from the base of the centre piece are now bound with fine copper wire as balance weights and these are painted white.

For reading off the revs., a form of dial is necessary, and this is provided by a bowl or dome, moulded from Perspex sheet and made to fit just inside the lip of the lid back plate. To do this, a turned wooden form was pressed against heated Perspex and through formed holes in plywood sheets. When held near an electric fire, or heated in a domestic electric oven, the transparent sheet softens and can easily be moulded, providing the job is done quickly. The sheet should be clamped lightly between the ply sheets to prevent the lot from becoming pushed through, or given the chance of becoming wrinkled. Acetate sheet can be used instead of Perspex, but, being softer, it scratches more easily.

When moulded, the dome is drilled to take the last item, which is the screw slotted tip, which retains the dome on to the back plate by screwing into the centre piece. Now the indicator is complete — but for calibration.



SECTION THROUGH REV. COUNTER.



DIAGRAMMATIC SECTION OF the rev.-counter at left shows how the unit is used to replace the normal airscrew nut. Calibration details are for large diameter belt drive, description on next page gives simpler friction drive details.

MOUNTED ON A FROG 150, the complete unit may be judged for size. Diameter is slightly less than $1\frac{3}{4}$ in., and the general contour resembles a transparent spinner.

Calibration

Assembled complete, the indicator should now be fitted to a suitable motor of known r.p.m. By means of a pulley system, described later, the speed of the indicator can be varied against the electric driving motor, and at each stage the white circle made by the out-flung bars of the wires can be measured for diameter by calipers and noted. Once the desired range is recorded, for example, from 5,000 to 12,000 r.p.m., the dome is detached from the unit and held reversed in a lathe or suitable device, when circles of appropriate diameters are lightly scribed by dividers inside the dome. These are then coloured to any scheme preferred, using coloured drawing inks. Different ranges could be made by altering the gauge of the spring wire or the copper wire balance weighting.

As shown in the diagram on the opposite page, the test rig and pulley system for calibration consists of a hardwood or metal series of pulley steps whose diameter is worked out to gear with the diameter of a wheel on the electric motor.

The calibration rig used by S/Ldr. Sholto Douglas differs slightly in that, instead of using a large flexible driving belt such as we have shown to go around both the electric motor pulley and the stepped pulley, and which, is much larger than the Hoover belt, a friction drive is used.

The Hoover belt on the original test and calibration equipment was fitted around a 4 in. diameter pulley on the electric motor. Known revs. of the motor were 1,450 and, with the belt fitted, the outside diameter of the driven pulley was $4\frac{3}{8}$ in. This is used to drive the stepped pulley by friction and as the latter pulley slides and can swing to any position, different speed ranges can quickly be arranged.

On test, the indicator gave no doubt as to its efficiency, as the white painted balance weights varied their throw with speed changes.

APPENDIX I

The need for Silencers

INTEREST in the subject of noise from model aircraft and the introduction of a rule making silencers compulsory for all model engines for S.M.A.E. members suggested the need for an investigation into just how much noise different types of models are making. This article describes the result of measurements made at the Nationals in 1964. It compares noise levels of different categories of model flying.

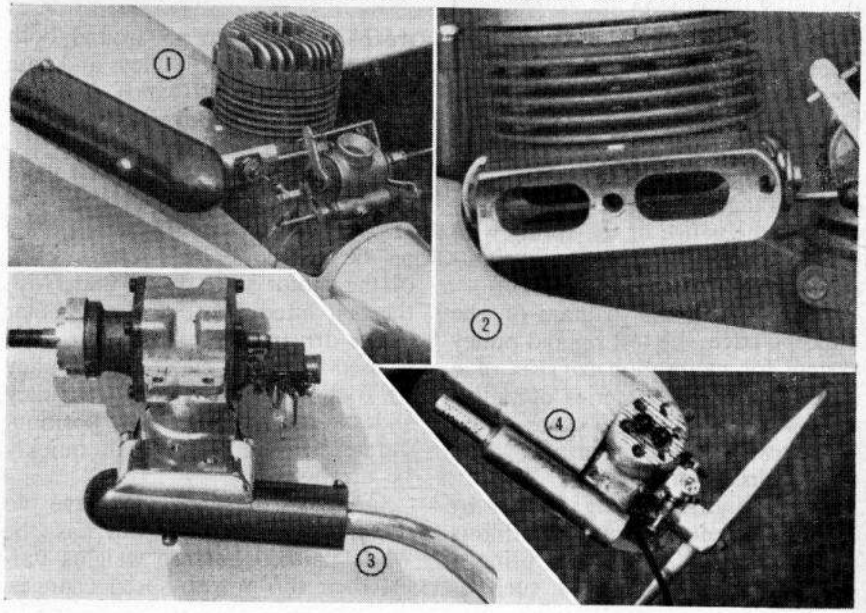
The measurement of sound is a complicated technical procedure with several different units to describe the physical characteristics of the sound such as Sound Intensity and Sound Pressure Level in decibels, and the sensation of loudness produced in the listener in phons and sones. (*Reference* 1). The decibel is a measure of the relative power of two sounds and it only becomes a unit of measurement when an arbitrary reference point is taken. As the loudest sound that we are likely to meet consists of several billion times the energy of the softest, a logarithmic scale is used. This approximates nearer to the response of the human ear than a linear scale. The Overall Sound Pressure Level in decibels is given by the formula:

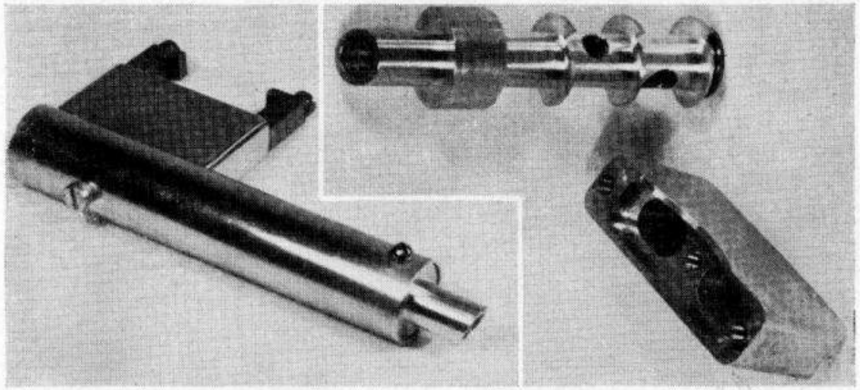
$$\text{O.S.P.L. dB} = 20 \operatorname{Log}_{10} \frac{P}{P_o}$$

Where P = measured sound pressure level and P_o = .0002 dyne/cm². The reference sound pressure.

In practical terms sound meters are

SPINAFLO SILENCER ON A SUPER TIGRE 56 R/C ENGINE WITH THE exhaust baffle incorporated, close up of the mount is in (2) where baffle can be seen. Photo (3) is a McCoy 60 with Johnson carb: and a suspended Spinaflo. Photo (4) is a Rogers-Merco 61, two plugs in the head and Rogers' exhaust silencer.





THE D.A.C. 'Spinaflo' silencer above has an adaptor block according to engine and an internal core with three baffles which rotate the gasflow, hence the name.

built to read directly in decibels and an increase of 6 dB indicates a doubling of the measured sound pressure level.

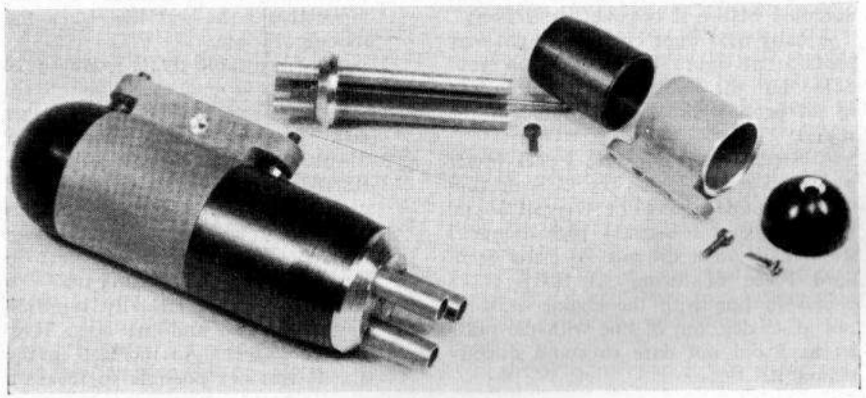
TABLE 1 gives a rough idea of common sounds in decibels:

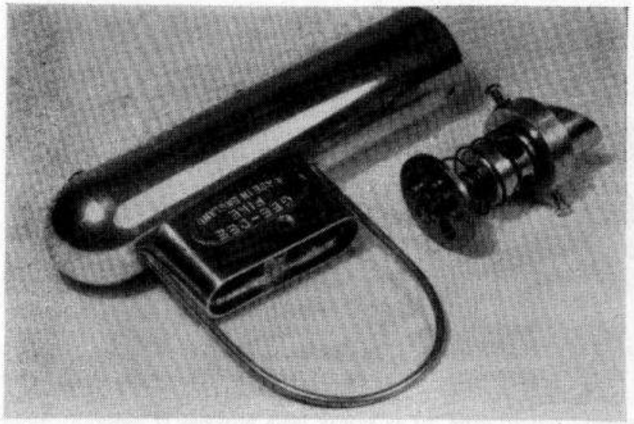
Bird song on a quiet summer evening	40 dB
Normal conversation	60 dB
Underground train	90 dB
Jet airliner take off from outside Control Tower	120 dB

This is by no means the whole picture, however, as sounds occurring in nature are made up of a mixture of frequencies, though one may predominate to give a recognisable "note" to the sound. To measure the sound fully the dB reading for each octave band should be recorded to show how the sound is made up of high and low frequency components. The ear is less sensitive to sounds at either end of the audible range, very roughly, below 100 c.p.s. or above 5,500 c.p.s.

Having now measured the physical characteristics of a noise we are faced with the problem that no two people will hear it exactly the same and that

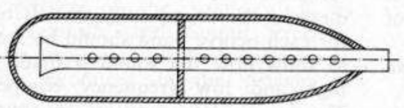
SWISS KOELLIKER SILENCER IS A SIMPLE EXPANSION CHAMBER made up of three pieces plus triple outlet tubes which break down the noise well.

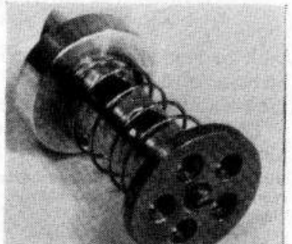




THE GEE DEE PIKE Silencer has an internal spring valve which operates on a resonance with exhaust pressure. See close up. This results in extreme fuel economy and quietness.

THE LINE drawing is a diagram to show what can be done with the plain silencer as originally supplied by O.S. A tube is pushed through from the rear and collects exhaust through holes.





their reaction to it will depend on many different factors. Just two examples quoted in the Wilson Report (*Reference* 2) will show this. A number of people, asked to judge in a controlled experiment, when a motor vehicle was "very noisy" selected 92 dB as the level, but in a similar experiment for aircraft passing overhead up to 115 dB was accepted before it became "very noisy". The baby next door crying at night may produce no more than 40 dB in your home but the annoyance caused is out of all proportion to the volume of the sound.

A Scott Sound Pressure Level Meter was used recording on the C Scale (unweighted) and giving the Overall Sound Pressure Level in decibels. Measurements were made at a distance of "one Standard Piece of String" (1 S.P.S. = 5 paces) in line with the engine exhaust, but at 45 deg. out of line with the pulse jet as I did not dare to stand directly behind it.

Results

1 Noise from control line flying decreases steadily with distance from the circle, and with stunt models particularly it was noticed at a distance that noise was less when the model was low.

2 Noise from radio flying averaged 70 dB on the ground over about 1 sq. mile and was much harder to get away from than with control line.

3 "Multis" produce little, if any, more dB than a "single" of the same engine type. The apparently anomalous result of a twin being quieter with both engines running was due to loss of power on warming up.

4 Speed models are usually run rich on the ground and the noise level can be expected to increase in the air. It was not possible to measure this increase with the equipment available.

5 For a given engine type, noise, as might be expected, increases with an increase of engine speed. The R/C engines seemed noticeably quieter than their C/L equivalent versions.

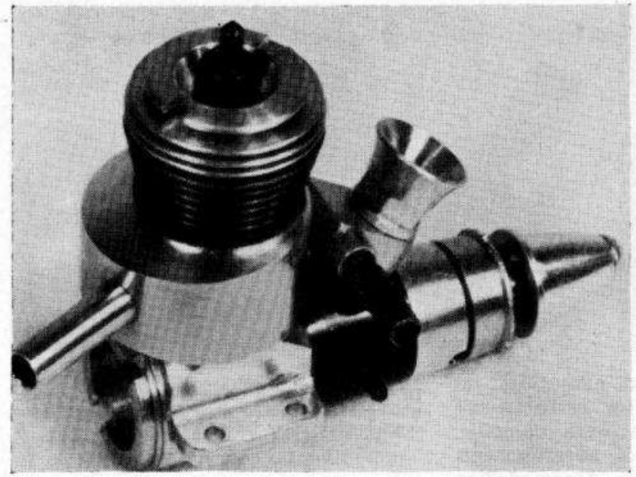
6 In September 1950 the AEROMODELLER reported on the Yulon 49 as unusually noisy. Apart from the Dooling and the pulse jet this still stands.

The dB as measured are no direct measure of potential annoyance. Frequency analysis would have helped as narrow band high frequency noise is more annoying than low frequency noise. However, it is reasonable to expect that, as with full size aircraft, maximum energy is in the frequency band covered by the motor speed in cycles per second, i.e., Stunt 200 c.p.s. and Speed 400 c.p.s. (*Reference* 3).

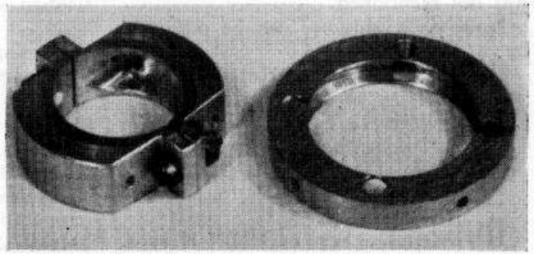
It seems probable that silencers reduce high frequency noise more than low frequency so that their effectiveness may be even better than suggested by the figures.

The reaction of people to noise is very complex and depends on duration, frequency of exposure and normal background noise as well as the character and sound pressure level of the noise itself. It is worth remembering that old people are more upset by any disturbance than young, and that what may sound like a mild buzz to a young man may be genuinely intolerable to an old lady.

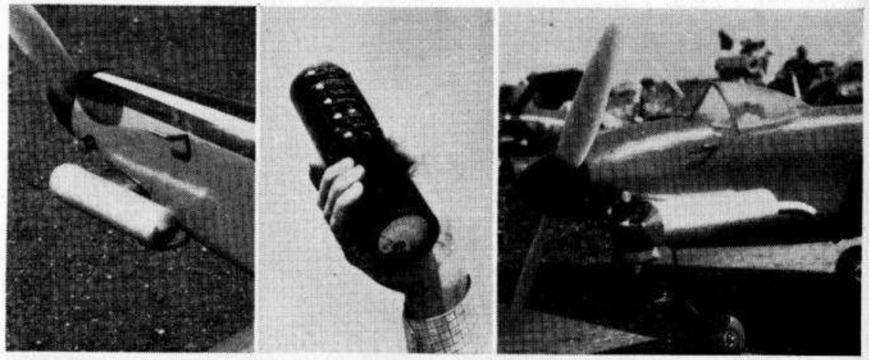
In the Wilson report it was stated that



SILENCER AT LEFT for a Cox .049 is by Russell Models, Worksop and Lincoln. Note eccentric manifold chamber which is rotated according to exhaust port position.



THE P. A. MOORE SILENCER for a Cox engine at left is a clamp fit that requires care in installation. At right is the P.A.W. ring for the P.A.W. 2.5, 3.5 and 19 diesels.



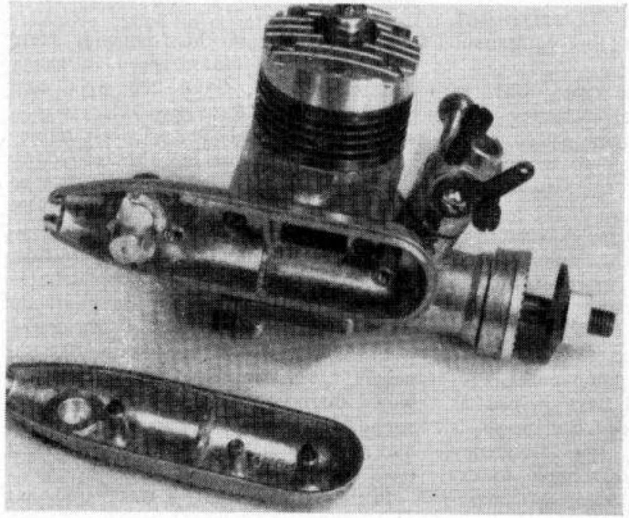
THE SCOTT METER AS USED BY AUTHOR Dr. M. F. Hawkins for figures in this Appendix is sandwiched between views of Geoff Higg's home constructed silencer units.

in one year 72 local authorities received 2,350 complaints of noise of which the commonest were domestic, motor vehicles, factories, advertisements and others in that order. Aeromodelling was not mentioned.

Noise as a nuisance can be mitigated by:

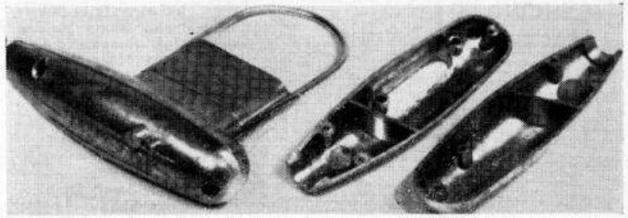
1 *Control at source. Silencers and/or limited flying hours.*
2 *Screening. This does not help with aeromodelling.*
3 *Keeping it at a distance. It is only common sense to fly well away from houses.*
4 **Prohibition.**

Damage to the ear by permanent loss of hearing can result from exposure to loud noises. Pulse jets produce noise of this level but the exposure of the ground crew is usually remarkably short. However, if ground running of more than four minutes in any one day is contemplated, then some sort of ear protection should be worn. To summarise:



THE O.S. SILENCER with rotary baffle inside connected to the carburettor to aid speed control.

THE MERCO Silencer at right, showing internally cast baffles.



1 Sufficient noise is produced by most forms of model engine to cause a public nuisance and unless action is taken to mitigate the noise, complaints, and therefore loss of flying grounds will increase in the future.
2 Efficient silencers greatly reduce the O.S.P.L. and the annoying character of the noise, but they do not eliminate it altogether and annoyance could still be caused by some forms of "silenced" model.
3 There seems to be little point in fitting a silencer to a "sports" engine of under 1 c.c. as their noise level is well below that of a silenced .35.
4 To quote again from the Wilson Report: "A noise problem must involve people and their feelings and its assessment is a matter rather of human values and environments than precise physical measurement."

REFERENCES

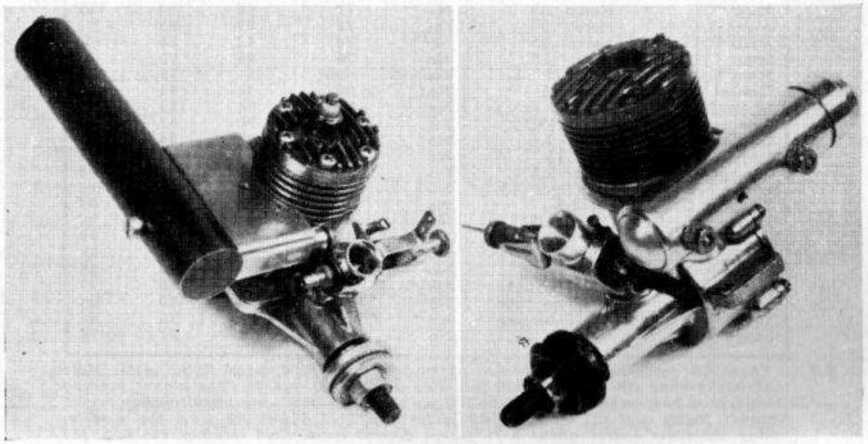
1 *Parrack, H. O. "Effects of Acoustic Energy" in Aerospace Medicine. Ed. Armstrong.* 284-323. *Balliere Tyndall and Cox* 1961.
2 *Final Report of Committee on Noise. Ed. Wilson H.M.S.O.* 1963.
3 *Gasaway, D. C. "Noise Associated with Operation of Miliary Aircraft". J. Aerospace Medicine.* 35.327. 1964.

Tests with Silencers

Engine	*Type of Silencer*	*Noise Silenced*	*Noise Unsilenced*
Merco 35	Fox	90	94
Merco 35	D.A.C.	80	94
Merco 35	Higgs home-made	80	94*
Merco 35	O.S. Jetstream	82	94*
Johnson 35	O.S. Jetstream	81	94*
Taplin Twin	Manufacturers	80	—

**Estimated*

SPINAFLO SILENCER ON A VECO 19 AND AUSTRALIAN BURFORD silencer on a Taipan 29 R/C. Note the hole for easier priming on the latter at side, and on Spinaflo above adaptor block.



SILENCING FACTS AND FIGURES

RELATIVE LEVELS OF TYPICAL NOISES

NOISE	DECIBELS	TYPICAL EXAMPLES
		Threshold of Feeling
	120	
Deafening	110	Jet Aircraft at 500 ft. Inside Boiler Making Factory Near Pneumatic Drill Motor Horn at 20 ft.
	100	
Very Loud	90	Inside Tube Train Busy Street Workshop Small Car at 24 ft.
	80	
Loud	70	Noisy Office Inside Small Car Large Shop Radio Set—Full Volume
	60	
Moderate	50	Normal conversation at 3 ft. Urban House Quiet Office Rural House
	40	

Nationals Recordings

All readings in dB. at standard distance from exhaust.

1—Stunt
Merco 35 — 96, 94
Fox 35 — 91
Super Tigre 35 — 86
Oliver Tiger — 84
Q.S. Max I. 35 — 97
O.S. Max III. 35 — 95

2—Radio
Merco 49 — 93, 95

3—Scale (Radio)
Merco 35 — 88
Super Tigre 56 — 96
Enya 35 TV — 86
Merco 49 — 92
Frog 500 — 82 (inside cowl)
Enya 29 — 94 (inside cowl)
Rivers 2.5 — 85
A.M. 1.5 — 80 (76 with cowl)
McCoy 60 — 98
Torp 19 x 4 — 88 (all running)

(Control line)
Yulon 49 — 97
E.D. 3.46 x 2 — 82 (83 one running)

(Free flight)
Mills .75 — 76
Mills 1.3 — 70 (very large prop)
A.M. 2.5 — 84
A.M. 1.5 — 80

4—Free Flight Power
Torp 19 — 92
Enya 19 — 90
Cox T.D. 15 — 94, 94
Cox T.D. .049 — 89

5—Team Race
Eta 29 — 94
Oliver Tiger 1.5 — 82, 84

6—Combat
Oliver Tiger 2.5 — 82, 86, 86

7—Speed
McCoy 60 — 98
Super Tigre 29 — 96
Johnson 29R — 97
Dooling 61 — 111
O.S. Pulse Jet — 120

Model type	Engine type	Silenced	Backgound Noise at site	Test dB readings 15 ft.	50 ft.	In air
Multi R/C	Merco .49	No	77	97	94	96
Multi R/C	Merco .35	Yes	78	89	83	91
Multi R/C	Merco .35	No	78	92	86	92
Stunt C/L	Merco .35	No	76	96	90	96
Stunt C/L	O.S. .29	Yes	48	81	79	82
Combat C/L	A.M. 3.5	No	80	98	91	94
Single R/C	O.S. .19	No	78	84	80	90
Combat C/L	PAW .19D	Yes	78	90	83	96
Combat C/L	Oliver 2.5	No	76	92	86	88
Single R/C	O.S. .15	No	78	96	90	94
Combat C/L	Oliver 2.5	Yes	50	81	78	80
Power F/F	Sup. Tigre .15	No	48	98	95	96
T. Racer C/L	Eta 2.5	No	48	93	89	97
T. Racer C/L	Oliver 1.5	No	47	84	82	86
Combat C/L	Cox TD .09	No	48	98	94	99
Sports F/F	A.M. 1.5	No	76	84	80	86
Sports F/F	A.M. 1.0	No	76	83	79	85
Single R/C	Cox TD .049	No	76	97	90	97
Sports F/F	Wen Mac .049	No	76	86	80	83
Sports F/F	Cox Babe Bee	No	76	84	76	86
Sports F/F	Mills .75	No	77	77	77	77

These dB readings were recorded at two well known London flying sites. Note that background dB readings fall into two groups, 40 and 75. The higher reading is due to a busy main road within 100 yards of one site, and simultaneous model aircraft flying. For all tests the dB meter was held downward from the models, facing the noise source about 5 ft. above ground level. Cold weather (40 deg. F.) conditions with slight drizzle prevailed.

APPENDIX 2

R.P.M. TEST FIGURES by TED MARTIN* ON VINTAGE U.S.A. ENGINES STILL IN REGULAR USE

TOPFLITE "POWER PROPS"

ENGINE (.19—.60)	10 x 8	10 x 6	9 x 8	9 x 6	8 x 8	8 x 6	7 x 9
54 Fox 19		11,500		12,000			14,000
Fox 25		12,000		12,800			14,200
Fox 29	11,500	12,400	12,600	13,700	14,050	14,800	14,650
Moir-Fox 29			13,200	14,250	14,600	15,400	15,300
Fox 29R	12,200	13,150	13,500	14,600	15,050	16,100	16,000
Fox 35	11,900	12,750	13,050	14,100	14,500	15,250	15,100
McCoy 29FR	11,400	12,250	12,500	13,600	14,000	14,800	14,500
McCoy 35	12,200	13,000	13,250	14,400	14,700	15,450	15,400
Veco 19		11,400		11,950			13,800
54 Veco 29	11,000	11,900	12,100	13,200	13,400	14,000	13,700
K. & B. 23		11,800		12,400			14,150
K. & B. 35	12,000	12,800	13,200	14,250	14,600	15,400	15,300
Johnson 29	11,100	12,000	12,300	13,400	13,500	14,100	13,800
Forster 29	9,000	10,400	10,600	12,500	12,750	13,700	13,900
Cameron 19		11,000		11,500			13,200
McCoy 60	14,750		16,100		17,050		

ENGINE (.049—.09)	6 x 5	6 x 4	6 x 3	5¼ x 5	5¼ x 4	5¼ x 4
Sky Fury 049	10,000	11,000	12,900	12,200	13,200	14,300
Sky Fury De Luxe ...	11,200	12,350	15,000	14,900	16,200	18,100
O.K. 049D	10,500	11,700	14,000	12,800	13,700	15,200
McCoy 049 Glow ...	10,500	11,750	14,250	13,000	14,500	17,000
McCoy 049 Diesel ...	11,200	12,400	14,700	13,750	14,250	14,800
Ohlsson 049 Midget ...	11,800	13,300	15,400	14,100	16,000	17,500
Royal Spitfire 049 ...	9,500	10,500	13,000	13,200	11,500	15,500
Wen Mac 049	9,000	10,400	13,100	11,500	13,350	15,600
Atwood 049	10,500	11,900	14,000	12,500	14,100	16,740
Shriek 049	10,850	12,250	14,400	12,900	14,500	17,200
K. & B 09	13,550	15,000	16,200	14,900		
Cox 049	12,000	13,650	16,400	15,050	18,000	21,200

*Noted British engine designer (Amco 3.5 P.B. & B.B.) currently designing high performance racing car engines.

APPENDIX 3

Torque Absorption Data

A limitation, common to all engine performance test reports to date, is a lack of co-relation between performance in terms of torque output and B.H.P. and performance in terms of r.p.m., with a given size of propeller. Some reports give torque and B.H.P. and no propeller-r.p.m. figures. Others give propeller-r.p.m. figures and no torque or B.H.P. measurements. Our own policy throughout has been to give both, but as a general rule derived under somewhat different test conditions. For that reason, and others which will be discussed, anomalies can appear. Engine "A" which, from the B.H.P. curve is seen to be more powerful than engine "B" does not give a correspondingly higher r.p.m. figure on a quoted size of propeller.

In terms of basic theory the torque absorbed by any particular propeller should be proportional to (r.p.m.)^2, so that the performance of any particular propeller should be capable of being expressed in the form.

Torque $(Q) = K N^2$

where K is a constant (torque coefficient) N=r.p.m.

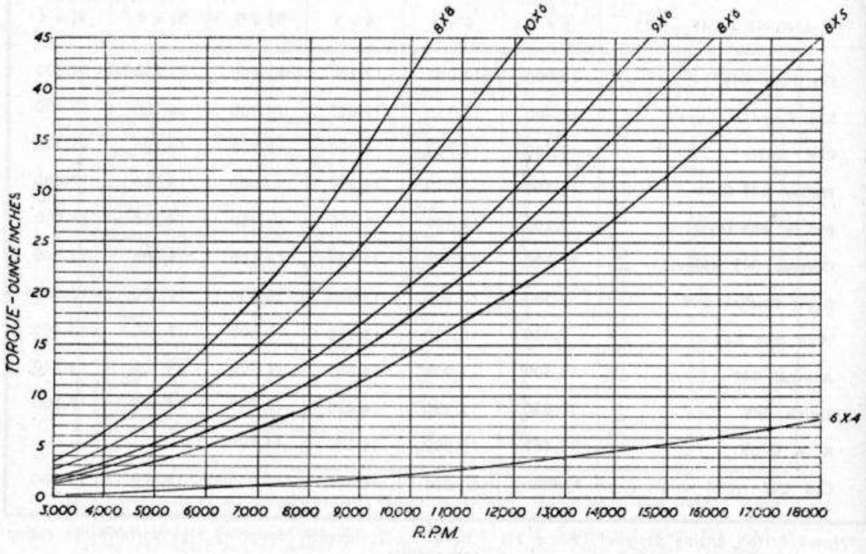
To be strictly true torque absorbed will also vary with the density of air and so a more accurate equation is

$$Q = C_q \sigma N^2$$

where Cq is the torque coefficient of the propeller, and σ is the relative air density.

Now unfortunately Cq is very dependent on the geometry of the propeller. Nominally identical propellers may have quite different values of Cq depending on differences in edge form and thickness, actual blade section, and so on, so there is one possible source of error. The fact that the relative air

TORQUE ABSORPTION : FROG PLASTIC PROPELLERS



density may be several per cent different on two different occasions for testing is another.

Other possible sources of error are largely concerned with measurement and adjustment—limits of accuracy of the measuring instrument used and in adjusting the engine itself to optimum settings on a particular load.

Dealing with direct measurement first. Liability to error in r.p.m. measurement can be as high as **plus or minus** 10 per cent with a good reed tachometer and up to twice this with a poor one. That could mean a matter of 1,500 to 3,000 r.p.m. at a nominal 15,000 r.p.m. In general, errors will be smaller than this but, in any case, reed-type counters are not used for our own figures. But either of the alternative standard types—a tachometer or a stroboscope—are still subject to limitations. The former absorbs a certain amount of power to drive and therefore gives a slightly low reading. The latter is subject to drift, possibly as much as 500 r.p.m. **either side** of a nominal value at times.

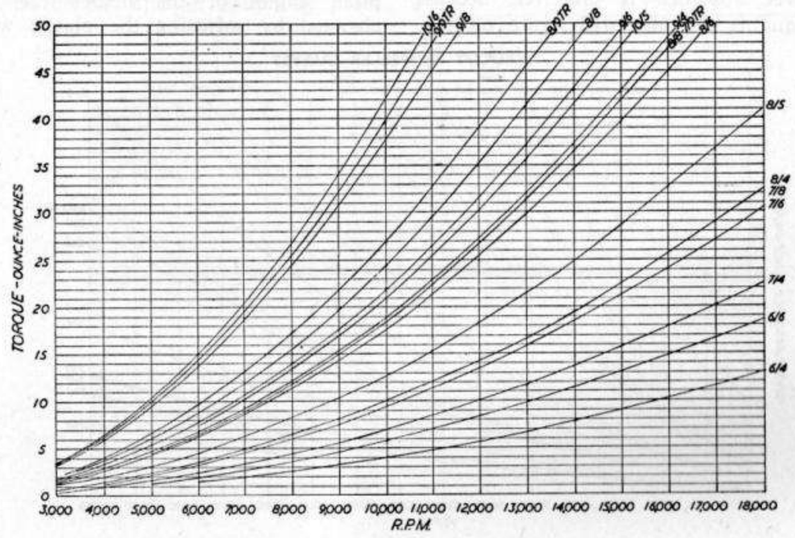
Add to this the fact that engine adjustment also plays a significant part. Also, of course, many engines tend to lose speed on warming up and show a consistent r.p.m. figure lower than might be obtained by measuring straight away after starting.

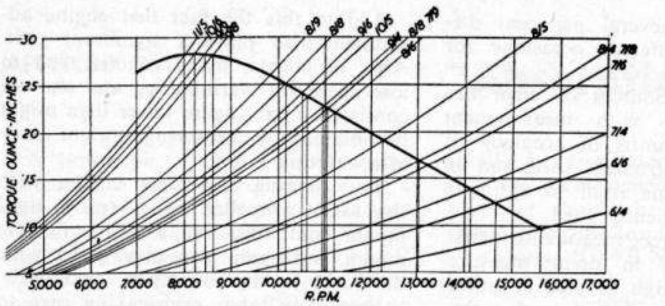
Try running the same engine with the same propeller on a really rigid mount and then on a fairly flexible mount and again you may get a wide difference in the two r.p.m. readings.

Sooner or later, even taking care to reduce reading and adjustment errors to a minimum, all the "plus" or "minus" errors are going to add up the same way and then you get a big discrepancy, which may well pass unnoticed at the time. Since by far the most difficult part of engine testing is in extracting torque figures corresponding to different speeds, i.e., at different braking loads, one is rather apt to regard the more direct measurement of how fast an engine will drive a particular prop, as more of an afterthought.

To the engine user, however, the propeller data is probably of more use

STANT PROPELLER CURVES





HOW TO USE the prop. torque charts. This graph shows an overlaid torque curve, taken from an AERO-MODELLER engine analysis, with figures for the Stant prop. range. Reading off estimates, the engine should deliver 14,200 r.p.m. on a 7 x 4, 10,500 r.p.m. on a 9 x 6.

than B.H.P. or torque curves and so for a long time it has been appreciated that a reliable tie-up between the two was necessary. By conforming strictly to the use of a selected batch of propellers only and averaging out results over a large number of engine tests it has, finally, been possible to produce a series of curves which show good co-relation with practical results.

Frog Prop Curves

These curves plot torque absorbed by each propeller against r.p.m. and are based on the simplified equation of torque=torque coefficient **times** (r.p.m.)^2 and assumes that air density is constant. The use of **torque** absorbed instead of **power** absorbed is preferred because torque is the **measured** figure on test.

The corresponding horse-power absorbed is found by

$$\text{horse-power} = \frac{\text{torque (oz.-in.)} \times \text{r.p.m.}}{1{,}008{,}000}$$

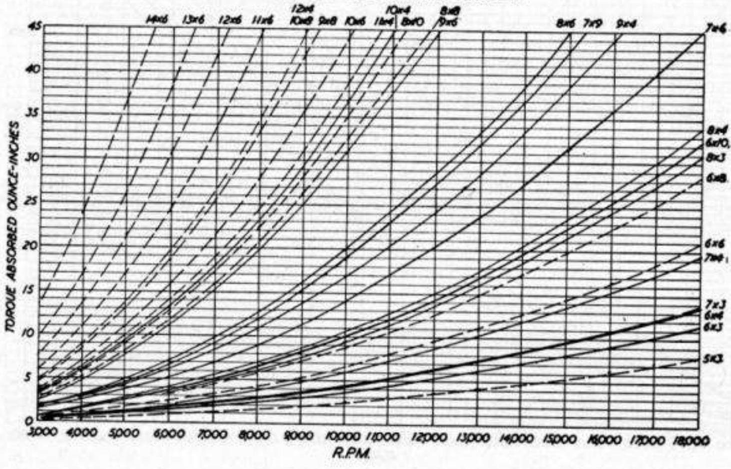
or with sufficient accuracy for most purposes.

H.P.=torque (oz.-in.) x r.p.m. **divided** by 1,000,000.

Thus thinking in terms of power, original errors are **multiplied** by r.p.m. and so exaggerated.

The Frog range of plastic propellers is actually moulded in high impact Polystyrene, Acetate and Nylon AF, the former in colours (mainly red) and the latter only in natural (translucent creamy-white). Both materials are thermoplastic which means that the pitch angles of the blades can be changed by softening the plastic with

TRUCUT PROPELLER CURVES



gentle heat and twisting. This is an advantage in many practical cases. For example, the standard propeller for the Frog "50" is the 6 x 4 which is a little too small for optimum performance on a control line model. A marked improvement can generally be realised by resetting the blades to a slightly coarser pitch angle (5 to 6 inches pitch) and, if necessary, trimming the blade diameter slightly. Such treatment provides an "intermediate" propeller size to fill the gap at present existing between the 6 x 4 and the next smallest size in the range (8 x 5).

The fact that the materials used are thermoplastic is also a disadvantage in that the pitch as moulded may be subjected to change on ageing. In fact the final pitch on any moulded propeller is largely dependent on the temperature of the product when initially removed from the mould. Thus individual examples do show differences in performance in practice, the main offender in the range being the 8 x 5 which has a somewhat thick blade section.

Data on Torque Absorption for the Wooden Prop Ranges

From the readers' point of view these should be of value in estimating probable r.p.m. figures with standard commercial propellers against an engine performance curve—subject to a rather wide range of tolerance and the fact that discrepancies may well show up related back to some individual test reports because of the different conditions under which some of these figures were obtained.

Table below gives comparison of figures measured from graph and actual readings taken on test bench from engine, discrepancies of up to 200 r.p.m. are insignificant.

Propeller	R.P.M. from graph	R.P.M. as measured
11 x 5	8,000	8,000
10 x 6	8,250	8,200
10 x 6	8,250	8,200
9 x 9	8,450	8,400
9 x 8	8,650	8,600
8 x 9	10,200	9,600*
9 x 6	10,500	Not tested
10 x 5	10,650	10,500
9 x 4	10,950	11,000
8 x 8	11,050	10,900
7 x 9	11,050	11,000
8 x 6	11,250	11,200
8 x 5	12,350	12,250
8 x 9	13,050	13,000
7 x 4	14,200	Not tested
6 x 6	14,700	Not tested
6 x 4	15,750	Not tested

* Test report reads . . . "motor was not happy with this prop".

APPENDIX 4

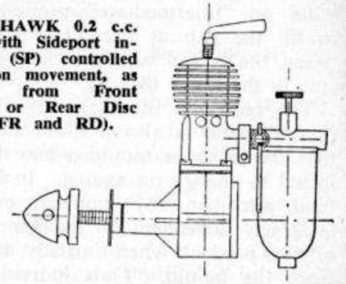
World's Model Engines

The old scheme of classifying engine utility against its capacity used to serve as a standard yardstick — until engine design unfolded new techniques, new power levels. For example, there are some 1.5 c.c. diesels equal in usefulness to others of 2.5 capacity. Conversely there are 2.5 c.c. engines with "slogging" power at lower, revs per minute, that equal much larger engines for use in a sport model, yet fall below requirements for 2.5 c.c. when employed for a contest model.

Grading the world's engines so that due allowance be made for differences in characteristics and power output has meant that no less than fourteen classes are used to segregate the vast range from .15 to 26 c.c. The classes are lettered from **A** to **O** and to find the grading for your particular engine, just follow the line against its name until you reach the "power coding" column.

Propeller Selection

Against each engine there are three sizes of propeller. These are basic dimensions derived from practice in the field, contest flying, sports flying and designer's advice. Use the size given if you have any doubt on your own selection — and remember — large airframes (72 in. for 2.5 c.c.) require an extra inch in prop diameter, keeping advised pitch, and smaller airframes (48 in. for 2.5 c.c.) can be cut by as much as half an inch on diameter.



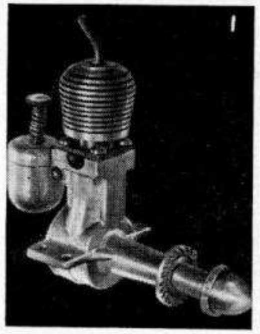
KEMP HAWK 0.2 c.c. diesel with Sideport induction (SP) controlled by piston movement, as distinct from Front Rotary or Rear Disc valves (FR and RD).

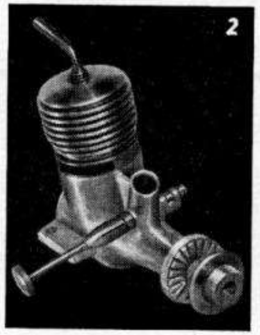
Gear the pitch of your prop against the rate of climb and engine r.p.m. (4 in. pitch for a fast climbing contest model and 12,000-13,000 r.p.m. engine) or step up the pitch for slower sports models (6 in. pitch for 7,000-9,000 r.p.m.). A good tip is to fit the prop back to front for first test flights when full thrust is not advisable.

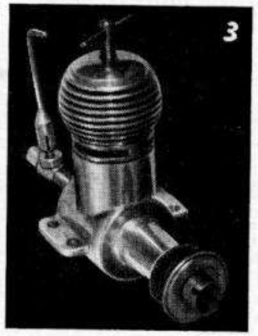
Above all: mount your engine firmly, and treat it with the respect it deserves.

ABOVE LETTER POWER CODING SYSTEM IS APPLIED TO ALL POWER DESIGNS IN THE AEROMODELLER PLANS SERVICE.

THREE DIFFERENT INDUCTION SYSTEMS are typified below. (1) is Sideport (SP); (2) is Front Rotary via the crankshaft valve (FR); and (3) is Rear Disc valve (RD).







WORLD'S MODEL ENGINES (excluding Soviet Products)

ENGINE	DISPLACEMENT cc.	cu. ins.	Cylinder Bore	Stroke	WEIGHT (ozs)	Useful RPM RANGE	RECOMMENDED PROPELLER Sport	Contest	C/L	MOUNTING•	• Induction	POWER GROUP
BRITISH												
Allbon Bambi ...	.15	.009	.21	.25	.75	10–14000	4x2	—	—	R or B 9/16	FR	A
Kalper	.32	.019	.251	.402	.875	9–11000	5x3	—	—	B 11/16	SP	B
E.D. Baby	.47	.028	.312	.375	1.4	10–12000	5x3	6x3	6x3	B 3/4	FR	B
Frog 50	.49	.030	.343	.33	1.25	11–13000	6x3	6x3	6x4	B 3/8	FR	B
Allbon Dart ...	.55	.036	.35	.35	1.25	9–14000	7x4	6x3	6x4	R or B 11/16	FR	C
Elfin .5	.536	.0327	.329	.385	1.5	10–12000	6x4	6x4	6x4	R	FR	B
A.S. 55	.566	.034	.350	.356	1.5	6–16000	7x4	5x6	6x4	B 11/16	FR	C
Cobra 049	.798	.0487	.406	.376	1 3/8	8–16000	6x4	6x3	6x4	B 3/4	RR	C
A-M 049	.83	.0506	.421	.364	1.75	9–18000	6x4	6x3	6x4	B	FR	C
D.C. Bantam ...	.762	.0465	.410	.352	1.5	10–18000	6x3	5x3	6x4	B 11/16	FR	C
Frog 049	.808	.0492	.400	.392	1.8	8–15000	6x4	6x3	6x4	B	FR	C
Frog 80 Mk II ...	.80	.049	.400	.392	1.9	7–14000	7x4	6x4	6x6	B	FR	C
Mills P 75	.75	.047	.335	.516	2	6 12000	8x4	7x4	6x6	B 7/8	SP	C
Albon Merlin ...	.76	.047	.375	.420	1.75	9–14000	8x4	7x4	6x6	R or B 13/16	FR	C
Amco .87	.87	.053	.375	.5	1.75	7–9000	8x4	8x4	7x6	B 13/16	SP	C
Allbon Spitfire ...	.975	.059	.425	.42	3	9–12000	9x4	8x4	7x6	R or B 7/8	FR	D
E.D. Bee	.98	.059	.437	.420	2.75	8–11000	9x4	8x4	7x6	B 7/8	RD	D
Frog 100 Mk. II ...	1.02	.062	.416	.460	3	6–14000	8x5	8x4	7x5	B 7/8	FR	E
A.M. 10	1.0	.061	.426	.430	3	11–14000	8x4	7x4	7x6	B 7/8	FR	E
M.E. Heron ...	.97	.059	.424	.420	2.4	7–14000	8x4	7x6	7x6	B 7/8	FR	D
Frog Viper	1.48	.09	.500	.460	4 1/8	8–16000	9x4	7x4	8x4	B 1 1/16	RD	F
Frog Venom ...	1.48	.09	.500	.460	3.75	8–12000	8x4	7x4	7x6	B 1 1/16	RD	D
P.A.W. 1.49	1.473	.09	.494	.469	3.5	7–18000	9x4	7x4	7x6	B 7/8	FR	F
Oliver Tiger Cub II	1.46	.089	.4659	.523	4 1/8	7–17000	9x4	8x4	8x4	B	FR	F
E.D. Super Fury ...	1.49	.092	.500	.462	3.75	8–17000	9x4	7x4	7x6	B 7/8	RD	F
Mills 1.3	1.33	.081	.406	.625	3.25	5–8000	9x5	9x4	8x6	B 1 1/8	SP	E
E.D. Hornet ...	1.45	.085	.531	.4	3.25	9–13000	9x5	8x4	8x6	B 7/8	RD	E
E.D. Fury	1.5	.090	.5	.468	3 5/8	8–14000	9x5	8x4	8x6	B 7/8	RR	E
A.M. 15	1.48	.089	.517	.430	3	7–15000	9x5	8x4	7x6	B 7/8	FR	F
J.B. Atom	1.47	.09	.536	.397	3.12	8–10000	8x4	7x4	7x6	R or B 15/16	FR	D
Frog 149 & 150 ...	1.49	.091	.5	.46	3.25	11–14000	9x5	8x4	8x6	R or B 7/8	FR, RR	F
Elfin 1.49	1.49	.091	.503	.466	2.6	11–14000	9x6	8x4	8x6	B 7/8	FR	F
Elfin 1.49 BB ...	1.49	091	.503	.46	4	10–15000	9x6	8x4	8x6	B 7/8	RR	F
AllbonJavelin&Sabre	1.49	.091	.525	.42	3	10–12000	8x6	8x4	8x6	B 7/8	FR	E
Elfin 1.8 PB	1.8	.110	.505	.562	3.25	8–12000	10x6	9x4	8x6	R	FR	F
Elfin 1.8 BB	1.8	.110	.503	.562	4.1	8–14000	10x6	8x4	8x6	B	RD	F
E.D. Comp. Special	2.0	.122	.5	.625	5.75	6–8000	10x6	8x6	8x8	B 1 1/8	SP	E
Allen-Mercury 25	2.4	.147	.57	.562	4	10–13000	10x6	8x4	9x6	B 1"	FR	G
E.D. Racer	2.46	.15	.59	.55	5.4	10–15000	10x6	9x4	9x6	B 1 1/8	RD	G
D.C. Allbon Rapier	2.46	.15	.578	.570	5	8–13000	9x6	8x4	9x6	B 1 1/16	RD	G
Elfin 2.49 PB & BB...	2.48	.15	.554	.625	3.4	9–13000	10x6	8x4	8x6	B or R	FR, RR	G
Frog 249	2.49	.151	.58	.568	6	8–16000	11x6	10x4	9x6	B 1 1/16	FR	G
Rivers Sil. Streak II	2.49	.152	.578	.578		8–17000	10x6	9x4	9x6	B	FR	G
P.A.W. 2.49 III ...	2.46	15	.592	.532	5	8–17000	10x6	8x4	9x6	B	FR	G
Oliver Tiger III ...	2.424	1479	.551	.620	5.5	8–16000	9x6	8x4	8x6	B	FR	G
ETA 15	2.48	.15	.558	.620	5.75	8–18000	9x6	8x4	9x6	B 1/18"	RD	G
P.A.W. 19D	3.128	.1912	.642	.590	5.5	7–16000	10x4	8x4	8x6	B	FR	H
Rivers Sil. Arrw 3.5	3.46	.211	.607	.642	7 1/8	8–17000	10x6	8x5	9x6	B 1 1/8"	FR	H
Frog 3.49	3.43	.209	.666	.600	6.5	8–14000	10x4	9x6	9x6	B 1 1/8"	RD	G
ETA 19 Mk 2 ...	3.254	.1985	.640	.617	4.5	9–18000	9x6	8x4	8x6	B	RD	H
D.C. Tornado ...	4.972	.303	.567	.585	10	8–14000	11x4	9x6	9x8	R		J
Amco 3.5 PB ...	3.42	.209	.687	.562	4.25	10–12000	11x6	10x4	10x6	R or B 1	FR	H
Amco 3.5 BB ...	3.42	.209	.687	.562	5.5	10–13000	11x6	10x4	10x6	B 1 1/8	RD	H
Allen-Mercury 35 ..	3.42	209	.687	.562	4.5	10–14000	11x5	9x4	9x6	B 1	FR	H
D.C. Manxman ..	3.43	.209	.687	.562	5.5	8–11000	11x6	10x6	10x8	B 1 1/8	FR	H
E.D. Hunter ...	3.46	.211	.656	.625	6.5	8–11000	12x6	10x5	10x6	B 1 1/2	RD	H
Miles Special ...	4.92	.3	.781	.625	10	8–14000	12x6	10x4	10x6	B 1 5/16	RD	J
ETA 29 (glow) ...	4.95	.3	.75	.672	6.5	10–14000	10x6	9x5	9x6	B 1 1/2	RD	J
Frog 500 (glow) ...	4.95	.3	.75	.68	7.75	7–11000	11x6	10x4	9x6	R or B 1 1/4	FR	J
Merco 35	5.794	.353	.800	.703	7.5	8–16000	11x6	10x4	10x6	B 1 5/16	FR	L
Taplin Twin.. ...	6.920	.420	.656	.621	15	6–10000	10x8	—	10x8	B 1 1/4	FR	J
NORWEGIAN												
David Anderson ..	2.46	15	.551	.630	5 3/4	6–12000	11x6	9x4	9x6	B	FR	
AUSTRALIAN												
Taipan 1.5	1.500	091	.511	.453	3 3/16	7–16000	8x4	7x4	7x6	B 7/8"	FR	E
Burford Sabre 15 ..	1.42	091	.503	.466	3	8–12000	9x6	8x4	7x6	R	FR	E
Sabre 250	2 46	15	.55	.620	4 1/4	8–14000	10x6	9x4	8x6	B 1" or R	FR	G
Taipan 2.5 BR ...	2.506	1529	.576	.587	5 1/2	8–16000	9x4	8x4	8x6	B 1 1/16"	FR	G
Glo-Chief 19 ...	3.30	.1994	.640	.620	6 1/8	8–16000	9x6	9x4	9x6	B	FR	H
Glo-Chief 29 ..	4.92	30	.739	.700	7 3/8	8–17000	11x4	9x6	10x6	B 1 1/4"	FR	J
Burford Sabre 19 ...	3.27	19	64	.620	6	9–14000	10x6	9x6	9x8	B	FR	H
Sabre 49 ..	8.2	49	.89	79	8	9–13000	11x8	10x6	10x6	B	FR	L

WORLD'S MODEL ENGINES (excluding Soviet Products)

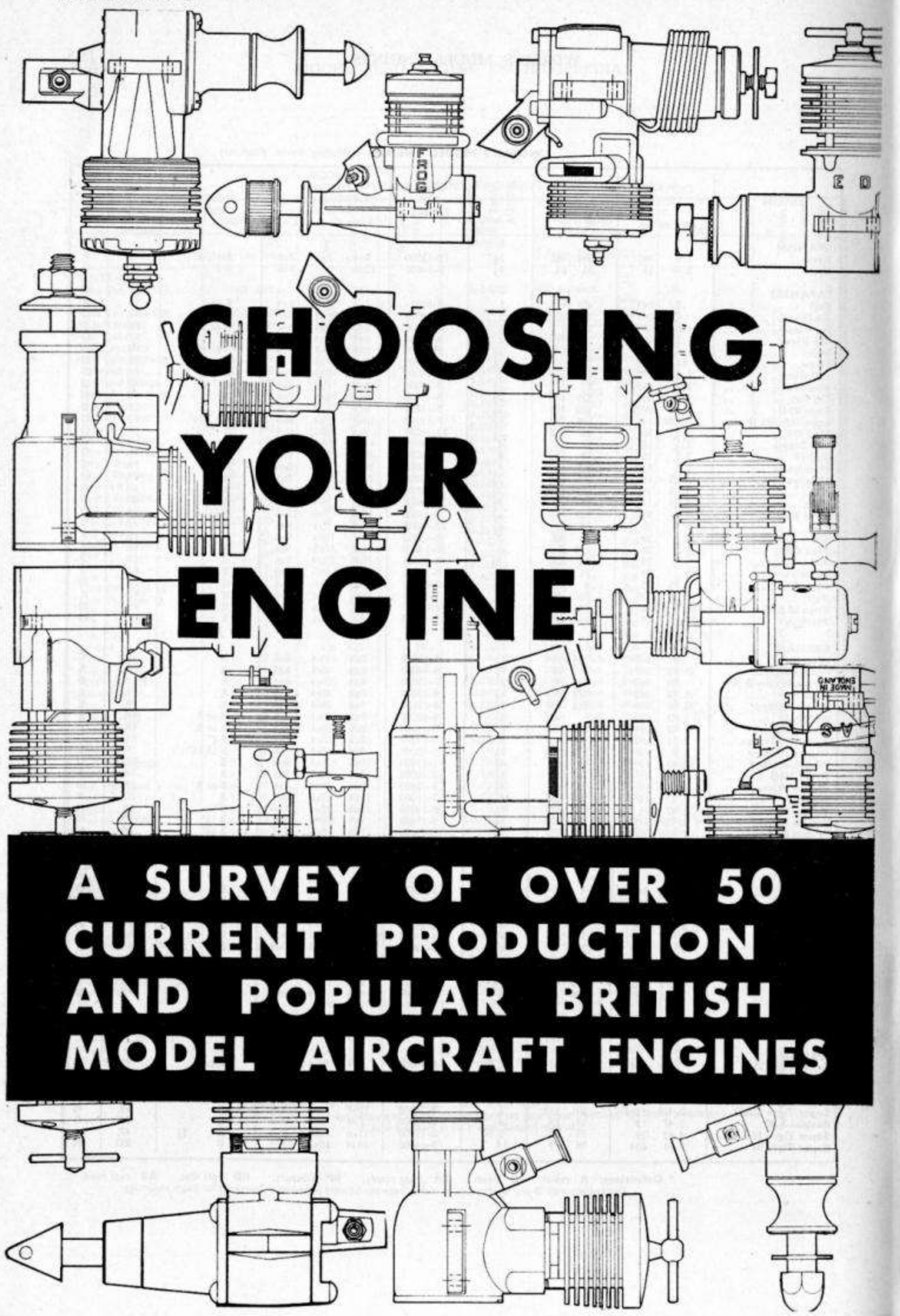
ENGINE	DISPLACEMENT cc.	DISPLACEMENT cu. ins.	Cylinder Bore	Cylinder Stroke	WEIGHT (ozs)	Useful RPM RANGE	RECOMMENDED PROPELLER Sport	Contest	C/L	MOUNTING*	* Induction	POWER GROUP
AMERICAN												
Cox Tee Dee 010 ...	.163	.010	.237	.226	½	16–34000	3½" supplied.			R	FR	B
Cox Tee Dee 020 ...	.327	.0199	.300	.282	.85	13–23000	Cox 3½" x 2¼" three Blade or Cox 4"x2½"			R	FR	B
K & B Infant ...	.327	.020	.281	.331	1	12–15000	5x2	5x2	—	R	FR	A
OK Cub	.6	.039	.39	.336	1¼	11–16000	5½x4	5½x4	5½x4	R	FR	B
Baby Spitfire ...	.72	.045	.375	.406	1	10–13000	6x4	6x3	6x3	R	FR	C
OK Cub (glow & D)	.8	.049	.420	.36	1¼	9–14000	6x4	5x4	5x4	B ¾" or R	FR	C
Atwood	.8	.049	.420	.36	1½	9–14000	6x4	5½x3	5½x3	R	FR	C
Holland Wasp ...	.8	.049	.420	.36	1½	9–14000	6x4	5½x3	5½3x	R	FR	C
Holland Hornet ...	.795	.048	.422	.35	2	8–15000	6x4	5½x3	5½x3	R	FR	C
K & B Torpedo ...	.8	.049	.396	.406	1¼	10–15000	5x3	5x3	5x3	R	FR	C
McCoy Diesel ...	.8	.049	.405	.386	1¼	7–15000	7x6	6x4	6x6	R	FR	C
Wen-Mac	.8	.049	.42	.37	1¼	9–16000	6x3	5x3	5x3	B ¾" or R	FR	C
Cox Babe Bee ...	.81	.0494	.406	.382	1½	12–16000	6x4	6x4	6x4	R	RR	C
Cox Golden Bee ...	.81	.0494	.406	.382	1¼	10–17000	6x4	6x3	6x4	R	RR	C
Royal Baby ...	.8	.049	.396	.406	1½	10–15000	6x4	6x3	6x3	B	FR	C
Allyn Skyfury ...	.8	.049	.420	.36	1½	9–14000	6x4	5½x3	5½x3	R	FR	C
Atwood	.83	.051	.420	.37	1¼	9–14000	6x4	5½x3	5½x3	R	FR	C
Royal Spitfire ...	1.06	.065	.44	.420	1¾	8–12000	7x4	6x4	6x6	B	FR	D
OK Cub	1.21	.074	.479	.415	1¾	10–13000	7x4	6x4	6x4	B ¾" or R	FR	D
OK Cub (Diesel) ...	1.23	.075	.48	.415	2	7–12000	8x4	7x3	7x6	B ¾" or R	FR	E
OK Cub	1.6	.09	.51	.5	2	8–13000	8x4	7x3	7x6	B 1" or R	FR	D
McCoy Diesel ...	1.61	.09	.5	.5	2.6	7–12000	9x5	8x4	8x6	B 5/16"	FR	E
Fox 09	1.639	.099	.530	.453	3	8–16000	7x4	6x4	6x6	R or B	SP	D
Fox 15	2.415	.147	.593	.537	4	9–16000	9x4	8x4	8x6	B 1"	FR	G
K & B 15R ...	2.485	.1516	.599	.537	4.9	10–18000	9x3	8x4	8x6	B 1⅛"	FR	G
Cox Olympic 15 ...	2.423	.1478	.585	.55	4	10–19000	9x4	8x3	8x6	B 1"	RR	G
Cox Tee Dee 15 ...	2.449	.1494	.585	.556	4	10–18000	9x4	8x4	8x6	B 15/16	FR	G
OK Cub	2.45	.14	.6	.530	2.75	9–15000	9x3	8x3½	7x6	B 1⅛" or R	FR	F
K & B Torpedo ...	2.43	.15	.595	.535	3.75	10–14000	9x3	8x3½	7x6	B 1" or R	FR	G
OK Cub	3.25	.19	.655	.59	3	11–13000	9x6	8x4	8x6	B 1⅛" or R	FR	G
Arden	3.25	.19	.635	.625	4.16	8–12000	10x4	8x4	8x6	R	FR	H
Veco	3.25	.19	.635	.63	6	11–14000	11x6	9x6	9x8	B	FR	J
K & B Torpedo ...	3.25	.19	.64	.62	5½	11–14000	11x6	9x6	9x8	B 1⅛"	FR	J
McCoy	3.25	.19	.625	.630	4	10–14000	11x5	8x4	8x6	B	FR	H
Cameron	3.25	.19	.64	.62	5	6–14000	10x6	9x4	9x6	B 1⅛"	FR	H
Fox	3.25	.19	.65	.6	4¾	11–12000	10x6	9x6	10x6	B 1⅛"	FR	J
K & B	3.75	.23	.68	.62	5.6	11–14000	11x6	9x6	9x8	B 1 3/16"	FR	J
Ohlsson	3.75	.23	.687	.625	5	6–12000	11x5	9x4	8x8	B 1¼"	FR	H
Fox	4.09	.25	.738	.6	4¾	8–14000	11x6	10x4	9x6	B	FR	J
Ohlsson	4.9	.29	.76	.660	5	7–11000	11x5	10x4	9x6	B	FR	H
K & B Torpedo ...	4.9	.29	.725	.724	7½	8–13000	11x5	10x4	10x6	B 1¼"	FR	J
McCoy	4.9	.29	.75	.672	6	11–14000	10x8	9x6	9x8	B	RD	J
OK Hothead ...	4.9	.29	.76	.660	7½	6–10000	11x6	10x9	9x8	B	FR	H
Forster	4.9	.29	.75	.67	6¼	8–13000	12x6	10x6	10x6	B 1 5/16"	FR	J
Veco	4.9	.29	.725	.724	7½	11–14000	10x8	9x6	10x6	B 1¼"	FR	J
Fox 29 & 29X ...	4.9	.29	.738	.7	5¾	10–14000	12x6	10x6	10x6	B 1 3/16"	FR	J
Dooling	4.9	.29	.8	.594	6¼	10–16000	10x6	10x4	7x9	B	RD	K
De Long	4.9	.30	.748	.680	8¼	8–12000	11x6	10x5	9x6	B	RD	J
Forster	5	.305	.760	.672	6¼	8–13000	12x6	10x6	10x6	B 1 5/16"	FR	J
Ohlsson	5.4	.33	.687	.687	5	9–14000	12x6	10x6	10x6	B 1¼"	FR	J
Fox	5.75	.35	.8	.7	5¾	10–14000	12x6	10x6	10x6	B 1 3/16"	FR	K
Veco	5.75	.35	.78	.725	6¼	10–14000	12x6	10x6	10x6	B 1¼"	FR	K
K & B	5.75	.35	.79	.72	7½	11–15000	11x6	10x6	10x6	B 1¼"	FR	K
Veco 35C ...	5.743	.3502	.785	.725	7¾	9–15000	10x6	9x6	10x6	B 1 5/16"	FR	K
K & B 35 (61) ...	5.78	.3574	.790	.719	8¼	9–17000	11x4	9x6	10x6	B 1 5/16"	FR	K
McCoy 35 ...	5.362	.327	.775	.743	7½	8–15000	12x4	10x4	10x6	B 1⅜"	FR	K
Fox 40	6.495	.3961	.800	.788	7¾	8–16000	11x4	9x6	10x6	B 1¼"	FR	L
Atwood Triumph ...	8	.49	.89 •	.79	8½	9–13000	14x6	12x6	11x6	B	FR	L
Pal Twin	9	.55	2x.72	.6	10	6–10000	14x6	12x6	11x8	B	SP	K
Fox	9.75	.59	.937	.860	9½	10–16000	14x6	12x6	11x8	B	RD†	N
Ohlsson	9.9	.604	.937	.875	9	6–12000	14x6	12x6	11x8	B	SP†	M
McCoy	10.0	.607	.940	.875	14	12–16000	12x8	11x6	11x6	B	RD	N
Dooling 61 ...	10.0	.607	1.015	.75	14	11–18000	12x8	10x6	9x11	B	RD	N
Super Cyclone ...	9.9	.604	.906	.937	9½	6–11000	14x6	12x6	11x8	B	FR	M
Anderson Spitfire	10.6	.64	.937	.937	12	8–14000	14x6	12x8	12x8	B	FR	N
Forster 99 ...	16.4	.997	1.0621	.125	14	3–8000	16x8	14x6	14x8	B	SP	O
OK Twin	20	1.208	2x9	.95	22	6–8000	16x8	14x6	14x8	B	RR	O
Avion Mercury ...	26.25	1.609	1.25	1.312	20	4–6000	16x8	14x6	14x8	B	SP	O

* Definitions: **R** radial. **B** beam. **FR** front rotary. **SP** sideport. **RD** rear disc. **RR** rear reed.
Dimensions with **B** are for distance in inches between bearers (crankcase clearance) for Beam mounting.
† Also available in FR (Front Rotary) versions.

WORLD'S MODEL ENGINES (excluding Soviet Products)

ENGINE	DISPLACEMENT cc.	DISPLACEMENT cu. ins.	Cylinder Bore	Cylinder Stroke	WEIGHT (ozs)	Useful RPM RANGE	RECOMMENDED PROPELLER Sport	Contest	C/L	MOUNTING*	* Induction	POWER GROUP
SPANISH												
Byra	1.5	.091	.494	.455	3¾	10–13000	9x4	7x4	7x6	B 15/16"	RD	F
Byra	2.47	.15	.56	64	4½	8–14000	10x6	9x4	8x6	B 1"	RD	G
JAPANESE												
Fuji	.8	.049	.402	.4	1	8–9000	6x4	5½x3	6x4	B or R	FR	B
Ko Diesel	.8	.049	.420	.36	2	9–14000	6x4	5½x3	6x5	R	FR	C
Ko Diesel	1.6	.099	.51	.5	3	9–14000	8x4	6x4	6x6	R or B	FR	E
OS Diesel	1.5	.095	.5	.49	3	8–12000	8x4	6x4	6x6	B	RD	D
Fuji	1.6	.099	.5	.5	2¼	9–12000	8x4	7x3	7x6	R or B	FR	D
Hope	.16	.099	.5	.5	1¾	8–14000	8x4	7x3	7x6	B	FR	D
KO Glow	1.6	.099	.49	.51	2	8–14000	8x4	7x3	7x6	B	FR	D
Mamiya	1.6	.099	.5	.5	1¾	8–14000	8x4	7x3	7x6	B	FR	D
OS Pet	1.61	.098	.529	.448	2¾	8–15000	8x4	8x3	7x6	B	FR	E
Enya 09-II	1.60	:0978	.500	.498	3¼	7–15000	8x4	7x4	7x6	B	FR	E
Enya 15D Mk II ...	2.448	.149	.589	.547	6¼	7–17000	10x6	8x4	8x6	B 1 3/16"	FR	G
Max OS	2.5	.15	.597	.540	3¼	10–14000	9x4	8x3½	8x6	B 15/16"	FR	G
KO Diesel	2.5	.15	.59	.55	4	8–14000	10x6	9x6	9x6	B	FR	G
Fuji 15	2.5	.15	.59	.55	4	8–14000	10x5	8x4	8x6	B	FR	G
Mamiya 15	2.5	.15	.577	.56	4	8–14000	10x5	8x4	8x6	B	FR	G
Mamiya	3.25	.19	.625	.630	3½	9–15000	10x5	9x4	9x6	B	FR	G
KO	3.25	.19	.625	.630	3¼	9–15000	10x5	9x4	9x6	B	FR	G
Enya	3.25	.19	.63	.63	4¼	12–16000	10x5	9x4	9x6	B or R	FR	G
Enya 29-3B	4.94	.3012	.735	.710	6¼	9–16000	10x6	9x6	10x6	B 15/16"	FR	J
Fuji	5	.29	.75	.67	6¼	12–14000	10x6	9x5	9x6	B	FR	J
OS	5	.29	.74	.68	7	8–13000	11x6	10x4	9x6	B	FR	J
Max OS	5	.29	.738	.7	6	10–14000	11x6	10x5	9x6	B	FR	J
KO	5	.29	.74	.68	5½	8–15000	10x6	10x4	9x6	B	FR	J
Mamiya	5	.29	.75	.67	6	8–15000	10x6	9x5	9x6	B	RD	J
Max OS	5.75	.35	.8	.7	7	10–14000	12x6	10x6	10x6	B	FR	K
Fuji	5.75	.35	.75	.75	6	13–15000	12x6	10x6	10x6	B	FR	K
Enya 60 & 63 ...	10	.60	.94	.875	14	10–16000	4x6	12x6	12x8	B	FR	N
Mamiya	10	.60	.94	.875	16	10–16000	4x6	12x6	9x12	B	RD	N
GERMAN												
Wilo Boy	.71	.043	.394	.354	1¼	6–12000	8x4	7x4	6x6	R	FR	C
Webra Piccolo ...	.78	.049	.41	.35	1¼	8–14000	8x4	7x4	6x6	R	FR	C
Taifun Hobby & RS	.99	.060	.424	.43	2	7–13000	8x4	7x4	6x4	B	FR, RR	D
Wilo Fox	1.36	.082	.472	.472	2¼	8–13000	9x6	8x4	7x6	R	FR	E
Taifun Record ...	1.49	.090	.512	.441	3¼	8–12000	9x6	8x4	7x6	B	FR	E
BWM 150	1.49	.090	.5	.461	3¼	8–12000	9x6	8x4	7x6	B or R	FR	E
Webra Record ...	1.48	.099	.512	.453	3	8–14000	9x6	8x4	8x6	B or R	FR	F
Taifun Hurrikan ...	1.51	.092	.507	.457	3.8	10–14000	8x5	7x4	7x6	B ¾"	RR	F
Webra 1.7 glow ...	1.7	.10	.513	.515	2¼	9–13000	8x4	7x4	7x6	B 15/16"	FR	E
Webra Mach. I ...	2.47	.15	.61	.51	4¼	11–14000	10x6	8x4	8x8	B	RD	G
BWM 250D	2.47	.15	.56	.64	5¼	8–10000	11x4	10x4	9x6	B	FR	F
Webra Winner ...	2.46	.15	.56	.64	3¼	8–12000	10x6	9x4	9x6	B or R	FR	G
Jaguar	2.47	15	.59	.55	5	7–10000	11x4	9x6	9x6	B	FR	F
Taifun Tornado ...	2.47	.15	.59	.55	5½	8–14000	10x6	8x4	9x6	B	FR	G
Webra Komet ...	2.454	.175	.551	.627	5¼	9–15000	9x4	8x6	8x6	B 1¼"	FR	G
Webra Bully ...	3.416	.208	.650	.627	5¼	8–13000	9x6	9x6	9x6	B 1¼"	FR	G
Taifun Bison ...	3.629	.2214	.631	.703	6	8–15000	9x6	9x4	8x6	B	FR	H
Taifun 3.5	3.44	.210	.65	.61	6¼	9–13000	11x6	10x6	9x8	B	FR	H
FRENCH												
Micron	.8	04	4	.4	2	5–10000	8x4	7x4	6x6	B	SP	D
Meteore	2.47	.15	.59	.551	4	9–14000	10x6	9x4	9x6	B 1"	FR	G
REA	4.79	29	.74	.66	6	8.12000	10x6	10x4	9x6	B	FR	H
Micron 28	5	305	.74	.68	6	9–14000	10x6	10x4	9x6	B	FR	H
Micron 29	5	.305	.74	.68	7½	11–14000	10x5	9x5	9x6	B	RD	J
Micron 10	10	.60	.95	86	8	9–13000	11x6	10x6	10x8	B	FR	K
HUNGARIAN												
Alag X-4	1.517	.0925	.512	.044	2½	8–16000	8x4	7x4	7x6	B ¾"	FR	E
Moki S-2	2.465	.1503	.590	.550	5	10–19000	9x4	8x4	8x6	B 1⅛"	FR	G
DUTCH												
Typhoon R250 ...	2.47	15	.59	.55	4.75	8–13000	11x6	8x4	9x6	B 1"	FR	G
ITALIAN												
Super Tigre 25 ...	.95	.059	.43	.40	2.5	8–14000	8x4	6x4	6x6	B or R	FR	D
Barbini B.38... ...	.99	.061	.4	.48	2	10–13000	9x4	8x4	6x6	B ¾	FR	D
Super Tigre 26 ...	1.49	.091	.52	.43	3	8–14000	9x6	8x4	8x6	B or R	FR	E
Super Tigre 20 & 23	2.46	.15	.59	.55	4	8–14000	9x5	8x3½	8x6	B 1"	FR	G
Super Tigre G20D...	2.482	.1514	.591	.552	6	8–16000	10x4	8x4	8x6	B	FR	G
Barbini B.40... ...	2.47	15	.57	59	4	8–12000	9x4	8x3½	8x6	B	FR	G
Super Tigre 19 & 21	4.82	.29	.75	.67	8.5	8–12000	11x6	10x4	9x6	B 1¼	FR	J
Super Tigre 24 ...	9.81	.604	98	.79	14	9–16000	14x4	10x6	11x8	B	RD	N

* Definitions: **R** radial. **B** beam. **FR** front rotary. **SP** sideport. **RD** rear disc. **RR** rear reed.
Dimensions with **B** are for distance in inches between bearers (crankcase clearance) for Beam mounting.



CHOOSING YOUR ENGINE

A SURVEY OF OVER 50 CURRENT PRODUCTION AND POPULAR BRITISH MODEL AIRCRAFT ENGINES

1965/6 PRODUCTION SUMMARY

A complete listing of British engines in current production with pertinent facts and prices. Basic price is for the benefit of overseas readers who are not liable to Purchase Tax. "D" means Diesel, "G" means Glowplug

Engine	Ignition	Displacement c.c.	Displacement cu. ins.	Cylinder Bore	Cylinder Stroke	Weight (oz)	May 1965 basic price	British price with p.t.
A-M								
10	D	1·00	·061	·462	·430	3·0	51/8d	60/10d
10 R/C	D	1.00	·061	·462	·430	3·25	62/8d	74/0d
15	D	1·48	·09	·517	·430	3·00	53/4d	62/10d
15 R/C	D	1·48	·09	·517	·430	3·25	64/2d	75/9d
25	D	2·4	·147	·57	·562	4·00	60/0d	70/9d
35	D	3·42	·209	·687	·562	4·5	61/8d	72/9d
E.D.								
Cadet	D	·98	·061	·437	·400	3·9	24/6d	29/6d
Hawk	D	1·48	·093	·513	·452	3·3	24/6d	29/6d
2·46 Racer	D	2·46	·15	·59	·55	5·4	70/5d	85/0d
2·46 Racer (tuned)	D	2·46	·15	·59	·55	5·4	89/1d	107/6d
2·46 Racer R/C	D	2·46	·15	·59	·55	6·00	89/1d	107/6d.
ETA								
15 Mk III	D	2·48	·15	·558	·620	6·25	125/0d	148/6d
29 Mk VIc	G	4·95	·30	·750	·672	6·5	129/6d	153/10d
FROG								
80 Mk III	D	·80	·049	·400	·392	1·9	46/3d	54/6d
100 Mk III	D	1·02	·062	·416	·460	3·00	48/3d	57/0d
150 Mk III	D	1·49	·091	·500	·460	3·25	50/4d	59/6d
249 BB	D	2·49	·151	·580	·568	6·0	67/4d	79/6d
349 BB	D	3·49	·208	·666	·600	6·7	74/4d	87/6d
349 BB R/C	D	3·49	·208	·666	·600	6·7	88/8d	105/0d
MAROWN								
M.E. Heron	D	·97	·06	·424	·420	2·4	50/10d	60/6d
M.E. Snipe	D	1·48	·09	·505	·455	4·1	56/3d	67/0d
M.E. Snipe R/C	D	1·48	·09	·505	·455	4·25	64/4d	76/6d
MERCO								
·29	G	4·75	·29	·734	·703	7·5	101/3d	119/6d
·29 R/C	G	4·75	·29	·734	·703	8·25	129/0d	152/6d
·35	G	5·79	·35	·800	·703	7·5	101/3d	119/6d
·35 R/C	G	5·79	·35	·800	·703	8·25	129/0d	152/6d
·49	G	8·00	·49	·880	·805	12·5	169/0d	199/6d
·49 R/C	G	8·00	·49	·880	·805	13·00	200/0d	236/8d
·61 R/C	G	10·00	·61	·938	·875	13·00	217/6d	256/9d
P.A.W.								
1·49	D	1·48	·09	·494	·469	3·5	72/10d	86/0d
2·49 Mk III	D	2·49	·151	·592	·532	5·00	83/0d	98/0d
19 D Mk II	D	3·128	·1912	·642	·590	5·5	88/6d	104/6d
19–BR	D	3·128	·1912	·642	·590	6·0	106/9d	126/0d
TAPLIN								
Twin Mk II	D	8·00	·49	·705	·625	17·00	158/4d	190/0d
OLIVER								
Tiger Cub Mk II/S	D	1·46	·089	·470	·523	4·13	108/4d	130/0d
Tiger Cub Mk II(sports)	D	1·46	·089	·470	·523	4·13	125/0d	150/0d
Tiger Mk III/S	D	2·424	·148	·551	·620	5·5	115/10d	139/0d
Tiger Mk III (sports)	D	2·424	·148	·551	·620	5·5	132/6d	152/6d
Tiger Mk III R/C	D	2·424	·148	·551	·620	5·5	145/10d	175/0d
Tiger Major/S	D	3·47	·212	·620	·705	6·00	129/2d	155/0d
Tiger Major (sports)	D	3·47	·212	·620	·705	6·00	145/10d	175/0d
Tiger Major R/C	D	3·47	·212	·620	·705	6·00	160/5d	192/6d
D.C. Quickstart								
Dart	D	·55	·036	·35	·35	1·25	63/9d	75/0d
Bantam	G	·76	·047	·410	·352	1·5	38/3d	45/0d
Bantam de luxe	G	·76	·047	·410	·352	1·5	49/4d	58/0d
Merlin	D	·76	·047	·375	·420	1·75	50/7d	59/6d
Super Merlin	D	·76	·047	·375	·420	1·8	55/3d	65/0d
Spitfire	D	·97	·059	·425	·420	3·00	54/4d	71/0d
Sabre	D	1·49	·091	·525	·420	3·00	63/9d	75/0d

From the smallest . . .

This survey of current production and recent British engines is offered as a buying and identity guide

Made under licence from **Wen Mac** the **A.M. .049** glow engine is no longer in production. This was the first British engine to have a fully enclosed starter called "Rotomatic" in the U.S.A. Front rotary induction with a plain bearing shaft and integral air intake made this a very light-weight engine. For beam or radial mounting the lugs were slotted and not drilled as in the normal manner. Space bearers $\frac{3}{4}$ in. apart. Sadly the **A.S. 55** also went out of production. This was a pity as it was a really nice easy starting and smooth running little engine. Bearers are spaced $\frac{11}{16}$ in. apart for beam mounting. A lively little engine is the red headed **D.C. Quickstart Dart** .5 c.c. diesel. It was one of the first "performance" front rotary induction miniature engines with the air intake cast into the crankcase and now has a coil spring and cam rewind starter with a transparent fuel tank as standard items. Beam mount only with bearers spaced $\frac{11}{16}$ in. apart. The **D.C. Quickstart Merlin** comes in both standard and de luxe versions (called the **Super Merlin**) the extras including a rear bolt-on transparent fuel tank and a red spinner nut and cylinder, front rotary shaft induction via integral air intake. They are easy starters, cheap to run and can be fitted by radial or beam mounting with bearers spaced $\frac{15}{16}$ in. apart. The **D.C. Quickstart Bantam** is also produced as a de luxe version, this including radial mounting rear tank as an extra. It has front rotary shaft induction with the air intake cast into the crankcase. Beam mounting can be used with bearers spaced $\frac{11}{16}$ in. apart. Out of production but still widely used, the **E.D. .46 Baby** is known for its docile handling and ability to run on practically any sort of fuel. Beam mount with the bearers spaced $\frac{3}{4}$ in. apart. Also no longer in production the **Frog 50** diesel was another with front rotary shaft induction and a turned metal tank. With a screw-in cylinder liner and head it proved very popular for small F/F models of up to 35 in. span. Beam mount only bearers spaced $\frac{5}{8}$ in. apart. The current production **Frog 80** incorporates a coil spring cam starter and can be used for all sport models. With a deeply finned steel cylinder, large but shallow exhaust stacks and built in nylon compression screw lock, it has much to commend it. Having front rotary shaft induction with integral air intake it is suited to 30-44 in. F/F or 30 in. R/C and 12-25 in. C/Liners. A glow plug version known as the **Frog .049** was once produced with an unfinned head and knurled spinner nut for pulley starting. Beam mount only on bearers spaced $\frac{11}{16}$ in. apart. The **Mills .75** must surely enjoy a hallowed place in many modellers' hearts for its ability to run for many years without needing any attention, easy starting and slogging power on large props. With side port induction that has the air intake screwed to the rear of the cylinder and a suspended clear plastic fuel tank it is easily distinguishable with its black crackle finish cranckcase and small diameter shaft housing. Beam mount only with the bearers spaced $\frac{5}{8}$ in. apart. Regrettably the Mills .75 is no longer produced. Also out of production the **Keil Kraft Cobra .049** was a rear reed valve induction glow plug engine with a neat rear mounted needle valve. Beam mount on bearers spaced $\frac{3}{4}$ in. apart. Alphabetically last of the small capacity group is the **Z.A. .92** a zippy little engine that starts and runs well. It went out of production only recently. Front rotary induction is arranged via an integral air intake that forms a square block in front of the cylinder. Beam mount only with bearers spaced $\frac{13}{16}$ in. apart. *All the engines in this group should use $\frac{1}{4}$ in.* x *$\frac{3}{8}$ in. hardwood bearers for beam mounting.*

17 Years of AEROMODELLER ENGINE ANALYSES

PEAK B.H.P. and R.P.M. FIGURES FOR 212 TESTS

1948		
E.D. Comp. Special	May	·109 @ 7,000
Frog 100	June	·058 @ 8,100
Mills Mk. II 1·3	July	·078 @ 7,250
AMCO Mk. 1·87	August	·046 @ 8,900
Eta '5' Diesel	September	·181 @ 6,250
Jagra Dyne 3	October	·103 @ 6,300
Kalper ·3 cc	November	·010 @ 10,250
Allbon 2·8	December	·144 @ 6,500
1949		
'K' Vulture 5 cc	January	·246 @ 8,900
2·8 cc Masco Buzzard	February	·108 @ 7,300
Nordec R.G. 10	March	·480 @ 11,200
E.D. Mk. II	April	·128 @ 7,800
Mills Mk. III	May	·167 @ 9,900
Weston 3·5 cc	June	·204 @ 7,500
Elfin 1·8 cc	July	·114 @ 12,100
Frog 160	August	·081 @ 10,850
Eta 29 Mk I	September	·370 @ 11,600
E.D. Mk I Bee	October	·062 @ 10,600
Yulon '30'	November	·310 @ 12,300
AMCO 3·5 cc PB	December	·260 @ 11,600
1950		
Allbon Arrow 1·5 Glow	January	·051 @ 11,500
E.D. Mk. IV 3·46 Hunter	March	·265 @ 13,300
Wildcat Mk. III 5 cc	April	·340 @ 10,000
Frog '500'	May	·381 @ 13,300
Elfin 1·49	June	·100 @ 13,700
Javelin 1·49	July	·099 @ 12,000
Fox 35	August	·623 @ 15,000
Forster G-29	August	·580 @ 16,200
Yulon 49	September	·820 @ 12,900
Frog 100 Mk. II	October	·071 @ 8,000
D.C. 350	November	·270 @ 11,000
1951		
Allbon Dart	January	·045 @ 13,300
E.P.C. Moth	February	·042 @ 9,700
Reeves, H.18	March	·104 @ 11,700
Super Tigre G.19	April	·485 @ 13,300
Super Tigre G.20	April	·240 @ 14,400
Sabre 2·50 cc	May	·225 @ 13,300
Elfin 2·49	June	·231 @ 12,300
Frog 250	July	·192 @ 10,700
E.D. 2·46 Racer	August	·260 @ 14,100
Frog 150	September	·121 @ 12,900
Atwood Wasp ·8 cc	October	·100 @ 15,400
REA 5 cc	November	·360 @ 10,900
Hot Top Super Hurricane	December	·279 @ 11,600
1952		
D.C. 350 (G)	January	·262 @ 11,100
D.C. 350 (D)	January	·281 @ 11,300
Frog 500 Glow	February	·420 @ 13,200
Metro 52	March	·225 @ 12,600
Typhoon 2·47	April	·241 @ 13,500
Mills P.75	May	·059 @ 11,350
E.D. ·46 Baby	June	·028 @ 11,000
Frog 50	August	·030 @ 12,300
Moteurs Micron 28	October	·225 @ 11,000
Super Tigre G. 20. S.	November	·260 @ 14,000
Oliver Tiger Mk. II	December	·305 @ 14,000
1953		
Allbon Javelin	January	·130 @ 11,000
E.D. 1·46	February	·150 @ 10,500
Typhoon 5 cc	March	·430 @ 13,000
Allbon Dart Mk. II	April	·042 @ 11,000
AMCO BB 3·5 cc I	May	·320 @ 13,000
Allbon Spitfire	June	·085 @ 11,000
O.K. Cub 2·5 cc	July	·195 @ 12,500
Super Tigre G.22	August	·095 @ 12,600
Typhoon R. 250	September	·290 @ 13,500
McCoy ·049	October	·080 @ 12,000
1954		
K & B Torpedo ·15	June	·142 @ 13,600
Oliver Tiger 1·5 Cub	July	·120 @ 12,500
Allbon Bambi	August	·007 @ 12,500
Webra 2·5 Mach 1	September	·218 @ 16,700
Webra 2·5 Winner	September	·162 @ 11,300
Allen Mercury 25	October	·181 @ 12,200
E.D. 2·46 Racer	November	·196 @ 14,650
Allbon Merlin	December	·058 @ 13,000
1955		
Elfin 1·49 BB	January	·158 @ 13,600
Webra 1·48	February	·133 @ 13,800
Miles 5 cc	March	·435 @ 13,500
Taifun 1 cc Hobby	April	·100 @ 13,400
Tornado 2·5 cc	April	·192 @ 14,000
Miles Special	July	·365 @ 13,000
Webra ·8 Piccolo	August	·058 @ 12,800
1 cc E.D. Bee Series 2	September	·068 @ 10,900
Jaguar 2·5 cc	October	·199 @ 12,750
Allen Mercury 35	November	·260 @ 11,400
Byra 2·5	December	·196 @ 12,000
1956		
Frog 249 BB	January	·206 @ 13,700
Frog 150	February	·108 @ 12,400
Allbon Sabre	February	·104 @ 13,300
Mamiya 15	March	·160 @ 12,800
Super Tigre G.20	March	·174 @ 13,400
J.B. Atom 1·5 cc	April	·090 @ 10,700
Carter 5 cc	May	·595 @ 18,300
E.D. 2·46 Racer (Buskell)	June	·271 @ 14,700
Frog 149 Vibramatic (D)	July	·122 @ 12,750
Frog 149 Vibramatic (G)	July	·078 @ 14,000
Allen Mercury 10	August	·113 @ 14,200
Schlosser 2·5 cc	September	·215 @ 14,000
Daru	September	·154 @ 11,000
OS MAX-15	October	·237 @ 14,650
Elfin 2·49 BR	November	·202 @ 13,200
Eta 29	December	·600 @ 17,200
1957		
Taifun Hurrikan 1·48	January	·154 @ 14,500
D.C. Manxman	February	·257 @ 10,700
Frog 80	March	·051 @ 11,000
Byra 1·5	April	·114 @ 12,000
Webra 1·7 cc	April	·090 @ 13,000
Barbini 1 cc	May	·104 @ 15,500
OS 29	June	·480 @ 14,800
D.C. Rapier	July	·213 @ 13,800
Barbini B. 40 T.N.	August	·189 @ 14,000
Spitfire Mk. II	September	·073 @ 12,400
Veco 19	October	·316 @ 15,000
Enya 15D	November	·252 @ 14,200
PAW Special	December	·249 @ 14,000
1958		
Alag X-3	January	·185 @ 12,700
Webra 2·5R Glo.	January	·202 @ 13,200
Frog 2·49 Modified BB	February	·253 @ 14,800
Taifun Hobby RS.	March	·071 @ 12,000
Fuji 29	April	·400 @ 12,400
Fox 29X	May	·465 @ 14,000
Fox 29R	May	·625 @ 17,500
E.D. 1·49 Fury	June	·132 @ 14,000
A.M. 15	July	·152 @ 14,000
O.S. Pet 1·6 cc	August	·133 @ 14,400
Super Tigre G-32	September	·097 @ 15,000
Frog 100 Mk. II	October	·103 @ 15,500
Kometa M.D. 5	November	·234 @ 13,000
1959		
Taifun Blizzard	January	·242 @ 13,000
Taipan 1·5	February	·110 @ 12,000
Alag X-4	March	·123 @ 13,000
Rivers Silver Streak	April	·277 @ 15,800
Taplin Twin	May	·290 @ 9,000
Glo-Chief 29	June	·495 @ 14,600
Fox 15	July	·218 @ 13,500
Webra Komet	August	·235 @ 13,000
Webra Bully	August	·200 @ 9,500
E.T.A. ·19	September	·300 @ 16,800
Enya 29-3b	October	·590 @ 14,000
Cox 15 Olympic	November	·287 @ 16,500
Frog 3·49	December	·303 @ 12,200
1960		
A.M. ·049	January	·052 @ 14,000
D.C. Bantam	January	·053 @ 15,000
Frog ·049	January	·046 @ 12,000
Merco 35	February	·550 @ 13,400
P.A.W. 1·49	March	·176 @ 17,000
A-S 55	April	·052 @ 12,000
McCoy "35"	May	·455 @ 12,000
E.D. Super Fury	June	·162 @ 14,000
Rivers Silver Arrow	July	·382 @ 15,500
M.E. Heron	August	·072 @ 9,500
D.C. Tornado	September	·397 @ 12,200
Cobra ·049	October	·052 @ 15,000
Enya 09	November	·115 @ 12,800
O.S. Pet 09	November	·119 @ 11,000
ETA 15 Mk I	December	·345 @ 16,600
1961		
Fox ·09	January	·084 @ 14,000
Rivers Silver Streak 11	January	·296 @ 16,000
P.A.W. 2·49 111	February	·318 @ 15,000
Enya 15D Mk II	March	·332 @ 15,000
Frog Venom	April	·075 @ 10,000
Frog Viper	April	·161 @ 14,800
Cox Babe Bee	May	·056 @ 13,000
Cox Golden Bee	May	·063 @ 14,000
Taifun Bison	June	·304 @ 13,300
Veco 19 R/C	June	·395 @ 14,000
Super Tigre G20D	July	·322 @ 15,000
Glo-Chief 19	July	·310 @ 13,800
Oliver Tiger III	August	·330 @ 15,100
Oliver Tiger Cub	August	·170 @ 14,000
P.A.W. 19D	September	·347 @ 15,000
Cox Tee Dee 010	October	·028 @ 32,000
Frog 80 Mk II	November	·057 @ 11,000
Fox 40	December	·596 @ 14,000
1962		
Cox Tee Dee ·15	January	·455 @ 18,000
Veco 35C	February	·538 @ 15,000
Moki S-2	March	·320 @ 18,000
K. and B. 15R	April	·480 @ 19,250
Taipan 2·5B	May	·245 @ 14,000
K. and B. 35 Series 61	June	·560 @ 14,000
Cox Dee Tee ·020	July	·031 @ 20,500
D.C. Bantam De Luxe	August	·053 @ 14,500
Marown Snipe 1·5	September	·138 @ 14,000
Merco 49 R/C	October	·720 @ 12,200
Cox Tee Dee ·049	November	·105 @ 22,000
Taifun Zyklon 2·5	December	·210 @ 12,500
1963		
O.S. ·49 R/C	January	·550 @ 12,200
Enya ·45 R/C	January	·550 @ 12,400
Cox Special ·15	February	·460 @ 18,000
Moki M-3	March	·530 @ 13,800
M.V.V.S. -1D	March	·132 @ 15,400
Rossi ·60	April	no B.H.P. reading taken
Taplin Twin Mk. II	April	·363 @ 9,500
Cox T.D. ·09	May	·235 @ 19,000
Cox Medallion ·09	May	·162 @ 16,500
E.D. Cadet	June	·028 @ 6,400
Rythm 2·5	July	·250 @ 14,400
E.T.A. ·15 Mk II	August	·350 @ 16,400
Webra Piccolo	September	·062 @ 13,500
Webra Record R/C	September	·134 @ 13,000
Taifun Orkan	October	·328 @ 16,400
Fox ·40 B.B.	November	·760 @ 15,600
Aero 35	December	·400 @ 12,800
1964		
Z.A. ·92	January	·082 @ 11,800
Taipan 1·5 cc	February	·101 @ 11,000
Jena 2 cc	March	·283 @ 13,800
Oliver Tiger Major	April	·386 @ 13,200
Moki S-4	May	·600 @ 17,800
O.S. Max ·19 R/C	June	·254 @ 13,300
Fox ·15 R/C	July	·220 @ 13,000
McCoy ·35 R/C	August	·438 @ 12,700
Enya ·09 R/C	September	·118 @ 12,750
Merco ·61 R/C	November	·860 @ 11,800
K & B ·19 R/C	December	·317 @ 13,400

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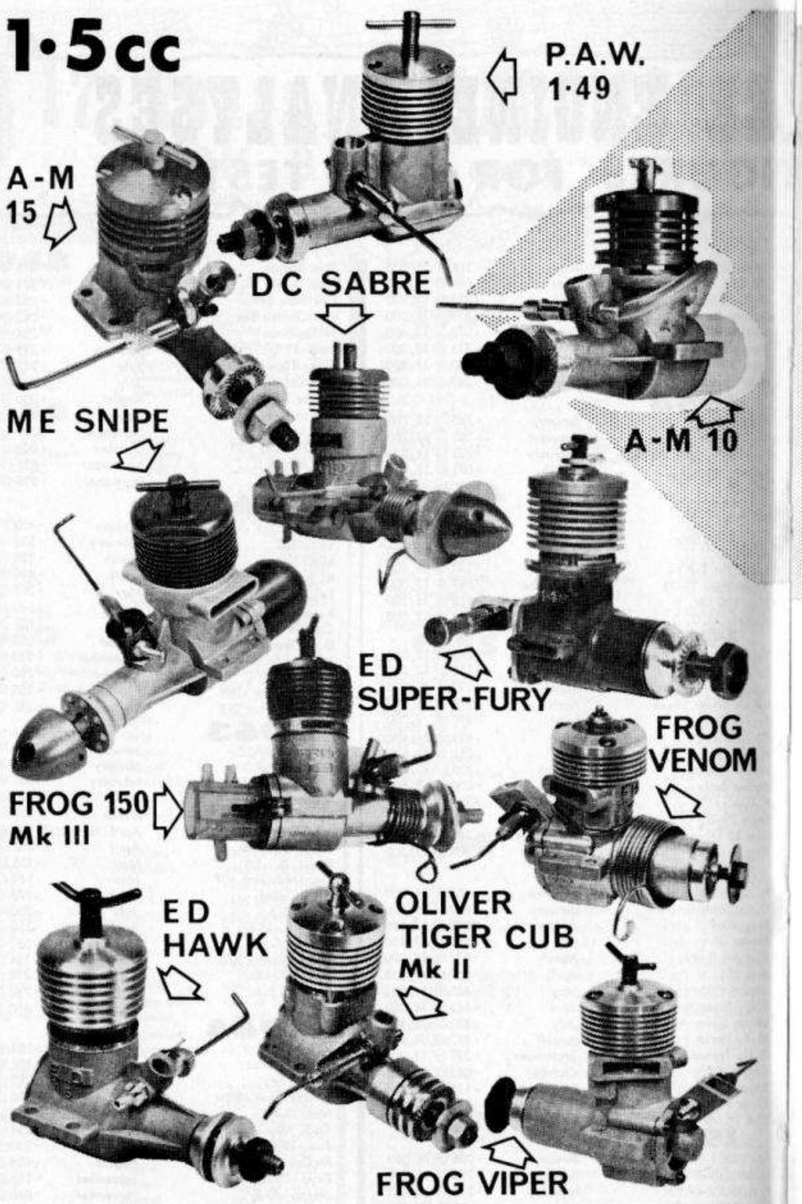

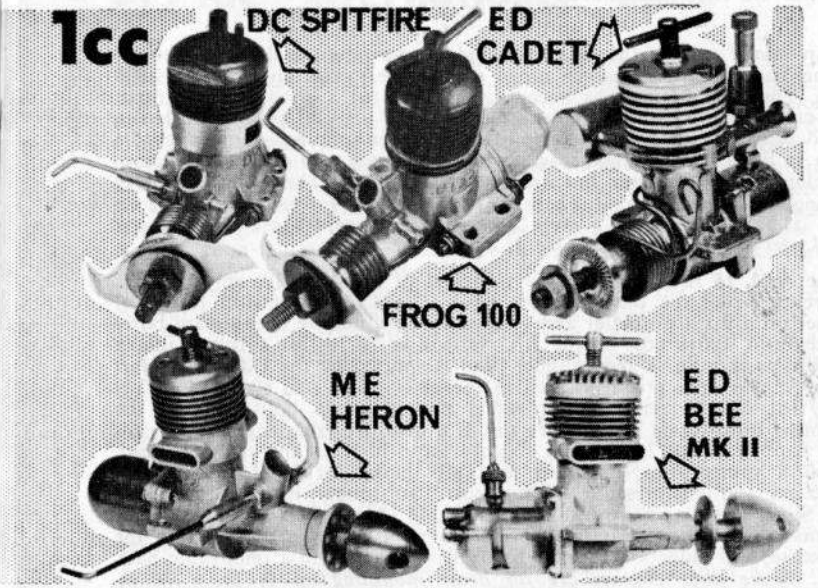

. . . the popular 1 cc to 1.5 cc

The **A.M. 10** is a universal purpose engine and was exceptional for its specific power output when introduced in 1956. Green anodising on the cylinder head distinguishes it from the blue headed A.M. 15, each sharing the same front rotary shaft induction via an integral cast intake in crankcase, also the same rear bolt-on nylon tank. A radio control version with a simple rotating barrel throttle is available at 12/5d. above the basic cost and with slight increase in weight. The A.M. 10 will slog at 6,000 or equally happily at 14,000 r.p.m. Developed over many years the **D.C. Quickstart Spitfire** is a very docile sports and beginners engine. Featuring a coil spring and cam recoil starter, with a blue cylinder head (it was once red) and transparent fuel tank bolted onto the mounting lugs it is robust and prang proof. A screw-in cylinder head retains the otherwise loose fit cylinder and the needle valve is swept back for greater propeller clearance. Tank mounting bolts can be used if a radial mount is preferred. Although no longer in production the **E.D. Bee** is still a favourite and there are thousands to whom this was a first engine. Featuring rear disc induction with the air intake running through the tank. **E.D. Cadet** was introduced as a replacement incorporating a coil spring recoil starter and silencer as standard items. The back plate mounted tank is turned from alloy and the rest of the engine is basically similar to the E.D. Bee with modification to side-port induction and solid cylinder head. Starting, with silencer proved to be less predictable. Mounting details are as for the Bee. A most forgiving engine for beginners, the **Frog 100 Mk. III** is recognisable by its red cylinder head and spring cam starter. It has front rotary shaft induction

(continued)

1 c.c. to 1.5 c.c. *(continued)*

via an intake cast on the crankcase and a transparent fuel tank is bolted to the backplate. The power range extends very usefully over 6-14,000 r.p.m. The **M.E. Heron** has become renowned for a good power output. The crankshaft is exceptionally strong and is unusual in that it has a honed bearing, making for longer life and better running. A raked needle valve is fitted to the intake cast on the crankcase and the Heron comes complete with a spinner nut and metal fuel tank. This is a very easy starting and reliable engine. *These* 1 *c.c. engines are for beam mounting on $\frac{1}{4}$ in. x $\frac{3}{8}$ in. hardwood bearers spaced $\frac{7}{8}$ in. except the Frog* 100 *and Quickstart Spitfire which require $\frac{15}{16}$ in.*

The **A.M. 15** is a bored out development of the A.M. 10, only external difference being the blue cylinder head. The 15 R/C version is a real slogger on a 9 x 4 propeller and vibration is low despite over-square bore/stroke ratio. Extra cost of the R/C version is 12/7d. for a rotary intake choke type throttle. A red cylinder head is sported by the **D.C. Quickstart Sabre** (it was once green) as well as the coil spring cam starter and fuel tank that can be removed for radial mounting via bolts through the mounting lugs. Again over square this is a good slogging engine that runs most happily at 10-12,000 r.p.m. With a loose fit liner and screw-in head, it has a spinner nut, integrally cast air intake for front rotary shaft induction and compression finding post. No longer in production the **E.D. Super Fury** is a very fast engine suitable for all sport or contest work on small propellers. Shaft is supported in twin ballraces pressed into the crankcase, and induction by rear rotary disc attached to a removable back plate. *These three engines are for beam mounting on $\frac{5}{16}$ in.* x *$\frac{7}{16}$ in. bearers spaced $\frac{7}{8}$ in. apart.*

Originally a Webra engine with front rotary shaft induction via cast intake and screw-in cylinder liner the **E.D. Hawk** runs most happily at 8-10,000 r.p.m. Easy starting and robust construction made this a good first engine. Both of 1.5 c.c. capacity the **Frog Venom and Viper** use the same major parts except for the alterations required to make the Venom for Glow Plug ignition and the Viper as a diesel motor. No longer in production they are both unusual with rear drum induction, a twin ballrace supported shaft and rearward slanting exhaust ports cast into the crankcase unit. The Venom was equipped with a spring starter. Another unusual recognition feature was the use of deep cooling fins on the bottom of the crankcase. Developed over many years the **Frog 150 Mk. III** has a bright blue cylinder head and front rotary shaft induction via an intake cast into the crankcase. A coil spring and cam starter is fitted together with a transparent plastic bolt-on fuel tank. Essentially a sport engine we have also seen many perform creditably in $\frac{1}{2}$A T/R and speed. *These four engines are beam mounted on hardwood bearers $\frac{5}{16}$ in.* x *$\frac{7}{16}$ in. spaced $\frac{15}{16}$ in. apart.*

A very "hot" 1.5 c.c. engine is the **P.A.W. 1.49** distinguished by its tall upright air intake situated close to the cylinder head. Front rotary induction and plain bearing shaft of rugged proportions make this engine suitable for all types of flying and contest work, where high revs are desired, space mounts $\frac{7}{8}$ in. apart. Last but by no means least of the 1.5 c.c. engines comes the **Oliver Tiger Cub Mk. II** with a twin ballrace supported shaft and front rotary induction, racing porting and performance to match. This engine is the standard for $\frac{1}{2}$A T/R flying. Performance outclasses some 2.5 c.c. engines. Long life and low fuel consumption make this an outstanding product for which one must pay extra. Beam mount on $\frac{3}{8}$ in. x $\frac{1}{2}$ in. hardwood bearers spaced 1 in. apart.

. . . the 'international' class

In production for many years and still firm favourites the **A.M. 25** and **A.M. 35** are easy to start, reliable moderate power output engines for general purpose use. They have also achieved many contest successes, especially for open power duration models. The A.M. 25 with black cylinder head was the first to be produced and the A.M. 35 came as a bored development recognisable by its red head. Both are front rotary shaft induction with the air intake cast into the crankcase, and screw-in back plates. Useful r.p.m. range is 10-14,000. Both are for beam mounting on at least $\frac{1}{2}$ in. x $\frac{3}{8}$ in. hardwood bearers spaced 1 in. apart. Now back in production the **E.D. 2.46 Racer** is available in three versions. Standard model is rear rotary disc induction with the air intake cast into the back plate assembly. Shaft is supported in twin ballraces and a hollow bolt is used to retain the propeller. The radio control version is identical with addition of a butterfly throttle. A tuned version with a higher power output is also available. Beam mount on hardwood bearers $\frac{1}{2}$ in. x $\frac{3}{8}$ in. spaced $1\frac{1}{8}$ in. apart. Designed for F.A.I. team racing the **Eta 15 Mk. III** has a twin ballrace supported shaft, rear disc induction and special porting for speed and fuel economy. This is a good starter for the novice or expert and has a specially cast cylinder. Beam mount on $\frac{3}{8}$ in. x $\frac{1}{2}$ in. hardwood bearers (preferably faced with $\frac{1}{8}$ in. steel) spaced $1\frac{1}{8}$ in. apart, or alternatively a cast metal pan. No longer in production the **Eta 19 Mk. II** is a lightweight racing glow motor with rear disc induction and a twin ballrace supported shaft. An unusual feature of this engine was the use of an integrally cast rear air intake with the front housing removable. Peak power is realised at high r.p.m. with small propellers. Beam mount on $\frac{3}{8}$ in. x $\frac{1}{2}$ in. hardwood bearers spaced at $1\frac{3}{16}$ in. Recognisable by its square front rotary shaft intake and red cylinder with head fins is the **Frog 2.49 BB.** A pace setter in its day, this engine is suited to combat or similar hard work. The twin ballrace supported shaft is set up very free so a plastic seal is used around the front housing to keep out foreign matter. Beam mount onto hard-wood bearers $\frac{1}{2}$ in. x $\frac{3}{8}$ in. spaced $1\frac{1}{16}$ in. apart Produced in a radio and standard version the **Frog 3.49 BB** is a rear drum induction, ballrace supported shaft motor with ability to slog at heavy loads without complaint. An unusual feature of this engine is the side stack exhaust port which was one of the first designed to accept a silencer. Firmly beam mount to absorb vibration on hardwood bearers $\frac{1}{2}$ in. x $\frac{3}{8}$ in. spaced $1\frac{1}{8}$ in. apart. A much respected name the world over is that of the Oliver Tiger. It is true to say that over the years they have won more team races and combat finals in their class than all the other makes put together. The **Oliver Tiger Mk. III** is a twin ballrace supported shaft, front rotary induction contest motor with a screw-in air intake just in front of the cylinder. This engine has both high and low speed capabilities. Latest addition to the range is the **3.5 c.c. Oliver Tiger Major.** Similar to the Tiger in layout this engine provides extra power for combat and peaks at a lower r.p.m. It is also ported for radio control use and for a silencer. Both of these engines are available in several versions these being the **/S** machined to accept a silencer, the **Sport** ready silenced and the **R/C** version complete with a barrel throttle and silencer. Mount both engines on $\frac{1}{2}$ in. x $\frac{3}{8}$ in. hardwood bearers, spaced $1\frac{1}{8}$ in. apart for the Tiger and $1\frac{1}{4}$ for Tiger Major. **P.A.W. 2.49 Mk. II** is also a contest engine and has a single ballrace supported shaft with front rotary shaft intake via an air intake cast in the crankcase. Very robust and tough throughout, this engine has been used and progressively modified with success in combat and team racing. Beam mount on $\frac{1}{2}$ in. x $\frac{1}{8}$ in. hardwood bearers spaced 1 in. apart. A development of this led to the **P.A.W. 19 D Mk. II** and **19 D BR** being produced to meet a demand for a larger engine. The same layout is used

2·5 - 3·5cc.

in each but the 19D is plain bearing. Each delivers quite extraordinary power for weight. Beam mount on ½ in. x ⅜ in. hardwood bearers spaced 1 in. apart. No longer in production but still to be seen in considerable use are the Rivers engines. Aimed at the contest modeller the **River Silver Streak Mk. II** was unique in having a roller bearing supported shaft. A front rotary shaft valve induction engine with the air intake cast onto the crankcase and tapped shaft for bolt-on propeller fitting, it is tough but had a habit of breaking connecting rods. The 3.5 c.c. **Rivers Silver Arrow** also used the roller race supported crankshaft and was of the same layout with the exception of a general beefing up and extended air intake for combat flying and stunt models. Mount both engines on hardwood bearers at least ½ in. x ⅜ in.

and the 'Big Stuff' up to 10 cc

Only surviving British racing .29 glow engine is the **Eta .29 VIc.** Developed over many marks this engine enjoys the classic racing glow ignition layout originated in the U.S.A. during the late 40's. A twin ball-race supported shaft with an extension collar is used in a detachable front housing. It is of rear disc induction, the air intake being cast onto the removable rear backplate. The large transfer passage and exhaust stack together with its black finned cylinder head are easily distinguishable features. A feature unique to the Eta is the use of a thick asbestos fibre gasket under the cylinder head. Decidedly happier at higher r.p.m., this is mainly a contest modellers' engine for F/F, speed and "B" T/R. Mount on hardwood bearers at least ½ in. x ½ in. spaced 1¼ in. apart. Out of production for some months but a leader in its time the **Frog 500 RG** has a closely finned cylinder head, and light crankcase with a bolt-on metal fuel tank and long exhaust stack. Employing front rotary induction via an air intake cast into the crankcase, it originated as a coil/spark ignition engine. Beam or radial mount, on ½ in. x ½ in. hardwood bearers spaced 1¼ in. apart.

The Merco range of large capacity engines started with the **Merco 29** and **Merco 35** using similar components. Front rotary shaft induction with the shaft supported in a plain bearing and a one piece crankcase incorporating the air intake, these engines are robust and suitable for all forms of aeromodelling, the 35 being the standard British choice for control line stunt flying for several years. Radio control "Multispeed" versions of both engines are available using a rotating barrel carb: with air bleed adjustment and a linked exhaust chopper. Beam mount on ½ in. x ½ in. hardwood bearers spaced 1¼ in. apart. The **Merco 49** and **61** also use the same crankcase, the 61 having a bored out liner. Construction is more complex than the smaller sizes with the shaft supported in twin ballraces and front rotary shaft induction air intake cast into the crankcase which only extends as far up as the top of the transfer passage. The liner has a light alloy cooling fin jacket and a finned cylinder head. The piston has two rings and extra ports cut in the side to match those in the liner to aid gas transfer. Standard version is available of the 49 for C/L stunt. The 61 is for radio control only and uses the same carb/chopper as the 49. The carb: is a progressive barrel type with air bleed adjustment via a small screw and is linked to a centrally pivoted exhaust chopper which is removed when a silencer is fitted. Mark II versions of both engines have honed bronze bushed piston bearings and connecting rod as well as two glow plugs. Although many twin cylinder engines have been produced in the U.S.A., Germany and in Great Britain, the Taplin is the only one to have proved continually popular and to have retained a fairly low price level. Current model is the **Taplin Twin Mk. II** of 8 c.c.

capacity, an in-line alternate firing twin with built up shaft supported in front by one roller race and by one ballrace at the rear. Induction is side port via a single screw-in adjustable throttle that offers remarkable speed range control. Exhaust is collected in a common manifold. With symmetrical timing the engine runs usefully in either direction. Heavy at 17½ oz. it comes into its own on large (14 in. dia.) propellers and peaks on an 11 x 4. Reliability and very low idle speed are two of its virtues for large sport models. Beam mounting only on ½ in. x ½ in. hardwood bearers spaced 1½ in. apart.