creased. This confirms the above explanation, since such an increase in cross-sectional area will reduce the force necessary to accelerate the gas already in the pipe and eliminate the pressure build-up.

The process of investigating the increasing diameter or 'stepped' pipe has shown another interesting development, in that the larger-diameter length vibrates in its own period, it being possible to combine the waves to give a very low pressure drop after the initial positive pulse.

The designer of the self-induction or harmonic-induction two-stroke diesel engine (fig. 5 : 13) which was developed over 30 years ago (and which depended entirely on pressure waves for the charging and exhausting processes) successfully applied the stepped-pipe

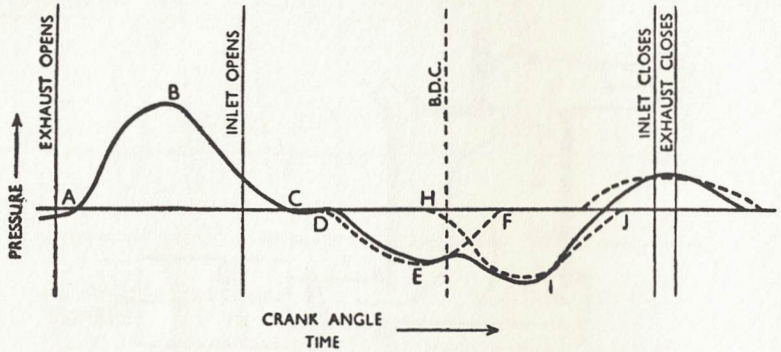


Fig. 5 : 14. Pressures in stepped exhaust pipe of self-induction engine.

principle to improve volumetric efficiency. It was stated (XVIII) that a plain length of exhaust pipe gave a short period of negative pressure whereas, by putting a step in the pipe, the exhaust ports could be kept open for much longer, thus enabling a greater charge volume of air to be inhaled. The negative reflection which returns to the port from the step is of smaller amplitude than would be the case from the pipe end; while this reflection is taking place however, the initial pulse continues down the pipe to the open end of the larger diameter and is reflected, this second reflection following closely on the one from the step. The result is to prolong the negative pressure. Next time the combined waves are reflected (from the extreme end), they are of course positive in sign, but by this time the exhaust ports have closed.

The curve reproduced in fig. 5 : 14 was taken from the harmonic-induction engine. The portion AB represents the initial pressure pulse as the port opens. From B to C the pressure falls to slightly below

that in the cylinder. At DEF is shown the wave reflected from the step in the pipe, while the second dotted line HIJ is the pipe-end reflection. The distance AD is the time taken for the wave to travel from the port to the step and back, while AH is that taken by the wave from the port to the pipe-end and back.

This engine was a comparatively slow-speed type, the torque-speed curve being shown in fig. 5 : 15, but there is every reason to assume that the trumpet-end or megaphone applied to many racing pipes owes much of its efficiency to having a similar effect to that of a stepped-pipe. This device is dealt with in more detail in Chapter 8.

The diagrams so far have been concerned with single straight-through pipes attached to one cylinder, and thus show the conditions

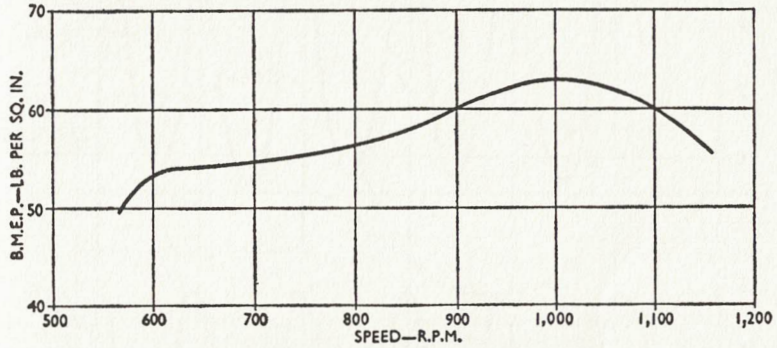


Fig. 5 : 15. Torque-speed curve of self-induction engine.

as would obtain in a single-cylinder engine. The data can be made use of in designing a multi-cylinder system where long branch pipes are used from each exhaust port. First, however, it will be interesting to note what has been shown to happen, as demonstrated by indicator tests on the conventional type of multi-cylinder manifold.

Multi-cylinder manifolds

Where the usual type of cast-iron manifold is employed, having a single outlet to a downpipe and short offtakes from the exhaust ports, the wave from each individual cylinder is seen quite clearly at very low engine speeds. However, as the speed increases the tendency is for the whole of the curve to become lifted above the atmospheric line and to show a fluctuating positive pressure at all times. Obviously under these conditions there will be appreciable back-pressure, particularly at high engine speed, and on the face of it the use of any overlap at all seems of doubtful value. Since however it is indisputable that overlap usually increases torque even at fairly low

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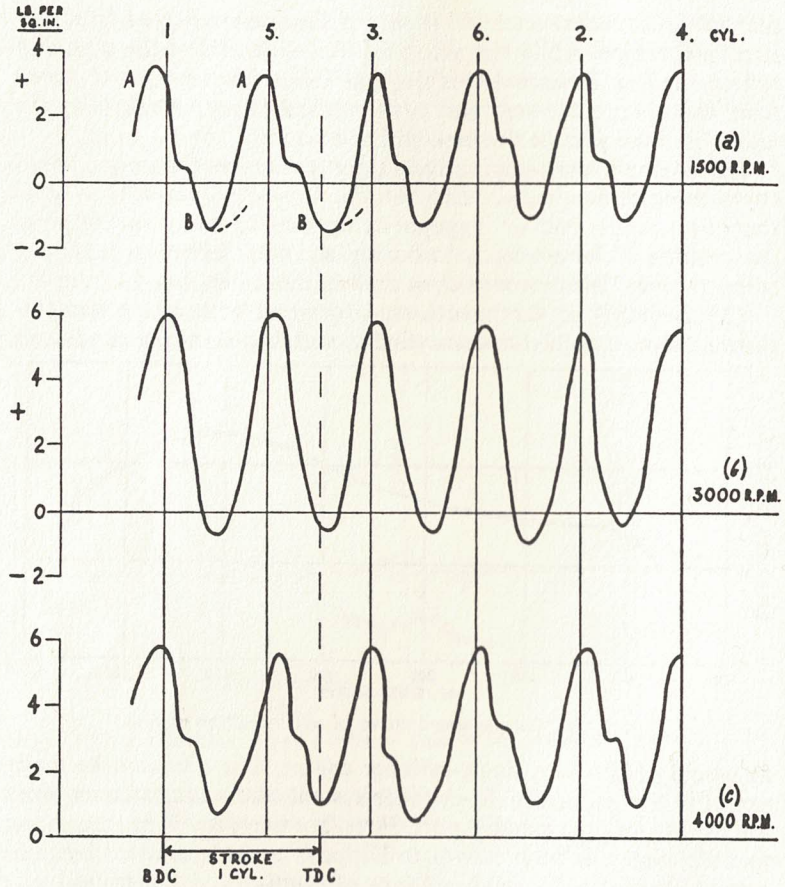


Fig. 5 : 16. Pressure recordings of six-cylinder engine with one-piece manifold bolted to ports.

engine rev/min it may be assumed that the cessation of upward piston movement at exhaust t d c and its immediate reversal to a downward movement has a favourable influence on the pressure in the immediate vicinity of the port. Otherwise the surcharge with exhaust gases (which could be under a manifold pressure, in a really poor design, of up to 6 lb/in²) can be very serious. This effect is shown on the curves in fig. 5 : 16, taken by the Morrison indicator on a six-cylinder 2·15 litre engine having a single exhaust manifold with one outlet and short offtakes to the ports. A single constant-diameter

pipe led to an absorption silencer and the pressure recordings are taken at the end of the manifold opposite to the pipe flange.

In curve (*a*) it will be seen that there is a pulse at A from each cylinder, just before the b d c, followed by a rapid drop to sub-atmospheric at B. At the latter point the dotted line shows how the curve would probably continue at below atmospheric pressure for a further time, were it not for the pressure pulse coming in from the next cylinder; this is repeated at each exhaust valve opening. Conditions for torque are good, as there is an appreciable pressure drop at t d c while each of the initial pressure pulses is reduced in amplitude by having added to it the negative part of the previous wave.

Curves (*b*), (*c*) and (*d*) show that as the speed rises these conditions

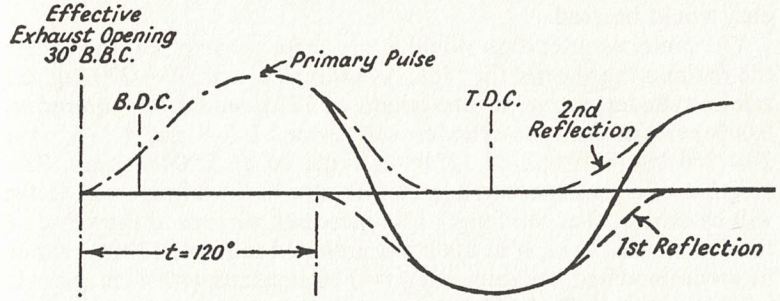


Fig. 5 : 17. Hypothetical curve of pressure at exhaust port when cylinder branches of graded length are used.

do not persist. There is a tendency for the whole curve to come above the atmospheric line, showing a fluctuating positive pressure at all times. This is the condition approximating to the use of a connecting pipe some 5 ft in length having a straight-through silencer at its end.

In the case of manifolds designed with individual long pipes from each point, it has been stated (V) that when the latter are used and correctly merged into the main pipe or pipes very little back-wave is evident and that a hypothetical curve can be drawn (fig. 5 : 17) showing the exhaust port pressure; this assuming cylinder branches of approximately equal lengths.

The curve fig. 5 : 17 is based on a time *T* equal to 120 deg. (see earlier calculation). The basis is that the primary pulses from each cylinder have negligible effect on each other, affecting only the waves in their own ports. Their reflections, however, will interact, *i.e.* the reflections from Nos. 2 and 3 and 4 will pass back to No. 1 port and the whole system will then resonate as the successive reflections from all four cylinders coincide. The amplitude of the wave as a result

will equal that which would be obtained with a separate short pipe on each cylinder (V). With a T of 180 deg. the positive and negative reflections will produce complete damping after the initial primary pulse has passed.

The following values for T will thus apply:

For damping $T = 180, 360, 720$ deg. (or multiples of 1, 2, 3, 4).
For resonance $T = 90, 270, 450, 630$ deg. (or multiples of 1, 3, 5, 7).

It is possible to calculate a main pipe length in the same way as described earlier for single pipes. Whatever value is used for T, the appropriate multiple is also used as above. Thus, for a four-cylinder with $T = 90$, the figure to be used in the calculation would be as above. For a six-cylinder with twin downpipes, a T value of 120, 360 etc., would be used.

The same consideration would apply as for single pipes; the higher the rev/min the shorter the pipe. As examples, with $T = 270$ deg. on a four-cylinder engine, a pipe length of 77 in. would be required at 6,000 rev/min. On a six-cylinder, sub-divided 1–2–3 and 4–5–6, with $T = 360$ deg. a length of 122 in. is required at 5,000 rev/min. The length is of course measured from the valve to the tailpipe end and it will be evident that this length also increases *pro rata* to the value of the T multiple. Thus, if in the last-mentioned example 122 in. cannot be accommodated, the same effect will be obtained with 41 in., as this corresponds to a T of 120 deg.

It must be emphasized that a pipe dimensioned in accordance with the foregoing may not give the desired performance and the same important reservations apply as were detailed in the case of the similar formula for single pipes. The unpredictability of a good part of the wave action is quite evident from the experiments detailed and it is quite impossible therefore to accept 'paper' solutions at their face value.

Sub-division of manifolds

The usual method of sub-division, as shown in the illustrations (fig. 5 : 18), is aimed at obtaining an exhaust discharge at equal intervals, at the junction of the branches serving the 'pairs' or 'threes', in a four and a six respectively. An eminent designer of racing 'singles' (XIX) has stated that inter-action can be used whereby one cylinder's exhaust-flow helps to draw out that from the next one in sequence, by suitably shaping and proportioning the system. While this generalization does little to indicate the layout in mind, it has been found from discussion with pipe-designers that the following is a popular theory.

Imagine a pipe formed like a letter Y with the two top ends connected to the outer cylinders of a four, the vertical leg forming the downpipe. The high-velocity discharge from the left-hand cylinder will flow down the appropriate pipe and as it passes the junction at

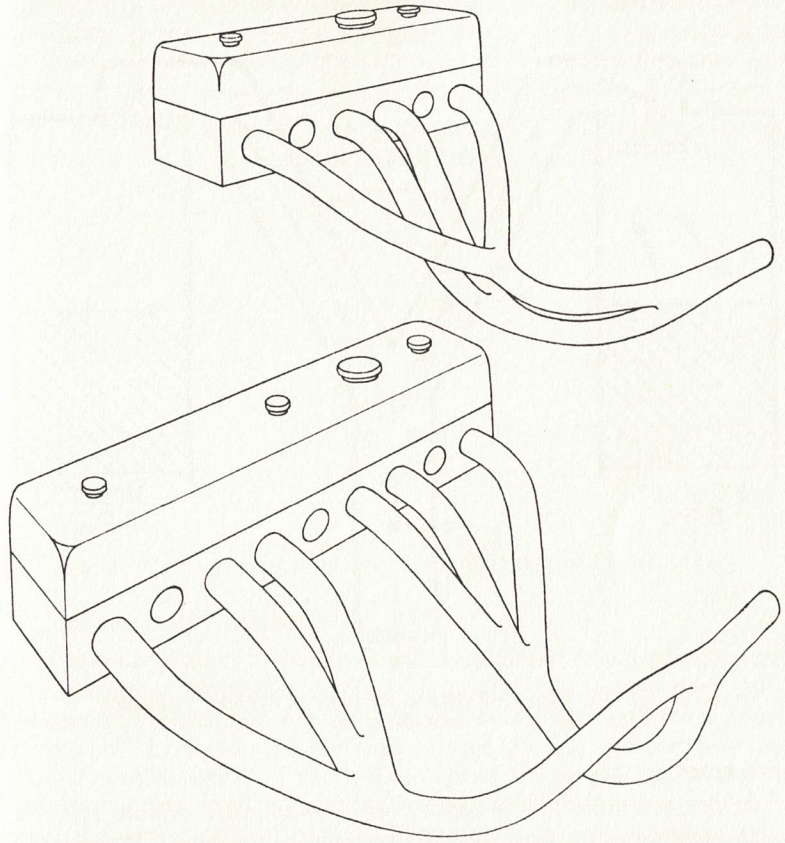


Fig. 5 : 18. Sub-division in manifolds to obtain equal discharge intervals at junction of branches.

the middle it will create a vacuum in the right-hand pipe; and, of course, vice versa when the right-hand cylinder discharges (see fig. 5 :19).

There is absolutely no evidence to support this theory. The indicator has shown that it is quite possible for a positive pressure wave from one cylinder to go 'round the corner' at the junction of the pipes and back up the other leg; the wave then acting in opposition to the

discharge when that valve opens. This is particularly likely to happen, naturally, if the junction is too close to the ports, *i.e.* if the branches are unduly short.

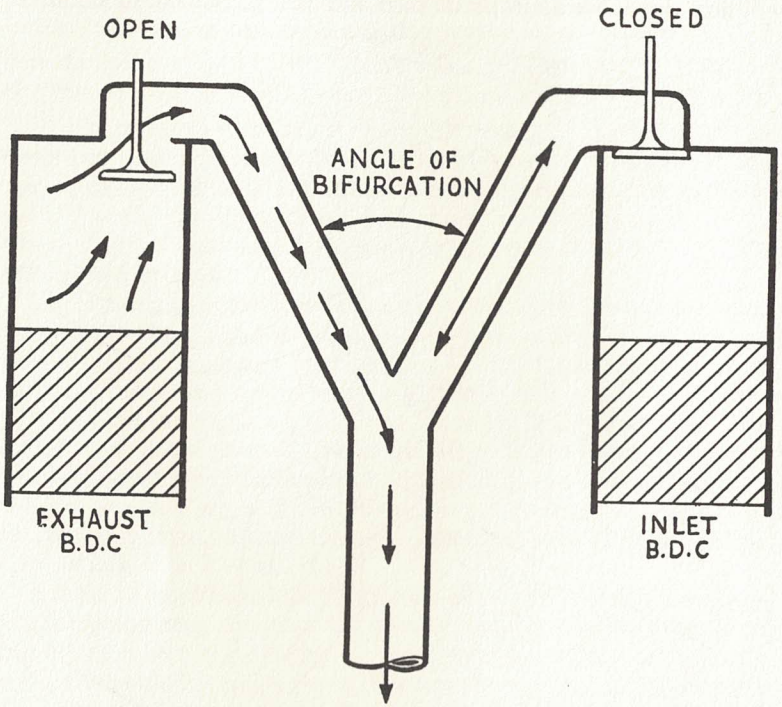


Fig. 5 : 19. Theory of assisted scavenge of cylinder by branch sub-division.

Siamesed ports

It is regrettable that a number of four-cylinder engines retain a siamesed centre port serving the two middle cylinders. Certainly, the discharge intervals are clearly separated, but the fact remains that the individual branch pipe from this port must carry twice the volume of gas in comparison with the others, in increments at double the frequency. The disadvantages of this in practical terms may be queried. But it must be said that, if the aim is to secure as nearly equal conditions as possible, as between cylinders (even if this is an impracticable ideal), a design point that unnecessarily prejudices this is bad. Perhaps of more concrete concern is the fact that the seated exhaust valve is inescapably exposed to the blast from the open one,

at the very time when it should be getting a chance to cool off. The possibility that with many existing manifolds the non-siamesed valves have an almost equally bad time is hardly an excuse.

It will however be obvious that a form of manifold which reduces this liability to extra heat is advantageous to the engine in respects additional to that of better scavenging. It is probable (XXXIII) that in many conventional systems excessive valve temperature causes a greater loss of power than does average back-pressure, both losses having the same root cause.

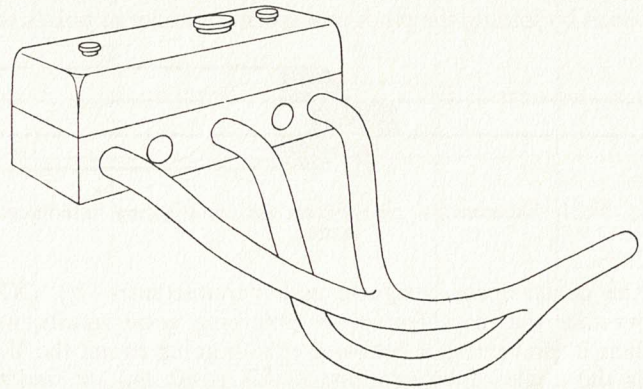


Fig. 5 : 20. Four-cylinder system with central siamesed exhaust port.

Independence of vibration

The conception of a single straight-through pipe from a single port being comparable with a stopped organ pipe has already been considered in this chapter, in regard to its theoretical length for a given frequency. However, the addition of any kind of silencer must be expected to change the period; a silencer of a highly obstructive type might well convert the pipe virtually into a double closed one, in which case the period would be halved (Chapter 3). It is possible to retain the 'independence' of the primary pipe from the cylinder by leading it into a large exhaust box, providing the latter is completely unobstructed inside and of sufficient capacity.

With one pipe discharging into such a box experimentally, a small reduction in the period of the pipe was noted with a box of 50 times the cylinder swept volume, but this capacity could be reduced to 15 times before there was any serious change in period. Thus, if a chamber of such capacity can be accommodated, it can be assumed that the separate cylinder pipes would retain a high degree of inde-

pendence. As shown in Chapter 6, a box of this kind does not have much silencing effect, but in using it for retaining pipe independence it has the further advantage of nullifying any appreciable effect, on the primary pipes, of the length of outlet pipe from the box (XXXIII). This outlet pipe can thus be taken through a silencer as described in Chapter 7.

Interference

We can next consider the results to be expected by a deliberate process of allowing 'interference' between exhausting cylinders, this being done by joining the pipes in a specific manner at points remote

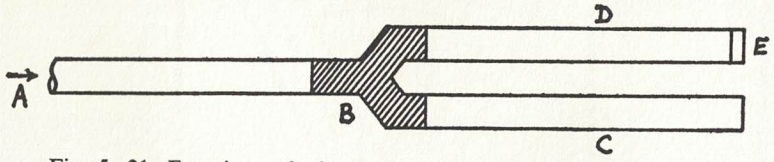


Fig. 5 : 21. Experimental pipe system for investigating 'interference' action.

from the ports. Some designers and manufacturers (V), (XXXV) consider that the main feature in obtaining good results, and in particular in preventing a back-wave from going round the Y-junction, is the angle of bifurcation; if this is 40 deg. or under, the latter effect will be negligible, and 30deg. is frequently used (see fig. 5 : 19). While it is true that a small angle might encourage mass flow in the right direction, experiments (XXXIII) have shown the angle to have negligible effect on the propensity of the wave for going round the corner. The whole action of the layout is in fact much more subtle, as will now be detailed.

If the diagram fig. 5 : 21 is considered, the intention is to find out what happens when the wave starting from the port at A travels to the open end of the main pipe at C. That is, how much of the wave goes into the other branch DB, and how this affects wave AC. The experimental set-up was arranged with the exhaust port connected to the point A. At B is a T-piece, one end of which connects to an open-ended pipe while the other is coupled to a pipe of identical dimensions but closed by a plug at its outer end E.

Pressure readings were taken at five points along the pipe system, the position of the indicator tappings being shown in fig. 5 : 22(*a*.) This also shows the pressure variations at seven different crank angles from bdc to 90deg. after tdc. Figure 5 : 22(*b*) shows these angles, together with the instantaneous pressures at each of the five points. In analysing the curves in fig. 5 : 22(*a*) the following should be noted:

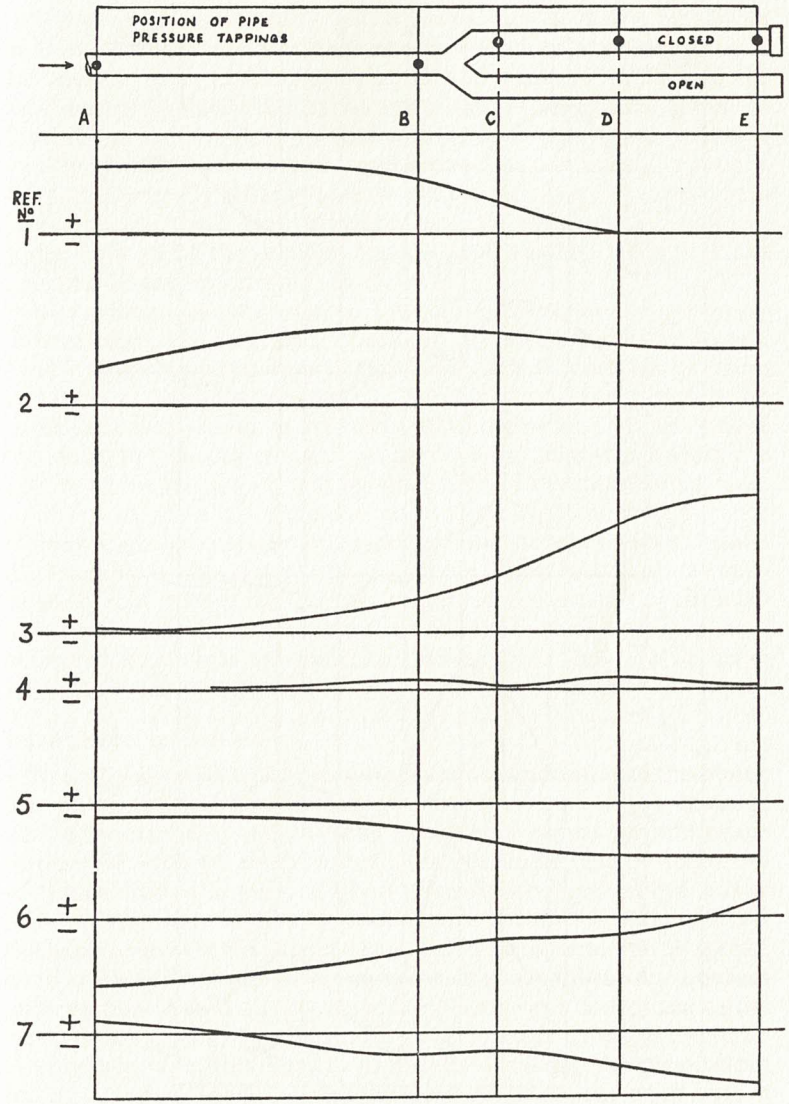


Fig. 5 : 22(*a*). Pressure recordings along pipe to end of closed branch at various crank angles.

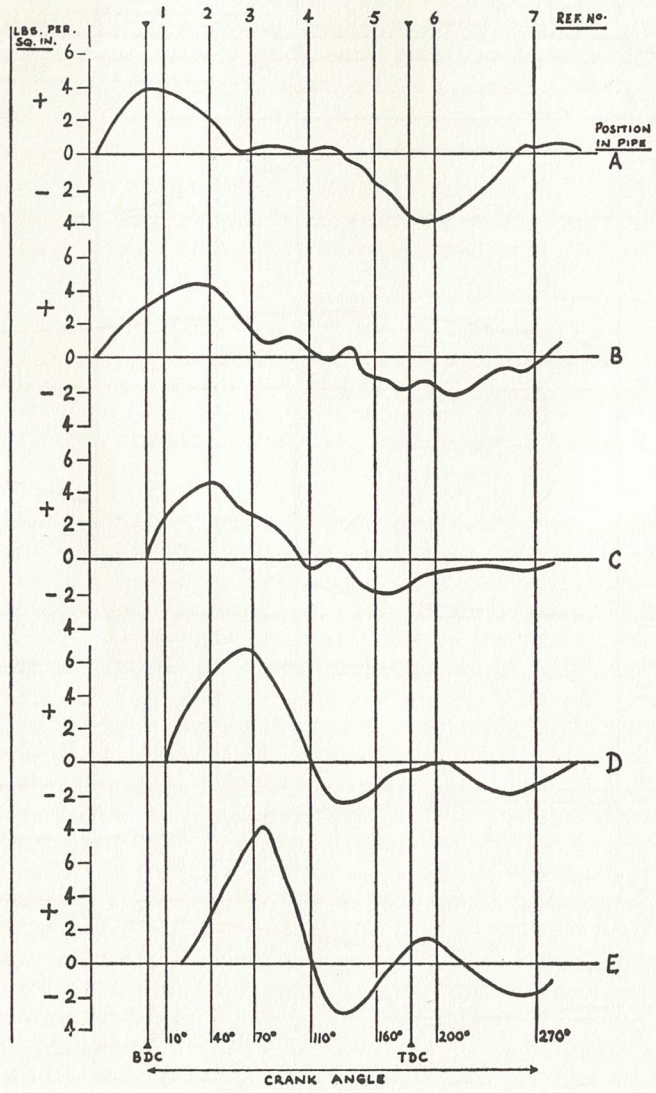


Fig. 5 : 22(*b*). Pressure recordings over 270 deg. of crank rotation at each pressure point.

In 1 the pressure is at a maximum at A, then decreases gradually to B and more quickly to D. In 2 it is dropping at A, is a maximum at B and rising at E. In 3 the pressure is down to atmospheric at A but rises to remarkable degree at D and again at E, despite the fact that presumably the initial wave had its amplitude halved at the bifurcation B. This phenomenon of a sound wave increasing greatly in amplitude when it reaches the stopped end of a pipe is of course generally referred to in works on acoustics (VI). It is useful to realize that such high pressures are met with and are quite in order. In 4 and 5 the pressure had dropped quickly, while in 6 it is negative all the way from A to D.

It is evident that the stopped length BE, which we call an 'interference pipe', has two main effects on the associated open length. First it diminishes the impact noise; secondly it alters the pressure variation at the exhaust port A, the sub-atmospheric part being much later than would be the case with a single pipe. The amount of this delay in terms of crank angle depends on the distance from the port A to the junction B and on the length of the interference pipe.

Obviously these latter dimensions are of considerable significance in designing a good system of a type in which the pipes merge at some distance from the ports; the foregoing experiments show conclusively that there is far more to it than merely a matter of ensuring unimpeded flow. In fact, within reason there is no objection to quite sharp bends, such as are sometimes necessary at the ports, to meet installation requirements.

Interference application

In applying the interference principle for practical use, the object is to obtain a favourable condition of low pressure over the scavenge period around t d c, and covering as wide an engine rev/min range as possible. At high engine speeds, using a single pipe of about 4 ft in length from the port, a good effect is obtained above about 3,800 rev/min, but with very poor results below 3,000 rev/min. At the latter speed the first return wave could be expected to be spread over the t d c period, thus impeding the scavenge. An obvious improvement would be to extend the positive portion of the initial wave, following this by a negative wave similarly extended. With an interference pipe, this may be achieved. Consider the pipe layout fig. 5 : 23 which shows an exhaust pipe of length BC fitted with a branch BA having its outer end blanked-off. The pressure pulse received at the entering end B will move both to the pipe outlet at C and also into the blanked-off branch BA, and a reflection from the blank end A will follow upon the pulse that is already part-way along

BC. There will thus be two wave peaks in sequence at b d c instead of one. This is shown on the lower curve, fig. 5 : 24. The pressures of the two wave peaks is lower than if the interference branch BA had not been fitted, and the following depression is smaller, but the important advantage lies in the fact that the depression is spread well beyond t d c. At 2,000 rev/min conditions are still good, as shown on the upper curve, fig. 5 : 24. This favourable condition will persist also to much higher speeds. Another good effect is that the two waves damp each other quite quickly.

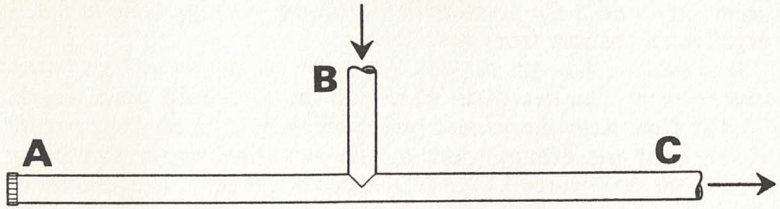


Fig. 5 : 23. Another experimental pipe system for investigation of 'Interference'.

The best experimental results have been obtained with an 18 in. long interference pipe attached to the exhaust pipe close to the valve-port; however, the length depends on the speed range required. If more than the dimension stated, the first wave is further extended, and the assembly would be suitable for a slower-speed engine.

Interference action

Now consider again the layout fig. 5 : 21. When the two exhaust pipes are merged as already described, there is a strong tendency for the 'working' branch (AB) up to the junction to vibrate in its own period, on top of the period of the whole length of the pipe; the same applies to the idle branch, or stopped 'interference' length. As stated, the influence of the length to the junction must affect the outcome of these phenomena and this has been investigated. In fig. 5 : 25(*a*) and (*b*), the dotted curve is the usual one taken at the engine port, the full curve being taken at the blank end of the interference branch. The remarkable point about the pressure rise at the blank end is the very short time taken for the pressure to reach its maximum. The calculated time for a wave to start from the engine and travel round the junction to the stopped end would be much greater, and quite logical for a wave only a few inches long; but the wave being dealt with extends over several feet of pipe, as already shown. What actually happens is that while the pressure is rising at

the engine end, it is also rising along the pipe and back up the other leg of the Y, so that the whole length is at a high pressure even before the maximum is reached at the engine. The high pressure

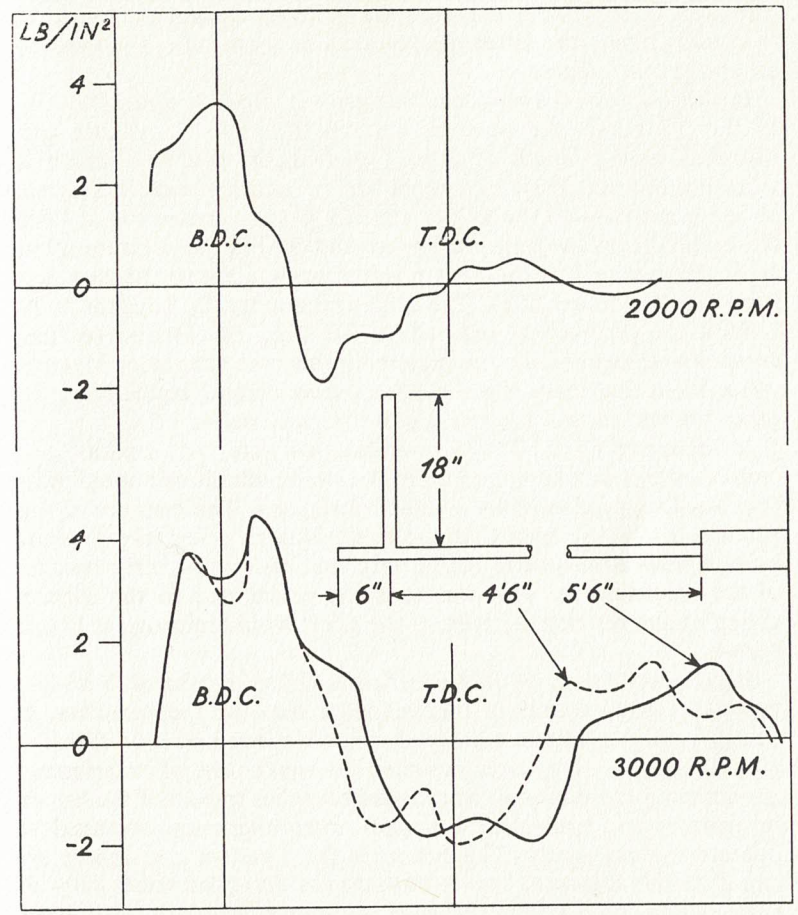


Fig. 5 : 24. Pressure recordings taken from system shown on fig. 5 : 23.

noted at the blank end has already been mentioned, and this persists for only a short time. The period of the small wave AB in fig. 5 : 25(*a*) is slightly shorter than that calculated for an 18-in. length. In (*b*), for a 3 ft 3 in. length, the periods vary somewhat but are approximately correct.

Test conclusions

The experimental set-up fig. 5 : 26 has given some useful information as it approximates to a quite popular type of divided system. The individual cylinder branches are quite short, and are joined to twin downpipes; the latter merge into one main pipe, followed by an absorption silencer.

In the first test, the pipe lengths were 4·25 ft for X and 3 ft for Y. In the full curve, the wave period AB from No. 1 cylinder corresponds to the length of pipe X and shows that this length is vibrating independently. There should be another wave following, of lower amplitude than B, but instead, a wave comes round from the branch from the centre cylinders and pushes the pressure up as at C. Following this the next positive wave adds to the new one from No. 4 cylinder at D. The same thing naturally happens at A.

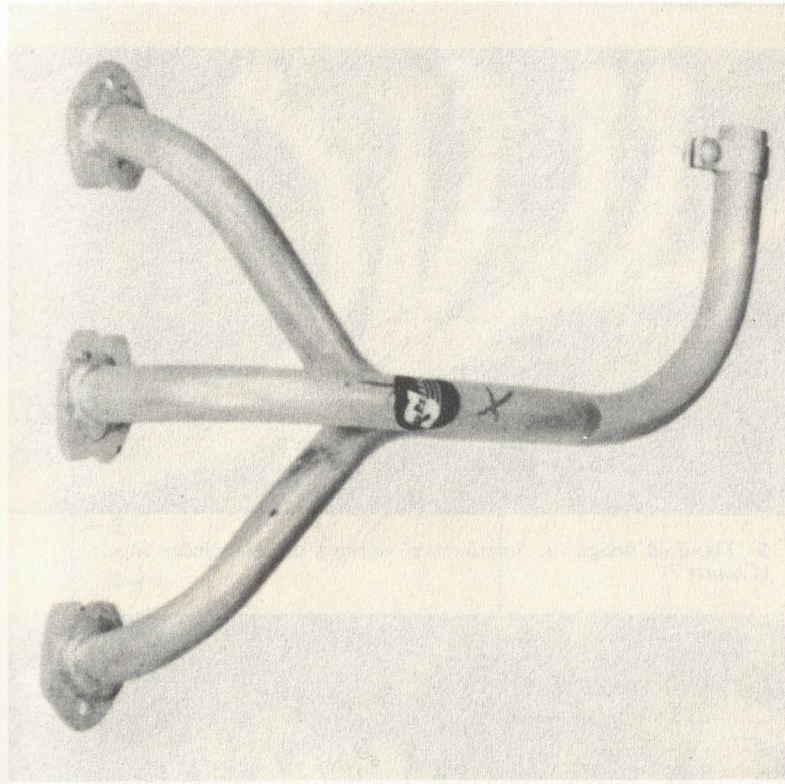
With the distance X reduced to 3 ft 2 in., the second (dotted) curve is obtained. While the periods in this case are rather obscure, it is evident that there is still a strong wave coming round from the other branch, so that the rise is still too early before t d c.

In the curve fig. 5 : 27 the two branches were kept separate and connected to a box having 15 times the swept volume of one cylinder, this box being followed by a standard silencer. The distance X was 6 ft 9 in. It will be seen that the effect is quite good, the pressure being below atmospheric before t d c, but rising too early because of the return wave. This was to some extent due to the silencer giving undue restriction; even so the effect would improve at higher speeds.

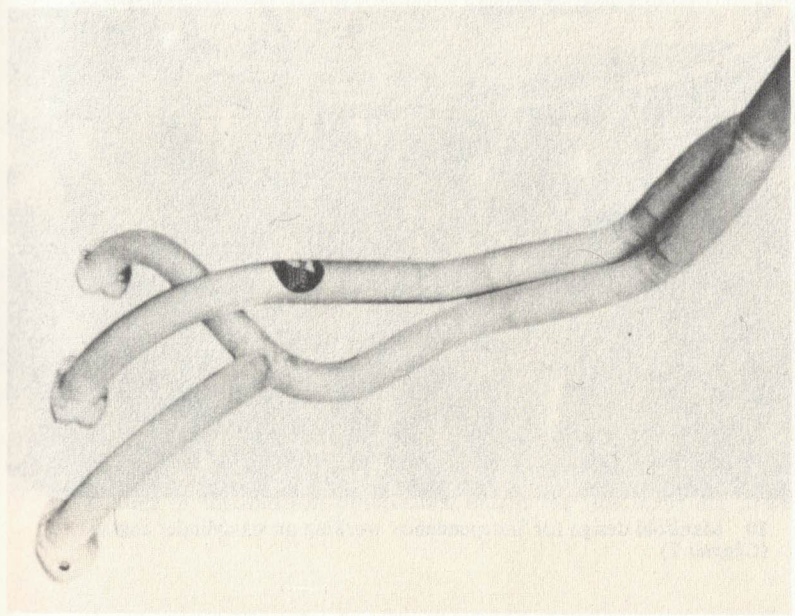
It was established by this experiment that the exhaust box does provide a good degree of independence between the branches, as shutting off one branch completely from the box had little effect.

A four-cylinder system designed to make use of interference phenomena would require its cylinder branches to be half the length, or less, of the equivalent branches when these are arranged to operate independently. The branches from outers and inners are united at this distance, the two downpipes being led either into an exhaust box followed by a silencer, or into a double-entry silencer incorporating such a box.

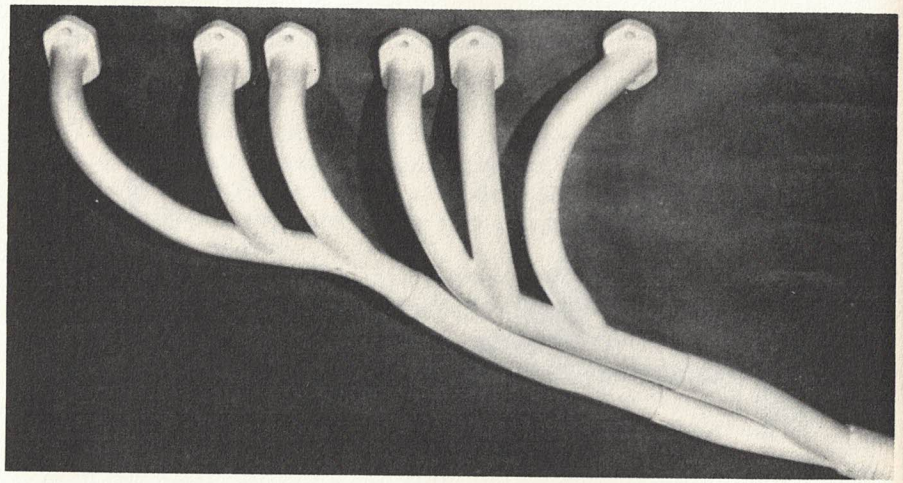
After the pairs of pipes coupling cylinders 1–2 and 3–4 have been united, the two resulting branches should not be joined together for a further appreciable distance. The system shown on plate 7 shows this feature whereas in that shown on plate 8 all the junctions are close together. In the latter case it is quite possible for the interference action to be prejudiced by a wave coming round the lower junction.



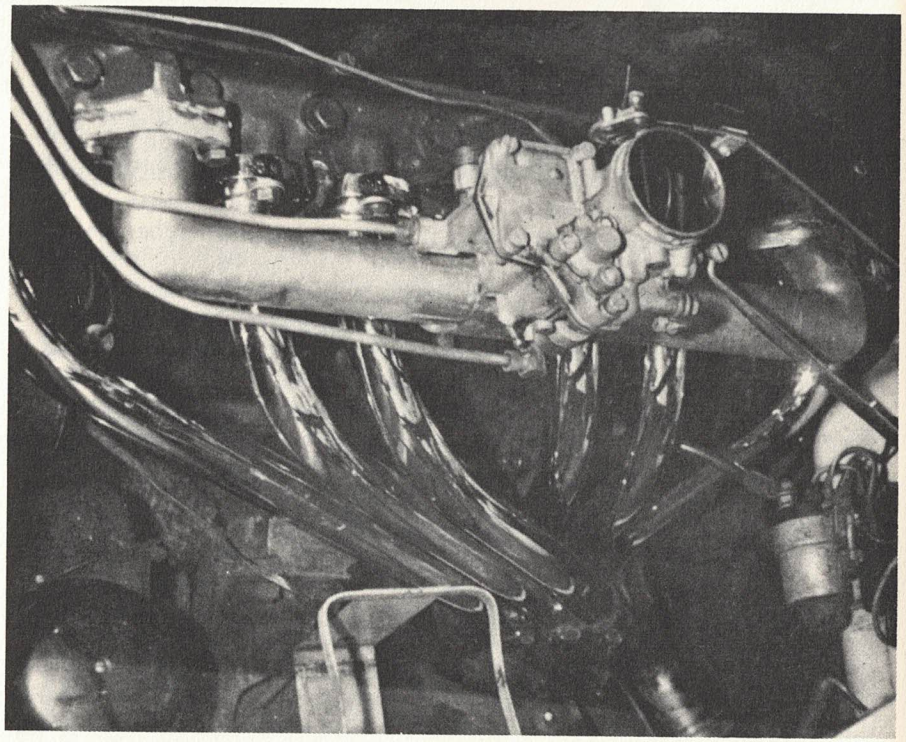
8 Manifold design for 'interference' working on four-cylinder engine with siamesed centre port (*Chapters 5 and 7*)



7 Manifold design for 'interference' working on four-cylinder engine with siamesed centre port. There is adequate pipe length before final junction (*Chapters 5 and 7*)



9 Manifold design for 'interference' working on six-cylinder engine (*Chapter 7*)



10 Manifold design for 'independence' working on six-cylinder engine (*Chapter 7*)

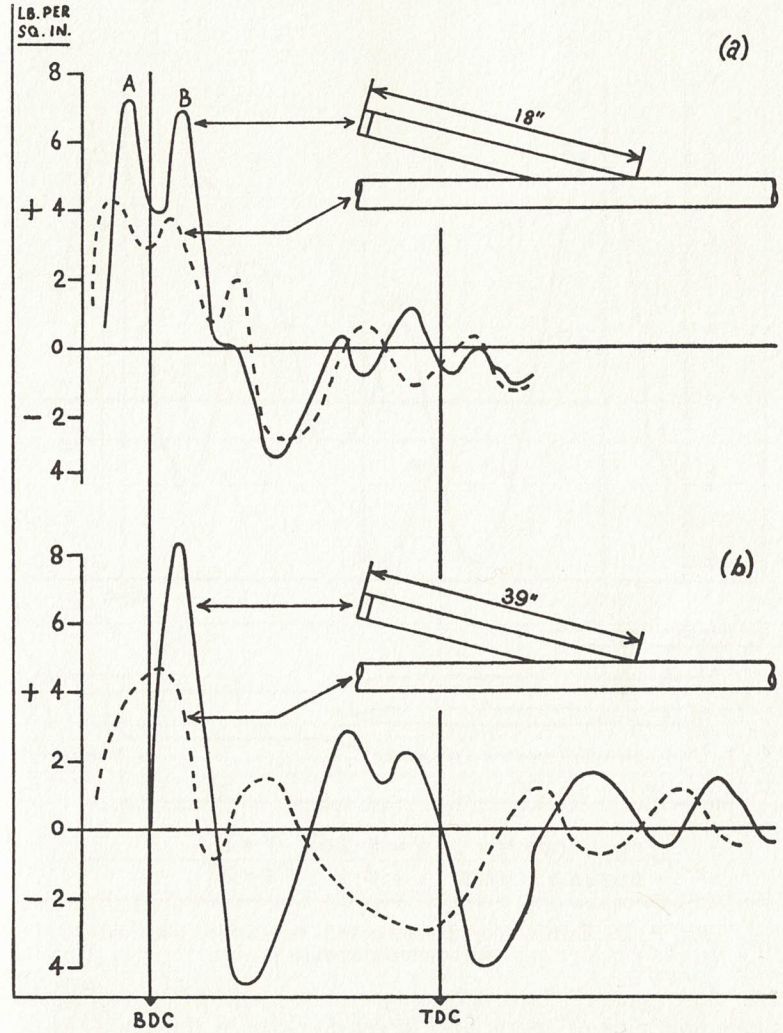


Fig. 5 : 25. Pressures at exhaust ports and at closed end of interference branch.

For example, with the exhaust valve of no. 1 cylinder about to close, such a wave could emanate from cylinder no. 3, whose exhaust valve is open and releasing gas at high pressure. It is obviously not always possible to arrange the pipe junctions in the ideal position because of installation difficulties, and if the junctions are unavoid-

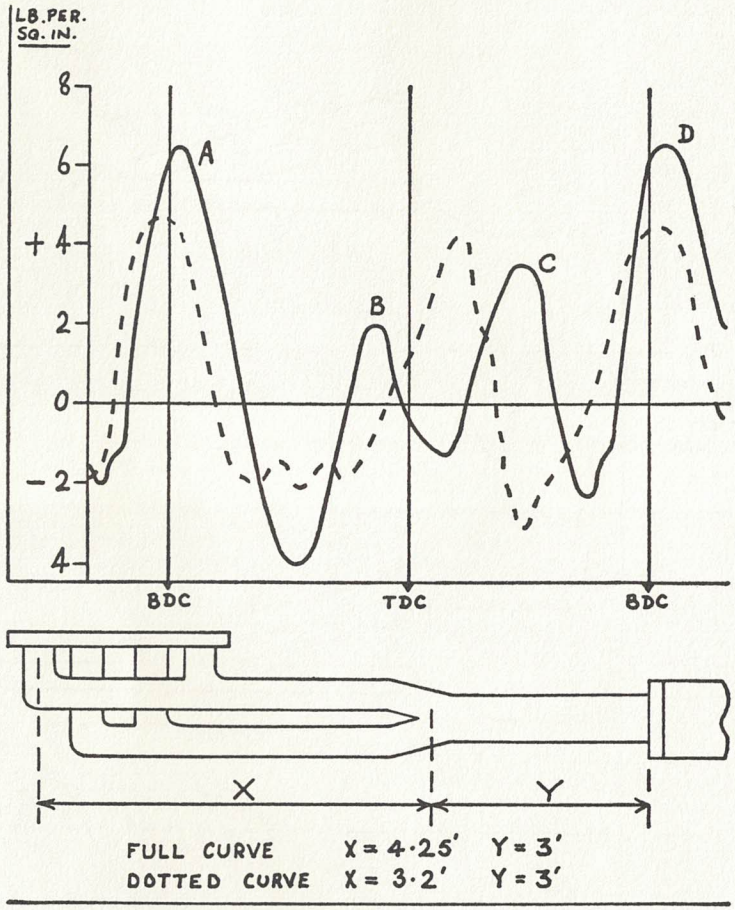


Fig. 5 : 26. Exhaust port pressures with sub-divided manifold—four-cylinder engine.

ably close together, the disadvantage can be mitigated by making the single final pipe of appreciably larger diameter at this point. More specific details of dimensions and layout are given in Chapter 7.

Six-cylinder engines

The objections to having six cylinders discharging into one manifold are obvious from the observations made in Chapter 1. Divided manifolds are essential but, even when these are used, the merging of the downpipes can have curious results and on the

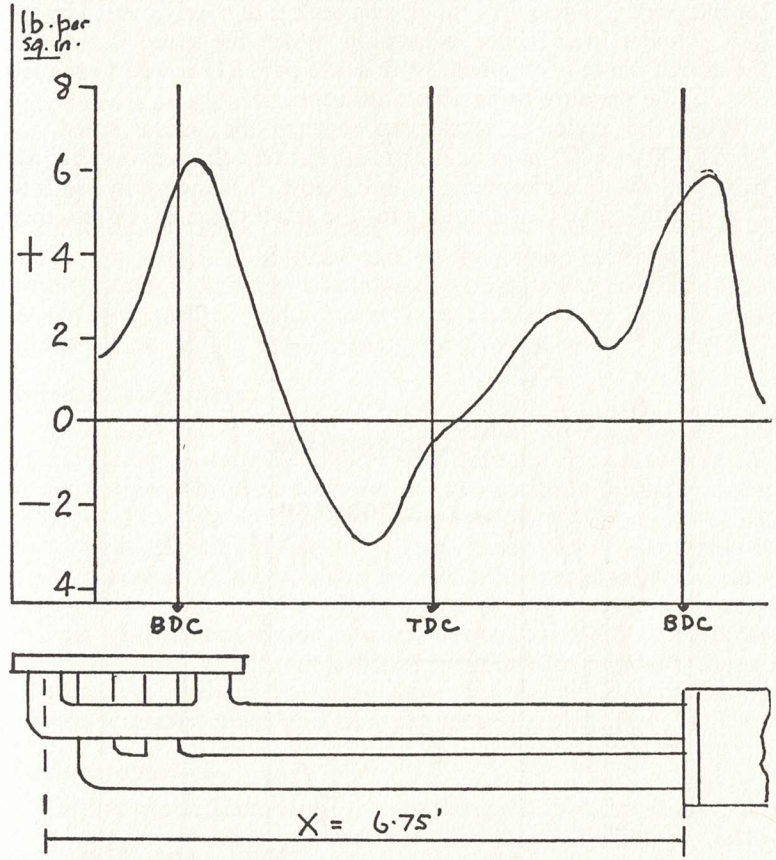


Fig. 5 : 27. Exhaust port pressures with separate down-pipes and box.

whole it is preferable to keep the two systems entirely separate if this can be arranged.

A good example of what can happen was shown in the case of a 3-litre high-speed car, on which the engine b m p suddenly increased from 116 to 126 lb/in² between 2,700 and 3,100 rev/min; while the surge of power when accelerating might not have been unwelcome, the disproportionate reduction in torque when slowing down in this short rev/min range was most disconcerting. Tests showed that the conditions as in the curves fig. 5 : 28 were obtaining. At 3,000 rev/min it will be seen that the peak at A is followed by a rapid drop to below atmospheric B, and that the rise from here to the

second peak C is also very rapid; this peak is of course caused by the next cylinder in sequence exhausting. When the speed is reduced, the dotted curve is obtained. In this, the part BD is well to the left of t d c, the pressure being above atmospheric.

When the engine is accelerated through the critical speed, *i.e.* from 2,000 to 3,000 rev/min, the pressure at t d c changes very quickly from well above atmospheric to well below. It is the rapid rise B to C of the full curve that accounts for the small range of rev/min over

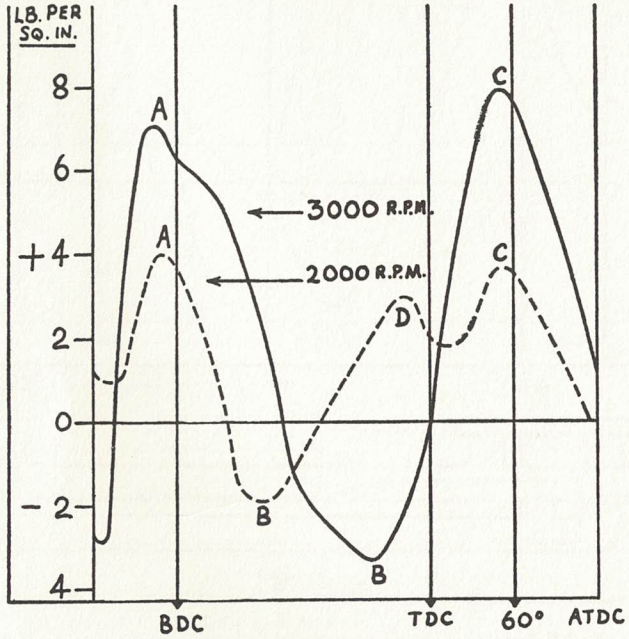


Fig. 5 : 28. Rapid change in wave characteristics over 1,000 rev/min engine speed.

which the change in b m p takes place; this is another manifestation of the fact already noted, that the wave coming round the junction of the pipes invariably causes a very fast rate of rise of pressure.

At 2,000 rev/min the point B is much earlier and a return positive wave is due to arrive somewhat before t d c, this being added to by the wave coming from the other downpipe and producing point D on the curve. Subsequently, this wave is in process of flowing out when the next cylinder in sequence begins to exhaust and thus the peak at C is much lower than in the previous case. Altering the length of the downpipe to the junction had little effect on the adverse

combination of waves. Complete separation of the two downpipes to give independence, either by means of an exhaust box or two completely independent silencers and pipe systems, is the logical answer.

Interference on a six

It is possible to incorporate in existing six-cylinder manifolds a device which will allow the advantages of interference to be obtained. Assuming that the engine has two cast-iron manifolds, each with three short branches, it is only necessary to connect a length of plain pipe, closed at its outer end, to the downpipe at its junction with the manifold. When a completely revised and branched system is involved, the port branches will be on the same lines as those shown in fig. 5 : 17. These layouts are given in detail in Chapter 7.

Effect on parallel-twin

An interesting modification to the exhaust system of a 650 cm³ twin-cylinder motorcycle engine with exhausting intervals at 360 crank degrees, involved coupling the two separate pipes by means of a cross-pipe located close up to the port outlets. The intention was to improve silencing by doubling the silencer capacity available to each individual cylinder, since flow in the cross-pipe would have such effect. However, the improvement in power output obtained by this simple modification was such as to indicate that the cross-pipe was functioning to some extent as an interference pipe, though the lessened resistance to mass flow as a result of doubling the silencer capacity could also have an advantageous effect.

Resistance method

Further experiments aimed at extending the initial pressure-pulse to advantageous effect, showed that this can also be done by introducing a resistance to flow in the exhaust pipe near the engine. As shown on fig. 5 : 29, the resistor device comprises a circular body with a length of 15 in. and diameter of 2·5 in. At the entry end of this, the pipe from the engine is reduced from 1·75 in. to 1·5 in. diameter for a length of 11 in., this length projecting into the larger tube. The pipe is plugged at its outer end, and drilled on the sides with 23 holes of $\frac{5}{16}$ in. diameter. The outlet from this device is connected to a conventional absorption-type silencer by a pipe 4·3 ft long and 1·75 in. diameter. As fig. 5 : 29 shows, the effect is similar to that of an interference system, in spreading the wave over t d c. With increase of rev/min however, it was found that the pressure increasingly remained above atmospheric to the end of the exhaust period because of the build-up of back-pressure on the greater flow of gas. As the

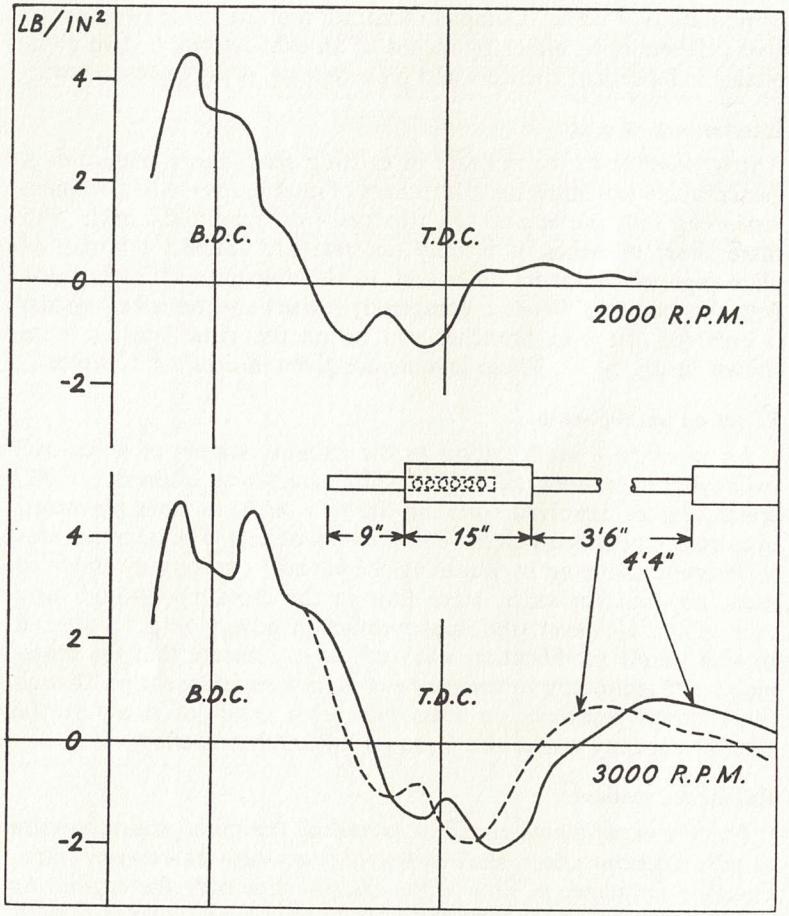


Fig. 5 : 29. Result of introducing resistance to flow.

curves show, this effect can be reduced by shortening the connecting pipe to the absorption silencer, and reducing the initial resistance, *e.g.* by drilling the entry pipe to the resistor with large holes. Though suitable for medium-speed engines, the scheme is evidently inferior to the interference system. It has the merit of simplicity however, and allows a standard exhaust manifold to be retained.

CHAPTER 6

SILENCER DESIGN

Sound reduction – The exhaust box – Restricted outlet – Absorption silencers – Design considerations – Tailpipe size – Silencing motor-cycle-type engines – Back-pressure silencer – The projected design – Test of back-pressure silencer – Further experiments – Addition to the 3rd Edition by John C. Morrison

Sound reduction

THE main problem in silencing is to reduce to an acceptable degree the noise emitted by the exhaust gas discharge, without prejudice to effective cylinder scavenging. Since even today many designs of silencers appear to have evolved on remarkably hit-and-miss lines, it will be useful to consider the theories applicable to the problem and how these can best be applied in practice.

It is authoritatively stated (II) that a sound can only die out in an enclosed space by the action of true dissipative forces, such as thermal conduction and viscosity; merely modifying the sound waves by irregularities is of no avail. Yet it will be obvious that there have been manufactured, and sometimes patented, silencers containing devices that attempted to do precisely the latter; their operation being described in terms of 'breaking up the sound'.

As regards thermal conduction, it should be noted that sound dissipation by this means does not simply mean cooling of the gas. Many engines eject water from the tailpipe after a cold start, which indicates a temperature low enough to condense the water vapour present. The effect on the noise is however negligible and hence any claim that a worthwhile noise reduction can be achieved by cooling can be discounted. In practical terms, silencers of the type with which we are concerned reduce the noise by friction, which converts the wave energy into heat.

In the earliest types of industrial engines, it was customary to lead the exhaust pipe into a brick-lined chamber or pit dug in the ground, from which a vertical pipe discharged the gas into the atmosphere at high level. An alternative was to have a large cast-iron 'pot' on the end of the pipe (I). In either case, the reduction in noise was due to confining the waves in a large space until they had spent their energy;

but obviously chambers of adequate size cannot be accommodated in vehicles and other means must be found.

Viscosity is a familiar enough term when applied to liquids, but is of equal significance in considering gas movement and the friction resulting therefrom. If we imagine a sound wave passing between two smooth parallel surfaces separated only by a small gap, the viscous effect can be made clear (see fig. 6 : 1). The layer of air in contact with each of the surfaces will be stationary, while at the centre distance, or halfway across the gap between the plates, there will be minimum drag on the air-flow. There is thus a velocity gradient caused by viscosity, varying from zero at the plates to maximum at

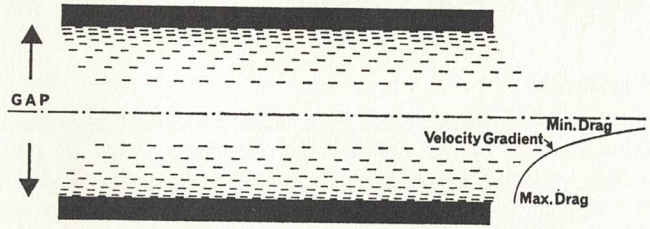


Fig. 6 : 1. Principle of viscous drag on sound wave.

the point where drag ceases to have an effect. The distance to which the drag penetrates is stated (II) to be given $h = 1{\cdot}29 \div N^{\frac{1}{2}}$ where the distance h is in mm and N is the frequency of the wave. From the above, it will be seen that if the plates are separated by a distance equal to $2h$, there will be no free movement of the air at all and the wave energy will be rapidly lost in friction. It is of interest to note a few values of h for different frequencies.

Frequency N	64	200	400	900	1,600	2,500
Distance h (mm)	1·6	0·91	0·6	0·4	0·32	0·26

This effect is made use of in silencers having small holes through which the gases have to pass; the smaller the holes, the better the damping. Typical sizes are:

Frequency N	64	240	900
Hole diameter (in.)	0·125	0·0625	0·0313

There are of course practical considerations regarding minimum hole size, which will be detailed later.

The exhaust box

The plain exhaust box, or expansion chamber as it is still called (incorrectly in most cases), probably originated as an attempt to

employ the traditional method of stationary-engine silencing to vehicle use. To make clear the matter of alleged 'expansion' of the gases in such a box, it will be evident that if the cylinder pressure at the time of exhaust valve opening is 60 or 70 lb/in² and the pressure in the pipe is around atmospheric, sudden opening of the valve will very quickly reduce the pressure, which is expended by propelling the gas through the valve port and into the pipe; in other words, the gas expands into the pipe. As is obvious from the indicator diagrams already reproduced, the pressure on the 'pipe' side of the valve is never more than a few pounds above atmospheric, even under the worst conditions of back-pressure plus return positive wave action.

The only correct usage for the term 'expansion chamber' would be in the case of an exhaust box fitted straight on to the cylinder exhaust

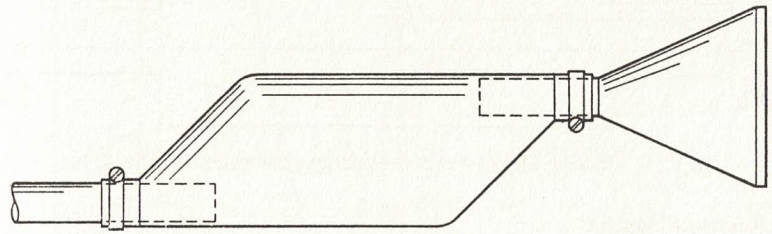


Fig. 6 : 2. Section of Brooklands silencer.

port, as is sometimes done in high-efficiency two-stroke engines; in such a case, the high-pressure efflux from the port does indeed expand directly into the chamber. In no other case, and decidedly not when the chamber is placed at the outer end of several feet of pipe, can the name be justified. However, the only reason for stressing the point here is to ensure that the device will not be credited with virtues that it does not possess, particularly in regard to silencing.

The attraction of the exhaust box from a silencing point of view is the virtual absence of any restriction to the gas flow, in the way of baffles etc. This was no doubt the reason for the adoption of the type by the Brooklands authorities when silencers were made obligatory shortly after the first world war (to their credit, they called it a 'receiver'). Even in this case, when it might have been assumed that the most expert opinions were called for regarding design, there is plenty of evidence of bewilderment in regard to flow conditions as, for example, the insistence on the inlet and outlet pipes continuing into the chamber for a specified distance, and being offset relative to one another – presumably to counteract the possibility of the gas leaping across from inlet to outlet, without 'expanding'. The degree

of silencing obtainable from a single plain chamber is quite small. Its effect can be ascertained by considering a box as shown in fig. 6 : 3. Ignoring the concept of a continuous flow of exhaust gas, imagine a pressure wave entering at A. The chamber will be filled at reduced pressure and thus if the inlet A and outlet B are of equal area it might be supposed that the wave energy would 'divide' and would return through A and pass out through B. This however will only occur if the chamber volume is sufficient to accommodate the whole of the wave (putting it in simple terms), but a box of this volume would be difficult to house particularly on a small car or motorcycle.

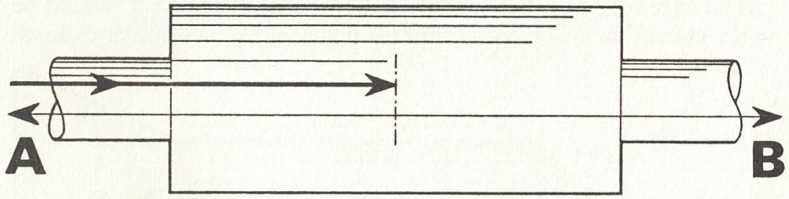


Fig. 6 : 3. Division and exit of wave energy after entering exhaust box.

Restricted outlet

The Brooklands silencer specification insisted on the addition of a fishtail outlet further to diminish the noise, and tests carried out (XXXIII) on a chamber with and without this fitting confirm what was suspected, *i.e.* that the fishtail had considerable effect in this direction. The curves fig. 6 : 4 show at (*a*), (*b*) and (*c*) respectively the sound wave with an open pipe, with box only and with box and fishtail. The reduction in wave amplitude compared with (*a*) was 25 per cent for (*b*) and 52 per cent for (*c*). The box used for this test had a capacity of 10 times the swept volume of the cylinder, but it is important to note that the outlet pipe was slightly greater in area than that of the inlet. Fitting the fishtail reduced the outlet area (via the fishtail slot) to 18 per cent less than the inlet, which partly accounts for the damping of the resultant wave.

It is quite evident that a reduction in the silencer outlet pipe area is relied on for the silencing effect to a very large extent, irrespective of the type of silencer.

Absorption silencers

The construction of the absorption type of silencer is generally familiar. The perforated main tube passing through the outer casing is usually encased in a glass-wool jacket contained in the annular

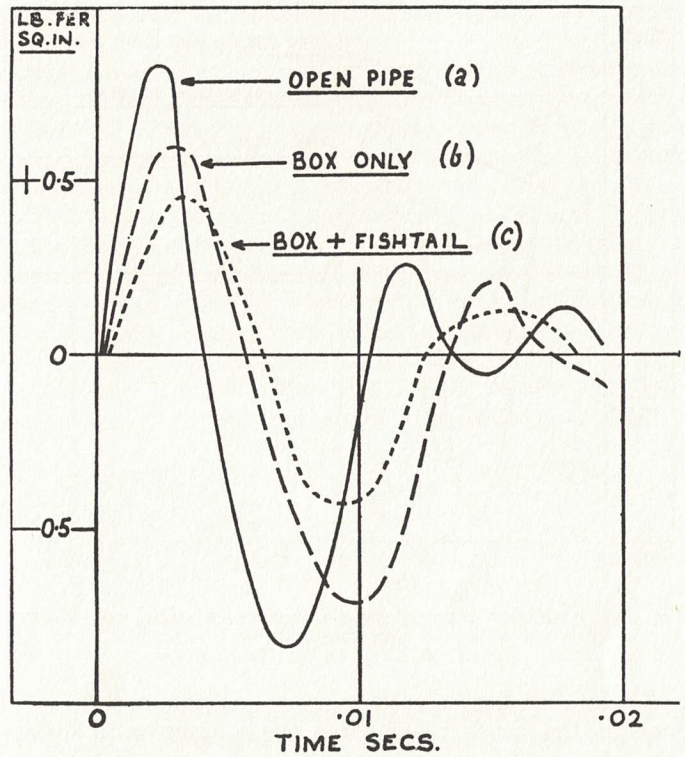


Fig. 6 : 4. Reduction in issuing sound wave with exhaust box.

space; other substances are sometimes used, such as steel shavings and wire-mesh. The operating principle is that the sound waves passing down the main tube penetrate into the close-packed absorbent material via the perforations and are there frictionally damped; the damping is usually very effective on the highest-frequency waves. The curve fig. 6 : 5 shows a test of a standard absorption silencer having a length of 18 in. and diameter of 4 in., the central pipe being 1·75 in. diameter. The reduction in wave amplitude is 26 per cent, while two similar silencers in series gave 61 per cent. However, the apparent noise reduction to the listener was much better than the figures suggest, because of the elimination of the high-pitched waves.

It is important to note that reducing the wave amplitude does not appreciably lengthen the period of the issuing wave (contrary to a

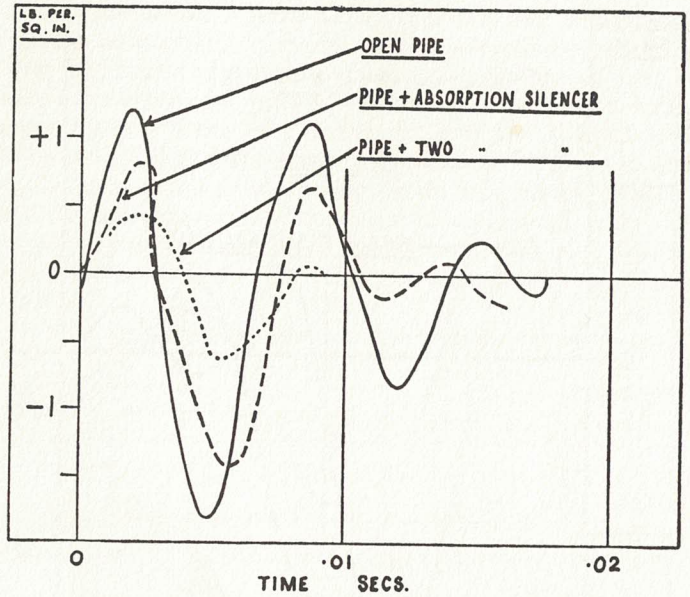


Fig. 6 : 5. Reduction in amplitude of sound waves issuing from absorption silencers.

general belief that damping of a harmonic motion must do so). This accounts for the fact that high-speed cars equipped with absorption silencers often emit a musical hum, which is caused by the presence of the individual pulses at sufficiently high frequency to provide the 'music'.

Design considerations

As shown in explaining the viscosity principle, the smaller the holes through which the wave has to pass, the greater the degree of frictional damping. However, irrespective of the silencer type such holes must not be liable to obstruction by deposits and this limits their minimum size. In the perforated main tube of an absorption silencer there is not so much tendency for fouling as when the holes are directly in the gas stream, but even in this case, 0·11 in. is the minimum desirable diameter, at a pitch of 0·25 in. If the holes are 'stabbed' towards the absorbent jacket instead of being clean-drilled, as shown in fig. 6 : 6, the resulting nozzle effect facilitates the flow into the jacket, but obstructs its return to the main throughway. This effect though useful must not be overrated since it appears that a wave can

pass through a hole with much greater alacrity than is the case with a continuous flow of gas.

As regards the quantity and disposition of the absorbent material, the energy content of the long wave may occupy anything up to 6 or more feet of 1·625 in. bore pipe at a mean pressure of say 3 lb/in² above atmospheric. If such energy is to be absorbed by the movement of a small proportion of gas passing into and out of the absorbent material, it will be clear that a considerable thickness of the latter is called for. This thickness or depth is represented by the difference in radii between the silencer casing and the main pipe; it should not be less than 1 in. and preferably over 1·5 in. While consistency of depth is desirable, which means a circular-section casing, it is often preferable for reasons of ground clearance to use an oval section. So long as the minimum depth is maintained there is no objection, but flat surfaces should be avoided because of their tendency to drum.



Fig. 6 : 6. 'Stabbing' of holes in main pipe wall gives superior effect in comparison with clean drilling (right).

For the same reason the outer jacket should be made of metal having an adequate thickness, particularly as it is also subjected to the ravages of moisture inside and out. Some constructions have incorporated a double outer container with walls separated by a small air space, the idea being to damp out drumming. Apart from the possibility of a minor bulge causing actual contact between the double skins, and thereby almost certainly adding to noise by resonant vibration, such relatively complicated anti-drumming measures are not considered to be worth the extra manufacturing difficulties and cost by many makers. A well-designed single-wall container should do all that is necessary to meet most requirements.

Tailpipe size

It is often thought that, because the main throughway is unobstructed by any baffle devices, the absorption silencer has no back-pressure effect. This is not true in practice even when the bore is unaltered from inlet to outlet. Some back-pressure must be created by the internal friction in the bore, however, with a constant diameter throughway there will certainly be very little but there may also

be insufficient silencing effect unless additional means are incorporated in the system. If the bore of the outlet and tailpipe is less than that of the inlet, the silencer can no longer be regarded as 'straight-through'. The reduction in outlet diameter obviously sets up an increased resistance, but its effect encourages the wave motion in the jacket and this, along with the smaller bore outlet, reduces the noise.

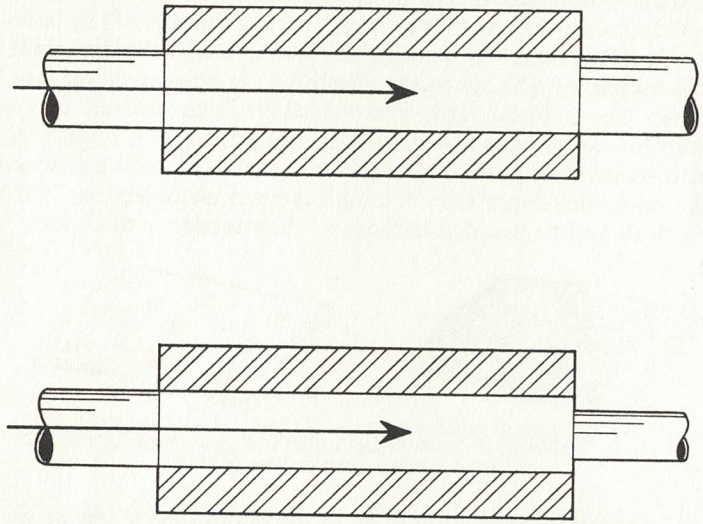


Fig. 6 : 7. Constant diameter throughway (top) gives minimum flow resistance, but reduced diameter outlet (bottom) adds to silencing effect.

If the reduction in outlet is kept within reasonable bounds, say from 1·625 in. entry to 1·5 in. exit or equivalent percentage, there should be no noticeable loss of power (see fig. 6 : 7).

Many designs incorporate plain chambers adjacent to the silencer proper and combined in the main casing. The intention of these is generally to reduce or eliminate the increase in noise that sometimes

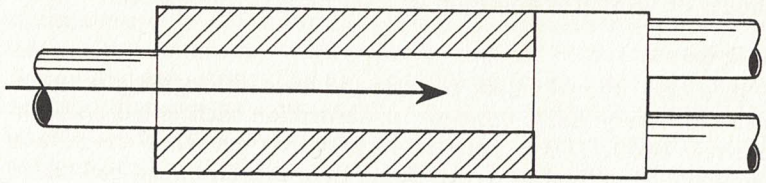


Fig. 6 : 8. Chamber incorporated in main body may improve silencing at periods of resonance, and also facilitate use of twin tailpipes.

occurs at certain engine speeds and superimposes a kind of 'hammering' resonant note on top of the normal smooth noise. Such chambers may also enable fashionable trends to be met, such as the use of twin tailpipe as shown in fig. 6 : 8.

Silencing motor-cycle-type engines

With its fewer cylinders and usually greater valve overlap, the motor-cycle engine imposes peculiar silencing problems. Since the engine type is also widely used in small cars and wheeled or tracked mechanical equipment it will be useful to detail some experiments carried out (XXXIII) and consider the possibilities of acting on these to improve on existing silencing standards.

Because of the limited room available in almost all cases, most motor-cycle silencers are of tubular type measuring from 12 to 18 in. in length and 3 to 3·5 in. in diameter. While the absorption type is not infrequently used, it is probably the worst possible design for such a purpose. Apart from the small size necessary, the intermittent nature, high velocity and sometimes oiliness of the gas discharge is prejudicial to both effective noise reduction by this means and to a satisfactory silencer life. Cases have been noted (XX) where, after an initial serious increase in the noise emitted, an examination of the silencer showed that the absorbent material, after gradual disintegration, had been blown out of the tailpipe via the perforations in the main tube.

Various alternative designs are however available, more robust internally and having baffles or shaped passages which 'direct' the gas flow. Whatever the design – and some are quite effective in most respects – the internal construction reduces noise simply by holding back part of the wave. Hence the type will now be referred to as a back-pressure silencer.

The first two curves in fig. 6 : 9 were taken (*a*) at the exhaust port and (*b*) at the entry to the silencer, with a back-pressure silencer assembled on the 500 cm^3 experimental engine referred to in Chapter 2. It will be seen that the initial wave A travels to the silencer, piles up at B, is reflected back to the engine at C, reappears at the silencer at D, and back to the engine again at E. The wave energy is thus partly dissipated in friction, the remainder escaping through the silencer, but not all at once. At higher engine speeds there is not time for more than one wave reflection, and if the second pulse has time to get away before the exhaust valve closes after t d c there is little harm done, though some 'negative' work will inevitably be added to the piston's exhaust stroke. This is clear from curve (*c*).

A remarkable feature of many such tests (XXXIII) with back-

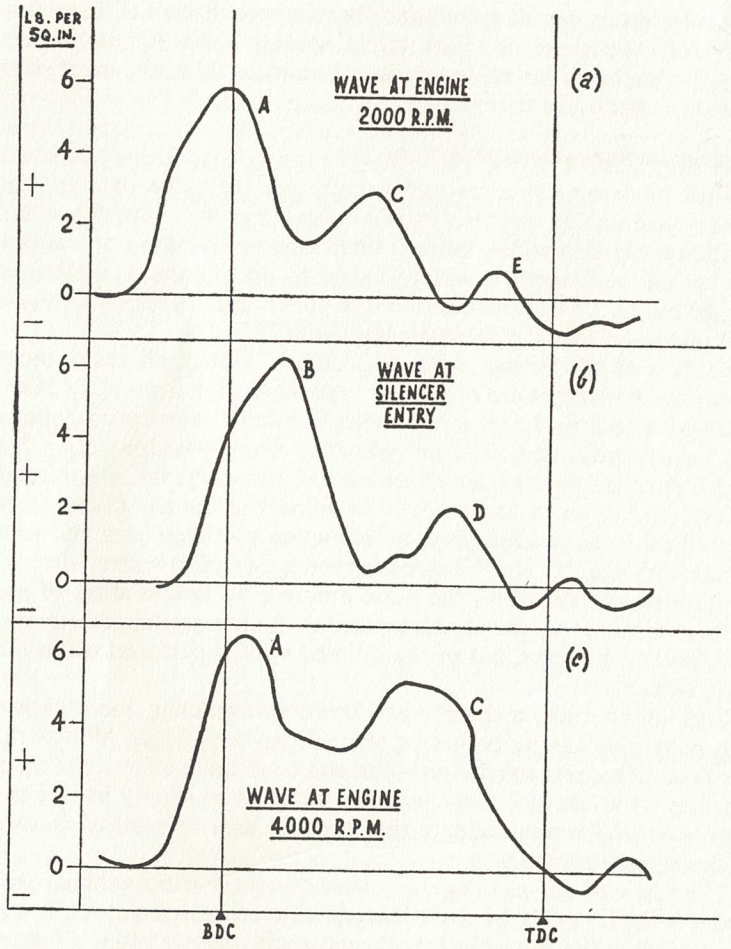
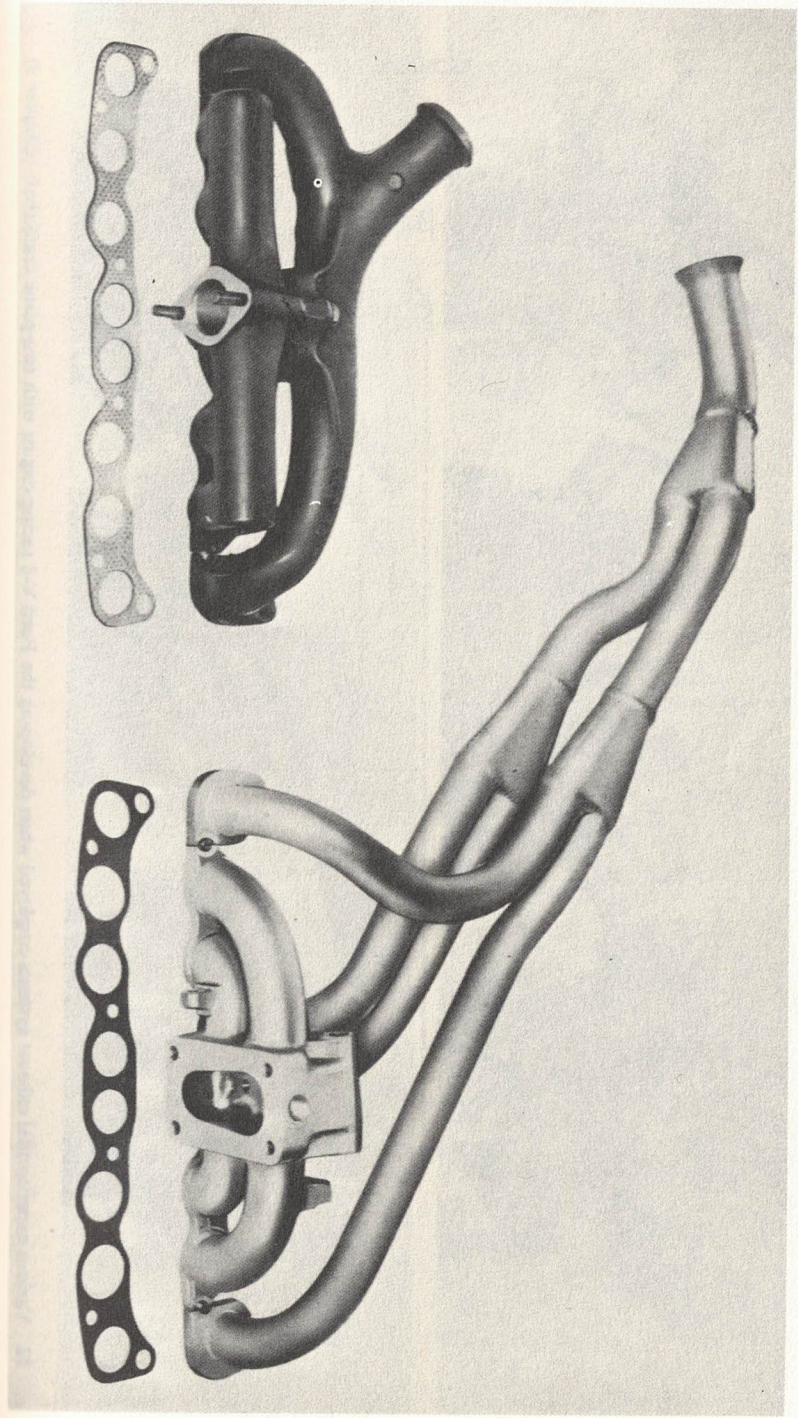


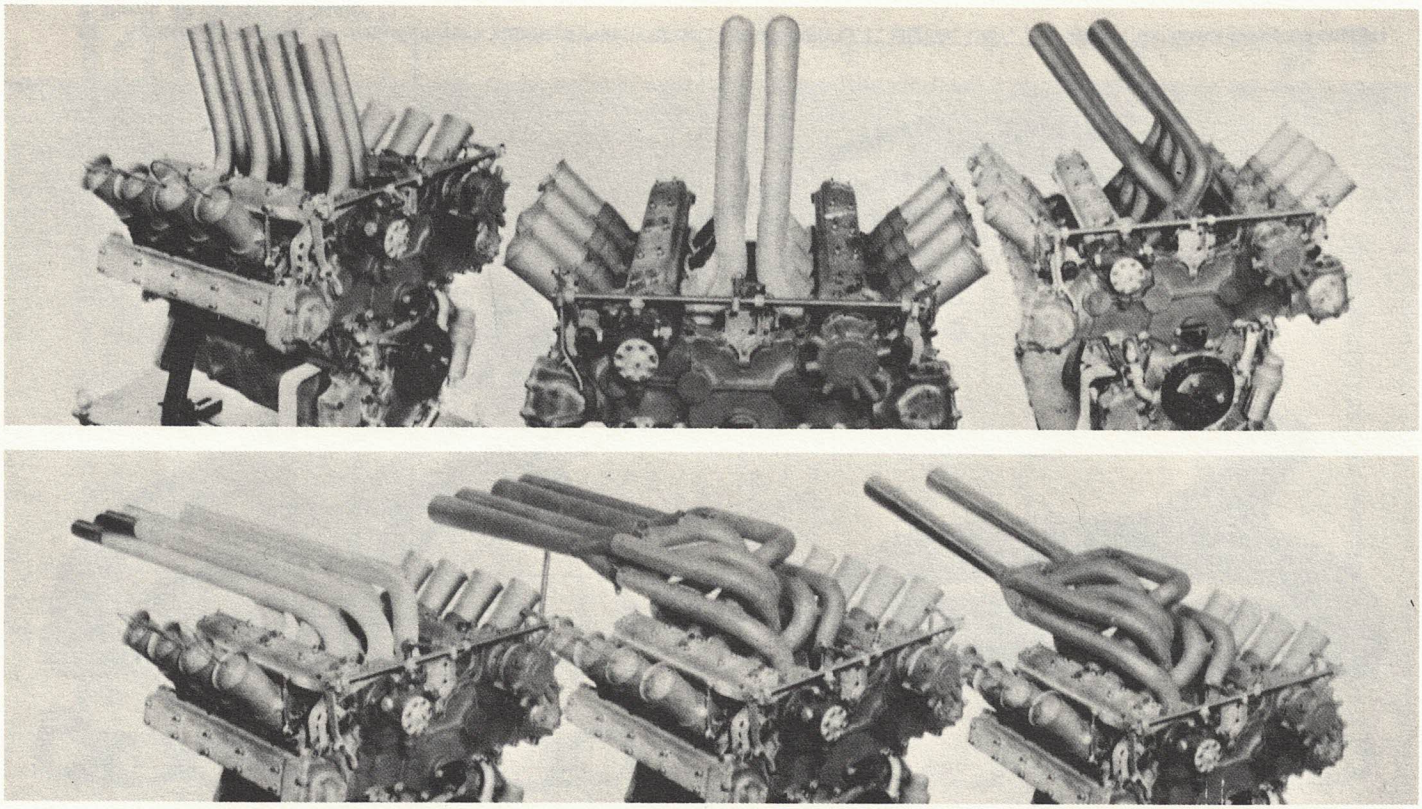
Fig. 6 : 9. Test of back-pressure silencer on 500 cm³ engine.

pressure silencers having widely varying internal construction is that all gave practically the same pressure curves and even similar noise curves, though to the ear the sounds were not the same. This was evidently due to the presence of high-frequency waves which some types suppressed better than others. Some of the types tested are shown in section in fig. 6 : 10.

It is quite feasible to design a practical back-pressure silencer which operates on logical principles and is at the same time sufficiently



11 An excellent design of manufacturer's standard manifold on Ford Corsair G.T. engine, 1967. The system for lower powered engines is shown on the right (*Chapter 7*)



12 Various experimental exhaust systems employed when developing the Ford V-8 racing engine with two-plane crankshaft (*Chapter 8*)

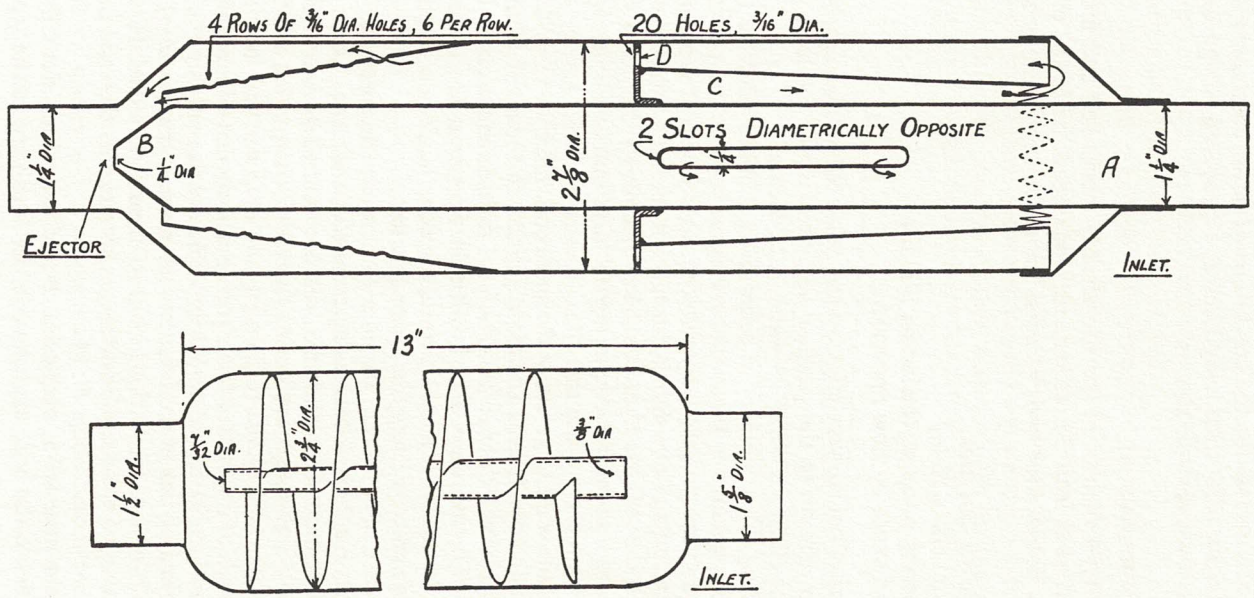


Fig. 6 : 10. Two types of back-pressure silencer of good design. Top – Scott-Howarth; bottom – Carbjector.

compact and robust for the conditions of usage common with this type of engine. The reasoning behind a proposed design and the design itself will now be shown.

Back-pressure silencer

Assuming a silencer of the size already mentioned and mounted at the end of at least 3 ft of exhaust pipe (measured to the cylinder port), the possible methods of noise reduction can be designated as follow:

a. Reduced exit area.
b. Chambers in series.
c. Tapering passages (including fishtail).
d. Interference by flow reversal.

The first is of course very effective but noise reduction must be considered in relation to the gas-flow and checking of the long waves, so that the effect on the short waves is incidental.

Regarding (*b*), chambers in series are far more effective than a single box, presumably because short waves become trapped in each chamber and are progressively reduced. In the case of tapering passages (*c*) including horn-shapes, the motion of sound waves in conical horns and curved passages has been extensively studied (VII). It appears that a sound wave entering the large end of a cone is partly reflected from the surface and thus returns on itself, leaving a proportion of the wave to go out of the small end. The increased intensity of this latter wave is thus not in proportion to the areas of the two ends of the horn. For example, when the ratio of end-areas is 7·8 : 1, the ratio of intensities is 3·1 : 1; with the former at 33 : 1, the latter becomes 7·9 : 1. This shows that as the cone angle is enlarged, the amount of reflected energy becomes greater.

In the case of a fishtail, its sides form a tapering passage which may or may not have a decreasing cross-sectional area. Experiments showed that even when the area was constant there was a distinct reduction in high-frequency noise. There is no doubt that the fishtail is a most effective adjunct to silencing, on both car and motor-cycle engines, and it is a matter for regret that it tends to be a victim of current fashion. In regard to (*d*) the difficulty in applying the interference principle lies in the fact that the noise is composed of notes of different frequencies, with which it is impossible to deal. However, a fair amount of noise suppression was obtained by the experimental arrangement shown in fig. 6 : 11.

In this the pipe AB is plugged at B and has a long slot CD in its wall parallel to the bore axis. A cover EF was fixed to the pipe, the

end E being open and that at F closed. The intention of the set-up was that a wave passsing from C to D as it issued from the slot would 'interfere' with the part that had already come out and which was returning under the cover from D to C.

With the cover positioned as shown, and the open end E facing the entering end A of the pipe, there was a definite decrease in sound. If however the cover was reversed so that the open end E was at the plugged end B of the pipe, there was no decrease. This would seem to show that the interference action did reduce the noise considerably and this was confirmed when tested on the engine. Unfortunately a good length is necessary for much effect but even a short

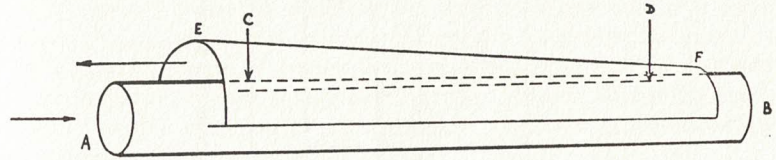


Fig. 6 : 11. Arrangement for investigation of reverse-flow silencing action.

length, such as could be put in a silencer, has some influence and is worth it on the principle of every little helping.

The projected design

The combination of the principles used in designing the back-pressure silencer shown in fig. 6 : 12 will now be detailed. The gas enters at A and emerges from the funnel B through two slots one on each side, into the cone-piece C. It escapes at the clearance-point D and leaves the outer chamber through a hole E, whence a perforated tube leads it to a fishtail F.

As already mentioned the pressure pulse from the engine has to be reflected back thereto and this reflection must take place as early as possible; consequently the volume of the first portion or parts of the silencer must be kept small. It is important in any silencer that none of the entering sound waves, whatever their frequency, should be able to impinge directly on the outer casing, unless this is of much heavier metal than is usual; the amount of noise that will pass through a metal wall is amazing. Thus, we have the gas first striking the funnel and then reversing through the slots into the cone C. The first real check comes at the escape point D where the passage area is 1·4 in^2. Both the gas and the sound waves are thus prevented from striking the casing with their full energy. In passing from D to E the

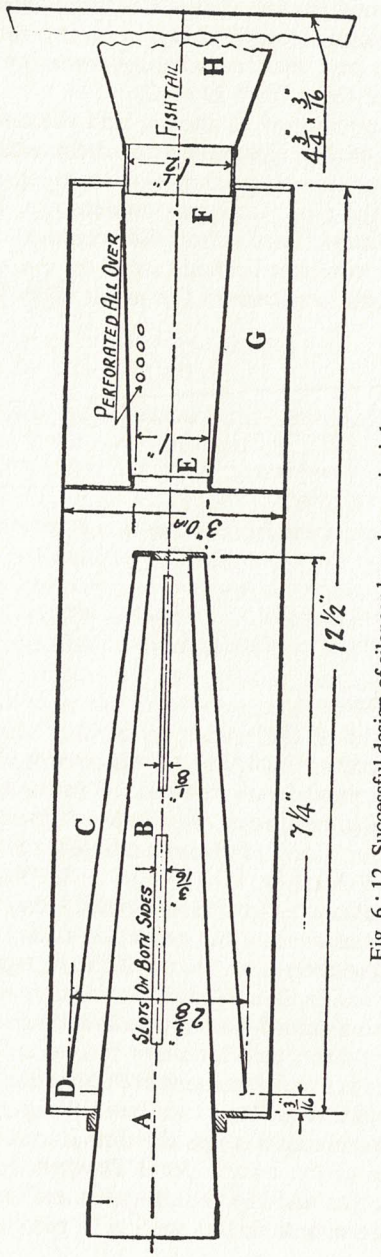


Fig. 6 : 12. Successful design of silencer based on principles enumerated.

gradually increasing cross-sectional area has the advantage of offering less inertia resistance to the initiation of flow.

The passage area at D is determined by its clearance of $\frac{3}{16}$ in. from the casing endplate. The clearance was arrived at by mounting the first assembly of the silencer, less the casing, on the exhaust pipe and running the engine at full throttle and 2,000 rev/min. As the cone C could be pushed near to, or away from, the end-flange, so the clearance could be varied. The amount of noise and the back-pressure as shown by the indicator were simultaneously observed and a compromise had to be struck between these two conflicting items. Less clearance than $\frac{3}{16}$ in. is undesirable, since any reduction from this due to deposits will have disproportionate effects. A little more clearance is of course in order, providing the noise level is kept within bounds. Fitting the casing with its baffle and central hole E adds considerably to the wave because of the small diameter of 1 in. at E. This diameter is probably on the small side but the object was to obtain the maximum possible silencing effect without the back-pressure becoming too great to be acceptable.

The final consideration apart from the fishtail was the small chamber G left at the end of the casing. It was obviously undesirable to add anything that would further increase the back-pressure and the solution was to connect E to F by means of a cone-piece perforated on its walls with holes $\frac{3}{32}$ in. diameter, which added a useful amount of damping.

Test of back-pressure silencer

Two of the silencers fitted to pipes 3 ft in length on a 500 cm³ o h v engine having two exhaust ports from a single valve gave the curves shown in fig. 6 : 13. It will be seen that the reflected wave produces considerable back-pressure during the latter part of the exhaust stroke, but gets away leaving sub-atmospheric pressure at t d c. This accounts for the fact that the torque measured on the brake was better than without the pipe or silencer, though of course an open pipe of a selected length for the speed would have given a higher figure. In any case, at speeds over 4,000 rev/min, the reflected wave was too late to give atmospheric pressure at t d c as will be seen by the curve in fig. 6 : 14.

When investigating the curves an interesting feature is that the brake torque values with the silencer in operation give a fairly flat curve, whereas an open pipe varies considerably. At 1,500 and 3,200 rev/min the silencers were better than the open pipes, but were not so good at 2,200 rev/min, or 4,000 rev/min and upwards.

To improve high-speed operation the only measure is to shorten

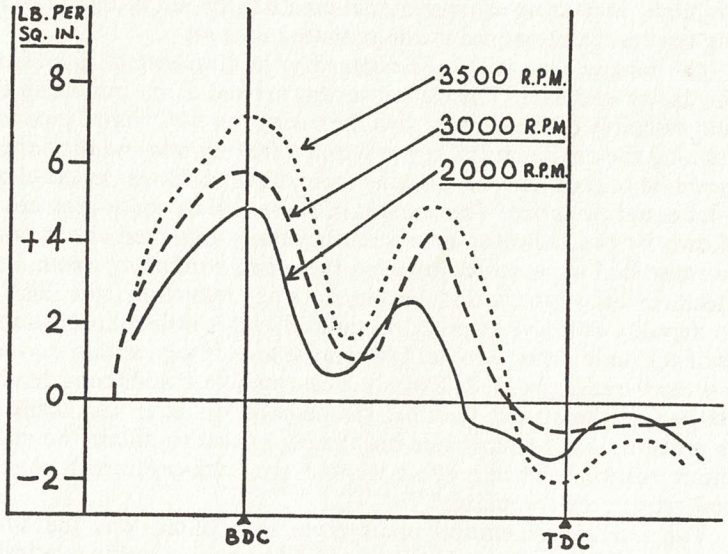


Fig. 6 : 13. Pressure curves at exhaust port with Morrison silencer in position.

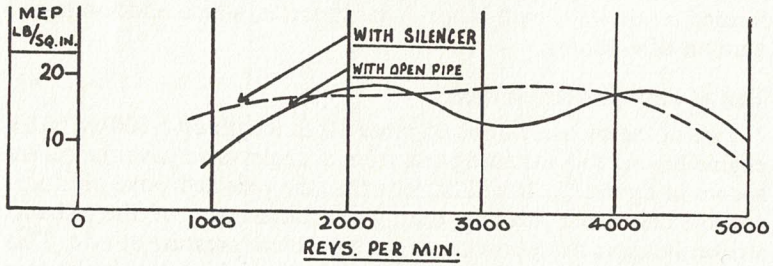


Fig. 6 : 14. Variation of mep with open pipe in comparison with silencer.

the pipes, which might make silencer installation awkward unless special provision is made. This might require an acute bend in the pipe near the exhaust port but as the pressure is high at this point the flow would not be affected appreciably.

Further experiments

In further developing this silencer, there are several possible lines on which to work. The fishtail could have an adjustable outlet slot to enable its area to be increased or decreased without damage. In this

way a good balance could be struck between the degree of silence and the loss in torque. It is however bad practice to drill holes in the fishtail sides or to indulge in intricate shapes for the slot outlet; an outlet area exceeding 1·5in² is unnecessary. The diameter of the hole at E could be increased to $1\frac{5}{16}$ in., this giving the same approximate area as at D. It will be obvious that experimenting with these aperture sizes is very largely a trial-and-error matter unless a test bed is available, but this should be no discouragement to those who are willing to take the trouble, using road performance, *vis-à-vis* noise, as the deciding factor.

Addition to the 3rd Edition by John C. Morrison

Silencers

This is not a treatise on silencers: that would need a whole book by itself, there have been so many of them.

Judging by the variety, it looks as if silencer design were more of an art than a science. In point of fact, the best silencer that the writer ever tested was made by a man who said he didn't know the reasons for his design – he just made it like that.

While in no sense calling himself a silencer expert, the writer proposes to offer some information derived from various experiments and to give details of the silencers used in the work on the single manifold. These silencers will be described first.

They are named 'interference' silencers because the reverse flow idea mentioned on page 129 is used.

In that case – of the back pressure silencer – the intention was to cause the primary pulse to be reflected towards the engine as soon as possible, whereas in the present case the object is to allow the pulse to occupy the whole volume of the box with a minimum of obstruction in order to give 'independence', see page 102, to the down pipe from the manifold. The volume of such a box should be 10 times the stroke volume.

The three-inch silencer (fig. 6 : 15)

The volume of this one is only seven times the stroke volume and this was found to be too small for use as a first silencer, but it was later used as the second silencer at the rear of the car; and another of $3\frac{1}{2}$ in. diameter was made, using the same internal design. It had the required 10 times ratio and did quite well.

Returning to the three-inch one, the incoming pipe was $1\frac{1}{2}$ in. diameter to suit the down pipe and so was the outlet, but the latter was found to be too large when used at the rear and it was reduced to

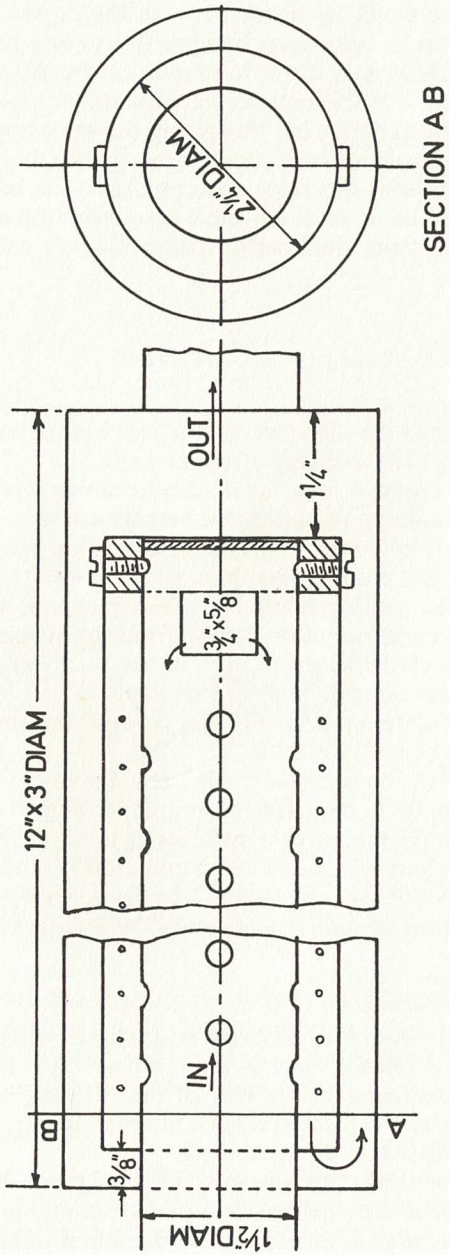


Fig. 6 : 15. The three-inch interference silencer.

$1\frac{3}{8}$ in. when the diagrams fig. 7 : 14 were taken. Later on it was decided to use $1\frac{1}{4}$ in. diameter pipes all the way because that is the size usually found on engines of this capacity. The effect is to raise the peak pressures of the waves, but it was thought advisable to deal with the more difficult case. The incoming pipe had 36 holes, $\frac{1}{4}$ in. diameter, giving an area of 1·76 in² and there were three slots $\frac{3}{4} \times \frac{5}{8}$ inch, giving 1·4 in² area, at the end where it was blanked off. This ensured that if the small holes became clogged there was still space for the gas to pass. At the end of this pipe a tube $2\frac{1}{4}$ in. diameter was attached so that the gas has to reverse direction here and also at the other end where the $\frac{3}{8}$ in. gap gives ample space. The cross section area between the first and second tubes is about 2 in² and that between the second and the 3-inch one is 2·25-in², so that there is no restriction to the gas flow. In the first model there were no holes in the $2\frac{1}{4}$ inch tube but it was found better to have holes in it. The total area of the $\frac{5}{64}$ inch holes is 0·92 in² which is less than the pipe area which is 1·4 in², the idea being to keep the gas flowing along the passages. Some of the gas will pass through the holes and interrupt the main stream and give some damping. This silencer could be used with the inlet and outlet reversed. In this way it was found that there was less resistance to the entry of wave motion but the damping was not so good.

The oval silencer (fig. 6 : 16)

The shape is not strictly an oval as it has flat sides and circular ends because that was an easy way to make it, but the name is convenient. To make the casing a plate was rolled and bent to shape and the end E fixed by small bolts, while the other end was fixed at F, thus dividing the box into two chambers connected by a slot as shown. It was 3 × $\frac{3}{4}$ in, giving an area of 2·25 in². There was about the same area between the secondary tubes and the casing so that the gas did not have to change speed. The secondary tubes had $\frac{5}{64}$ inch holes, giving the same area of 0·92 in² as before, and the tubes were fixed to the end plates by lugs. The inlet pipe had $\frac{1}{4}$ inch holes as before and could have a gap G or be pushed against the end. This was tried on test and seemed to make no difference. The outlet pipe instead of being drilled had 24 saw cuts $\frac{1}{16}$ in. by 0·7 in. long, giving 1·05 in² area. The length of the body is 8 in.

The ends were rectangular plates $\frac{1}{4}$ inch thick and wider than the casing. They were pulled together by long bolts through the corners and sheets of jointing material made the ends of the casing gas tight. This method was used with all the silencers for ease in taking them to pieces.

It is not to be expected that anyone will make silencers like these

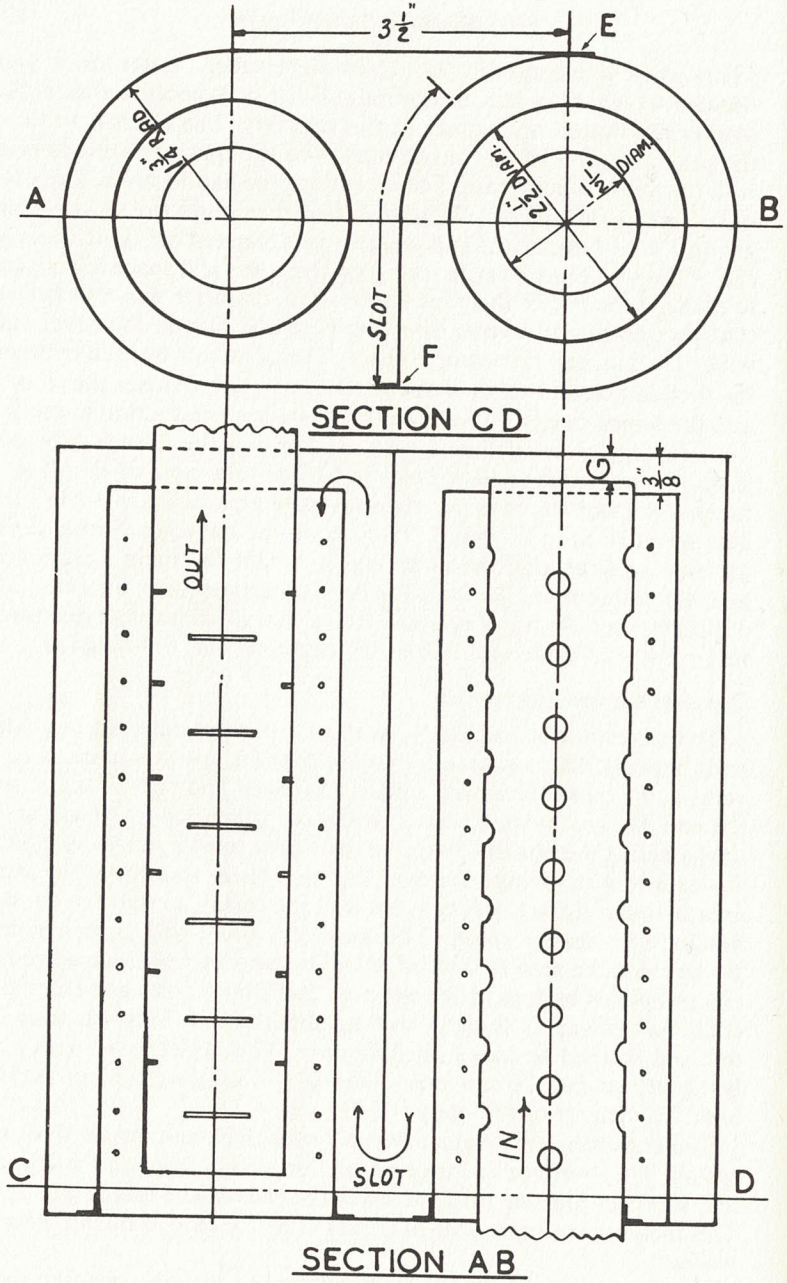


Fig. 6 : 16. The oval interference silencer.

because people with sufficient enthusiasm to make silencers always have ideas of their own. Well, good luck to them, and here are some facts that may be useful.

Tests with absorption silencers

In the discussion on absorption silencers it is stated on page 123 that the material should not be less than 1 in. thick and preferably 1½ in. This was the writer's experience but he thought it would be interesting to get actual evidence of the penetration of material by a pulse. The tests were made on the 2-stroke with the rig in fig. 6 : 17.

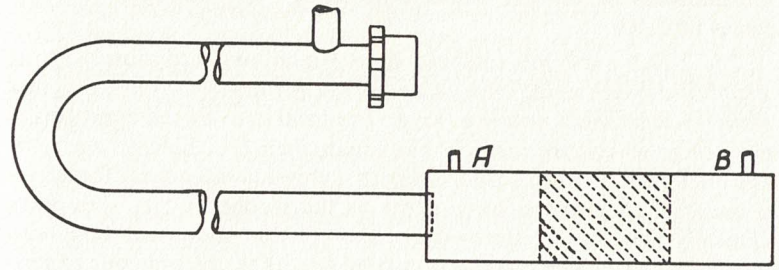


Fig. 6 : 17. Rig for testing pulse penetration.

The exhaust pipe was bent round and brought back so that the 2-inch tube to which it was attached was near the indicator and connections could be made to the offtakes A and B. There was no outlet here for the exhaust gases which were taken off in a small tube near the engine. The effect was that a good pressure pulse was sent down

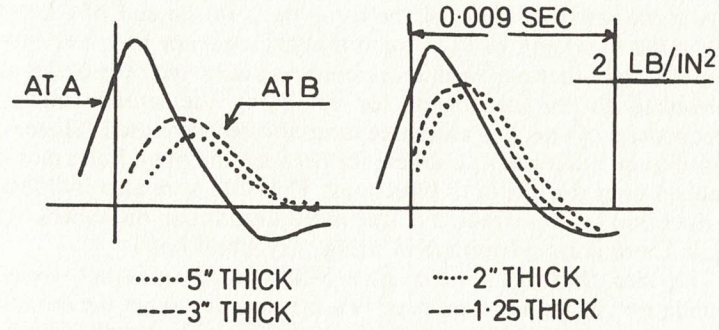


Fig. 6 : 18. Showing penetration of pulse through various depths of glass wool packing.

the pipe and into the 2-inch tube. The ends of this tube were detachable so that it could be packed with glass wool to any desired depth. With the engine running at 2,200 rev/min diagrams were taken at A and B for depths of 5, 3, 2 and $1\frac{1}{4}$ inches (fig. 6 : 18). At 5 in. the penetration is less than 50% and at 3 in. is just over that, although the delay of the returning wave – which is important – is good. At $1\frac{1}{4}$ in. the penetration is 80% but the delay is poor. At 2 in. it is 74% and the delay is better. So it looks as if $2\frac{1}{2}$ in. might be the maximum useful thickness and $1\frac{1}{4}$ in. the minimum.

Comparisons of the performance of baffles with different forms of holes

The 2-inch tube was replaced by a 3-inch one, and closely fitting metal discs were made with holes drilled $\frac{3}{16}$ in., stabbed holes $\frac{3}{32}$ and slots 0·025 in. wide and $\frac{1}{2}$ in. long. The total area of the apertures in each plate was approximately one square inch, which was the area of the inlet pipe. The slots were cut with a chisel and the sides forced on a mandrel to give the same effect as the stabbed ones. They were carefully sized but all the evidence showed that they must have been a little oversize. The tests were carried out as in the previous experiment. The plates with the stabbed holes and slots were tested with the flat side towards the inlet and then were reversed.

The diagrams showed no difference between the $\frac{3}{16}$ in., the $\frac{3}{32}$ in. holes and the stabbed holes in reverse. There was a slight improvement with the latter in the streamlined direction and this was also the case with the slots. Stabbing the holes has an advantage but not much. The small holes were no improvement.

This test is only for the large waves and tells nothing about the high pitched ones with frequencies of one to ten thousand per sec.

A static test was made of the same discs in the end of the tube where the blind end had been and a high frequency note was introduced at the other end. A microphone was placed near the outlet and connected to an instrument for recording the sound intensity. Frequencies of one, two and three thousand/sec were used. There was found to be practically no difference between the plain holes and the stabbed ones tried in both directions. The slots were also indifferent to direction but transmitted a little more sound than the others. And again there is no advantage in using very small holes.

The idea that small holes give better damping is not without foundation, as is shown on page 118, but in order to get the damping effect the hole would need to have appreciable length which it cannot have in a thin plate. Also, for high frequencies the hole would have to be minute. The glass wool satisfies these conditions admirably.

Flow tests

Air was supplied by a small blower to pass through the discs, and the pressure behind the discs was measured by a manometer.

The results were similar to those in the first experiment, the streamlined direction having again a small advantage. The slots were distinctly better than all the others.

To quote from Horace Lamb (II):

'It appears that a circular aperture transmits the fraction 0·816 of the energy propagated across an equal area in the primary waves . . . . In the case of a long narrow slot . . . the energy transmitted is again comparable with and may even considerably exceed that corresponding to an equal area of wave front in the primary waves.' All very depressing!

So it would seem that slots are out and small holes unnecessary.

By way of excuse for the use of small holes in the writer's silencers it may be mentioned that these experiments took place after the silencers were made.