

SCIENTIFIC DESIGN OF EXHAUST & INTAKE SYSTEMS

Third Edition

Air Essentials – Exhibit 2013
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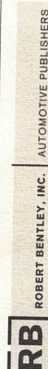
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Member of Automobile Division Council, I.Mech.E. and
Chairman of Scottish Centre, 1961-62

who has also edited and revised this edition



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PUBLISHERS' NOTE ON THE THIRD EDITION

FROM the extent of the distribution of this book since it was first published in 1962 (this is the seventh impression) it is reasonable to suppose that the work of the late Philip H. Smith and his collaborator, Dr John C. Morrison, has exerted a considerable influence on intake and exhaust system design in many parts of the world. It is hoped that this new edition will continue to point the way ahead.

The publishers have been fortunate in persuading Dr Morrison to revise the original text and to make additions of considerable interest from the hitherto unpublished results of his recent original research.

In particular, new work on silencer design is described and analysed in an addition to chapter 6. An addition to chapter 7 deals with research with the, still commercially popular, simple cast iron manifold on a four-cylinder engine; experiments were carried out to determine the most effective pipe lengths and the best positions for tandem silencers in such a system. Added to chapter 13 will be found new factors affecting the induction process.

Chapter 14 has been rewritten by a pollution research engineer to keep the reader up to date on the increasingly far-reaching requirements of fiscal policy in this respect (particularly in USA) and what is being done to meet them.

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important period over exhaust t.d.c. in relation to engine power. So far we have considered several basic but important features of pipe design largely on the assumption for simplicity that a smooth uncomplicated gas-flow is involved. There is little doubt that many engine designers, while paying lip-service to the presence of pressure-waves and their possible harnessing to good account, still tend to regard the exhaust flow in relation to the kinetic energy build-up which is caused by the moving gas-column as a whole. It would seem that in the case of the Crossley gas-engine, though mention is made of 'waves of pressure' in its description, the main reliance for the scavenge is vested in the high-speed plug of exhaust and its retention in the pipe until of no further use.

These phenomena are considered in full detail in Chapter 5, and from a study of this, it is not difficult to make a rough calculation based on the pipe length used by Crossley, indicating that a favourable wave action was in fact obtained. However before dealing with the influence of pressure-wave action in pipe systems, it is necessary to study the formation and characteristics of waves in relation to sound energy. This is the subject of the next chapter.

CHAPTER 3

SOUND AND ITS ENERGY

Basic theory – Propagation of sound waves – Wave length and velocity – Phase and path difference – Pressure variation – Interference in sound waves – Velocity of sound – Effect of pressure and temperature – Progressive and stationary waves – Particle velocity and pressure variations – The organ pipe – Resonance and harmonics

Basic theory (VI)

IN this chapter the basic theory of sound and its energy is considered in relation to what occurs in engine pipe systems. It must be emphasized that there is no easy way of explaining the various phenomena and careful study is essential for those who wish to obtain a firm grasp of the essentials. However, it can be stated with some confidence that all matter not strictly concerned with the application of sound energy as an aid to engine efficiency has been omitted, so that any extra effort will certainly not be in vain.

For ready reference, the following list of abbreviations is given, the same symbols being used throughout the chapter.

y	= displacement	E	= modulus of elasticity (of medium in which sound is travelling)
a	= amplitude	d	= density (of medium in which sound is travelling)
t	= time of wave travel	p	= pressure of a gas
T	= time of one cycle	γ	= 1.4 (ratio of specific heats of gas)
f	= wave frequency	m	= mass of a gas
λ	= wavelength	R	= gas constant
V	= velocity of wave	T°	= absolute temperature
		ntp	= normal temperature and pressure
		Algebraic sum	= addition of two quantities whether of same sign or not
		Integer	= a whole number

When something moves or vibrates to produce a sound, the vibration is transmitted to the air in immediate contact with the moving medium. The simplest vibratory motion, known as 'simple harmonic motion', is closely approached by the action of an ordinary tuning-fork. Its mechanical counterpart is the pendulum's bobweight swinging at constant time intervals alternately to left and right of the central position.

To explain the various definitions associated with wave motion, we can consider the pendulum as swinging or vibrating over a certain distance, displacement, or amplitude on either side of the vertical or mean position. Movements to one side are positive, those to the other are negative. If the variation of 'swing' over a period of time is plotted graphically, a sine curve is the result; this shows that simple harmonic motion can be defined by the equation

$$y = a \sin \frac{2\pi t}{T}$$

where y = the displacement and a and T are constants as explained below. A complete swing from positive to negative positioning is known as a cycle. In the diagram fig. 3 : 1 this is shown to occur

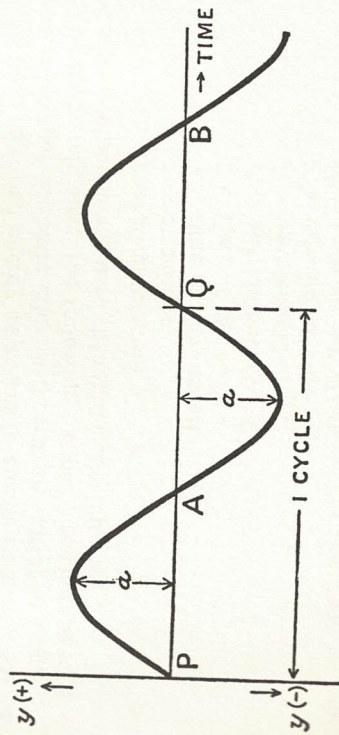


Fig. 3 : 1. Representation of a sine-curve sound wave.

in the intervals PQ or AB; from this it is seen that the vibration reaches its limit twice in a cycle.

The amplitude a represents the maximum distance on either side of the mean position. As the maximum value of the sine of an angle is 1, it follows that the amplitude is a constant in the above equation. The other constant T is the period of time taken by one complete cycle, i.e. from P to Q or A to B in the diagram. If the number of cycles per second, or frequency, is f we then have $f = \frac{1}{T}$.

Although the tuning fork is the nearest approach to simple harmonic motion of sound, other vibrations of greater complexity can be analysed into a number of simple harmonic vibrations; thus the equation given is usually employed in mathematical discussions of sound phenomena.

Propagation of sound waves

If the prong of a tuning fork is considered as vibrating as in the diagram fig. 3 : 2, we can consider its effect on the air, which for

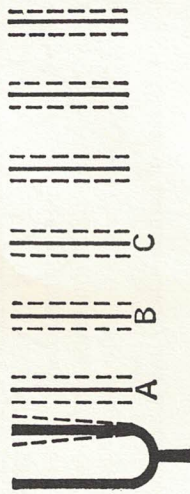


Fig. 3 : 2. Vibration of tuning-fork, and air movement.

convenience can be regarded as in 'layers' ABC etc. When the prong moves to the right, the air next to A is compressed and is thus moved to the right. This correspondingly displaces B to the right, and then C, and so on. When the prong returns to its mean position and then beyond it to the left, pressure on A is first reduced and then further diminished. Thus A now expands to the left,

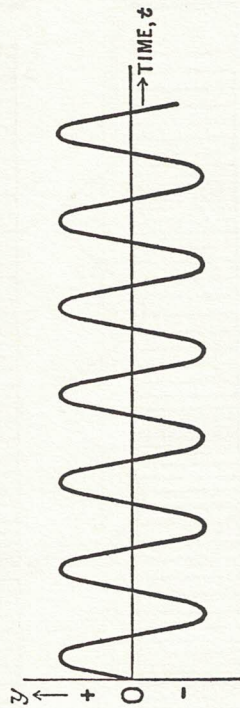


Fig. 3 : 3. Simple harmonic displacement-time curve produced by tuning-fork.

followed by B, C and so on. The movement of each layer is evidently at the same frequency as the prong of the fork. Because however the pressure disturbance takes a limited time to pass from A to B, etc., some layers may be out of phase with each other but there will also be positions where the displacement of a layer is in simple harmonic motion, depending on its distance from the fork. In that case the

amplitude plotted against time would be as shown on fig. 3:3 in which the left-to-right movements are considered as compressive or positive, and the right-to-left as rarefactive, or negative.

The velocity of the air particles during the passage of the wave is given by the rate of change of the displacement y in respect of the

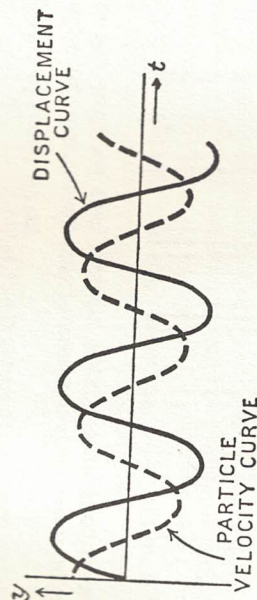


Fig. 3:4. Curves of particle velocity and displacement.

time t . Thus, the particle velocity is proportional to the gradient of the displacement-time curve, or $y - t$ curve, at the corresponding instant. The velocities are shown by the dotted curve in fig. 3:4 which is superimposed on the displacement curve. It will be seen that the maximum velocity occurs when the displacement is zero.

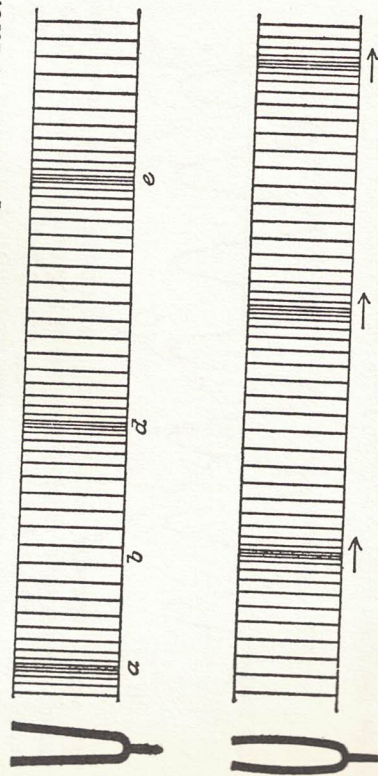


Fig. 3:5. Graphic impression of layers of air at a given instant.

This corresponds to the 'mechanical' analogy of the pendulum where the speed of the bobweight is at its maximum as it passes through the mean or vertical line.

Pictures taken of the layers of air at a given instant would be similar to those shown at fig. 3:5. As the fork moves to the right, the layer at a is passing through the mean position while that at b

has reached the limit of its vibration. Other layers close to a or b will be at different positions, being out of phase with a or b . But at d or e it will be seen that the layers are vibrating in phase with a .

A little later, with the fork moving the opposite way, the layers will be as shown in the lower diagram. The two pictures show how the wave is transmitted along the air by the transfer of kinetic energy of a vibrating layer (originally started by the tuning fork) to the layer next to it, which in turn hands on the energy to the next one. Some of the energy is of course 'lost' (*i.e.* converted to heat) at each transfer so that the sound eventually becomes inaudible. It will be clear that with the air undisturbed, the lines representing the layers would be equidistantly spaced instead of closing up and opening out as in the vibrations. This shows that the mean positions of the layers do not alter, irrespective of whether the air is undisturbed or has a wave travelling along it.

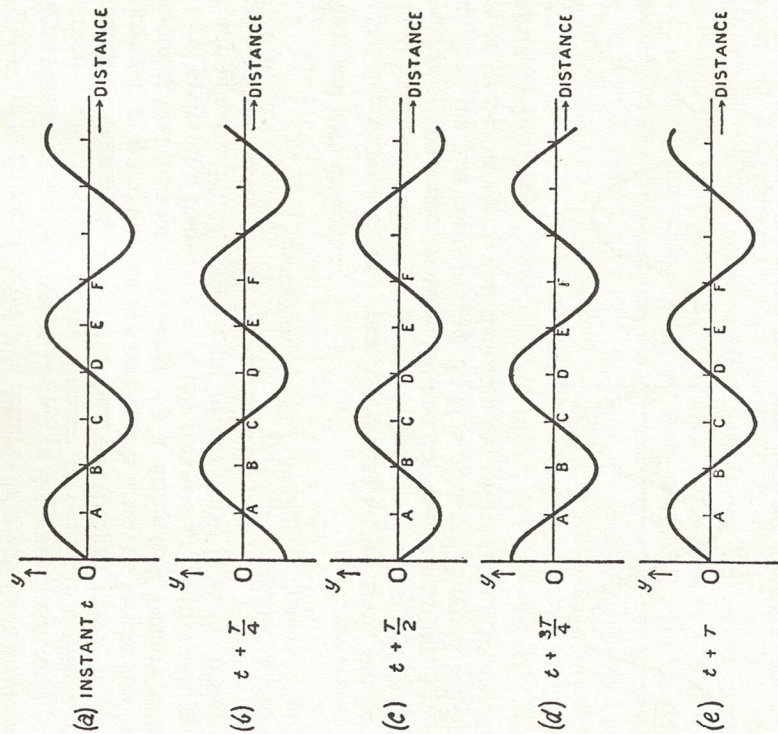


Fig. 3:6. Displacement-time curves, at various instants.

Wave length and velocity

Next we can investigate curves of displacement y at some instant of time t . These will be sine curves as already shown. Taking curve (a) (fig. 3 : 6) there is a positive maximum displacement at position A on the horizontal line, while at position B, displacement is zero, and at C it is a negative maximum. Curve (b) shows the situation one-quarter of a period later, at a time represented by $t + \frac{T}{4}$. Curves (c) (d) and (e) show respectively the positions at times of $t + \frac{T}{2}$, $t + \frac{3T}{4}$ and $t + T$.

It will be seen that all these variations of the positions of maximum displacement along the 'distance' line remain in step, or in phase with each other. That is, in curve (a), A and E are in step; in (b), B and F are in step and so on. The distance between these positions where vibrations are in phase is known as the wavelength, with the symbol λ , and is evidently the distance between the positions of successive wave-crests above the line, or troughs below it. The velocity V of a wave is the distance which it travels in one second. Assuming that the source of energy, e.g. a tuning-fork, has generated several cycles at a frequency of f , this means that f cycles will pass at any given point. As the length of one wave is λ , the distance occupied by f waves will be $f\lambda$, and thus we have:

$$\text{Velocity of waves } V = f\lambda$$

Phase and path difference

The sine waves can be drawn by plotting angles along the horizontal line instead of time. The sine of 0 degrees is 0, of 90 deg. is 1; the sine of any angle between 180 and 360 deg. is negative, and so on. Thus the first curve (a) (fig. 3 : 7) is obtained. Beyond 360 deg. the curve begins again, and therefore the values from 0 to 360 deg. constitute a cycle.

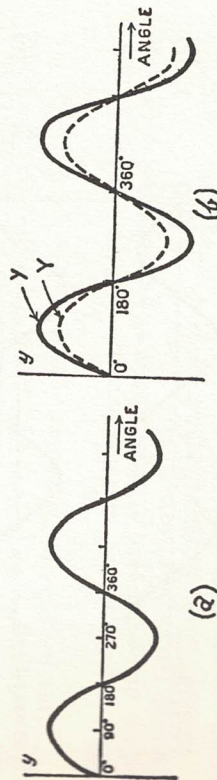


Fig. 3 : 7. Single wave, and two waves in phase, plotted on angle base-line.

On the second curve (b) two waves X and Y are shown, having different amplitudes but otherwise superimposed. These are said to be in phase with each other. If the vibrations occur exactly oppositely as in fig. 3 : 8 (a), they are 180 deg. out of phase; the last curve (b) shows them 90 deg. out of phase.

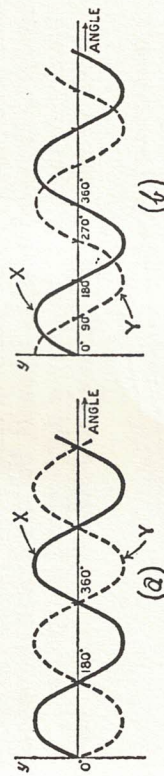


Fig. 3 : 8. Two waves out of phase, at different angles.

Referring again to fig. 3 : 6, it will be seen that the vibrations at AE or BF are a wavelength apart and in phase with each other. Those at BD are 180 deg. out of phase and half a wavelength or $\frac{\lambda}{2}$ apart. Those at AB or CD are 90 deg. out of phase and a quarter of a wavelength or $\frac{\lambda}{4}$ apart.

This establishes the relationship between phase and path difference of the waves, which is important when investigating the results of having a multitude of waves of varying characteristics all operating simultaneously, as happens in the exhaust gas discharge.

Pressure variation

We can next consider how the displacement of the layers of air affects the pressure in the air. The diagram fig. 3 : 9 (a) shows an example of the position at a given instant. Taking the top curve as representing the sound wave, and reading along the distance line, it will be seen that the displacement y at L on the line is zero, rising leftwards up the curve (positive) and falling rightwards down the curve (negative). Thus the layer of air at L is compressed (b), because the layer just to the left of it must be moved to the right by the positive displacement, while that just to the right of it must be moved to the left at the same instant by the negative displacement. Thus (as shown on the bottom curve (c) of pressure) the pressure of the air at L is higher than normal. Now consider the position of M on the top curve. Here again the displacement y is zero, falling leftwards down the curve (negative) and rightwards up the curve (positive). The layer at M is therefore decompressed, or rarefied,

because the air just to the left of it is moved left and that to the right of it is moved right. We thus obtain a lower pressure than normal on the bottom curve. At the point N on the distance line, *i.e.* midway between L and M, the layers on either side have been displaced by the same amount, to their maximum 'positions'. Thus the pressure here is normal. The same situation arises at the point P, the layers having been moved by equal amounts, but this time in the opposite direction to those at N because the displacement is opposite.

It will be seen that the pressure curve obtained in this manner is

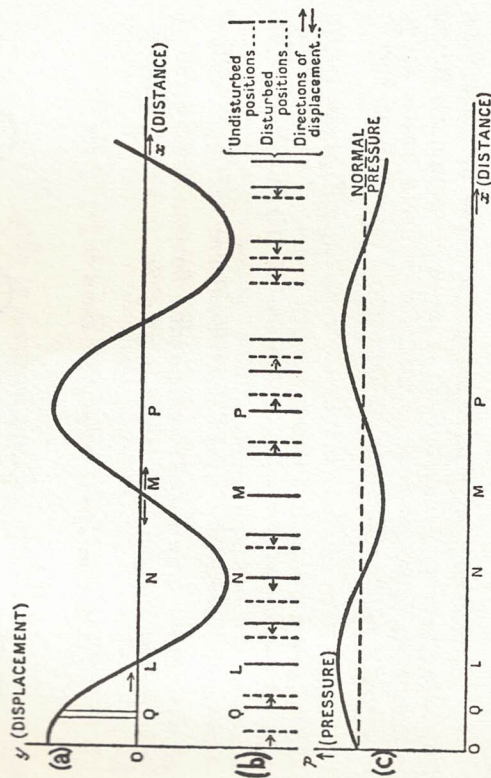


Fig. 3 : 9. Air-pressure variation caused by wave.

a simple harmonic one. The compressions and rarefactions occur at points where the displacement y of the sound is zero; normal pressure is obtained where y is at a maximum. It is not therefore strictly correct to state that exhaust pressure waves are sound waves. They certainly derive directly therefrom; the point is of course largely academic, as physical recordings of exhaust-pipe phenomena are concerned strictly with pressures.

Interference in sound waves

Interference is an important factor which affects both exhaust system efficiency and effective silencing. Referring to fig. 3 : 10, say two sources of sound A and B have vibrations in phase with each other. Their combined effect at a distant point X has been established as being equal to the algebraic sum of the vibrations at X due

to A alone and the vibrations at X due to B alone. If X is the same distance from A and B, the vibrations at X will be in phase because the distance AX traversed by wave A is the same as BX traversed by wave B; the sources being assumed to have the same frequency. The energy of a vibrating source is established as proportional to the square of its amplitude (it is not proposed to investigate the

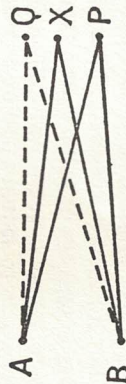


Fig. 3 : 10. Interference effect of sound from two sources.

theory of this statement) and so it follows that the wave formation will be shown on the curves.

The point Q can next be considered, this being so situated (fig. 3 : 11) that BQ is greater than AQ by a whole number of wavelengths. The vibration at Q due to A is in phase with that due to B and the result is a loud sound. This will be the case at any point Q

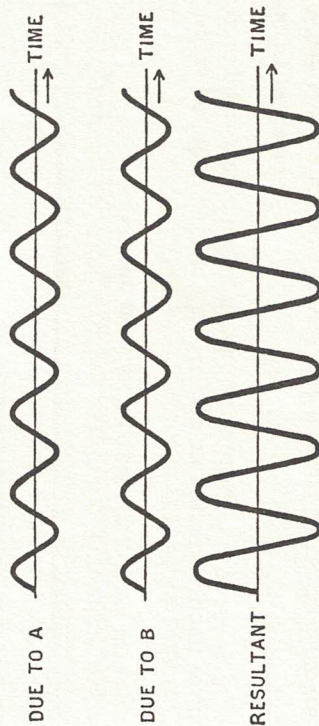


Fig. 3 : 11. Vibrations of individual sources and resultant wave at point X on fig. 3 : 10.

if the path difference $BY - AY$ is given by $BY - AY = n$ where n is an integer. Now, considering the point P, say this is situated in such a place that BP is half a wavelength longer than AP, *i.e.*

$AP - BP = \frac{\lambda}{2}$. The vibration at P due to B will be 180 deg. out of phase with the vibration due to A (as shown at fig. 3 : 8 earlier). The effect at P will now be zero, as the displacements at any instant are equal and opposite to each other. No sound is thus heard at P (fig. 3 : 12).

It will be evident that if the path difference were $\frac{3\lambda}{2}$ or $\frac{5\lambda}{2}$ instead of $\frac{\lambda}{2}$ silence would also result at P, as the vibrations would again be 180 deg. out of phase.

To summarize this, it can be stated that the waves will cancel if the path difference is an odd number of half wavelengths and will add if the path difference is a whole number of wavelengths. The total energy in either case is equal to the sum of the sources and can thus give positions of loud noise or silence for the reasons given.

When considered in terms of pressure, the same effect will obviously give greater or lesser pressure, depending on the combination

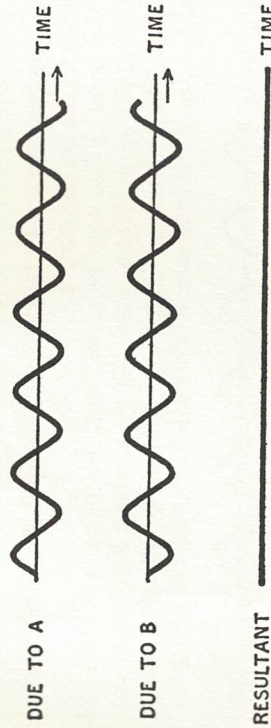


Fig. 3 : 12. Vibrations of individual sources and resultant cancellation of wave at point P on fig. 3 : 10.

of waves and their source. This is again an important point to consider in pipe-system design, as will be detailed.

Velocity of sound

When the sound wave travels, the medium in which it moves is subjected to stresses and strains. The speed or velocity of the wave must thus be partly dependent on the modulus of elasticity E of the medium and its density d . Given E in dynes per sq. cm., and d in grammes per cc, the velocity V will be in centimetres per second and is given by

$$V = \sqrt{\frac{E}{d}}$$

(For V in feet per sec., E must be in poundals per sq. ft and d in lb per cu. ft.)

In the case of a 'solid' such as a steel bar, E is of course Young's Modulus of Elasticity. The same formula holds good for gases, E in

this case being the bulk modulus of the gas and d its density. With the stresses and strains in the gas taking place so rapidly when the wave is passing, it was considered (Laplace, 1816) that an adiabatic situation could be assumed. As E , the bulk modulus, is actually the pressure of the gas, the formula now becomes (with pressure p)

$$V = \sqrt{\frac{p}{d}}$$

As p is adiabatic, we also bring in the ratio of specific heats at constant pressure and constant volume, or the standard symbol γ .

The formula now becomes $V = \sqrt{\frac{\gamma p}{d}}$.

The velocity in air, as stated by Laplace's correction, can be worked out taking $p = 76 \times 13.6 \times 980$ dynes per sq. cm., and d as 0.00129 grammes per cc at normal temperature and pressure and γ as 1.4. $V = \sqrt{\frac{1.4 \times 76 \times 13.6 \times 980}{0.00129}} = 33,100$ cm. per sec., or 331 metres per sec. This is about 1,100 ft per sec. and agrees quite closely with the results found by experiment.

Effects of pressure and temperature

While the foregoing applies to the conditions in air at $n\tau p$, the exhaust system is dealing with high-temperature gas, under an average pressure in the system usually higher than atmospheric. If we take the mass of the gas as m and its volume as v , its density will be $\frac{m}{v}$, and thus the velocity of sound will be given by $V = \sqrt{\frac{\gamma p}{d}} = \sqrt{\frac{\gamma p v}{m}}$.

If R is the gas constant for 1 gramme of the gas and T° is its absolute temperature, we have $p v = m R T^\circ$, and $\frac{p v}{m} = R T^\circ$, and hence $V = \sqrt{\gamma R T^\circ}$.

Now as γ and R are constants for any particular gas, it follows that the velocity of sound in a gas is independent of the pressure if the temperature remains constant and that it is also proportional to the square root of its absolute temperature. For example, if by experiment it is found that the velocity in air at 16 deg. C is 338 metres per sec., the velocity at 0 deg. C will be (with absolute zero as -273 deg. C)

$$\frac{V}{338} = \sqrt{\frac{273}{273 + 16}}$$

from which $V = 338 \sqrt{\frac{273}{289}} = 328.5$ metres per sec.

Thus it is seen that the speed of the wave increases with an increase in the gas temperature.

In the case of exhaust gases the speed of sound therein is considerably higher than in air. The broad reason is that in the former, much greater movement of the gas particles is called for; this need not be considered in detail. Further information on this aspect will be found in Chapter 5.

Progressive and stationary waves

So far we have been considering sound waves moving unidirectionally in air, or what are known as plane-progressive waves. It is necessary also to investigate the case of stationary waves, such as are caused by reflection. (The latter is of course a highly significant factor in exhaust and intake design.) If a progressive wave a is considered as travelling along OW (fig. 3 : 13(a)) and meeting a reflecting surface, such as a wall, at W, a reflected wave will be returned as shown at b . The condition of the air on the route OW is then due to the combined effects of the original and the reflected waves, a and b . The air in immediate contact with W must be at rest. If the displacement of the layers of air caused by a at that instant are as on the sine curve shown, this means that at W there is maximum displacement. This layer at W being at rest, the reflected wave must also displace the layers, as on curve b , at the same instant; otherwise the net displacement at W, which is the algebraic sum of WR and WH, would not be zero. Thus, as shown, the reflected wave b will be 180 deg. out of phase with the original wave a .

If the amplitudes of a and b are equal, and the same wavelength, the algebraic sum or resultant S must be zero, anywhere along OW. However, if we consider the position at a time $T/4$ later, the displacements will appear as in the second diagram. In this the two waves a and b have advanced respectively to the right and to the left, each by a quarter of a wavelength; the algebraic displacement is now shown by the curve S (fig. 3 : 13(b)). At the end of another $T/4$ period, the position is as shown at fig. 3 : 13(c), with the advance of a further quarter of a wavelength in opposite directions; the algebraic sum is again zero at any point along OW. After another period $T/4$ the position is as shown at fig. 3 : 13(d). In each case the resultant wave S is known as a stationary wave.

Referring again to the line OW, it will be seen that with a stationary wave, some layers at N are permanently at rest, while others at A vibrate through twice the amplitude of either of the two waves. The points at N are known as nodes, those at A as antinodes.

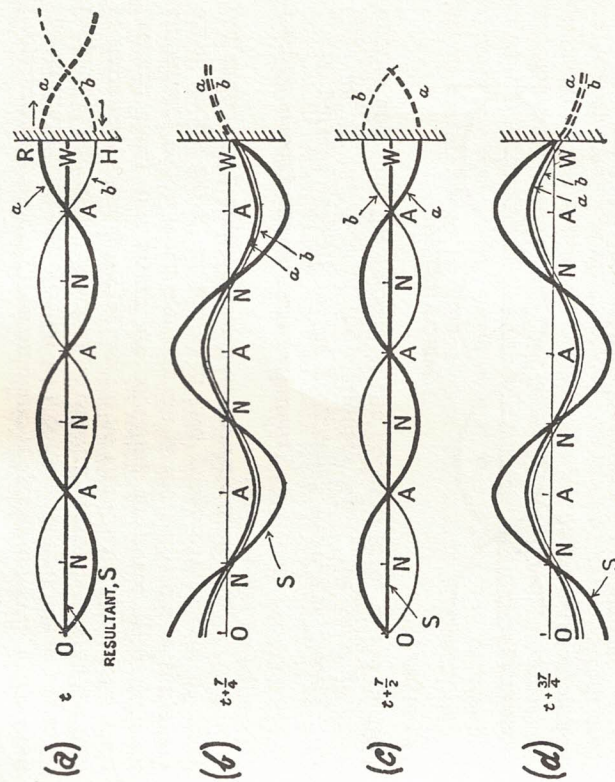


Fig. 3 : 13. Formation of stationary reflected waves.

The layers between nodes are vibrating in phase with each other, the amplitude of vibration varying from zero at N to maximum at A. In fig. 3 : 14, the displacement of the layers is shown along OW at five different instants numbered 1 to 5. From this it is seen :

Distance between consecutive nodes $N = \frac{\lambda}{2}$

Distance between antinodes $A = \frac{\lambda}{2}$

Distance from node N to next antinode $A = \frac{\lambda}{4}$

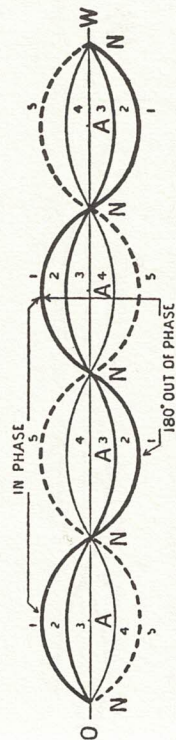


Fig. 3 : 14. Displacement of air layers at five different instants.

From previous observations on plane-progressive waves, it has been shown that each layer vibrates with constant amplitude at the same frequency, and that it is out of phase with nearby layers. In the case of stationary waves however there is no vibration at the nodes, while between them the layers are vibrating in phase with different amplitudes which reach a maximum at the antinodes, a stationary wave always being set up when two progressive waves of equal amplitude and frequency travel in opposite directions.

Particle velocity and pressure variations

The particle velocity of the progressive wave was discussed on page 46. In the case of the stationary wave, the same principle applies and thus the velocity at the nodes is zero, as the particles

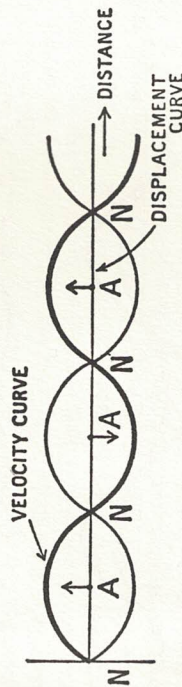


Fig. 3:15. Curves of particle velocity and displacement.

are at rest. The velocity at the antinodes is the maximum, *i.e.* where the particle passes through its mean or original position; the particles at the antinodes are moving in opposite directions at any particular instant (fig. 3:15). The pressure variation can readily be obtained from the displacement curve (fig. 3:16). The principle is similar to that followed for progressive waves. Fig. 3:16(a) shows the dis-

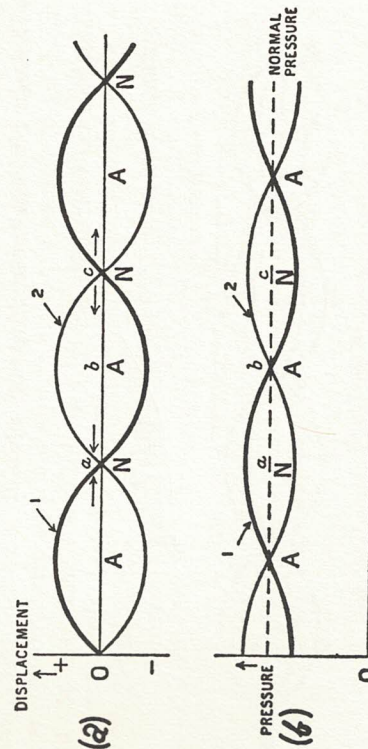


Fig. 3:16. Air-pressure variation caused by stationary waves.

placements at an instant where they are at maximum. The layer to the left of node *a* is displaced towards *a*, since displacement is positive; the layer to the right of *a* is also displaced towards *a*. Therefore the air at *a* is compressed. At the antinode *b* the displacements are both at a maximum to the left and thus the pressure is normal. At the node *c* the air on its left is displaced away from *c*, and that on its right is also displaced away; here again then, the air is rarefied and pressure lowered. The pressure variations will thus appear as on fig. 3:16(b).

The pressure is evidently always at a maximum at a node, and at zero at an antinode in the case of a stationary wave. In a plane-progressive wave however it has been shown that the pressure variation is the same at every point, as reference to fig. 3:9 will confirm.

The organ pipe

The foregoing covers the basic theory of sound waves in the atmosphere. When applying this to engine pipe systems it is usual to make the comparison with the operation of a normal organ pipe of the 'stopped' or closed-end type. There is of course an essential difference. The organ pipe receives its initial wave propagation from the open end, the wave then being reflected from the stopped end. In the case of the exhaust pipe, the wave is started by the pressure discharge from the cylinder, *i.e.* at the closed end, and the wave is reflected from the open end.

The organ-pipe principle is shown in fig. 3:17(a). Air entering at P strikes a bevelled lip to cause the vibration. The wave travels up the pipe to the closed end Q and is reflected as a stationary wave. From previous observations it will be clear that Q is a node, and A (the lower end) an antinode; thus the wave will in its simplest form be as shown at fig. 3:17(b). The displacement *y* of the layers is shown to vary along the line *x* at different distances from the reflecting end Q. As the length *l* of the pipe is the distance between N and A of the stationary wave, and $NA = \frac{\lambda}{4}$ it is evident that the wavelength in a closed-end pipe must be $\frac{\lambda}{4} = l$, and $\lambda = 4l$.

The frequency is given by $f = \frac{V}{\lambda}$, where *V* is the velocity of sound in air (page 44). The frequency must then be $f = \frac{V}{4l}$.

This is the lowest note obtainable in the pipe, and is called the fundamental. If the air blast from P is increased in strength, notes of

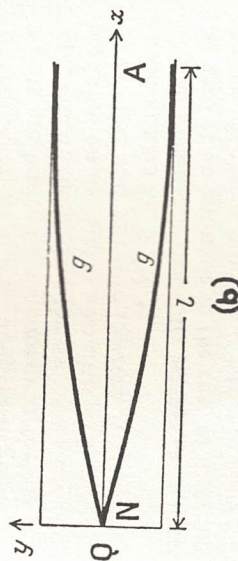
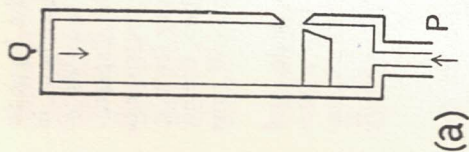


Fig. 3 : 17. Principle of closed organ pipe, and wave formation.

higher frequency are obtained which are simply multiples of the fundamental.

Although as stated the stopped organ pipe is the one related to our design problems, it is desirable to mention also the case of the organ pipe open at both ends as this shows an essential difference between the reflections. The open-pipe diagram is shown at fig. 3 : 18(a). It will be seen from this that there is now an antinode at each end, since both ends are open; the node is positioned halfway down the pipe.

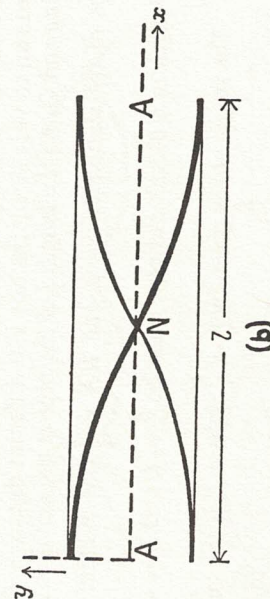
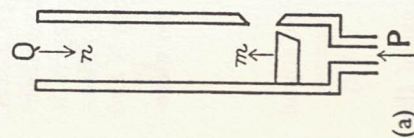


Fig. 3 : 18. Principle of open organ pipe, and wave formation.

From the same reasoning as was applied to the stopped pipe, it will be evident that in the case of the pipe open at both ends, the wavelength will be $\frac{\lambda}{2} = l$, and $\lambda = 2l$ and the frequency is given by

$$f = \frac{V}{\lambda} = \frac{V}{2l}$$

this again being the frequency of the fundamental note.

The above shows that when considered in terms of pressures, open-end and closed-end reflections have an essential and important difference. A pressure wave emerging at an open pipe end is reflected as a rarefaction which enters the open end, while a rarefaction wave is reflected as a positive pressure wave. In the case of the closed end however the reflected wave is of the same sign as the original, *i.e.* positive is reflected as positive, and negative as negative.

Resonance and harmonics

Resonance is the term used for sympathetic vibrations between objects whose wave frequency is similarly timed, as when for example striking a note on the piano causes an ornament in the room to emit a noise. The natural frequency of vibration of the latter is evidently in tune with the fundamental wave. Similarly, the air in a pipe will resonate in the length of the pipe and, if favourably timed to the periodic force, *i.e.* the engine speed, will increase the amplitude of the vibrations.

The final factor to be considered is the harmonic. In a sound wave produced by a tuned string for instance, a note of double the pitch is obtained if the string is trapped halfway along its length; each half-string will vibrate at double the frequency of the original. Even without trapping however, it is quite usual for a string to vibrate not only in its fundamental tone but also in one or more over-tones or harmonics. Similarly in a pipe, the wave may be accompanied by harmonics superimposed thereon.

Obviously, electronic apparatus of the type outlined can be used for a great many other investigations on engine performance, when using additional equipment, and it will be appreciated that for the purpose of this book the description is confined to its application to our particular purpose.

CHAPTER 5

PRESSURE PHENOMENA AND THEIR APPLICATION

Exhaust pressure – Wave propagation – Gas speed – Column inertia – The ideal curve – Organ-pipe theory – Speed of the wave – What the curves show – Varying pipe length – Pressure and pipe length – Pressure and engine speed – Resonance – Pressure in long pipe – Multi-cylinder manifolds – Sub-division of manifolds – Siamesed ports – Independence of vibration – Interference – Interference application – Interference action – Test conclusions – Six-cylinder engines – Interference on a six – Effect on parallel-twin – Resistance method

Exhaust pressure

WE can next apply the conception of sound energy to exhaust pipe pressure fluctuations. From Chapter 2, in which the flow was considered in its simplest aspects, it was commented that many designers tended to regard the kinetic energy of the gas discharge as the main factor in scavenging; earlier statements give evidence of this. This kinetic effect is easily explained and, if it were the sole criterion, efficient scavenging would no doubt be simplified.

The diagram fig. 5 : 1 will explain the theory. When the exhaust valve opens, there is a high-pressure release of gas into the port and pipe. This imparts kinetic energy to the column in front of it; as the 'bung' or slug of pressure reaches the end of the pipe, the energy is transferred to the region of the exhaust port, causing a rarefaction which pulls out the residuals concurrently with a rise in port pressure as the kinetic effect decreases.

It is by no means inferred that this proposition is incorrect; it forms an essential part of the process. But superimposed as it were on the flow as a whole is the presence of the pulsating wave motion, in consequence of having an elastic medium to deal with. Whereas the gas column as a whole may be moving in the pipe at an average speed of 200–300 ft per second, a much higher speed is obviously attained at the wave pressure peaks.

In a great many writings on engine design, emphasis is put on the kinetic energy of the exhaust outflow as an aid to auto-extraction of the residuals. While it is possible that, particularly in the past, auto-

mobile engineers have tended to emphasize this effect while largely ignoring the significance of the wave motion, this may have been due to the kinetic effect being more explicable. It is worth while looking a little further into this.

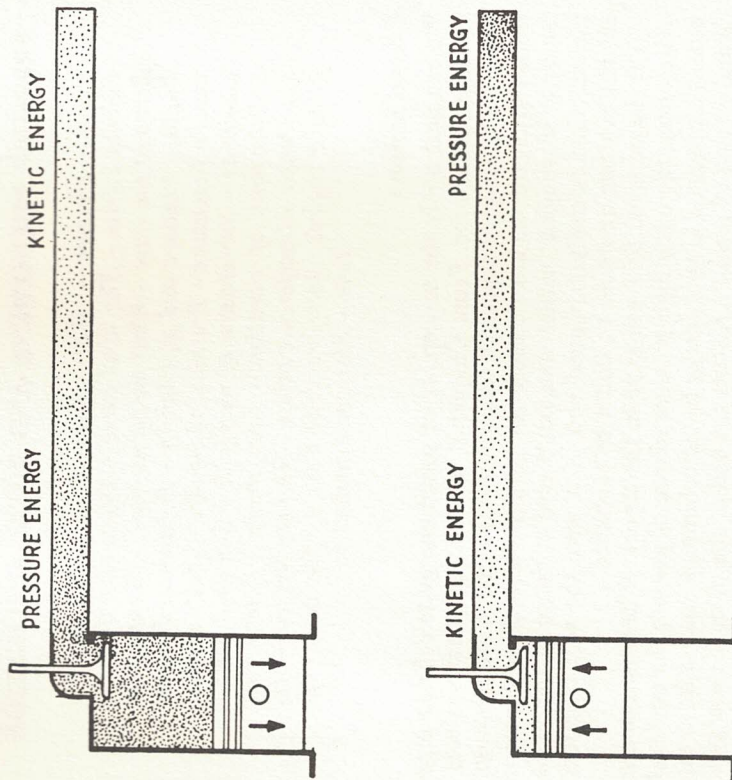


Fig. 5 : 1. Kinetic energy theory of scavenging.

Wave propagation

The pressure waves in the outflowing gas are formed by sound waves and move at the speed of sound (the theory of which is the subject of Chapter 3). In order to start such a wave, however, a sudden impulse is necessary; this can be illustrated by a simple laboratory experiment which is here shown diagrammatically, fig. 5 : 2.

A long coil spring of thin wire is suspended freely on threads. If a sudden blow is given to the left-hand end as at A, the end coils will close up momentarily; these closed coils will then proceed rapidly down the length of the spring as at B, always trying to equalize the

pressures before and behind them, until they arrive at the far end of the spring. (The whole spring will also move bodily to the right, of course.)

If, instead of the left-hand end being knocked to start the impulse, a few end coils are held compressed by a stop, and then suddenly released, as at C, the same effect will be obtained, i.e. the compressed

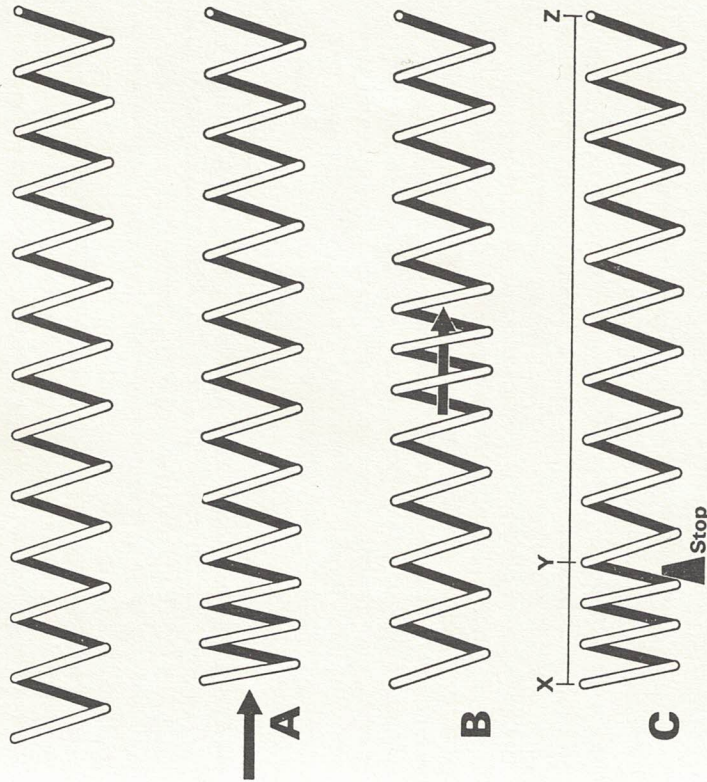


Fig. 5 : 2. Coil-spring experiment illustrating wave propagation.

coils will travel down the spring. In this diagram C, the compressed part x-y can be regarded as the pressure in the cylinder and y-z as the pressure in the pipe at the other side of the exhaust valve. Releasing the coils at y is the same as opening the exhaust valve.

Now it will be evident that the more suddenly the compressed coils are released, and the higher the difference in the spring compression between x-y and y-z, the more violent will be the propagation of the wave; carrying the analogy to its logical conclusion it is readily seen that the angular timing of the valve, rapidity of its opening and

cylinder pressure at the time of opening, will all have a profound influence on the nature of the wave.

It will be remembered that in Chapter 2 mention was made of Crossley and Atkinson's emphasis on wave motion as an aid to scavenging; their engine seems to have been a pioneer example of successfully applying the principle, notwithstanding its low speed. In modern engines on the other hand, it is quite indisputable that when the exhaust valve opens at around 45 deg. before b d c, the considerable cylinder pressure existing at that time – up to about 70 lb/in² – is such as to push a very large proportion of the exhaust gases through the valve port before the piston reaches b d c, at which point the valve will be at about half lift, and this must set up a pressure wave of considerable magnitude; the actual speed of the gas through the port at this moment appears to defy measurement but must be very much higher than the mean speed; in any case it would vary as between different engines, for the reasons stated above.

Gas speed

The conventional calculation for gas speed through the valve and port is however useful in so far as it gives some indication. This takes piston speed as the primary factor, which is, of course, correct as far as the theoretical exhaust stroke, extending from b d c to t d c, is concerned. If we imagine the engine turning with the cylinder head removed, there will be a column of air moving in and out of the cylinder at the same speed as the piston. If the head is now replaced, this column has to pass through the restricted area of the valve port and its speed through this restriction will be increased in proportion to the cross-sectional areas of the cylinder and the valve port respectively. Each of these areas is obtained by the usual formula:

$$\frac{\pi D^2}{4}, \text{ where } D \text{ is the diameter of the cylinder or port.}$$

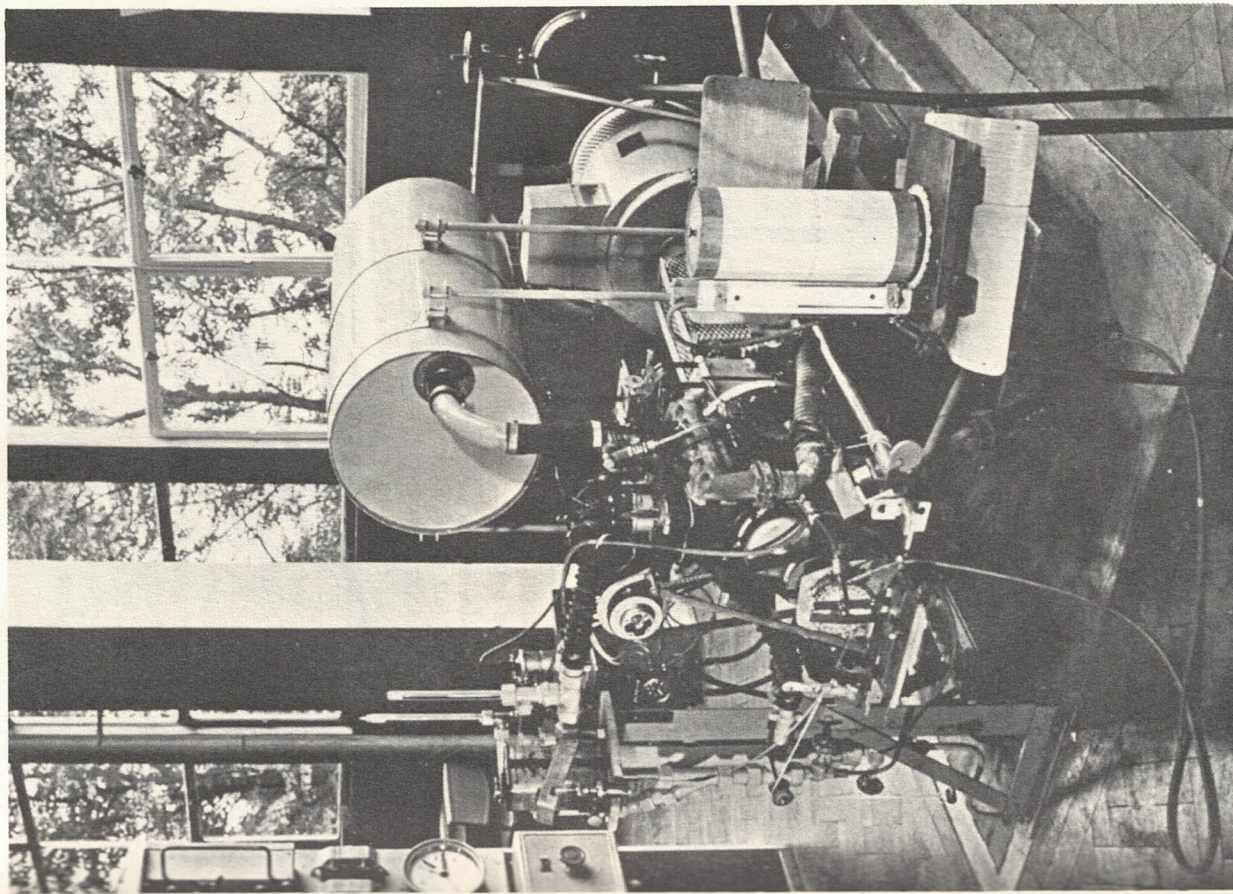
Thus, if the cylinder diameter is D and the port diameter d , the gas speed through d is given in feet per second, by

$$\text{Piston speed (ft per sec)} \times \frac{D^2}{d^2}$$

This is only true for the valve when at full lift, when this lift is equal to rather more than one-quarter of the diameter (so allowing for the seat angle and giving an equal area to that of the port) and when the coefficient of flow is unity.

Column inertia

The exhaust gas speed obtained from the foregoing formula is of course the mean, i.e. it is zero at top and bottom centres and at its



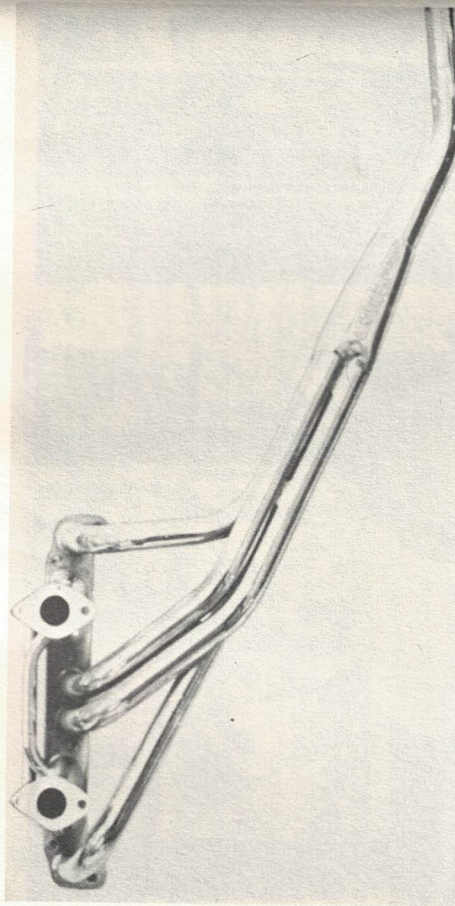
4 General view of laboratory test-bed and Morrison single-point indicator (Chapter 4)

maximum at about half-stroke. At peak engine revolutions, it is around 200 to 300 ft per second. This is sufficiently high for a useful build-up of energy, which can be preserved to advantage by using an individual pipe of appropriate length from each port, as illustrated and described earlier. This adds yet another advantage to the use of these extended branches, all totalling up to a better exhaust system. It is obvious that any gas speed based on piston speed must be zero (in theory at all events) at the overlap period at exhaust t d c because the piston speed is also zero.

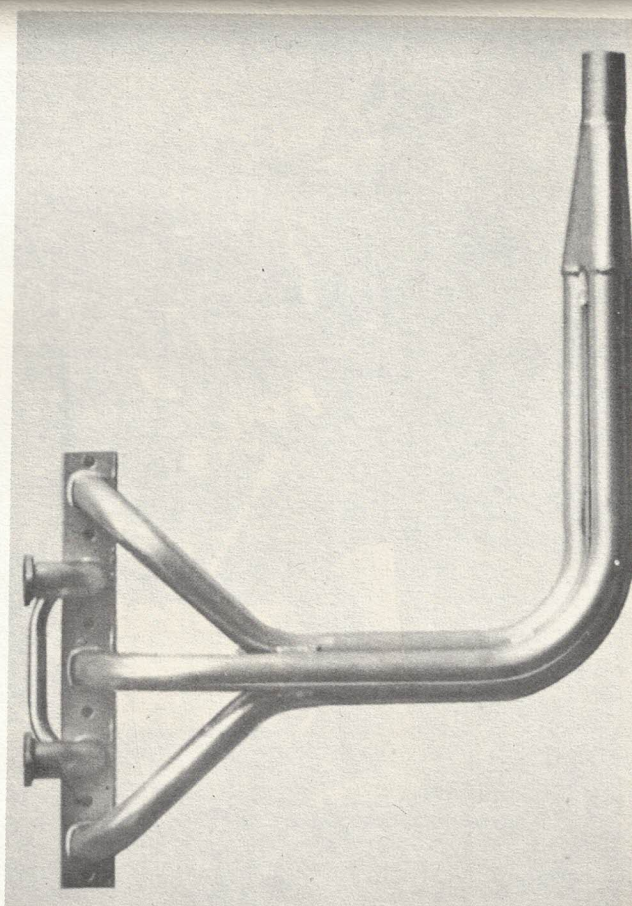
We can hope to preserve some kinetic effect, or 'column inertia' right through the exhaust stroke to this point, but a wave moving at 1,400 ft per second or more in the gas is certainly going to have a decisive effect on gas travelling at a mean of 200 or so. If the wave is moving *away* from the port it will help the inertia effect; if it is moving *into* the port it will easily overcome the inertia and recharge the cylinder with residuals. In using long branch pipes we may have with advantage removed some of the aids to this recharging that exist in conventional manifolds, as already explained. Thus, even if the wave is highly unfavourable, its effect is likely to give less of a power loss when using a special exhaust system – another reason why many special pipes work quite well. From the general designing apparent in marketed systems (other than from the aspect of separation etc., already gone into) there appear to be two schools of thought regarding wave motion. First, that it is a nuisance, to be rendered innocuous as far as possible, particularly in relation to the creation of a high pressure at the port at exhaust t d c. Second, that the energy is so obviously of considerably potential value, in that it does in fact create pressures below atmosphere in the system at certain times, it would be folly to throw away this advantage.

The ideal curve

The two curves shown in fig. 5 : 3 indicate firstly what happens in many an exhaust system, and secondly what should happen if everything went according to plan. In the full curve, the pressure wave at b d c of the explosion stroke, with the exhaust valve fully open, has produced a commendably low pressure or rarefaction at A, which relieves the piston of back-pressure during a large part of the exhaust stroke. Unfortunately, this is followed by another compression wave which shoots the pressure again well above the atmospheric line to B, and this just about the time at which both valves are overlapping at t d c of the exhaust stroke. By the time the pressure is again sub-atmospheric, at C, the valve is almost shut. Thus, although the pulses have assisted to reduce the total back-pressure by their behaviour at



5 Manifold design for 'independence' working on four-cylinder engine with four exhaust ports (*Chapter 7*)



6 Manifold design for 'independence' working on four-cylinder engine with siamesed centre port (*Chapter 7*)

A, the much more important scavenge period has, if anything, been hindered by the action at B-C.

Now consider the dotted curve. In this case, thanks to reflection, resonance, harmonics or all three, which cause the peculiar kink at D, the pressure is held to a much lower value of rarefaction than before, *i.e.* the curve does not fall much below the atmospheric line. Therefore, there is more back-pressure on the piston than was the case in the top curve. But the rarefaction, though less in terms of 'minus-pressure', is prolonged right across the t d c period of overlap,

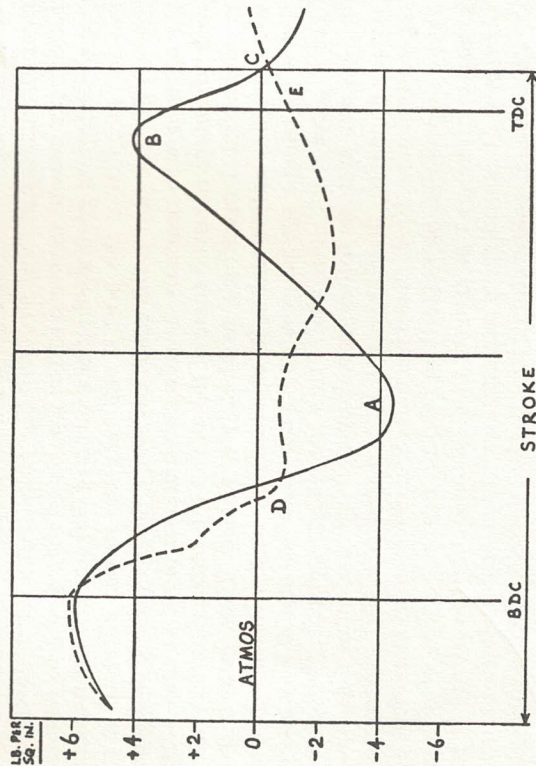


Fig. 5:3. Theoretical pressure curves of poor (full line) and good (dotted line) scavenging.

as at E. This cannot fail both to give effective scavenge and to help to induce the fresh inlet charge.

To achieve a curve such as this has been the ambition of exhaust system researchers for many years. In one or two cases, test-bed figures for torque and h p have been quoted as justification for drawing theoretical curves such as that shown in fig. 5:4, which is taken from a single-cylinder 500 cm³ o h v engine. In regard to this, it is stated that the exhaust pulsations may give low or high pressure at the port, at the moment of valve closing. The former means good scavenging and a purer inlet charge while a high pressure results in dilution of the new charge by exhaust gas. In the torque curve shown (fig. 5:5), it is seen that at 1,100 rev/min the column inertia favours

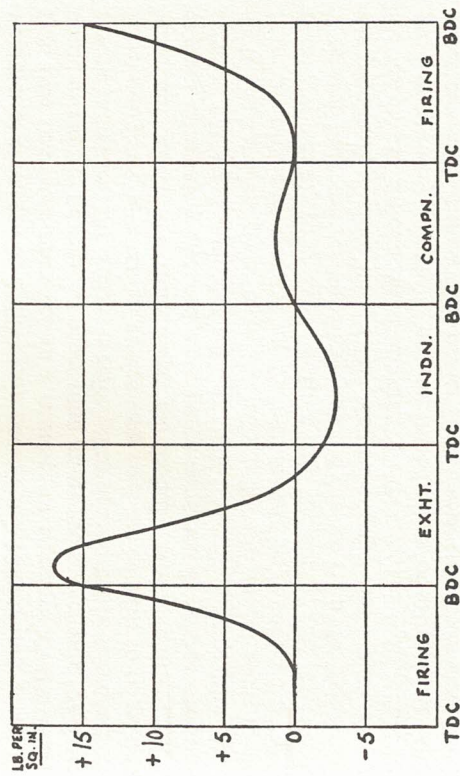


Fig. 5:4. 'Assumed' shape of exhaust port pressure curve.

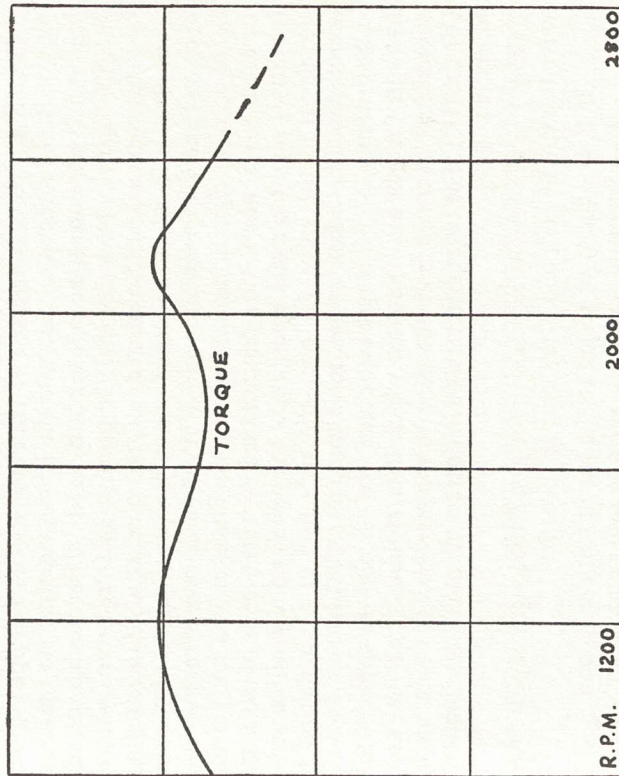


Fig. 5:5. Torque curve on test of 500 cm³ o h v engine, with peaks at multiples of rev/min.

volumetric efficiency and this again occurs at 2,200 rev/min, a condition that has been observed in a number of engines (XVII). There could obviously be reasons other than improved scavenging for these torque increases; the matter is referred to again later in this chapter. Most designers have tended to the view that the benefits to be gained were not worth the difficulties, particularly when multi-cylinder engines adequately silenced were being considered. As has been indicated, the single-cylinder with straight-through exhaust pipe presents the simplest picture, as the only reflection would seem to be from the pipe-end, and the pipe length should be capable of adjustment fairly easily to resonate in time with the engine revs. Hence, it is natural that the racing single was the first to reap the benefit of the pipe-tuning, since with a few lengths of pipe and a hacksaw in the test-house, an enormous amount of experimental work could be got through very quickly. (The later introduction of megaphones and other pipe-end fittings, still further to augment the power output on such engines, is considered in Chapter 8.)

Organ-pipe theory

If such a pipe is considered in the same way as a stopped organ pipe, it is not difficult to calculate the length necessary to obtain a pulse of rarefaction at the exhaust port, at tdc of the overlap period. From Chapter 3, we know that the positive pressure pulse initiated by the valve opening towards bdc of the power stroke (and starting from what is in effect the closed end of the pipe) emerges from the open end and scatters to atmosphere. A reflected wave of rarefaction, *i.e.* of opposite sign, instantaneously enters the open end and travels up to the valve, where it is again reflected but this time as of the same (negative) sign. On again emerging at the open end, a positive wave travels up to the valve, *i.e.* open-end reflection of a rarefaction is accompanied by a flow of gas down the pipe (XVIII).

Any calculation for length must take into account the speed of the wave in the gas and the time which it is required that the wave should take to travel from the valve to the pipe-end and back again (V), (XVIII). The latter is expressed in degrees of crankshaft rotation, as under:

Taking V as the velocity of sound in the gas, L the length of pipe from valve to extreme end, and T the time in seconds:

$$\begin{aligned} \text{If } V &= \text{velocity of sound in the gas in feet per second} \\ L &= \text{length of pipe from valve to outer end, in inches} \\ T &= \text{time for pulse to travel from closed to open end and back, in seconds} \end{aligned}$$

$$\text{Then } T = \frac{L}{6V}$$

In terms of crankshaft degrees, this becomes

$$T = \frac{\text{rev/min} \times L}{V}$$

Using a value for T equivalent to 120 deg. from the time the exhaust valve opens sufficiently to start an effective wave, this may be assumed for calculation purposes to give favourable conditions at the port (V). The speed of sound is difficult to determine, as will be apparent from later observations. If as suggested (V), (XXII), a speed of 1,700 ft per second is taken as representative, the pipe length to give the above conditions for any fixed engine speed may be obtained. For example:

rev/min for desired maximum torque = 4,000

$$V = 1,700 \text{ ft per second}$$

$$T = 120 \text{ crankshaft degrees}$$

$$\text{Pipe length } L \text{ in inches, } \frac{V \times T}{\text{rev/min}} = \frac{1,700 \times 120}{4,000} = 51 \text{ in., or } 4 \text{ ft } 3 \text{ in.}$$

Obviously with any increase or decrease in engine speed, T becomes shorter or longer respectively, and thus L must be shortened or lengthened in proportion. This is exemplified in the table below:

<i>rev/min for max. torque</i>	<i>pipe length</i>
4,000	4 ft 3 in
6,000	2 ft 10 in
8,000	2 ft 1.5 in

There is no doubt that single cylinder pipes calculated initially on the foregoing lines have shown good results and the formula serves as an admirable basis for starting experiments. An obvious snag is of course that even if the wave behaves as well as shown in fig. 5:3, at one engine speed with the calculated length of pipe, there is no guarantee of what it will do at other speeds. For example, a 'good' pipe length under the above conditions might so lower the torque at another point in the power curve as to offset any advantage.

Also, a fundamental flaw in the principle is seen by the fact that the rev/min for maximum torque could be increased to a point at which there is no pipe length left at all.

A further discrepancy is added by the fact that when the valve is open the cylinder is in effect included in the pipe length and, as the

piston is moving, so the resonant length of L is also varying. As regards the best figure to use for T the only accurate method of obtaining this would be by direct measurement of the pressure.

Speed of the wave

As shown in Chapter 3 the wave speed is affected considerably by the gas temperature. Thus, considerable variation is found in state-

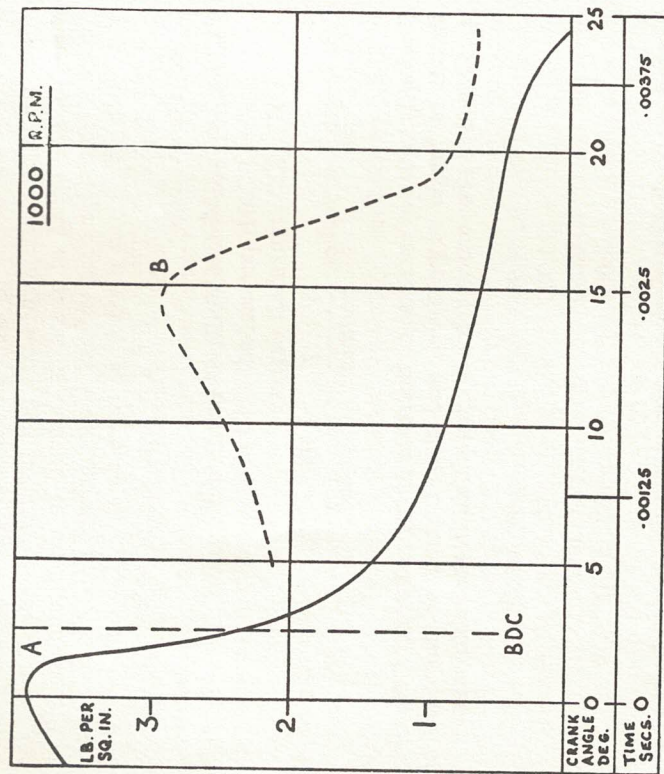


Fig. 5 : 6. Time taken for pressure pulse to traverse measured length of pipe from A to B.

ments relating to this rather important aspect. The pipe length calculation already detailed suggests a speed of 1,700 ft per second, while the curves obtained on the Morrison indicator, to be investigated later in this chapter, give a speed of just over 1,300 ft per second by calculation, which is confirmed by the shape of the curves. A further experiment, involving taking pressure readings on the pipe at a stated distance apart, showed that the wave took 0.0025 sec. to travel a distance of 3.5 ft, which gives 1,400 ft/sec. See fig. 5 : 6. It is evident that the speed of the initial pulse may be considerably higher

than average, and in fact as high as 1,968 ft/sec. has been recorded near the source (XXXIII). Such high velocities however are rapidly reduced to a normal value. A further investigation showed that there is a variation from about 1,500 ft/sec. at low rev/min to 2,000 ft/sec. at higher speeds; it is considered that this is due to the increased temperature of the gas at high engine revs.

In general, it seems advisable to assume an average speed not far above that of sound, ignoring the very temporary behaviour of the first pulse which can obviously be much greater. Thus the range 1,300–1,700 ft/sec. should cover most engines produced on a commercial basis. Very high speed racing units, particularly those of small cylinder capacity, are in another category and may show quite unpredictable results, as mentioned in Chapter 8.

We can now proceed from calculations to actual experimental indicator diagrams of pipe pressures, taken on the Morrison indicator under research conditions.

During investigations (XXIX), (XXXIII) extending over a period of some 40 years (which incidentally shows how little the basic problem has altered despite improvements in engine design) a large amount of quantitative data has been amassed. The outstanding value in this is that careful study gives an excellent general picture of causes and effects. From this, it is possible to avoid doing things that are prejudicial to power output and it should perhaps be made clear that not very many standard systems meet this observation. The applicable indicator curves give a first-class indication of what happens inside the system and undoubtedly enable some intelligent conclusions to be drawn about what size, shape and so on a good exhaust system ought to be. To begin with, we shall investigate the four curves in fig. 5 : 7, which refer to a single-cylinder with a straight-through pipe of just over 5 ft in length.

Each curve is obtained by joining up points of pressure obtained from the indicator described in Chapter 4 at regular and close intervals of crankshaft rotation, over two revolutions of the engine. (The portion reproduced covers only that period of the engine cycle with which we are particularly concerned.) Thus, the symmetry or otherwise of the shape depends entirely on the indicated pressure at that particular point in the rotation of the crank and which may be the result of the algebraic sum of several waves.

What the curves show

With the exhaust pipe of a length of 5 ft 2 in., and assuming a wave speed of 1,300 ft/sec., then f is equal to 65 waves/sec., according to the calculation detailed in Chapter 3. Taking fig. 5 : 7 curve (a) at

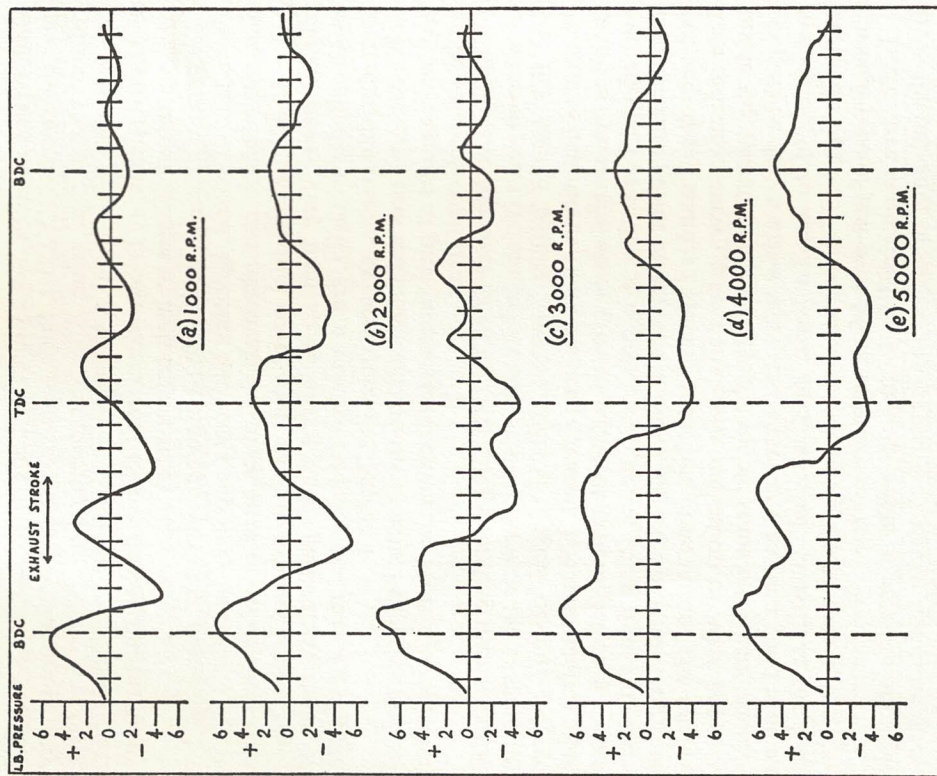


Fig. 5 : 7. Pressure recordings in 5 ft 2 in. open pipe at various engine speeds.

1,000 rev/min there should be about 3.8 waves per revolution, which can be seen to be the case. The symmetrical wave shape, already commented upon, is due to it consisting almost entirely of the fundamental; it is free from kinks, and because of the low engine speed, it almost dies out before the next pressure pulse, caused by the valve opening again, comes in. The maximum pressure occurs just before b d c and this is followed by a rarefaction of almost the same ampli-

tude. Subsequent plus and minus pressures of lesser amount occur just before t d c, the latter being quite favourably placed. At t d c the pressure is almost atmospheric.

At twice the speed, fig. 5 : 7, curve (b), maximum pressure moves slightly ahead of b d c, again followed by a considerable rarefaction drop. Unfortunately there is after this a steep rise, well above atmospheric at t d c, which persists throughout the overlap period; thus scavenging will be poor and the engine torque at 2,000 rev/min can be expected to suffer. The number of fundamental waves in the revolution is of course halved in comparison with curve (a), because of doubling the speed (this factor should be borne in mind when considering curves (c) and (d)), and various kinks indicate the presence of reflections, residual waves or an additional pressure pulse from the engine coming in before the first one has died out.

In fig. 5 : 7, curve (c), at 3,000 rev/min the maximum pressure has again moved ahead at b d c but the rarefaction is much more favourable to scavenging and this partly accounts for the higher initial pressure at b d c as the valve opens, the cylinder pressure being greater than at 2,000 rev/min. Again, there is evidence of the residual wave influencing the following new one and the behaviour of the pressure after the valve has closed shows considerable disturbance in the pipe which is likely to affect the fundamental wave to an unpredictable extent when the valve opens again.

In curve (d), at 4,000 rev/min, the initial pressure is again high and has moved further ahead of b d c but there is a rise during the exhaust stroke which implies back-pressure. The drop at t d c is better than at 3,000 revs., though it occurs unfortunately rather late; however, it is prolonged until well after the valve has closed. In curve (e) at 5,000 rev/min, a similar state of affairs is shown, with increased pressure during the exhaust stroke.

A study of the curves tends at first to indicate that there is hardly a single factor of similarity between any of them. The curves certainly show what is already known, *i.e.* that it is extremely difficult to obtain the pressure we should like at the desired time in the cycle and over the best part of the speed range for our purpose. However, there are some useful general conclusions to be drawn, which will now be considered.

It will be seen that the maximum positive pressure occurs, as is to be expected, at about b d c, when the exhaust valve is open. At low speeds it is a little before, at higher speeds a little after, and the pressure itself also increases slightly with engine speed at this point.

As the speed, and thus the frequency of exhaust valve opening increases, the presence of a residual wave in the pipe from the pre-

vious outflow of gas, plus other possible disturbances, make the fundamental wave more difficult to recognize in comparison with the low-speed curve (a). However, all these odd waves appear to lower the pressure, taken as an average over the whole of the exhaust stroke. The conclusion can be reached that in the case of these four curves, the first one, being not much above idling speed, would be of little significance. The second one (fig. 5 : 7(b)) shows that scavenging is bad due to a positive pressure at t d c, but here again the rev/min would be below that for maximum torque on a modern engine. At 3,000 rev/min (c), and above, conditions for scavenging at t d c are good, though at 4,000 and above, there is a disadvantageous high pressure persisting during the exhaust up-stroke itself. It is evidently quite feasible, with a given length of pipe, to obtain a low pressure at overlap t d c over a good half of the engine speed range, which (in these days of high-speed engines and close-ratio gearboxes, even on family saloons) is well worth having. This may, however, mean the presence of high-pressure peaks during the exhaust stroke, as at curves (d) and (e), and the hope must be that these do not persist for more than a limited amount of both the stroke and the rev. range.

Varying pipe length

Evidently, therefore, we cannot obtain the same conditions over a large speed range with a fixed pipe length. The next logical investigation is to see what happens with a fixed engine speed and varying lengths of pipe. Again, some reproductions of Dr Morrison's indicator diagrams are shown (fig. 5 : 8), all at 2,000 rev/min engine speed. (a) for the 5 ft 2 in. pipe is of course the same in fig. 5 : 7 for this speed.

With the pipe shortened to 3 ft 4 in., the curve (b) is obtained. While this has a pressure peak of short duration during the exhaust stroke, there is an excellent low-pressure drop at overlap t d c in direct contrast to the curve (a) which was positive at this point over an appreciable period. There also seems to be much more symmetry in the shape of the curve, while the maximum pressure at about b d c occurs earlier than with the longer pipe.

With a pipe length of about half (b), or 1 ft 7 in. (c), the conditions are not altogether unlike those at (b), if it is remembered that this shortening of the pipe will about double the number of waves in the same period of time. The peak pressures are however longer, so that scavenging will not be so good, while maximum positive pressure occurs some time before b d c where the valve is not at its best opening.

Curve (d) is of interest, because this is taken for a pipe length of only 8.5 in., which could well correspond to the exhaust stubs which

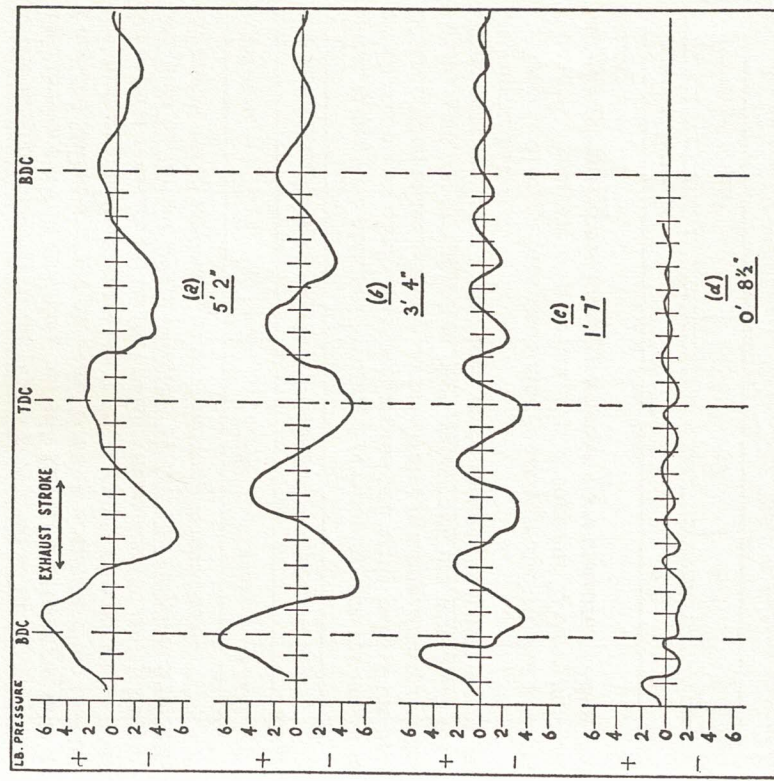


Fig. 5 : 8. Pressure recordings in four different pipe lengths at 2,000 rev/min.

have been used from time to time on racing cars, as an alternative to full pipe systems. The fairly drastic reduction in both positive and negative pressure peaks will be noted; the behaviour of the negative wave at overlap t d c is satisfactory, and the total back-pressure on the exhaust stroke quite low. When all this is combined with considerable weight reduction and freedom from all anxiety about the exhaust system falling off the car, it is obvious that exhaust stubs are by no means to be despised; in fact, their action is quite possibly more predictable than that of longer or more complicated pipe systems.

Pressure and pipe length

In considering the complete set of curves, it will be noted that when using a fairly long pipe the maximum pressure at about b d c tends

to rise at higher speeds. If the pipe is shortened as in fig. 5 : 8(b), a high pressure is still recorded, but still further shortening results in a decided lowering of the pressure. It would seem therefore that for any given engine speed there is a 'critical length' of pipe, which gives the highest pressure of the first pulse; increasing the length beyond this gives no further rise in pressure. The critical length decreases as the engine speed increases, as the following values show:

rev/min	1,000	2,000	3,000	4,000
Critical length, ft	3.75	3.0	2.5	1.9

These values were obtained by using a telescoping pipe the length of which could be varied while running and will apply only to the particular engine tested, since the rate of valve openings, amongst other things, is an important factor. Slow valve openings would call for a longer pipe while for a very high speed engine the length would be well under 2 ft.

The critical length was also found to increase slightly, but not very much, with increase of pipe bore. At a constant 2,000 rev/min the same length gave the following maximum wave pressures for different bores:

Area of bore, in ²	1.0	1.5	2.0	2.5
Pressure, lb/in ²	12.5	9	6.3	4.7

For the diameters of pipe most commonly used, therefore, the maximum wave pressure is approximately inversely proportional to the cross-sectional area of the pipe. The table also gives a practical guide to suitable pipe bores. For 2 and 1.5 litre four-cylinder engines the individual cylinder capacities would be respectively 500 and 375 cm³. The pipe bore corresponding to the cross-sectional area required would be 1.625 in. and 1.375 in. in each case. This dimension however would also be determined to some extent by the engine port size, and the matter is gone into in greater detail in Chapter 7.

Pressure and engine speed

An important feature also shown by the curves is the variation of maximum pressure with engine speed. Figure 5 : 9(a) shows the primary positive pulses for various speeds plotted on an rev/min base. Figure 5 : 9(b) shows the form and pressure of the first pulse on a time base. From these it will be seen from (b) that, as the duration of the pulse decreases, the maximum pressure and the rate of rise both

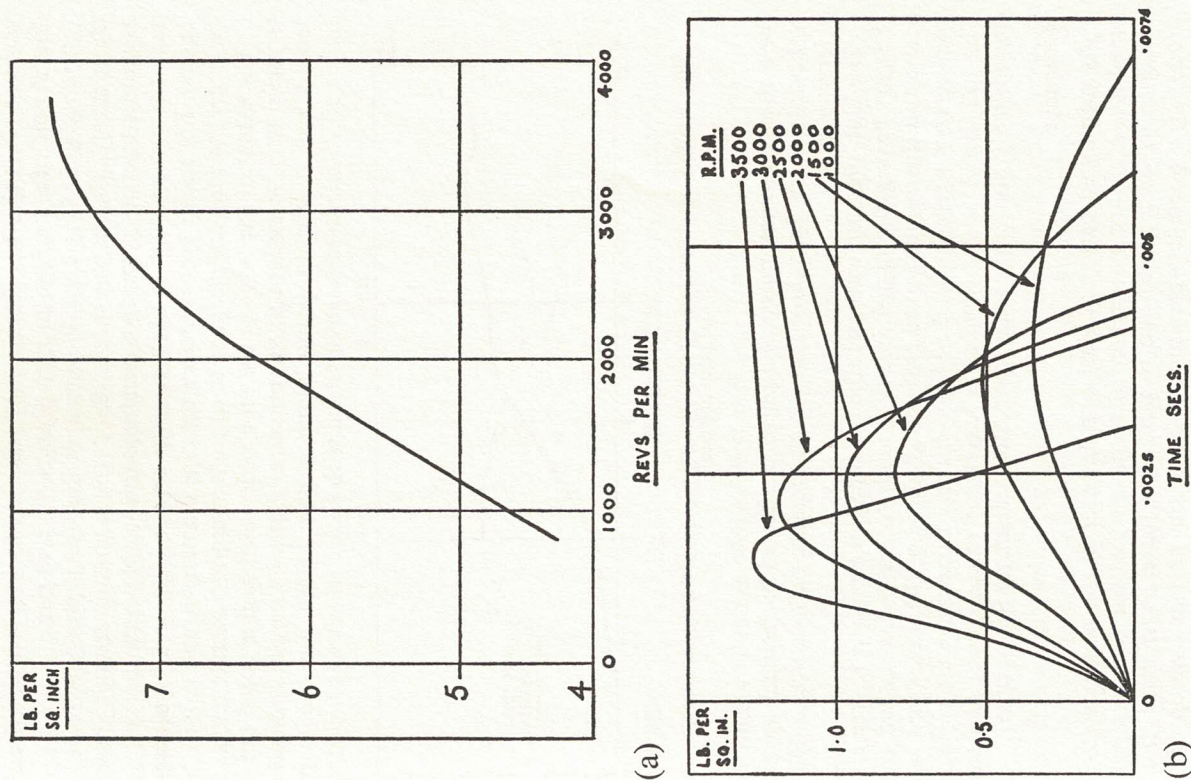


Fig. 5 : 9. Variation of primary pulse pressure with engine speed and time.

increase. It is this last feature that accounts for the sharpened and 'hard' note of a straight-through pipe as the revs rise.

From fig. 5 : 9(a) it would seem that the rise of pressure tends to decrease at higher engine speeds. But as the maximum rev/min shown is quite moderate by modern standards this is not conclusive and, in fact, the curve would be likely to keep on climbing on a high-speed engine. For such a type therefore the pipe diameter would require to be larger than for one designed to operate at more moderate rev/min.

The extent of the wave in the pipe varies with engine speed but it is useful to have an idea of the pressures along its length. Indicator tests taken at various points along a pipe length are shown in fig. 5 : 10. While these were obtained with a pipe which was too long

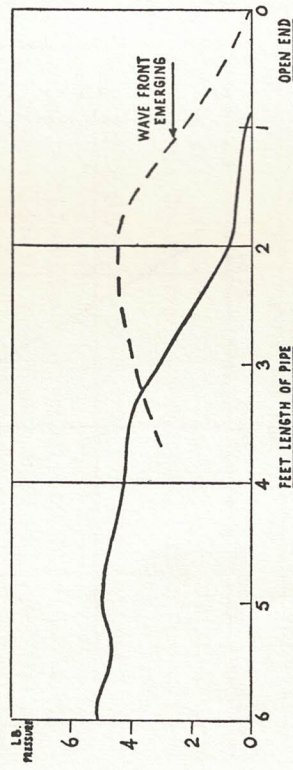


Fig. 5 : 10. Pressure recordings along length of pipe, showing persistence of pressure peak.

for a correct representation of a pulse, unaffected by residual waves, to be obtained it does show that a high pressure persists over a considerable length.

In a high-pressure wave, whether confined in a pipe or in free air, the part having the higher intensity tends to travel faster than the lower and this results in the pressure gradient of the wave-front becoming very steep. In fig. 5 : 11 two forms of one wave are shown. The full line A is the wave as it leaves the exhaust port, and the dotted line B shows the same wave after traversing 9 ft of pipe.

Resonance

As mentioned earlier at least one designer in the past has attributed a useful torque increase to the theory that the pipe was pulsating in its own period and in time with the exhaust valve, thus producing a good low-pressure period at the port at overlap t.d.c. Obviously, this can happen, as is seen from some of the curves. A more doubtful claim, however, is that such effect will then occur at

multiples of the engine speed; that is, if there is a good boost in torque at 1,000 rev/min, there will also be one at 2,000 (XVII). This cannot be confirmed from any of the curves; if we look at fig. 5 : 7(a) for 1,000 rev/min it bears no resemblance whatever to (b) at 2,000, whereas at 3,000 and 4,000 (curves (c) and (d)) there is at least a sub-atmospheric pressure where we want it.

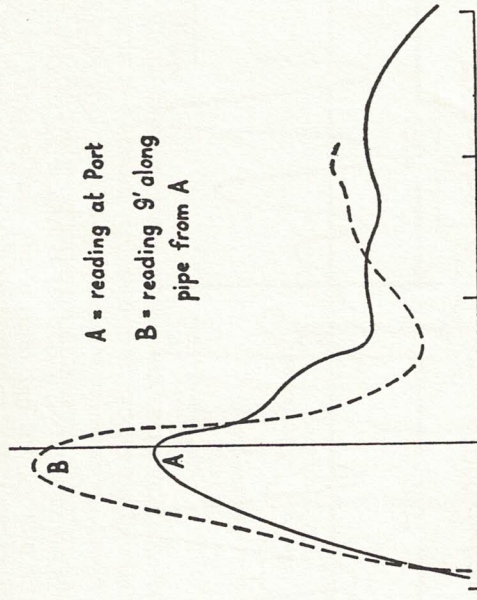


Fig. 5 : 11. Curves showing steep wave front at distance from source.

Cylinder vibration with exhaust stub only

On page 131 in the description of the test on the writer's silencer, referring to the curves on page 132, it is stated that the engine torque was rather better with that exhaust system than without it. Some diagrams were also taken of the pressure inside the cylinder when there was only the exhaust stub, 6 in. long. It discharged into a 5 gallon drum to reduce the noise, but that would have made no difference. Tests were made at 2,000, 2,500, 3,000 and 3,500 rev/min and two are given in fig. 5 : 12.

It is obvious that the period of vibration of the cylinder and stub, considered as a vibrating system, should vary greatly with the piston position, becoming much smaller as the piston nears the top. But this is evidently the case only to a small degree and it is interesting to calculate the periods of vibration for the half and quarter stroke positions. The expression for the frequency of a resonator with a

$$\text{short neck is } f = \frac{1}{4\pi} \sqrt{\frac{A}{v(L + 0.89\sqrt{A})}}$$

where V_s = velocity of wave in ft/sec.

v = volume of resonator in ft^3 .

A = cross section area of neck in ft^2 .

L = length of neck in ft.

Using a velocity of 1,400 ft/sec, the periods work out at 270 and 410 per sec for the half and quarter positions. From the curves the frequency at 2,000 rev/min varies from 240 to 270, and at the other speeds is about 300 per sec so the actual frequencies are less than the calculated ones.

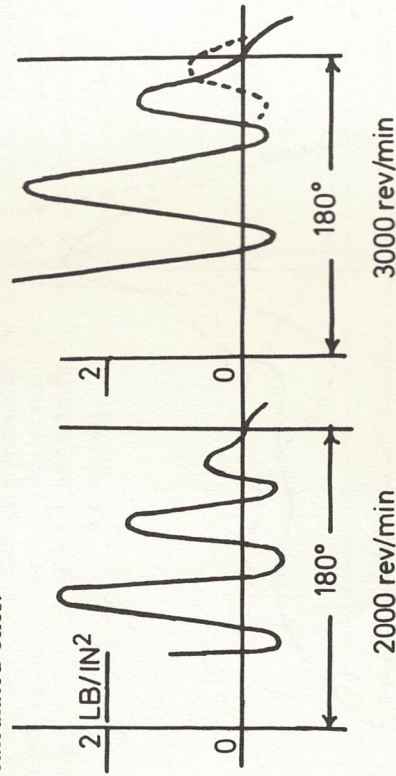


Fig. 5 : 12. Pressures in the cylinder with only a 6-inch exhaust stub.

Another point of interest is that the pressures are mostly well above atmospheric and the pressure at tdc depends on the timing of the final wave, e.g. at 2,000 and 3,000 rev/min it finishes on the atmospheric line but at 3,500 rev/min, shown dotted, it is a little above. By contrast, the diagram taken with the silencers at 3,500 rev/min shows an appreciable drop below atmospheric, which accounts for the slight increase in torque.

Pressure in long pipe

In considering fig. 5 : 7(d) it has been mentioned that after the initial pressure rise there is another rise during the exhaust stroke. At lower speeds the comparable curves do not show this second rise maintained to the same extent, the pressure dropping quite early from positive to below the atmospheric line. The magnitude of the rise at higher speeds is such that it cannot be the result of a residual wave, as such a wave would have had its effect earlier, and would in any case be too small. There is a good deal of evidence to show

(XXXIII) that at high engine speed the gas rushes into the pipe with a considerable velocity head. In so doing, it has to accelerate the gas already in the pipe and must therefore translate part of its velocity head into a static head. Further, the supply of this high-speed gas is limited, and soon tails off; this causes a decrease of pressure near the engine, and an increase 2 or 3 ft along the pipe. A pressure wave is thus started back towards the engine, and is added at the observed point on the curve.

Now if the pipe is increased in diameter by about 50 per cent at the point where the pressure increase takes place, the wave of increased pressure will almost disappear; the pressure drop is much more rapid, and this holds good even when the pipe length is in-

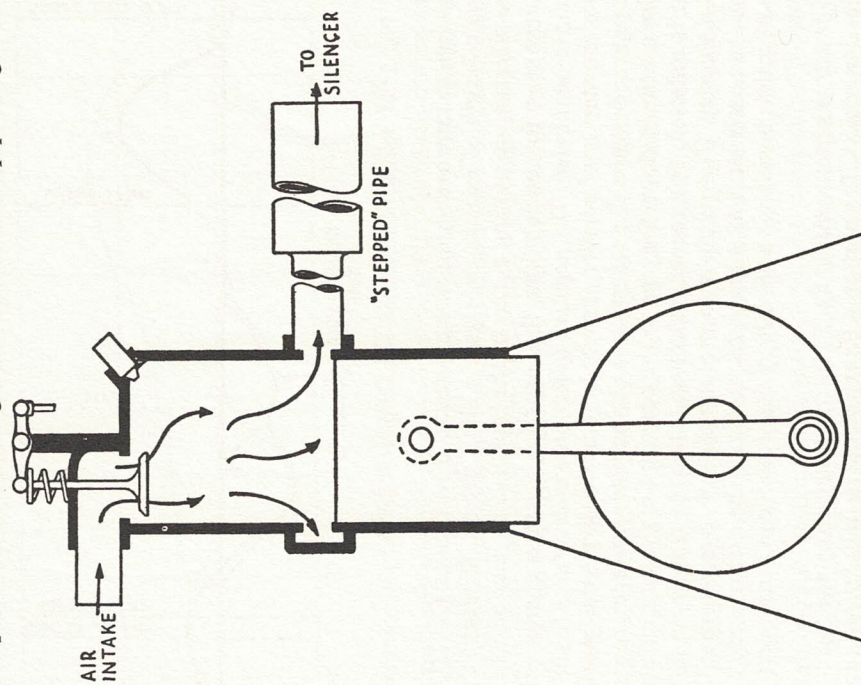


Fig. 5 : 13. Principle of self-induction diesel engine.