PART TWO

chapter seven

Port Design

When tuning the average production engine, it is logical to set about the task in an orderly manner. The first aspect to consider is what modifications will give the greatest power increase or at least have the most significant effect on power when combined with other modifications. Since this chapter is on ports and port design, we will consider which of the two ports, inlet or exhaust, is the most beneficial to perfect first. Before that question is answered, we must consider the basic function of each port. The job of the inlet port is to pass air from the mouth of the carburetter or injector, which ever the case may be, into the cylinders of the engine. To be an efficient port it must do this with the least amount of pressure drop between the carburetter and cylinder, assuming a constant gas speed at this stage. This means that the higher the inlet pressure or the nearer it is to the external air pressure, the more power we obtain.

The prime function of an exhaust port is to pass the gases out of the engine and, assuming basic conditions again, this is done most efficiently when the least back pressure is created or presented to the exhaust.

Having established the basic functions of each port, we can now get down to establishing some criteria on which to base any mods. Using as a basis an ordinary production engine, we find that if we increase the inlet pressure by 1lb per sq in, ie simulate an increase in port efficiency, we get between 20 and 25 times the increase in power than that obtained by simulating an increase in exhaust efficiency by reducing the back pressure by 1lb per sq in. The obvious conclusion to draw from this is that the greatest gains in power are to be had by improving the inlet port or inlet system as a whole. In practice we find that the efficiency of an inlet system varies from engine to engine. Take, for instance, the inlet system of a twin cam Lotus-Ford engine. This has ports which have no abrupt curves, feed into a hemispherical chamber and each cylinder has its own carburetter barrel. All this goes to make up an efficient inlet tract. If the standard valves and carbs are retained, then a really good port job on the inlet ports only, results in

a power gain of about 5%. In this particular engine, we find that there are greater inefficiencies in the exhaust system, mostly caused by the silencer. If the silencer is removed, we find a very similar power gain to that produced by a modified inlet port.

With a more mundane engine such as a series 'A' BL engine as fitted to Minis, A40s etc., increased induction efficiency whilst still retaining standard valves and carbs can result in a power gain of 30% whilst extensive work on the exhaust side will give only 10–11% power increase.

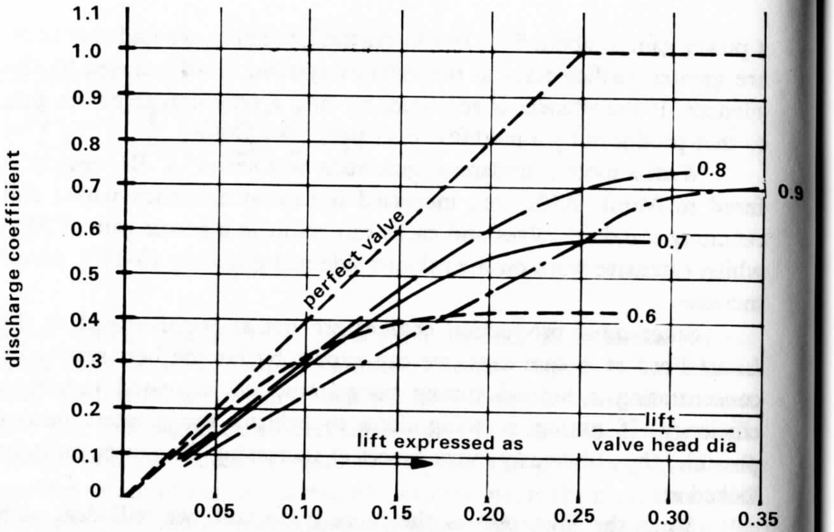
Since most production engines are not as sophisticated as the Lotus-Ford twin cam unit, we can expect by far the best results by concentrating a limited tuning programme on increased induction efficiency. If tuning is being done to achieve the greatest output possible, then obviously every aspect of increasing the power must be looked at.

Since the inlet port is the more important, we will deal with that one first. Before we get too detailed on this, it should be pointed out that other factors such as chamber shape, valve profile and the type of valve seat used have a distinct influence on the design of the actual port so not only will the port be dealt with but also these other relevant factors.

One of the first things the unenlightened head modifier does is to enlarge the inlet ports without thought as to why they need to be larger, if indeed they do need to be larger. Granted, a large diameter pipe will pass more air than a small pipe, but an inlet port is not as simple as a pipe. It has a reciprocating valve at one end of it and this tends to invalidate much, when a comparison is made between the two.

To determine the size of port required, we need a reference point to which we can relate it. In this instance the ratio of valve size to port size has a distinct effect on mass air flow. Thus if we know the valve size we are going to use, the optimum port diameter can be calculated. Fig. 29 is a graph which shows the discharge coefficient (C/D) of various ratios of port to valve head diameters. The diagrammatic layout shows the shape of the valve, port and chamber that produced the curves shown in Fig. 29. As can be seen, the best gas flow was obtained when the port size was made 0.8 times the diameter of one valve. This immediately dispels the theory that the bigger the ports are the better they are. Before we can apply the optimum port size shown by Fig. 29 to an actual port on an engine, we must consider that any variations of the test conditions will invalidate the results shown in Fig. 29.

You will notice in this figure that the valve opens on to a plain



surface whereas in a conventional pushrod engine we have the chamber or cylinder bore wall in close proximity to part of the circumference of the valve. This causes shrouding and prevents a full flow of gas around the entire circumference of the valve. Fig. 30 shows how shrouding is caused by either the chamber wall or bore wall. The shrouding caused by a chamber wall need not bother us unduly, since this is one of the things we can reduce to a minimum. The shrouding caused by the bore of the engine cannot, in most cases be changed, especially if the bores are close together. Assuming the bores cannot be materially altered, then the shrouding so caused represents the lower limit of shrouding. When modifying a head, we must attempt to get the shrouding factor of the chamber equal to that caused by the bore. The shrouding factor, it should be pointed out, is a percentage measurement of the reduction in efficiency caused by the close proximity of the chamber or bore wall. An example here may serve to clarify matters. Take, for instance, a crossflow Ford head—since it has no chambers in the head, there cannot be any shrouding caused at this point. If we test the quantity of air that the head will flow while it is on the block, we find that it is 16.5% less than when it is flow tested by itself. The shrouding factor of this head then is 16.5%.

It may seem as if the subject has changed from ports to chambers. We are in fact still dealing with the topic of ports, but it is difficult to deal with one without making some reference to the other. As has been pointed out previously, the curves given in Fig. 29 are for a completely unshrouded valve. Since, in a conventional pushrod engine this unshrouded condition does not exist, we must make due allowances

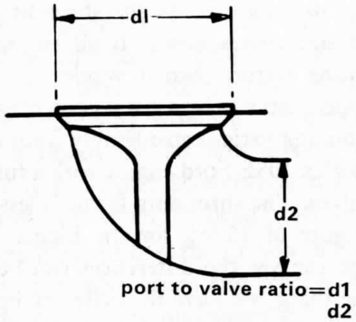


Fig 29 Determination of optimum port to valve ratio

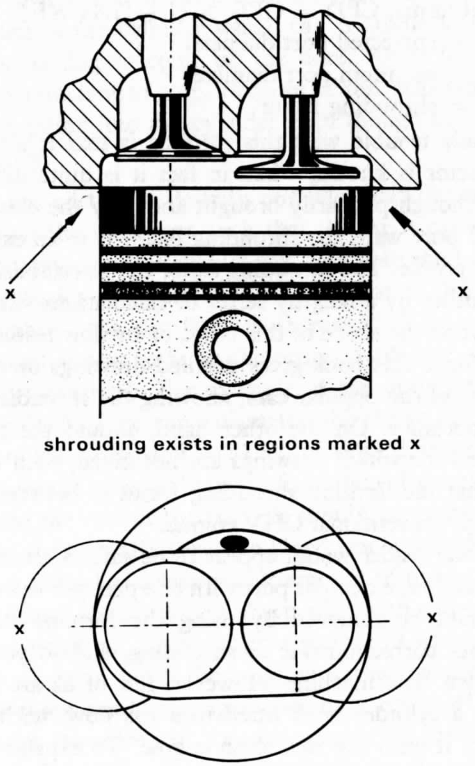


Fig 30 Shrouding points

to arrive at the optimum port size. As the shrouding factor goes up, so the optimum port size comes down. If all engines had the same lower limiting shrouding factor, then it would be simple to reduce our optimum unshrouded port size by a given percentage. Unfortunately the lower limit of shrouding varies considerably from engine to engine. Take, for instance, a 997cc 105E Ford engine with a fully modified head using standard size valves. The shrouding factor is less than 5%. When compared with the figure of 16.5% for the bigger crossflow engine mentioned earlier, you can see the differences can be quite large. All this boils down to one thing, we have to revise the optimum port size when the chamber is unshrouded to get the optimum port size when chamber shrouding exists. The size correction is quite simple, we make a percentage reduction on the optimum port size numerically equal to the square root of the shrouding factor times 1.4. For example, if we have a shrouding factor of 16% we reduce the port diameter from the optimum given in Fig. 29 by 4% × 1.4 = 5.6% to put it into more mathematical terms, $CPD = OPD \times (1 - [1.4\sqrt{SF}])$

where CPD = corrected port diameter
OPD = optimum port diameter
SF = shrouding factor

The only trouble with this formula is that it assumes that the shrouding factor is known, when in fact it is quite difficult to put a figure to. Although primarily brought about by the close proximity of chamber and bore walls the shrouding factor is to an extent altered by the camshaft profile. The shrouding factor can be established reasonably accurately, either by a lengthy series of calculations which are strictly speaking beyond the scope of this book, or by flow testing under cyclic conditions. Since this book gives detailed drawings on how to modify heads of most of the popular cars, knowing the shrouding factor is not of great importance. On the other hand, should the reader wish to modify a head for which drawings are not given, then it is fairly safe to assume that the limiting shrouding factor is between 15 and 20% for an average conventional OHV engine.

So far the reader would appear to be safe in assuming that with a flat roof chamber, a circular port with one port per valve the optimum gas flow would be achieved by using the formula 0.8 times valve diameter times correction due to shrouding, and so you would, on a steady state gas flow machine. However, few of us are likely to drive around with a cylinder head fitted to a gas flow machine, the usual thing is to fit it onto the rest of an engine. To expand on this it will be necessary to briefly look at the mode of operation of a gas flow machine. The usual method employed to determine the flow character-

istics of a head is to draw or blow air into or out of the relevant port. If the test is being done on an actual head, then a cylinder is placed under the head to simulate the cylinder bore of the engine (Fig. 31). The gas flow is then tested with the inlet or exhaust valve, whichever the case may be, opened by various increments up to the full lift value. Under these conditions the air flow is said to be in a steady state condition since it is neither accelerating nor decelerating and these conditions do not exist when the head is on the engine.

Under actual conditions, the air flow in the inlet port is pulsating. It accelerates as the valve opens and reaches its maximum speed just after the piston reaches its maximum speed down the bore. As the piston reaches the bottom of its stroke, the speed of the air flow drops. When the piston passes bottom dead centre, and starts its journey up the bore, the air flow into the cylinder, especially at high revs, does not cease. That is, of course, assuming that the inlet valve is still open. The reason for this is that at high revs, the piston travels down the bore so fast that it leaves the air in the port behind. This causes a partial vacuum in the cylinder which can continue to be filled whilst the piston is starting to come up the bore. Apart from this, air has inertia and is reluctant to stop. Because of this reluctance to stop, the air can go on piling into the cylinder even though the piston is on its upward journey.

What has all this to do with our inlet port design? Simple. We wish to design the inlet port such that the air it carries contains the maximum energy, and that we use this energy in the best possible way to fill the cylinder. To this end we find that if the valve throat is enlarged slightly, the air flow is increased. The optimum throat diameter would appear to be in the order of 12–14% greater than the optimum port diameter. For example, if the optimum port diameter proved to be 78% of the valve diameter, the throat diameter should be 78 + 9.5% of valve diameter = 87.5% of valve diameter. The enlarged valve throat should extend about 1in from the valve seat and then blend smoothly into the rest of the port. It should be pointed out that it is also essential for the valve throat to blend smoothly into the valve seat. This point is critical as a measurable amount of power can be lost if at this point the valve seat forms a sharp edge with the valve throat.

At this point we can build up a picture of what our port should look like, bearing in mind the conditions that have been laid down. To go over these conditions again, we have one port per valve. The port is circular, the valve seat is a conventional 45° seat and the chamber roof is flat. In Fig. 32 we have a port which, for the sake of clarity, has been straightened out. This shows the valve seat, the enlarged valve

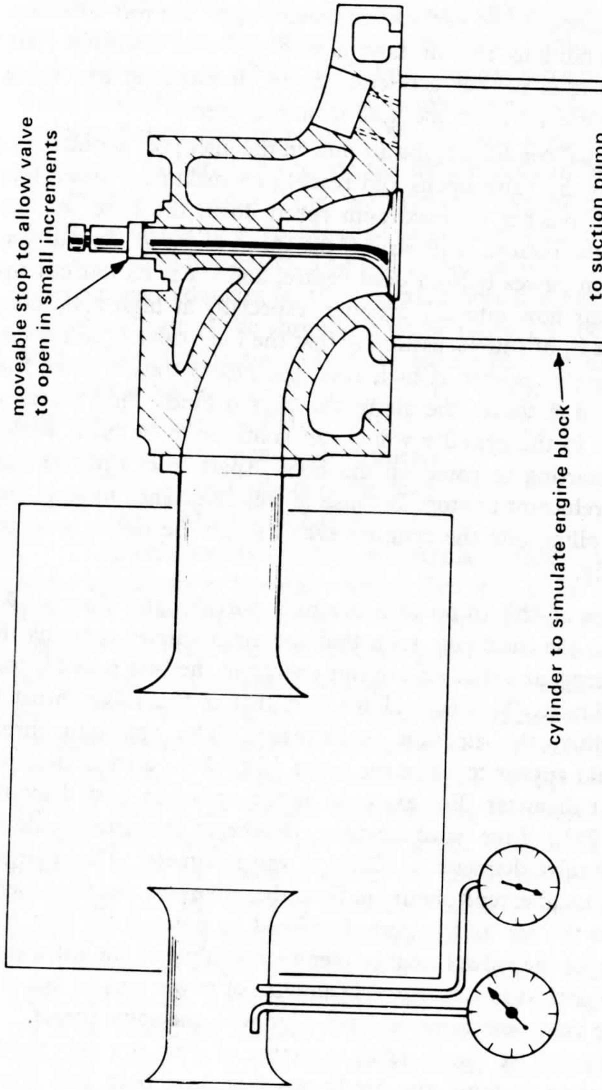


Fig 31 Flow testing a cross flow Ford

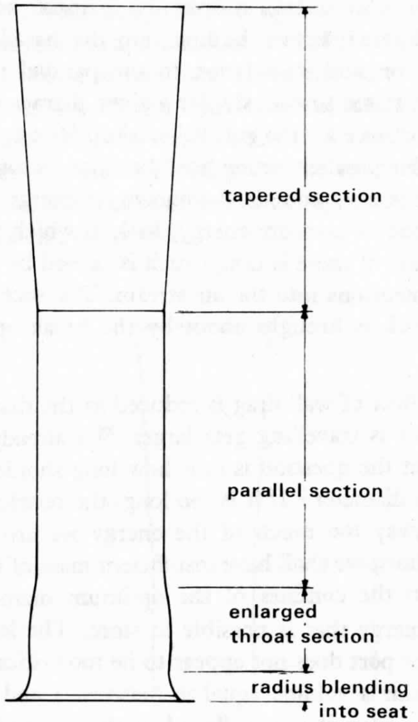


Fig 32 A straightened inlet port showing the various changes in form

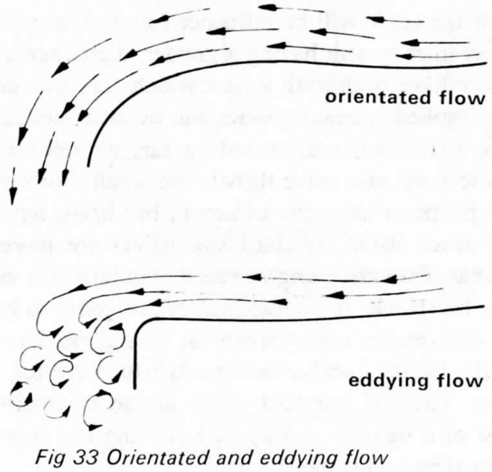


Fig 33 Orientated and eddying flow

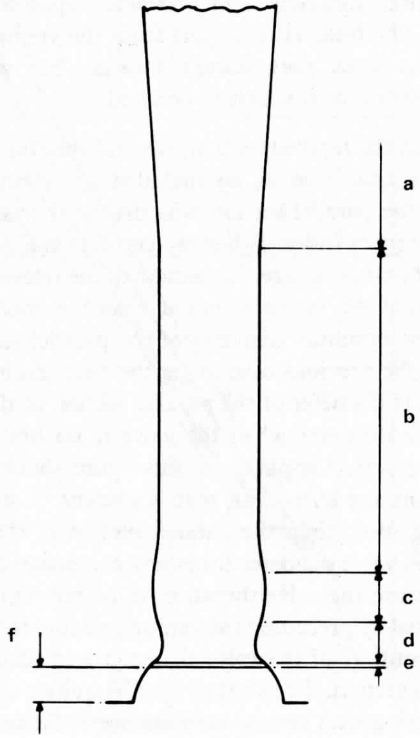
throat, the parallel section at optimum diameter and one more thing—a shallow tapered section leading into the parallel section. It is the tapered section and its relation to the parallel section we will now consider. As stated previously, for a given diameter of valve there is an optimum port size for the parallel section. However, there is a limit to the length this parallel section need be. The reason for this stems from the fact that it must contain the maximum energy possible; two things destroy kinetic or pressure energy, both of which are interrelated.

The first of these is drag which is caused by the walls of the port and any protrusions into the air stream. The second one is caused by eddying which is brought about by the break-up of orientated flow (Fig. 33).

The effect of wall drag is reduced as the diameter of the port in which the air is travelling gets larger. We already have an optimum port size, but the question is now how long should the port remain at its optimum diameter? If it is too long, the restriction caused by drag will drain away too much of the energy we are trying to preserve. If it is too short, we shall have insufficient mass of air travelling at high speed within the confines of the optimum diameter to contain the maximum energy that is possible to store. The length of the parallel section of the port does not appear to be too critical. If it is made such that it contains a volume equal to between $\frac{1}{8}$ and $\frac{1}{5}$ of the volume of one cylinder, then it is usually adequate. In practice this works out quite conveniently, for with many heads we find that the parallel section is contained in the head whilst the tapered section is within the carburetter or injector manifold. The included angle of the tapered section should be within 4°–8°. If 8° is exceeded, we start to lose out. The final choice of the angle will be influenced by the length of the intake system, a short inlet system having a greater angle than a long one.

So far we have dealt with a port which can, to a certain extent, be practically applied to many production cylinder heads. To produce a port to the specification discussed so far, we need a head which initially has the ports and valve throats too small. Ports which are too small for our purposes are quite common, but heads having undersize valve throats when using standard size valves are more of a rarity. This means that if we are going to retain standard size valves, a compromise must be struck. If oversize valves are going to be fitted, then the problem of oversize valve throats is, to a great degree, removed. Some heads do, unfortunately, have ports which are too large for the standard valve. Here, if standard valves are to be retained, we must make the best of a bad job and avoid removing any more metal than is necessary to achieve a smooth port.

On occasions, head designers use square ports, usually because there is insufficient room to accommodate round ports of adequate area. When confronted with a square port where one port feeds one valve, we should endeavour to achieve a cross-sectional area 10% greater than the equivalent round port, the reason being that a square port restricts air flow in comparison with a round port, due to increased wall drag and turbulence. The increase in area over the optimum size round port does, to a small extent, offset this.



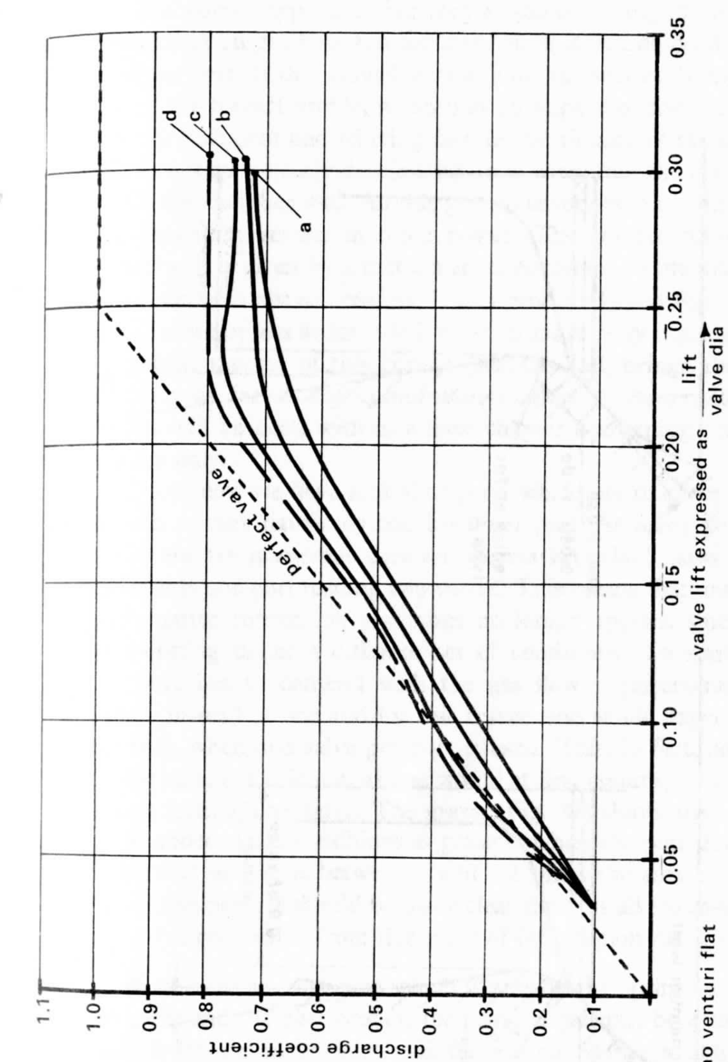
a tapered section
b parallel section
c enlarged throat section
d venturi section
e valve seat
f chamber wall

Fig 34 Make-up of venturi port, straightened out for clarity

Although the type of port so far dealt with represents a fair improvement over many standard ports, it is by no means the best that can be produced with present knowledge. The venturi port, which we are about to discuss, does show a significant increase in air flow over the more basic type of port discussed so far. One of the most important parts of an induction system as far as breathing goes, is the area just before and just after the valve seat. It is by subtle changes in this region that the venturi port gains in efficiency when compared with a more basic port. Fig. 34 shows the basic make-up of a venturi port with its form exaggerated for the sake of clarity. With this type of port we have an entirely different set of parameters upon which our optimum port is based. The basic shape, apart from the venturi section, will be as for the more basic port discussed earlier but we have a greater number of variables with which to contend.

Starting at the tapered section, we find that this will be the same as the previous case, that is, an included angle between 4° and 8°. The length of the parallel section will also be the same at $\frac{1}{8}$ and $\frac{1}{5}$ of the volume of one cylinder. When we come to the optimum diameter we find the first sign of change. Inspection of the relevant graph (Fig. 35) will reveal that if we use the curved chamber roof together with a venturi port, the optimum diameter of the parallel section needs to be increased over the previous case to get the best results. With a curved chamber roof the diameter of the parallel section of the port should be 0.84 times valve diameter when the valve is unshrouded. To get the port size for a practical application, this figure should be corrected to take into account any shrouding that is present. If a curved chamber roof isn't being used, then the parallel section of the port should be about 0.82 times valve diameter times any correction due to shrouding.

Moving on to the valve throat, or to be more precise, the part of the port immediately preceding the venturi, we find the diameter should increase to about 0.97 of the valve diameter and then smoothly blend into the venturi section. The venturi itself is quite a complex affair and after experimenting with various combinations of diameters, angles and radii, it was found that the profile shown in Fig. 36A gave the best results. The valve seat life, however, was short, acceptable for a racing engine, but not so for a road engine. The wear suffered by the seat did not impair the sealing but did have the effect of deforming the radius, thus destroying the flow properties of the seat. To achieve a longer seat life the profile shown in Fig. 36B can be used. It is only marginally inferior to type A but does not suffer such rapid seat deformation. On the debit side, seat B is much more difficult for the amateur to produce than seat A.



a conventional 45° seat no venturi flat roof 0.8 port/valve ratio

b venturi port with flat roof 0.82 port/valve ratio

c venturi port with curved chamber roof 0.8 port/valve ratio

d venturi port with curved chamber roof 0.84 port/valve ratio

Fig 35 Optimum port to valve ratio for venturi port with curved chamber roof

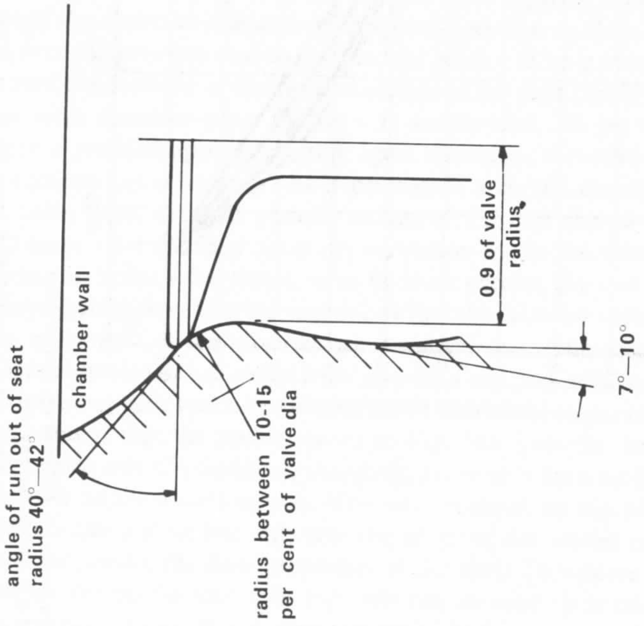


Fig 36A Valve seat profile—racing engine

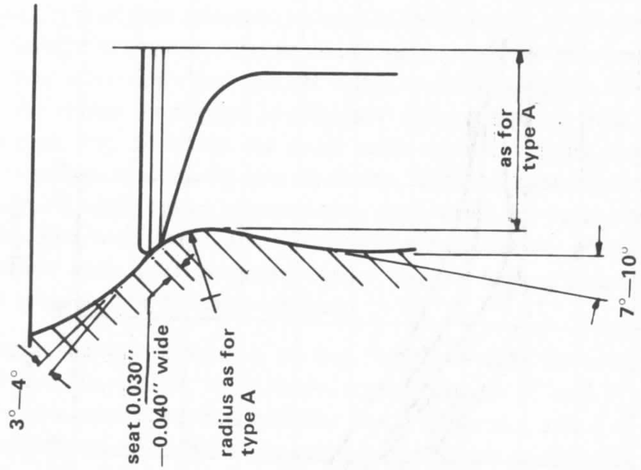


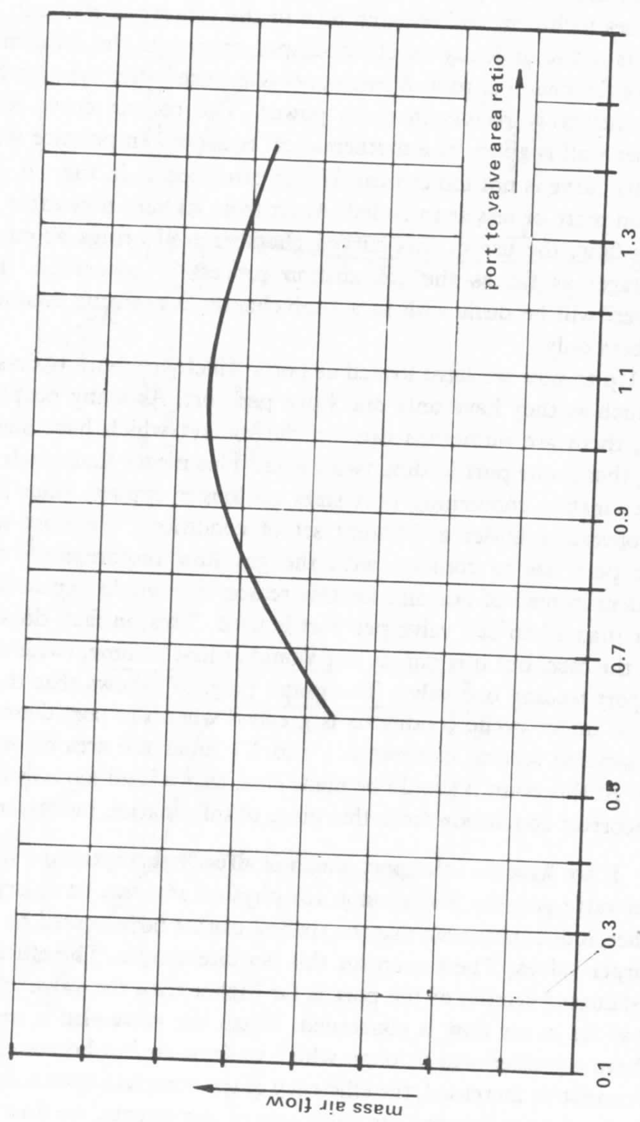
Fig 36B Valve seat profile—road engine

The curved type chamber roof as shown in Fig. 36 has quite a distinct effect on gas flow and for this reason it will be dealt with as a part of the port. If this curved section joins the seat at the right angle and is of the correct profile, it can play an important role inasmuch as it reduces turbulent and eddying flow in the vicinity of the seat. Also, the air is not so violently decelerated upon entry into the cylinder as it is with a flat chamber roof. All this goes to ensure better cylinder filling which inevitably results in more power. The correct curve for the chamber wall is given by a mathematical equation. In practice we find that this curve is not too critical. If it is produced as in Fig. 36, it will function more or less as intended. Apart from its very noticeable effect on gas flow, the use of this curved chamber wall brings about other advantages as far as the comubstion process is concerned. These, however, will be dealt with in a later chapter concerning combustion chambers only.

Up to now we have looked at ports which are more or less ideal inasmuch as they have only one valve per port. As many people well know, there are numerous cars on the market which have siamesed ports, that is one port feeding two valves. This means that much of the subject matter concerning port sizes no longer applies, since we are now operating under a different set of conditions. To start with, a single port has to contend with the gas flow requirements of two cylinders instead of one and for this reason one would expect it to be larger than when one valve per port is used. This, in fact, does prove to be the case, but it is not, as one would at first assume, twice the size of a port feeding one valve. The graph (Fig. 37) shows that the mass air flow under cyclic conditions is greatest when the port diameter of the siamesed section is between 1 and 1.1 times the area of one inlet valve. At this point it should be made clear that it is all too easy to draw an incorrect conclusion from this piece of information just given.

If we have an inlet port which is already the optimum size for a given valve and the port cannot, for physical reasons, be enlarged any further, it does not mean that the air flow cannot be increased by putting in larger valves. The reason for this is quite simple. The efficiency of the siamesed section of the port is far higher than the valve and valve seat as far as air flow is concerned. When the valve size is increased, we have a greater area through which to draw air but because the port size cannot be increased, the efficiency of the complete system drops off. If we keep on increasing the valve size in increments, we find that the time comes when efficiency is falling off faster than the valve area is increasing. When this point is reached, we can no longer get any increase in air flow. The point at which it appears to be impractical to increase

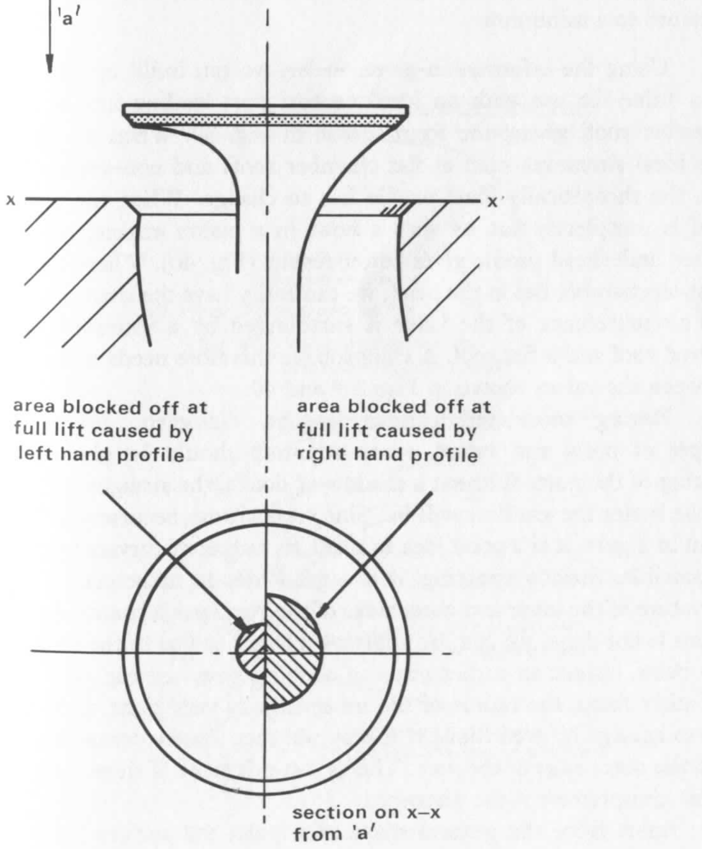
Fig 37 Determination of optimum port to valve ratio with siamese ports



the inlet valve size any further occurs when the area of one valve is 50% greater than the area of the siamesed section of the port. As far as venturi valve throats, radius seats and curved chamber walls are concerned, these will apply more or less as before.

Another factor vital to good gas flow is the shape of the inlet valve. In days past, it was thought that a well-tuliped valve gave the best gas flow by virtue of its streamline shape. This, however, was a fallacy. It is, in fact, possible to show by geometric means that if the underhead radius exceeds 0.25D (where D is the valve dia.) the part of the underhead radius that is level with the valve seat is causing a reduction in valve throat area (see Fig. 38). If that isn't clear, then the best way to visualise it is to assume that the underhead radius is very large, say about equal to the diameter of the valve.

Fig 38 Valve profile



When the valve is at full lift, an appreciable amount of the tulip section is still within the confines of the valve throat. Since the tulip diameter, which is level with the valve seat at full lift is of larger diameter than the valve stem, it is, in effect, partially blocking the hole through which the air is expected to pass. To avoid any such restriction, the underhead radius must not exceed 0.25 times valve diameter. Apart from attention to the tulip part of the valve, we can, by careful attention to profile, achieve a secondary venturi action at the valve seat which increases the discharge cofficient considerably at low valve lifts. This is of considerable importance, since it achieves the same effect as using a cam with a faster opening and closing motion with a conventional seat and port. The advantage of this secondary venturi effect is that we achieve the gains of the camshaft without any extra stresses imparted to any of the relevant components. Aside from any considerations of gas flow, the valve must be strong enough to withstand the loads expected of it, yet be as light as possible to keep the valve spring pressure to a minimum.

Using the information given so far, we can build up the profile of a valve for use with an ideal venturi port leading into a curved chamber roof, giving the form shown in Fig. 39. When we come to less ideal situations such as flat chamber roofs and non-venturi ports etc, the theoretically ideal profile has to change. When the chamber roof is completely flat, as with a bowl in a piston engine, we find a flatter underhead profile gives better results (Fig. 40). Where the combustion chamber lies in the head, we can easily have the situation where the circumference of the valve is surrounded by a shrouded area, a curved roof and a flat roof. A compromise therefore needs to be struck between the valves shown in Figs. 39 and 40.

Having considered various lengths, diameters, profiles and shapes of ports and valves, some attention should be given to the routing of the port. Without a shadow of doubt, the straighter the port is, the better the gas flow will be. Since it is always necessary to have a bend in a port, it is a good idea to make its radius of curvature as large as possible. Strictly speaking, it is a good idea to make sure that the curvature of the inner and outer sides of the port have a common centre. If this is not done, we can, by applying a larger radius to the inner side of a bend, induce an earlier onset of eddying flow (see Fig. 41). If, on the other hand, the radius of the inner edge is very tight, then it can pay to enlarge it, even thought it may not then have a common centre with the outer edge of the port. This is just one more of those situations where compromise is the password.

Apart from the general shape of a port, the surface finish can

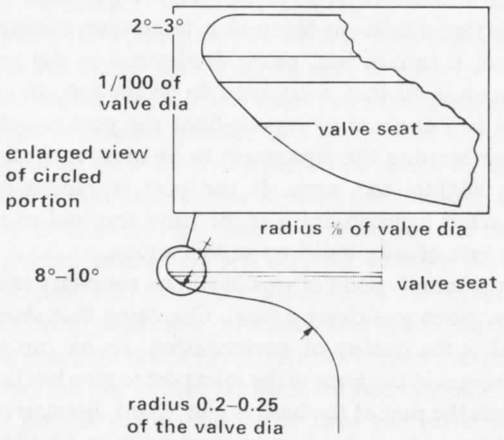


Fig 39 Valve profile for use with venturi ports and curved chamber roof

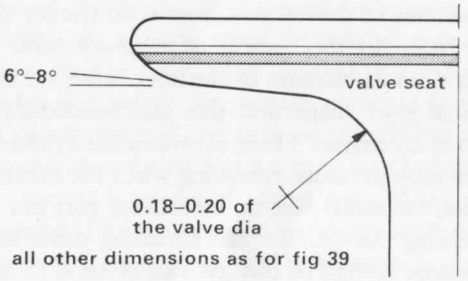


Fig 40 Valve profile for use with flat chamber roof

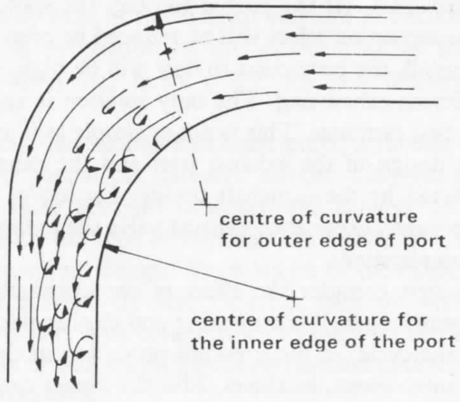


Fig 41 Eddying in a port

bring to bear a measurable influence on the gas flow. The popular conception is that a brilliant finish akin to chrome plating is the best. This is not so. It can, in fact, prove detrimental to the power output. A smooth finish is all that is required to do the job. In some cases it can prove to be beneficial to vapour-blast the port to achieve a matt finish. Vapour-blasting the port seems to be most applicable when the inlet port is slightly too large. If the port is vapour-blasted, it is advisable to retain a polish finish at the valve seat and its near vicinity to retard the rate of coke build-up in this region.

From the tuner's point of view it will be necessary to accept many compromises, when modifying a head. One thing that should never be compromised is the quality of workmanship. As we can see, it needs only small changes in the form of the inlet port to give less than optimum results as far as the part of the head is concerned. Because of this, every effort should be made to produce as good a job as possible for it is of little use having the knowledge if it isn't put into practice.

Exhaust ports

The function of the exhaust port is to convey the burnt gases from the cylinder to the outside atmosphere with the minimum hindrance. In a similar fashion to the inlet valve, we find that careful consideration of port shape and size can beneficially influence the power output of an engine. There is, within the cylinder of an engine, quite a considerable pressure prevailing when the exhaust valve opens. Because of this, the initial flow in the exhaust port can be likened to a slug of gas leaving the cylinder and travelling down the port at high speed. The kinetic energy of this gas can be used to help extract the remaining exhaust from the cylinder, thus relieving the piston of some of the work, as well as assisting in scavenging the chamber when the piston reached TDC. If the port is too big, the speed of the gas will drop and the extraction effect will be reduced or even lost. Should the port be too small, the restriction to flow will be high, which again will lead to inefficient exhausting. The only solution is to strike a balance between the two extremes. This is not as simple as it at first may seem, for the final design of the exhaust port and the exhaust system as a whole is affected by the camshaft timing. On top of this, we can, as with the inlet valve, experience exhaust valve shrouding which can lead to further complications.

Let us first consider the effect of the camshaft timing. As the cam timing gets longer, ie the opening and closing points occur sooner and later in the cycle, so the pressure prevailing in the cylinder when the exhaust valve opens, increases. Also the longer cam timing implies that the engine is to operate at higher revs, thus the speed of opening

of the valve with respect to time, is faster.

From these two points we can see that the initial slug of gas from the cylinder will occur with a more well defined front and will be travelling at a higher speed. This means that as cam timing is increased, so the amount of exhaust handled in a given time is increased during the critical exhaust blow-by period before the piston forcibly expels the gases. As one would expect, this dictates the need for a bigger bore exhaust port or exhaust pipe than is required for an engine using a cam with shorter timing. In practice, the exhaust port diameter is between 1.15 and 1.25 times the diameter of the exhaust valve, the lower figure being applicable to road cams while the higher one is intended for competition cams. A cam with semi-competition timing will, of course, fall somewhere between the two extremes. Obviously if the exhaust port is to become larger than the valve, it must taper outwards from the valve size or rather, to be more precise, from the inner valve seat diameter to its intended size. If the port tapers out too quickly, the flow will inevitably break away from the port wall. This will cause a loss of energy due to increased turbulence and a rise in back pressure. An ideal angle to taper the exhaust port to is between 2° or 3° inclusive, but in many cases we find that the port does not reach its intended size before the exhaust manifold is reached. This need not be a disadvantage if the exhaust manifold is made to suit. On the other hand if the exhaust port has to be matched to a proprietary manifold of the correct bore size, the taper angle of the port may well have to be increased. The approximate maximum angle of taper before the flow breaks away from the port wall is between 7° and 8°, and this should be regarded as the upper limit for exhaust port taper.

A reduction in the required port size is brought about by any shrouding that may exist at the exhaust valve. The reduction in diameter should be in proportion to the square root of the shrouding factor, ie if the shrouding factor is 9% the port diameter reduction will be $\sqrt{9} = 3\%$.

When a head has siamesed exhaust ports, a further size increase is required in the port at the siamesed section. The area increase over that of a single valve per port is somewhere between 25% and 40% depending again on the type of camshaft used and, to a lesser degree, on the exhaust manifold design. If the length of the exhaust pipe from a siamesed port is relatively long before it joins up with others, the optimum exhaust port area comes down slightly. If the length of the exhaust pipe to its first join is progressively shortened, then the optimum port area goes up slightly.

Curves in an exhaust port should be dealt with in the same fashion as the inlet, ie the radius of curvature of the outside and inside should have the same centre. Only when the inside curve is very tight should one attempt to enlarge it. Even so, this must be done with caution so as not to overdo things.