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Noise and Engineering Design

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Noise and engineering design

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1. Introduction

The paper presents a short account of various investigations carried out on the basic mechanism of noise produced by machines involving solid structures.

The emitted noise of the machine is generally due to bending vibration of either the outer or inner surfaces or both. In vehicles, for example, consideration of both the internal and external surfaces is of importance: vibrations of the internal surfaces produce noise inside the cab or car saloon and this affects the driver and passengers, while the external surfaces produce the noise which affects the community.

The vibrations which produce excessive noise are very seldom detrimental to the operation or the life of the machine. In many cases a noisier machine is more efficient and often has a greater life expectancy. A typical example is the comparison of the noise produced by a petrol and a diesel engine. The diesel engine operates at higher peak cylinder pressures and higher rates of pressure rise resulting from combustion. These factors are the primary causes of considerably greater noise. The diesel engine, however, is not only the most efficient prime mover, but also, because of its compactness, its life is *several times* that of a petrol engine. Operational life of over 200000 miles without appreciable maintenance is not uncommon.

There are many aspects of present-day machine design trends which follow general economic considerations, but which result in greater emitted noise. In transport, every effort is made to reduce the weight of the vehicle for the same load carrying capacity; examples of this are air cooling, the use of light-weight materials and higher operational speeds. Similar trends can be easily recognized in the production industry.

In general, the improvement in efficiency and economy of the machines results in higher levels of noise. This trend imposes greater difficulties on effective noise control. Thus in every case a clear understanding of the machine design, its operation and environment, is required; only then will worthwhile reductions of noise be achieved within the economic framework.

2. Mechanism of generation of vibration and noise in machines

When considering a simple single degree of freedom vibratory system there are two factors which determine vibration amplitude: (*a*) the characteristics of the force, and (*b*) the characteristics of the vibratory system. The emitted noise, however, will also depend on the size of the vibratory system in relation to the frequency of vibration.

For machines, which are generally very complex structures and which are set into vibration by numerous exciting forces of complex nature, the same basic principles apply. It could be stated that machine noise is determined by the form, the magnitude and the repetition rate of the exciting forces and the over-all structural-acoustical response, or to a very great extent by the relation between the predominant frequency response range of the machine and the characteristics of the exciting force.

The major and the most difficult task in the study of the machine noise is the determination and precise measurement of the exciting forces. The exciting forces in machines, apart from out of balance self-generated inertia forces of sinusoidal nature, are in general impulsive repetitive forces like the sudden pressure rise resulting from combustion in internal combustion engines, sudden loads applied in punch presses, and the sudden application of load between the gear teeth on making contact. These sudden applications of load induce either transient or resonant vibrations of the whole machine structure while its outer or inner surfaces emit the noise.

Two examples will illustrate the relations discussed. One of the examples is chosen where the repetition frequency of the force is some 1/20 to 1/100 of the frequency range of the predominant response, of the structure, while in another example the repetition frequency is only just below or equal to the predominant frequency range of the structure.

The first case applies to machines where the design is primarily governed by rigidity considerations, and such machines usually employ rigid cast structures, such as engines, gearboxes, machine tools, etc.

In figure 1 the origin and characteristics of a diesel engine noise are illustrated. Extensive investigations have shown conclusively (Priede 1966) that the predominant noise in a diesel engine is produced by the rapid rise in cylinder pressure resulting from combustion as shown in figure 1*a*. Since the force-time diagram is repetitive its excitation propensities can be fully described by Fourier series. If the actual force is recorded (electrical signal from a transducer proportional to force—in this instance cylinder pressure gauge) it can be readily analysed by an electronic wave analyser and thus a spectrum of the force as shown in figure 1*b* is obtained.

In the low-frequency range individual harmonics can be clearly distinguished, the level being primarily determined by the peak pressure. With increasing frequency the individual harmonics can no longer be resolved and the spectrum of high density is obtained in the high frequency range. The level of these high-frequency harmonics is determined by the actual form of the cylinder pressure diagram (Priede 1961). The amplitude of harmonics shows a general decay at the rate of some 30 dB per tenfold increase of frequency.

Response in terms of emitted noise of the engine structure to a constant sinusoidal force input of varying frequency is shown in figure 1*c*. Maximum response is in the frequency range between 1000 and 2000 Hz which is typical for a machine employing a rigid cast-iron structure. Thus the ratio of the frequency of maximum response to repetition frequency of the force is of the order 50 to 100:1; and the relevant part of the force spectrum which is responsible for the predominant noise is that of a high density.

The resultant spectrum of this diesel engine noise is shown in figure 1*d* which, as can be seen, is simply determined by both the characteristics of force and characteristics of structure.

It is of interest now to consider the effect of the operational speed. It is quite common that load or force diagrams when considered on the degree basis of the shaft rotation remain constant. This means that the machine is made to perform the same events during the period of one rotation irrespective of the speed of the machine. Thus, also in engines the cylinder pressure force (pressure diagram) tends to remain of similar form on a degree basis. Therefore the force spectra at higher speeds will be geometrically similar but with a shift parallel to frequency axis corresponding to the change of speed (Priede 1961). Because of this, the actual magnitude of the increase of the level of force, and thus the

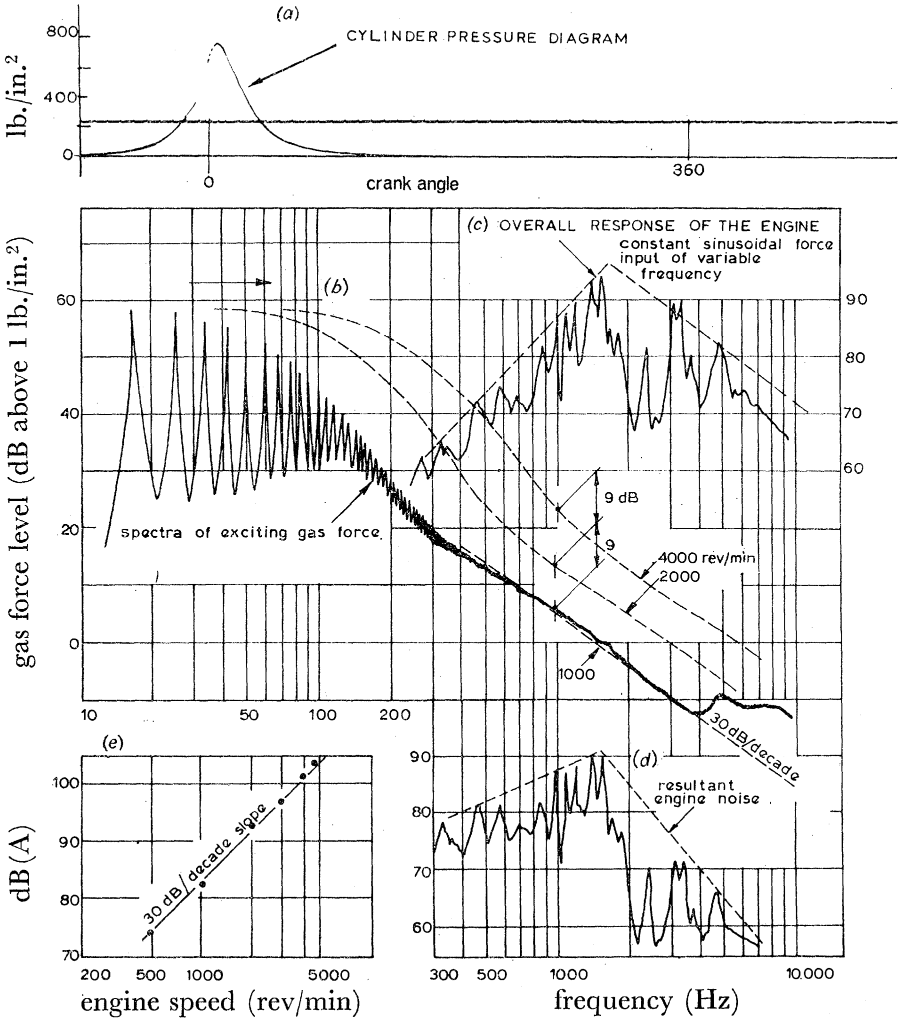

Figure 1. Characteristics of existing force, response and resultant noise of a diesel engine.

engine noise, will depend on the general slope of the cylinder pressure-force spectra. For example in this case the slope of the cylinder pressure-force spectrum is 30 dB/decade (as it is in most diesel engines) and therefore a 30 dB increase of noise with a tenfold increase of engine speed (or 9 dB for a twofold increase) can be expected. As can be seen, this is confirmed in figure 1*e* where measured over-all sound-pressure levels (dB(A)) of the engine are plotted against the logarithm of the engine speed.

The second case, where the ratio of the repetition frequency of the exciting force to the predominant frequency range of the structure is between 1 and 5, applies to noise produced inside vehicles. The predominant noise inside the vehicles is in the low-frequency range from 50 to 200 Hz which in very broad terms, particularly in the case of diesel vehicles, can be ascribed to torque fluctuations produced by the engine as shown in figure 2*a*. The form of the torque diagram (apart from some effect of the inertia forces at higher speeds) again remains the same on a degree basis. The repetition rate of this exciting torque is at the firing frequency of the four-cylinder engine.

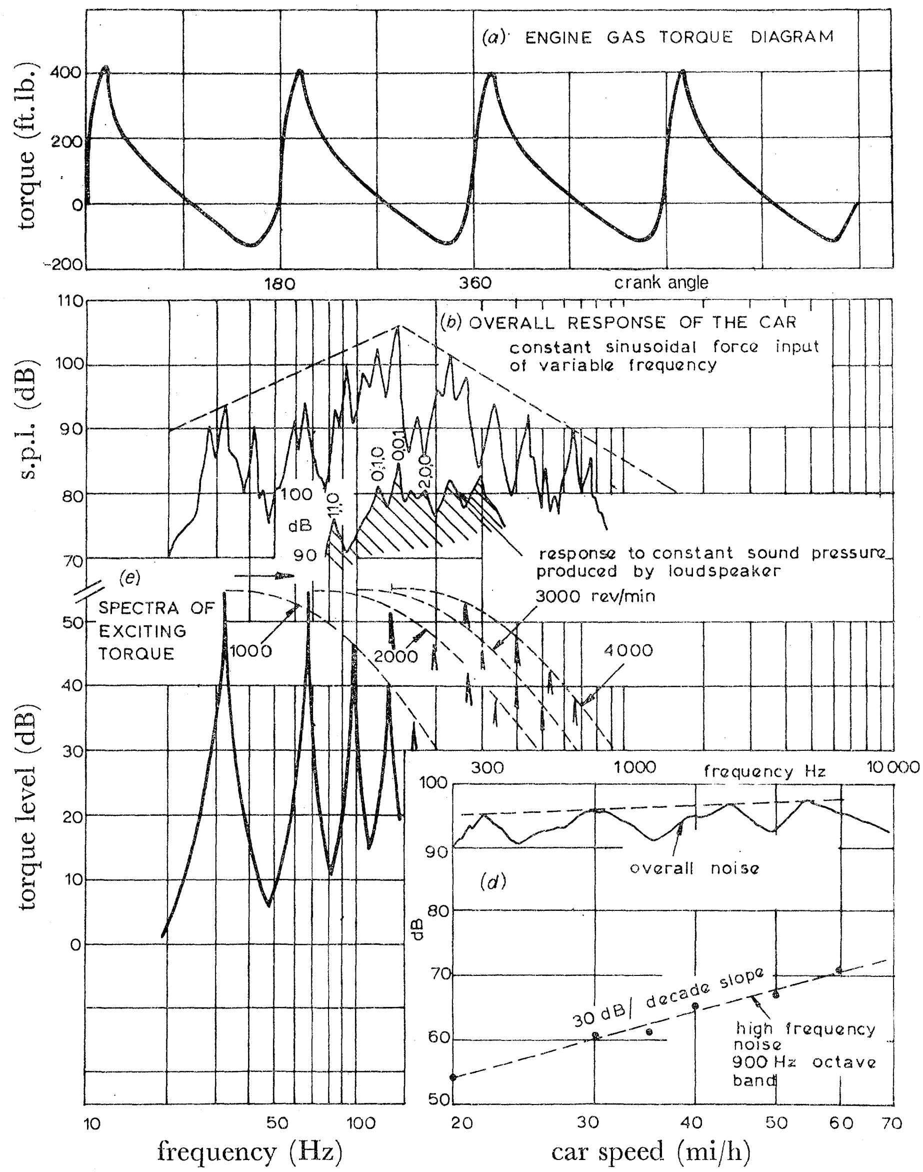

FIGURE 2. Characteristics of exciting force, response and resultant noise inside a car.

The response of the car structure in terms of the emitted noise is shown in figure 2*b*. The dashed envelope curve gives the general characteristic and its maximum is in the low-frequency range around 100 to 200 Hz. This is a typical response for structures employing thin sheet metal. As can be seen the acoustical modes of the car cavity also lie within the same frequency range.

The spectra of the torque fluctuations are shown in figure 2*c*. Also illustrated is the effect of increase of speed which is similar to the discussed in the previous example. The low-frequency excitation of the car structure results from the first few harmonics and thus over the whole operating speed range the excitation tends to be at a constant level. As the speed increases, different natural modes of the car structure are being excited in turn and for this reason a number of resonances can be expected which, as shown in figure 2*d*, give pronounced fluctuations in the over-all level of noise with speed. The general increase of noise inside the car, however (as shown by the dashed curve) between 20 and 60 miles/h is only 3 dB. In all cars or vehicles measured so far the increase of noise with speed very seldom exceeds 10 dB.

The high-frequency noise inside the car, which is due to transmitted airborne noise of the diesel engine through the car bulkhead, is found to increase by 30 dB per tenfold increase of engine (or car) speed, i.e. it is in agreement with the previous example already discussed.

It may be concluded that in the case of the car there is a most unfortunate choice of design factors which determine the noise, namely the repetition frequency of the exciting force coincides with the maximum response of the car within its operating range. Furthermore, the actual size of the car is such that its acoustic response lies within the same frequency range.

3. Effect of operating variables on the machine noise and its relation with the form of exciting force

Effect of speed

Numerous investigations carried out on a wide variety of machines employing rigid structures ($\omega_{\text{exc.}}/\omega_{\text{nat.}} = 0{\cdot}01$ to $0{\cdot}2$) have shown that the major factor which determines the increase of noise with operational speed is the form of the exciting force.

Some of the measured results are summarized in figure 3. For a diesel engine (figure 3*a*), as already discussed in §2, the increase of noise is 30 dB per tenfold increase of speed which is equivalent to the slope (30 dB/decade) of its force spectrum. The intensity of noise therefore is proportional to the cube of the speed, i.e. $I \approx N^3$, where N is engine speed (revolutions per minute). In a petrol engine (figure 3*b*) the pressure rise due to combustion blends gradually with the pressure rise resulting from compression and thus a very smooth force-time diagram is obtained. The force spectrum has a greater slope of 50 dB/decade and the resultant noise is found to increase for a petrol engine by 50 dB per tenfold increase of speed or $I \approx N^5$.

In hydraulic reciprocating pumps the pressure rise or release is very sudden owing to the use of virtually incompressible fluid as shown in figure 3*c*. The force spectrum, therefore, has an average slope of 20 dB/decade. The noise of such pumps increases by 20 dB per tenfold increase of speed or $I \approx N^2$. A similar relation is obtained between noise and speed of meshing gears. This reveals that the origin of gear noise is due to sudden application or removal of load between the meshing teeth. Many machines employ spring-operated valves. The seating velocity is determined only by the available spring force which does not change with operational speed. Thus the magnitude of the force impulse produced, as shown in figure 3*d*, does not change with repetition rate. The force spectrum is constant over the frequency range and noise has been found (peak values) to be independent of speed.

Noise from ball bearings (figure 3*e*) increases at a rate of 50 dB per tenfold increase of speed. It may be deduced therefore that the force spectrum should have a slope of 50 dB/decade. The most important parameter (Berry 1962) that determines the ball-bearing noise is the ball sphericity. Since the exciting force can only be due to eccentric ball rotation the resulting force–time curve should be reasonably smooth, which would account for the observed relationship between noise and rotational speed.

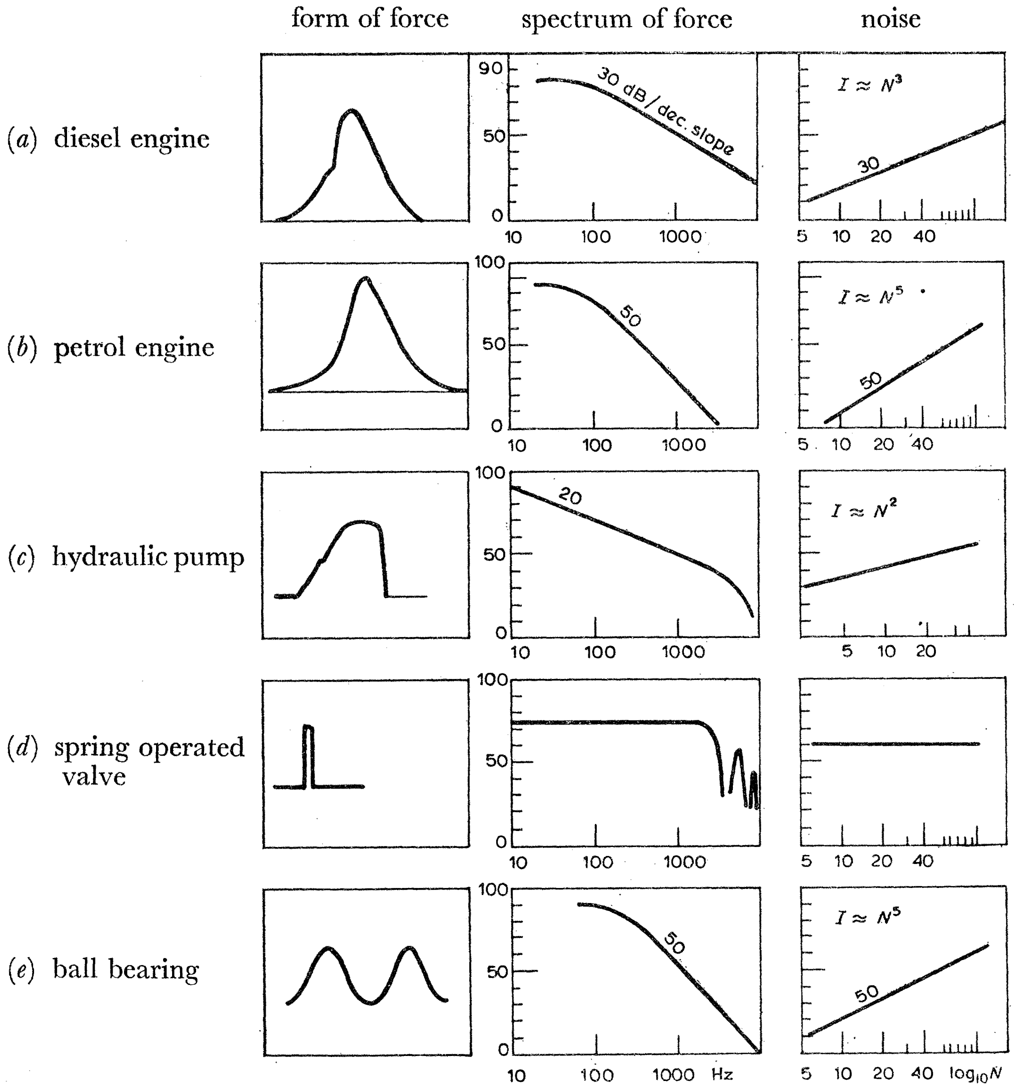

FIGURE 3. Relation between force–time diagram, force spectrum and noise.

Effect of load

Changes in the load of the machine generally produce marked changes in the exciting force–time diagram. These changes, however, do not necessarily increase the exciting propensities of the force. There are even many instances where the exciting propensities of the force are reduced with increasing load.

Some of such effects are illustrated in figure 4. For example, in diesel engine, noise is primarily produced by the rapid initial pressure rise. This initial pressure rise is more or less independent of the total amount of fuel injected (it is produced by the burning

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of first fuel droplets injected), and thus is more or less independent of load as shown in figure 4*a*.

The major differences in the pressure diagram with load is in the latter part of the pressure diagram during the expansion stroke where the pressure is already reducing. During this part most of the actual work is done, but its effect on noise is virtually negligible. In many cases, as a result of cooling effects at low loads, the initial pressure rise may be even steeper and thus a noisier condition is obtained.

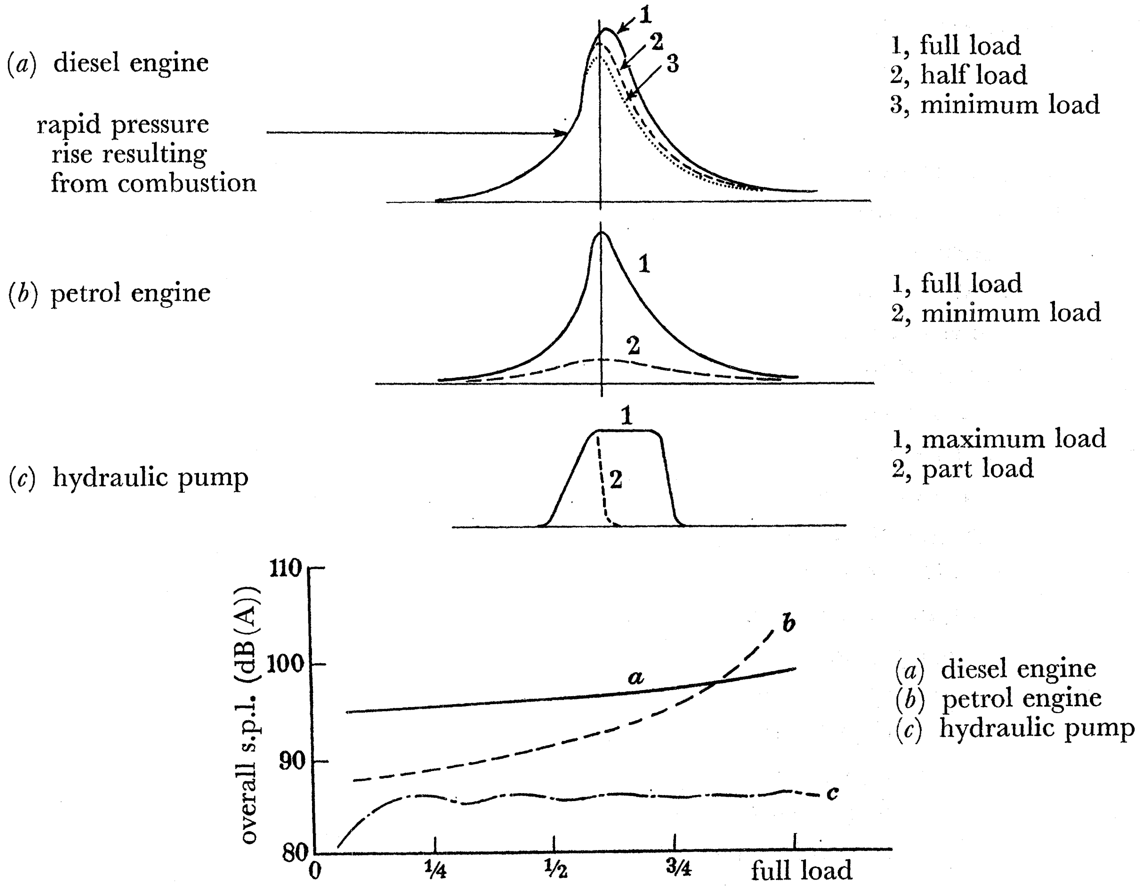

FIGURE 4. Effect of load on force–time diagram and noise.

In a petrol engine (figure 4*b*), however, the load is reduced by throttling the air intake thus reducing the cylinder pressure considerably. In petrol engines therefore the increase of noise with load is appreciable.

In hydraulic pumps (figure 4*c*) the peak pressure is about the same with increasing load, but the pumping time is increased. The relevant parts of the force–time curve which produce noise, namely the pressure rise and fall, remain exactly the same and thus the noise remains independent of load.

Figure 4*d* shows relation between sound pressure level (dB(A)) and load for various machines.

From the example discussed it can be concluded that there is no necessity for the noise of the machine to increase with load, but can, in some circumstances, reduce with reduc-

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tion of load. This contradicting statement can be explained by considering as an example the case of the petrol engine. Here the noise reduces as the load is reduced, but this, however, is achieved by paying the penalty of reduced efficiency. It would be far more economical to maintain higher pressures in a petrol engine at part load (employing variable compression ratio) and maintaining the noise at the same level.

Similar characteristics can be attained also in a diesel engine by retarding the injection timing with reduction of load. Great advances have been made recently on high output diesel engines by high supercharging, thus obtaining three to four times greater output at the same engine speed. It is interesting to note that in no case is there an increase of noise.

4. Effect of machine size

When a greater output of the machine is required which cannot be obtained by increasing the operational speed a larger machine has to be considered. The larger machine structure will be excited by considerably greater forces (in an engine, piston area is increased while the pressure per unit area is maintained the same) and therefore the whole structure is made stronger to keep the stresses at about the same level.

It is quite reasonable to assume (and has been verified by numerous experiments) that the level of vibration of the outer surfaces of machines of different size is about the same when operating at the same speed. Therefore an increase of noise by 13·3 dB per tenfold increase of the machine volume is expected due to the increase of the size of the radiating area. Experimental data have revealed (Priede & Grover 1966) that the increase of noise amounts to some 15–17 dB (i.e. an increase of sound pressure by 6 to 7 times) per tenfold increase of the machine volume.

The work output per unit time is, however, about 15 times per tenfold increase of the volume thus increasing at a greater rate than the emitted noise. This leads to a very common observation that for the same work done per unit time or horsepower the larger but slower machine is considerably quieter. In any machine there is always some design parameter which limits its operational speed. In rotating machines the limit is imposed by the tip-speed of the rotor mainly to keep the stresses at a reasonable level. In reciprocating machines the limit is piston speed. Thus the larger machine has to operate at a lower speed, for example, for diesel engines a relation between the rated engine speed and cylinder capacity has been derived (Priede 1958) giving maximum acceptable speed for an engine of any size:

$$N = 2500\, V^{-0{\cdot}475}$$

where N is the number of revolutions per minute and V is the cylinder volume in litres.

Owing to the combination of the effects of the various design and operational parameter already discussed it is generally found that the noise emitted by any type of machine tends to be of the same level and is independent of its horsepower or the work done per unit time.

In figure 5*a* are shown the levels of noise emitted by Rootes compressors of various size. The noise in the Rootes compressors is produced by a vibrating surface presented to incoming air which is formed by the rotor rotation. When the noise data (dB(A)) are plotted against tip speed it is found that the noise is of about the same magnitude within some 5 dB. The result is that with a large compressor about 6000 ft^3/min free air delivery can be made at the same level of noise as a smaller one delivering only 180 ft.^{3}/min which is a ratio of the work output per unit time of the order of 30:1.

From figure 5*b* it can be shown that the same argument applies to diesel engines when the level of noise is plotted against maximum piston speed. As can be seen, a 30 l./cylinder engine running at 500 rev/min develops 6000 h.p. at the same level of noise as a 0·37 l./cylinder capacity engine developing only 42 h.p. at 4000 rev/min; the ratio of power is 143:1.

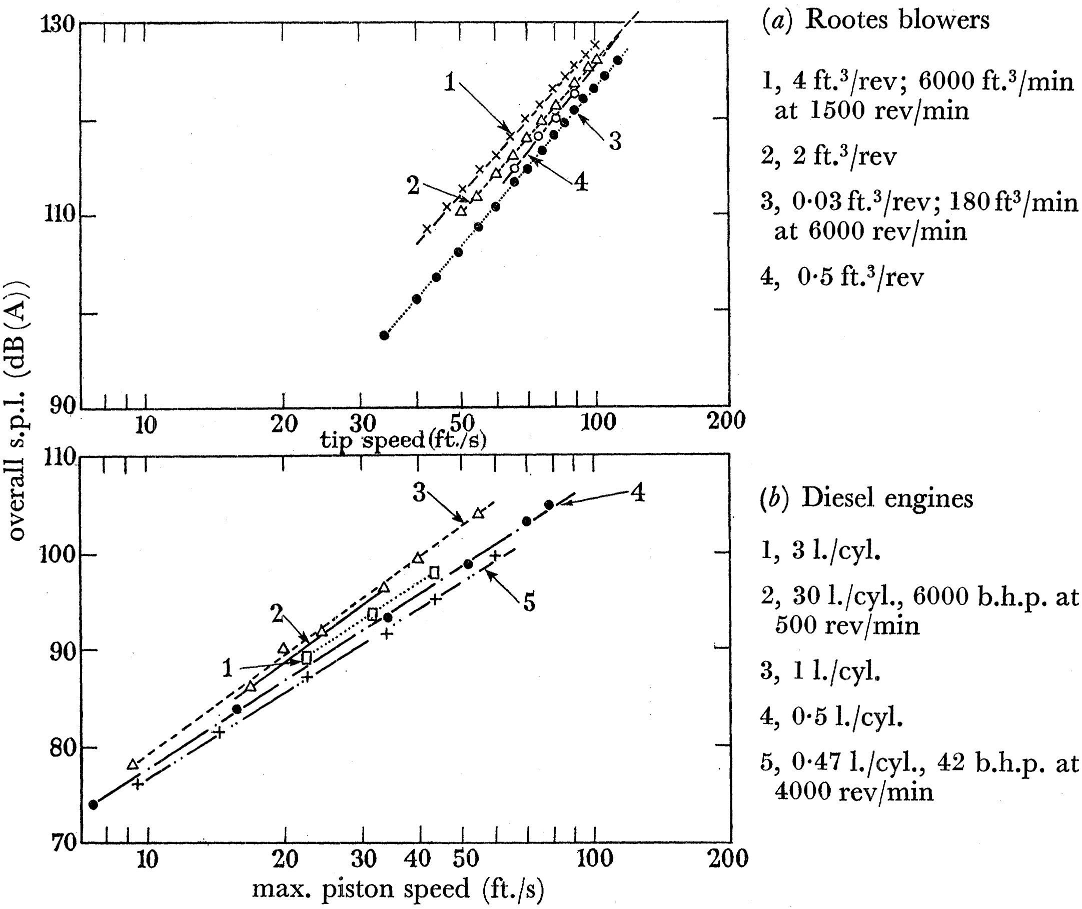

FIGURE 5. Noise of (*a*) Rootes blowers and (*b*) diesel engines of different size and output.

This phenomenon tends to be true for many appliances producing noise like ventilation systems where the noise again is only functional of the flow velocity (V^6) and not the volume of air displaced per unit time.

It could therefore be concluded that noise is generally independent of the volume of work done per unit time, or horsepower, but the main criterion is the operational speed or how short is the time interval within which the operation of one cycle of events is being performed by the machine.

5. Basic principles of noise control

Two aspects of noise control require consideration, namely, reduction of exciting propensities of the force or reduction of structural response. Which of the two may be more effective depends on very careful analysis of the relation between the force and machine structure in each individual case. The efficiency of the machine surfaces as sound radiators is a factor of importance, but it is primarily determined by the physical size of the machine which, for an intended design, can seldom be sufficiently altered.

Reduction of noise by change in the form of exciting force

It has been shown already that the exciting propensities of the force can be fully described by Fourier series (spectrum) and therefore any reduction of the force spectrum gives a corresponding reduction in noise.

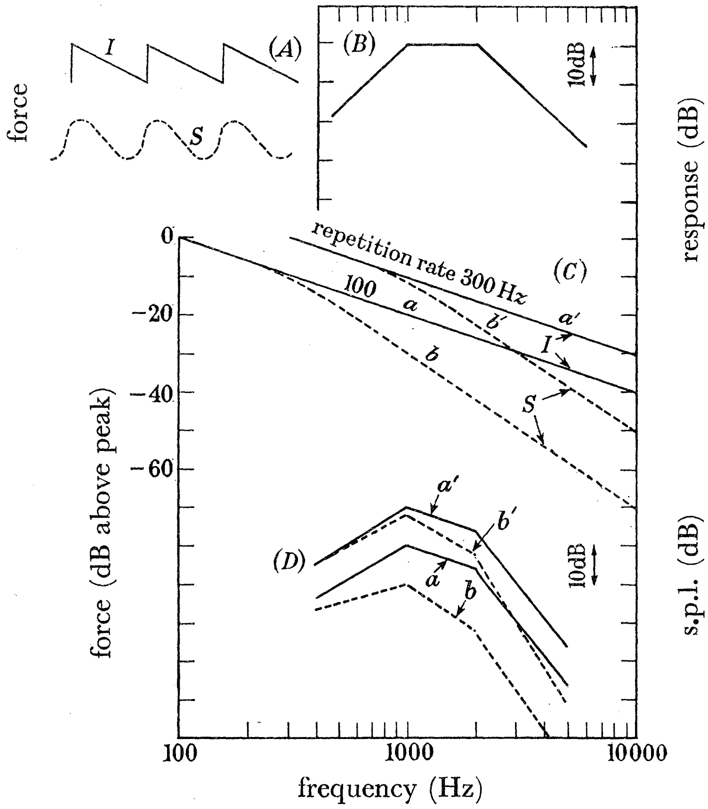

FIGURE 6. (*A*) Force-time waveforms; (*B*) structure response; (*C*) force-time waveform analysis; (*D*) resultant noise emitted. (*I*, instantaneous; *S*, smooth.)

There are, however, some limitations which require consideration: very seldom is an effective reduction of the magnitude of force possible (i.e. its peak value) since the machine is designed to carry out particular prescribed work. The only effective method therefore is the modification of the form of the force (i.e. smoothing) which reduces the high-frequency level of the force spectrum. There are, however, many instances where this procedure has been adopted and the results have been disappointing because the exact relation between the repetition frequency of the force and the frequency range of maximum response has not been taken into consideration. This may be illustrated by a hypothetical example shown in figure 6.

Two force–time waveforms which can be found in machines are considered: (*a*) one with a sudden application of force at a repetition frequency of 100 Hz, and (*b*) one where by careful design of the operation of the machine an extremely smooth rise of force is obtained performing the same task and operating at the same repetition frequency.

Analyses of these two waveforms (spectra *a* and *b*) show that, apart from the first five harmonics, there is a substantial difference in the exciting propensities of the two different force–time diagrams. The harmonics decrease for the applied force (*a*) at the rate of 20 dB/decade, while for more gradually applied force (*b*) by 40 dB/decade.

Also shown in the figure is an outline of the response (in terms of emitted noise) of a machine structure to a constant sinusoidal force input of varying frequency. Maximum response assumed is in the frequency range 1000 to 2000 Hz which is typical for a machine employing rigid cast-iron structure. The resultant noise emitted by this structure is determined by the two spectra of the applied forces.

Compared with the suddenly applied force (*a*) there is a substantial reduction of noise with the smooth force (*b*)—10 dB at 1000 Hz and 17 dB at 2000 Hz. It can be calculated from this that the loudness of the machine is reduced by half by the smoothing of the exciting force.

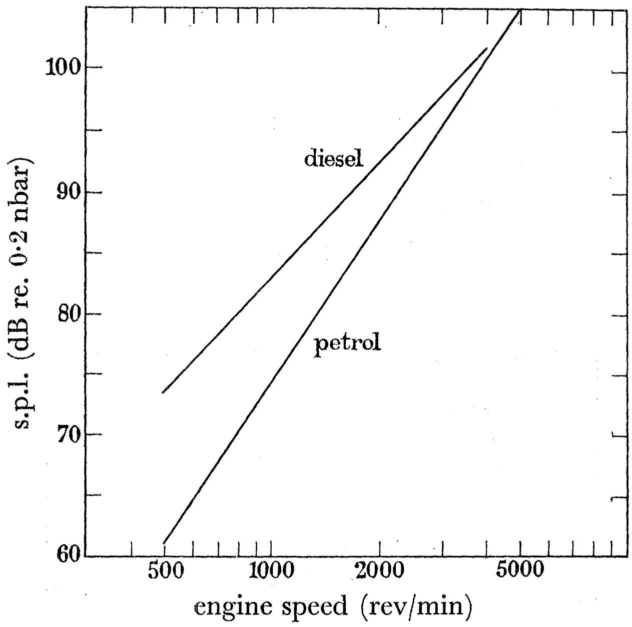

FIGURE 7. Variation of noise with speed for petrol and diesel engines (3 l.).

If it is decided, however, to operate the machine at three times the speed (three times the repetition frequency, i.e. 300 operations/second) the form of the force–time waveform will remain the same, and it will be bodily shifted along the frequency axis as shown by analyses *a* and *b*. The differences in levels of the two force spectra in the frequency range 1000 to 2000 Hz (maximum response of the structure) are considerably smaller.

In both cases the noise emitted by the structure increases due to the higher repetition rate, but the advantage of the smoothing of the force is now very small: 1 dB at 1 kHz and 5 dB at 2 kHz. Further increase of the repetition rate will diminish the advantage even more.

It can be concluded that a worthwhile reduction of noise by attention to the form of the exciting force can only be obtained provided the repetition frequency of the operation is low in comparison with the frequency range of the maximum response of the structure. At high repetition rates the form of the exciting force becomes less important.

The discussion above directly applies when comparing the noise of a petrol and diesel engine (smooth against abrupt pressure force). Figure 7 compares the variation of noise with speed for the same engine structure running as petrol and diesel engine, i.e. the ultimate effect of the smoothing of the force–time curve. As can be seen there is a marked difference of noise at low speeds (where the form of excitation is important) but at high speeds the noise is the same.

Control of noise by machine design

Control by isolation

In cases where the control of exciting force is impracticable it can be isolated from the outer surfaces of the structure. Often high vibratory forces originate from small components which do not emit high levels of noise because of their small physical size. Nevertheless, they can be a source of large vibratory forces. Examples are: isolation of small electrical motors and compressors in domestic appliances—refrigerators, washing machines, vacuum cleaners, etc.

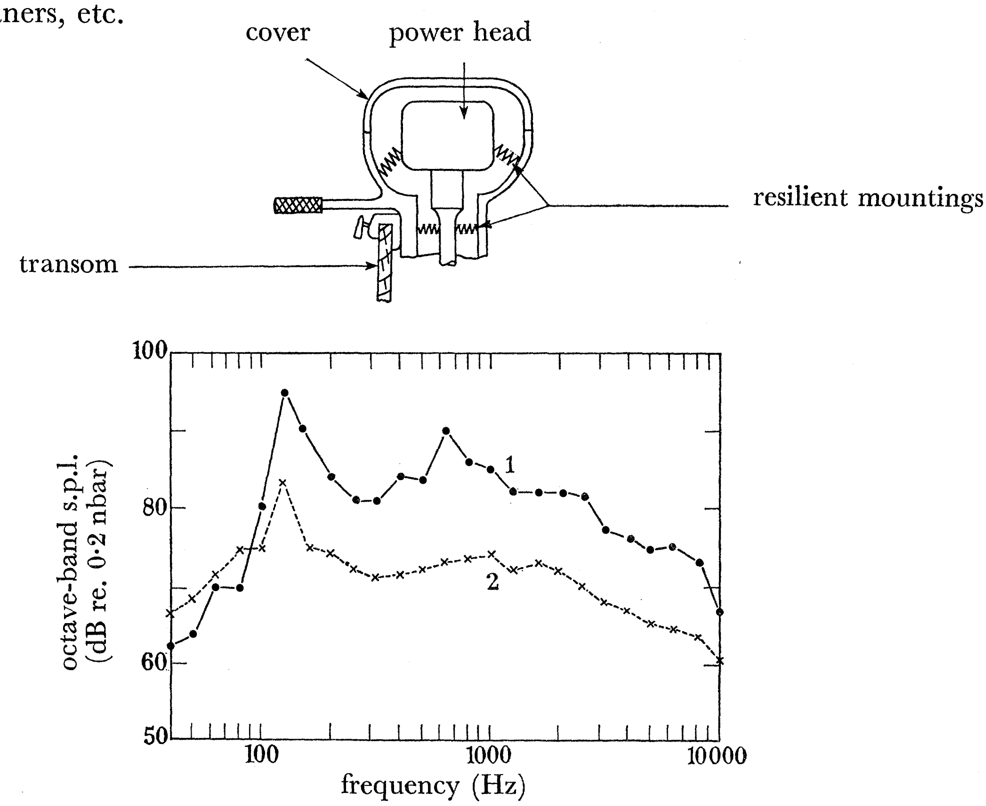

FIGURE 8. Effect of mounting the power head of an outboard motor resiliently in cover (curve 2); rigid power head, curve 1.

Isolation of vibration is generally achieved by mounting the component resiliently on the machine structure so that the frequency of the component on its mountings is very low compared with that of the exciting forces.

Limitations of isolation occur when any relative movement of component parts, no matter how small, cannot be tolerated such as in precision gear transmissions.

Another approach is to isolate the entire outer surfaces of the machine from the fully assembled working mechanism. One of the most successful applications of this method giving large reductions of noise is that of the outboard motor as shown in figure 8. The working mechanism, i.e. power head (engine) and transmission are flexibly mounted inside a light alloy casting which is rigidly connected to the boat. A reduction of noise by 10 to 15 dB is obtained over most of the frequency range compared with the outboard motor of conventional design. The space inside the cover also serves at the same time as an effective air inlet silencer.

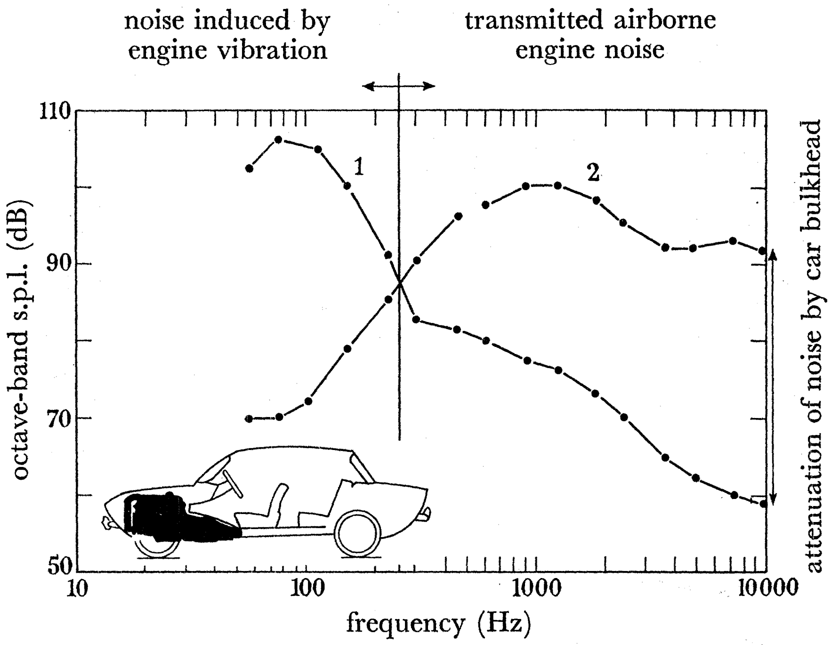

FIGURE 9. Comparison of noise of a diesel car and its engine on a test bed. 1, Noise inside the car (distance from engine approx. 4 ft.). 2, Engine noise on test bed 3 ft. distant.

In road transport vehicles, isolation of engine vibration, although effective at high frequencies, has limited success at low frequencies. An engine having its inlet and exhaust adequately silenced emits very little low frequency noise (up to 200 to 300 Hz), because of its small size, as shown in figure 9. Inside the car, however, the low-frequency noise in this frequency range is greater by some 20 dB because of transmitted vibratory forces through the rubber mountings. The failure here to attenuate these vibratory forces is due to the low stiffness of the car structure.

Control by structural design

In machine design of earlier days moving parts and mechanisms were often left exposed. In modern practice totally enclosed machine design is demanded by the use of forced lubrication necessitated by the higher running speed, protection of the mechanisms which are more intricate and for generally pleasing appearance. Figure 10 illustrates this design trend for a lathe. The practice 100 years ago shows a solid and functional design with a relatively small surface area. The modern practice has led to box-type design of a lathe with large noise emitting surfaces. This form of construction usually incorporates heavy sections to carry the main loads (ribbed bosses for journals, heavy flanges for attaching component parts) and thin walls which form the actual enclosure which are also integral parts of the casting. The thin walls exhibit considerably higher levels of vibration than the main load carrying members of the structure since vibrations impressed at the edge of a thin-walled section are enhanced towards the centre. An example is shown in figure 10*c* for a box-type cast structure. There are clear nodal lines along the thick vertical members while vibration of the thin walls can be as much as 20 dB higher.

The trend described above is inevitable since such structures have considerable advantages from other points of view. Similar changes have occurred in vehicle and car design where the whole body is made as a load carrying structure. Therefore if any reduction of noise is to be achieved it has to be within the framework of these trends.

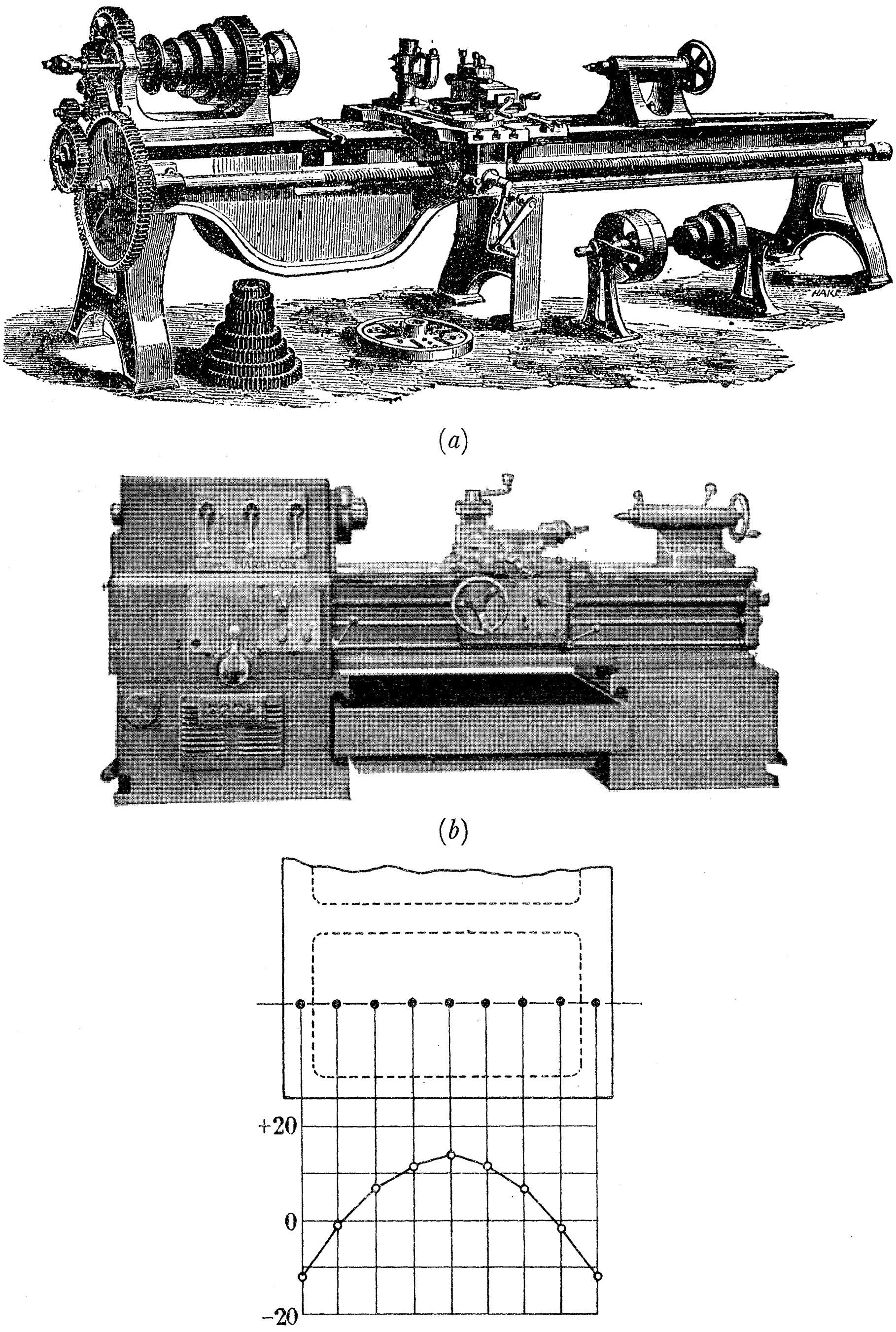

FIGURE 10. Trends of lathe design. (*a*) Lathe of 100 years ago; (*b*) present-day lathe; (*c*) variation of high frequency vibration (800 to 10000 Hz) over box-type structure.

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A designer has three factors at his disposal to control vibration, namely mass, stiffness and damping, which will be discussed in the following sections.

Control by mass

An increase of the total weight of a machine is not desirable since it represents a direct increase in material cost and in some applications such as transport where increase of weight also affects the economy of operation. Already thicknesses of a modern box-type machine design are made as thin as possible and any components attached to these will set up vibration as with the sounding board of a musical instrument. It is important, therefore, to fit any component that originates vibratory forces to the more rigid and heavier sections of the machine structure. Control of vibration by local concentration of mass is therefore obtained.

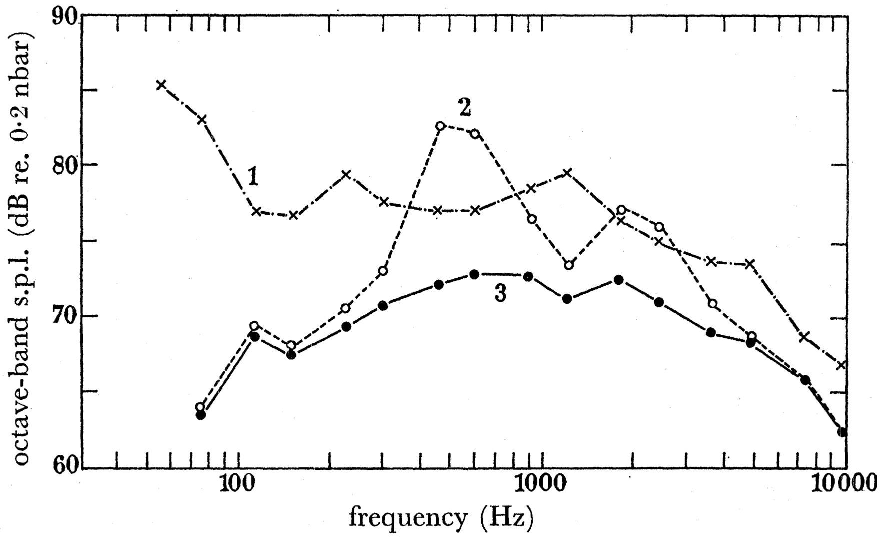

FIGURE 11. Effect of modified mounting table on noise of fuel injection pump test machine. 1, Running 9 l. engine, 500 rev/min; 2, original testing machine, 250 rev/min; 3, modified testing machine, 250 rev/min.

The noise of fuel injection pumps are often judged by the noise they emit on testing machines. Some of these testing machines employ thin T slotted mounting tables of large surface area producing high levels of noise. It has been shown (Priede 1958) that by using a smaller mounting table of substantial mass a reduction of noise can be obtained without increasing the total weight of the machine. Figure 11 shows the octave-band noise spectra of the original and modified machine. A reduction of noise by 20 dB at 500 Hz is obtained and a somewhat smaller reduction at higher frequencies.

Control by damping

Damping is an inherent property of a material. Unfortunately materials with high strength (steel, high-quality cast iron, aluminium alloys, etc.) have very little damping, while materials with high damping (lead, rubber, soft plastics) have very little strength and thus cannot be regarded as structural materials. In an assembled machine, however, another factor of greater importance is that of frictional or slippage damping induced by rubbing between various surfaces of the machine.

As can be seen for the control of vibration a designer has three possibilities, employing inherent damping of the structural material, high damping which can be obtained by special composite materials employing highly damped materials (non-structural) in conjunction with structural materials or employing slippage or frictional damping.

Slippage damping is found in machines with many sliding surfaces and bolted sections. Even with a highly resonant basic structure (made from steel or cast iron) enormous damping is introduced into the system by sliding surfaces when component parts are fitted. A bare engine crankcase shows a dynamic magnification factor Q of over 200 but the same crankcase in a fully assembled engine shows a Q of only 16. In this case it would be doubtful whether the engine would be very much quieter if the crankcase were made from a material with a higher damping coefficient than cast iron. All the experiments carried out so far by the writer indicate that in assembled machines the inherent damping of the structural material is completely irrelevant. Tests carried out on gears subjected to impulsive torque showed no difference in emitted noise between high-quality steel ($Q = 4000$ of an individual gear wheel) and copper manganese gears ($Q = 30$). The assembled gear train in both cases had damping equivalent to $Q = 12$. This illustrates the important part played by damping due to sliding.

Composite materials employing constrained damping layers (sandwich type materials) have shown considerable promise. When these are used as wall materials on a skeleton-type load-carrying structure it is found that generally vibration is considerably attenuated with distance from the point of excitation. Laboratory experiments of vibration of various plates clamped on a rigid framework and excited by an impulsive vibratory force show that plates with low inherent damping emit higher levels of noise. As shown in figure 12 the noise emitted by undamped plates increases towards the centre by 10 dB ($Q = 15$) and reduces towards the centre in the case of sandwich-type damped plates by 20 dB ($Q = 2{\cdot}5$). A total reduction of noise at the centre of the plates of 30 dB is obtained.

A diesel engine was built on this principle with a skeleton-form load-carrying structure with bonded outer walls (Priede, Austen & Grover 1964) of sandwich-type material. A reduction of the predominant high-frequency noise of 12 to 15 dB was obtained.

Control by stiffness

Since most of the noise is produced by bending vibrations of the outer surfaces, another way to reduce the emitted noise of the machine is to increase the bending stiffness of the basic structure and its outer walls. This approach has been tried from time to time by a number of engine manufacturers. In each case conventional methods of stiffening have been adopted, i.e. additional webs and heavy ribbing. Such experiments have so far all been disappointing and even resulted in a noiser engine in some cases. This can be reasonably explained by the fact that the natural frequency of a crankcase wall could at best only be raised by a factor of 2 by this procedure, for example from 1000 to 2000 Hz. The vibration levels of the crankcase may well be somewhat lower, but the efficiency of sound radiation will improve with frequency by more than any reduction in vibration levels.

Observations have also been made in many instances when a quieter machine is obtained with walls made of very low-bending stiffness, i.e. low-natural frequency, particularly when the wavelength of sound is large compared with the size of the machine surface. This has been verified also in a number of applications on various cover designs for engines and machines. Invariably it was found that cast covers made from aluminium (thickness of about $\frac{3}{16}$ in. to $\frac{1}{4}$ in.) emitted more noise particularly at high frequencies than light weight pressed steel covers.

This means that in a design where there is a bad region of chosen stiffness for a machine wall, improvement can be expected either by increasing or reducing stiffness.

The investigation of various wall materials for an engine as shown in figure 13 illustrates the points made.

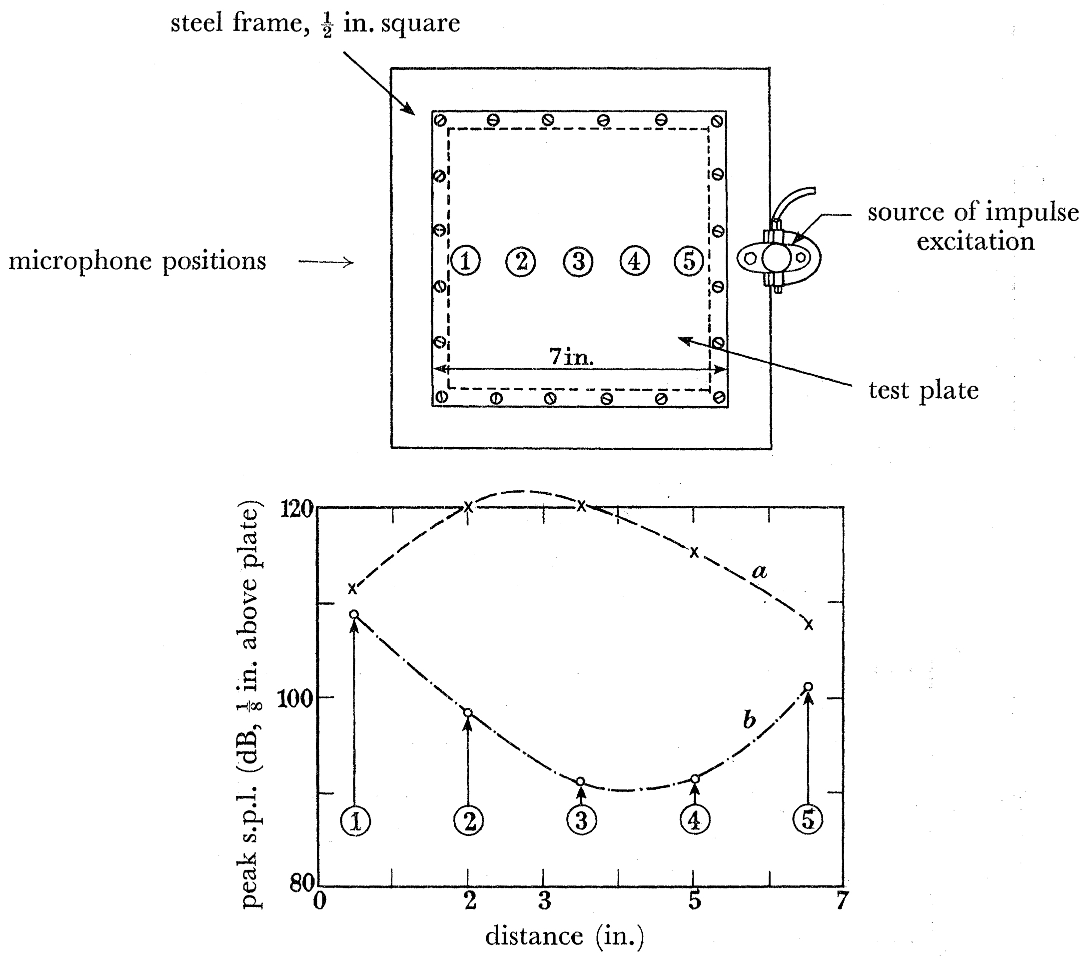

FIGURE 12. Effect of damping on noise emitted by clamped plates when excited by an impulse vibratory force. (*a*) $\frac{3}{32}$ in. aluminium plate; (*b*) $\frac{3}{32}$ in. damped plate (Al/rubber/Al sandwich).

As can be seen a wall made from thin steel sheet ($\frac{1}{32}$ in. in thickness) does give rise to a marked peak in noise at its natural frequency of 500 Hz (measured at very close microphone position of $\frac{1}{8}$ in. from the surface thus representing the vibration of the surface), but at frequencies of 1000 Hz and upwards the emitted noise is as much as 6 to 10 dB lower than the $\frac{1}{8}$ in. thick steel or magnesium plate or the $\frac{3}{16}$ in. wall of the normal cast-iron engine.

With the thin-wall material it was found that very little damping material is required to reduce the low-frequency peak at 500 Hz.

Investigations were also made on an engine design to increase the stiffness of the wall and its natural frequency to a far greater amount than is normal practice. By using a low-density material such as magnesium it was possible to design an engine having wall thicknesses of about five times those of an equivalent cast iron one without increasing the weight. As shown in figure 13 the noise emitted by the engine walls is again substantially

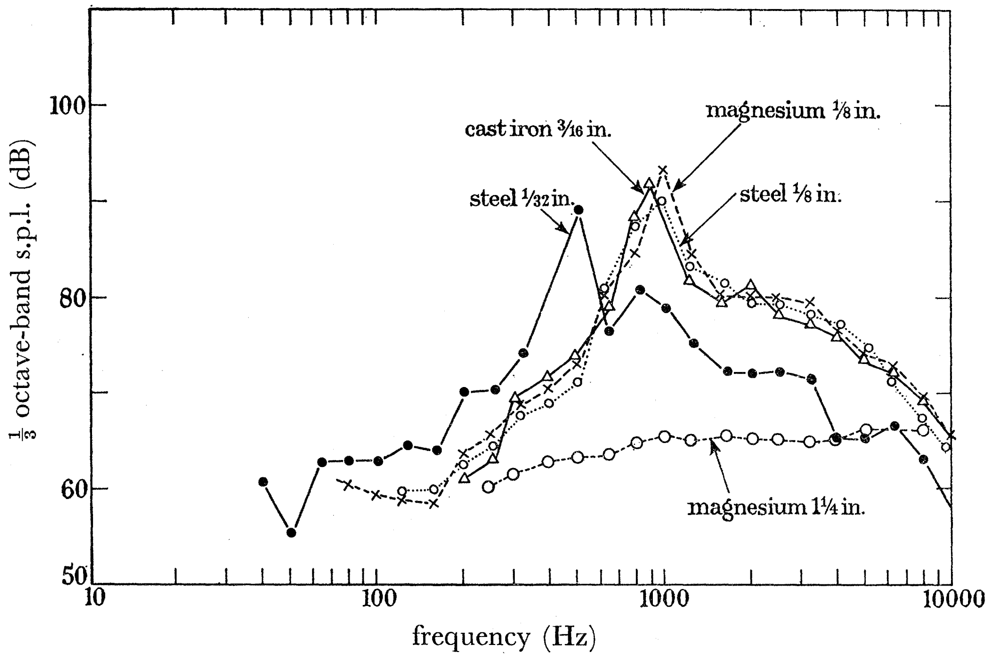

FIGURE 13. Relation between the engine wall material and its thickness and the noise emitted.

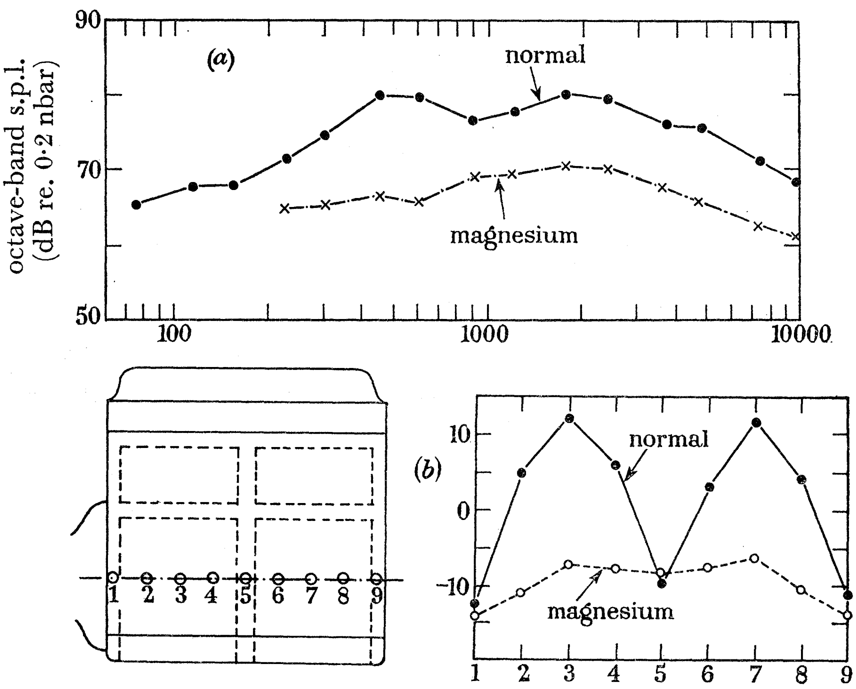

FIGURE 14. Comparison of noise and vibration of normal and magnesium engines. (a) Noise at 1000 rev/min; (b) vibration at 2000 Hz.

reduced. Figure 14 shows the characteristics of magnesium engine wall vibration and over-all reduction of noise.

It may thus be concluded that two possibilities emerge for the basic machine structure design: (*a*) skeleton or space frame load carrying structure with thin walls which can be readily damped; or (*b*) solid structures of large wall thicknesses of low density materials.

Both methods require a great deal of research to provide the designer with the necessary precise design data, and also acceptable manufacturing techniques need to be developed.

In the first case where the outer walls are regarded as separate entities from the load-carrying structure the optimum methods of fixing the walls require solution (bonding, bolting or cast-in sheet steel or other wall materials).

In the second case the light-weight material magnesium is very expensive when compared with cast iron and it has many inherent disadvantages such as liability to galvanic corrosion and low ultimate tensile stress. Therefore research is required on new materials which are cheap and easy to produce and which would satisfy stiffness and weight requirements. Work is in hand to develop expanded cast iron with an impervious skin resembling in section expanded rubber or plastic. With such a material noise problems in many machines could be simply and cheaply dealt with.

6. Conclusions

There is a definite need to deal with the problem of increasing machine noise at the design stage to offset the present trend of increasing density of machines and faster operational methods. Richards (1965–6) has shown that noise levels in many industries have already reached a danger point and that further increases cannot be tolerated if the hearing of those working in the factories as machine operators is to be conserved.

It is shown that there is no relation between the work done per unit time and the noise emitted, but the major factor which is responsible for increasing the noise is the operational speed.

Two approaches for the reduction of the noise have been discussed, namely the control of the exciting forces which set the machine structure into vibration and the control of the response of the machine structure, in particular the outer surfaces.

The primary aim should always be the control of the characteristics of the exciting forces but this becomes progressively less important as the operation speeds increase.

The paper outlines also a number of possibilities of machine design involving various combinations of the use of these properties, namely mass, stiffness and damping. It is shown that reductions of the order of 10 to 12 dB can be obtained by structural design alone.

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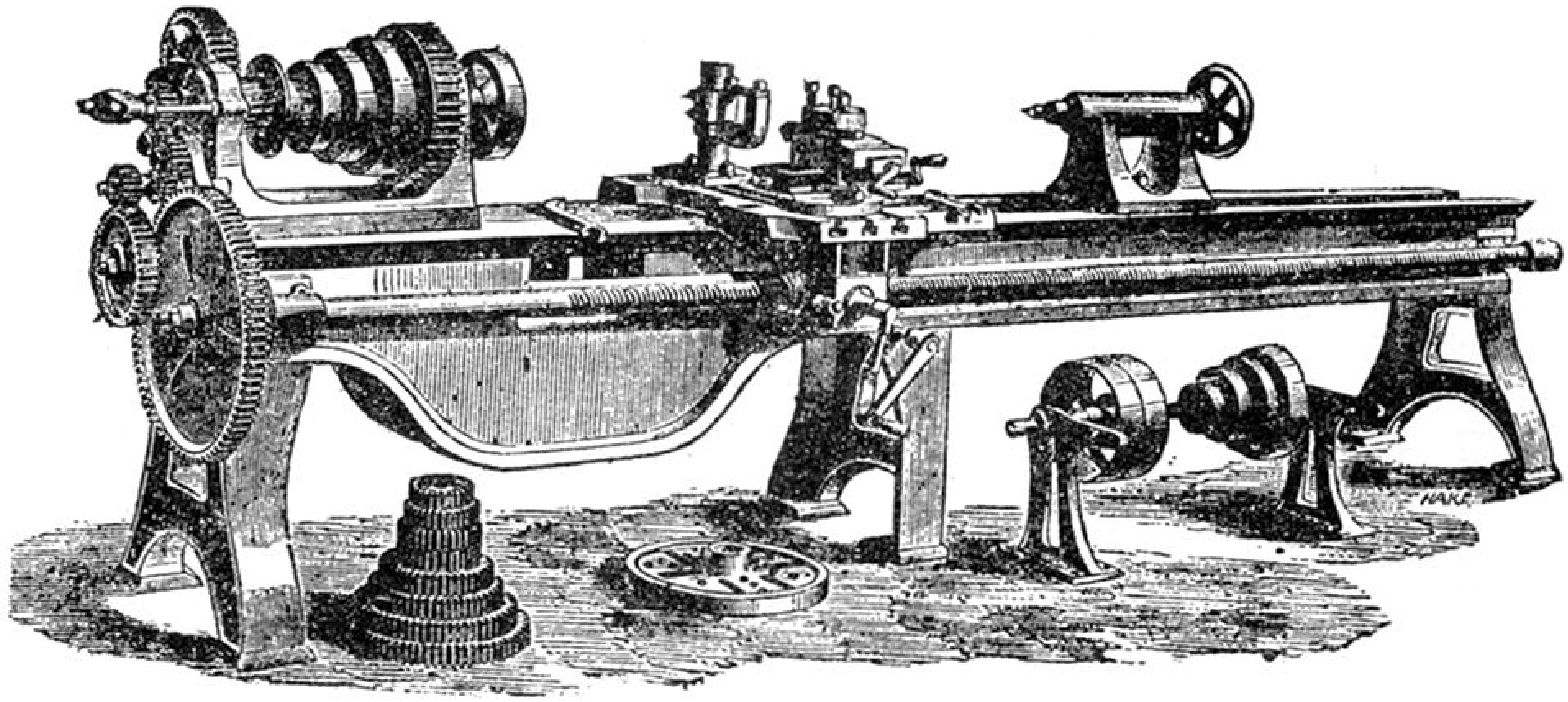
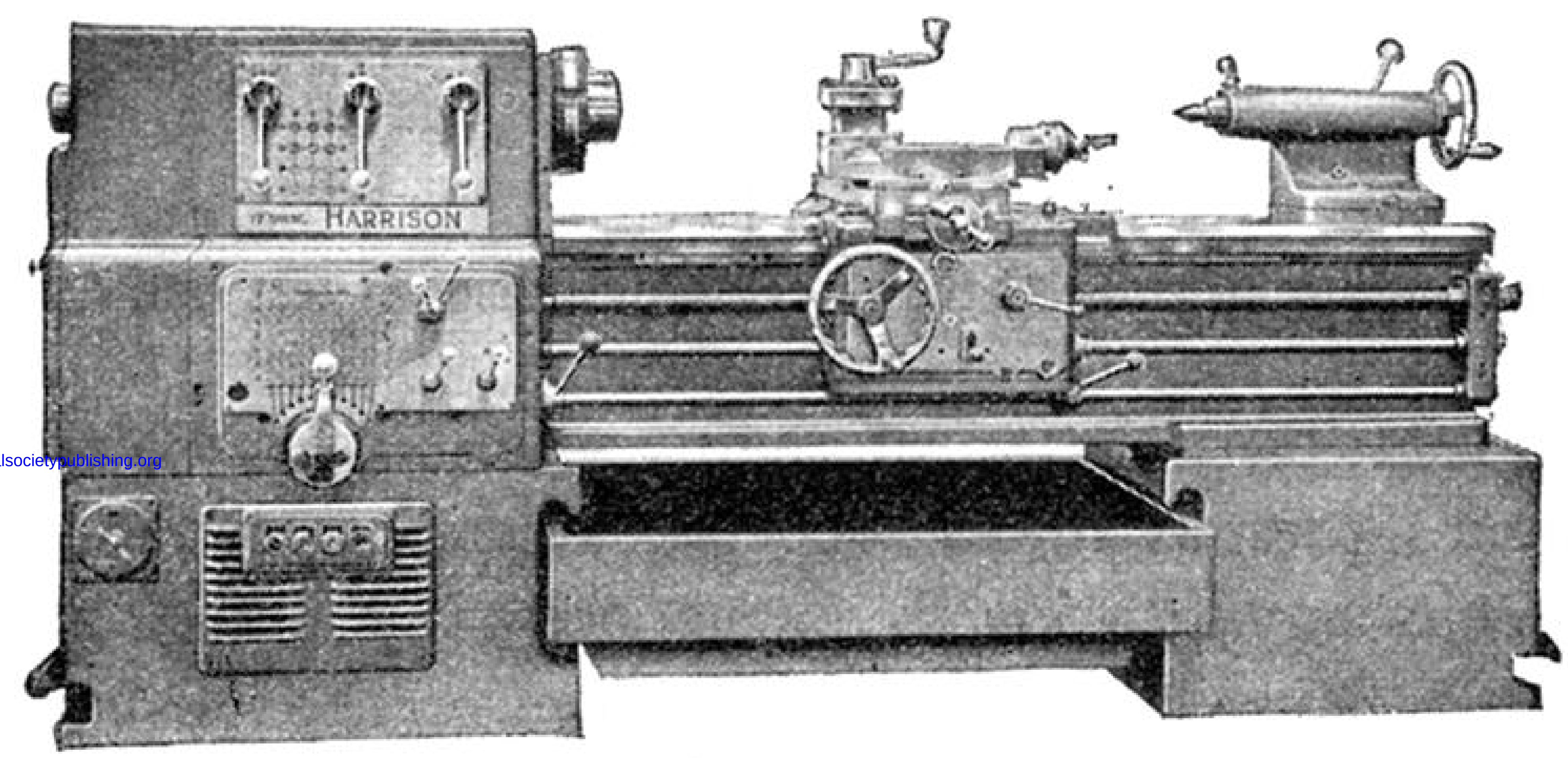


(*a*)



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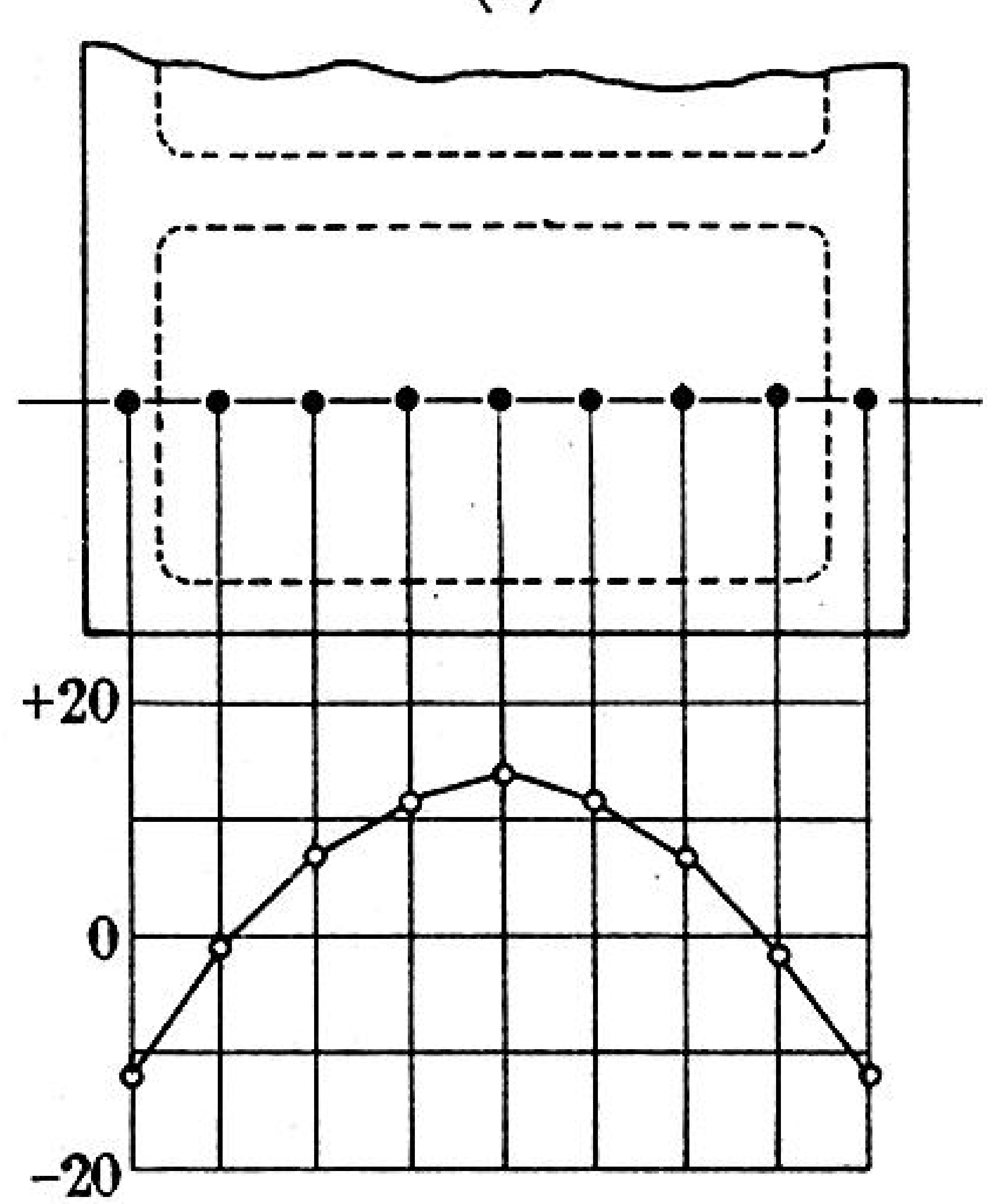

(*c*)

FIGURE 10. Trends of lathe design. (*a*) Lathe of 100 years ago; (*b*) present-day lathe; (*c*) variation of high frequency vibration (800 to 10 000 Hz) over box-type structure.