CHAPTER 10

MECHANICAL NOISE SOURCES

The noise sources described in Chapters 3, 4, 7, 8 and 9 have been predominantly hydrodynamic. The present chapter discusses basic mechanisms by which mechanical sources generate structure-borne vibrations, the transmission and radiation of which were covered in Chapters 5 and 6. Phenomena such as mechanical unbalance, electromagnetic force fluctuation, impact and friction are often the mechanisms whereby noise is generated by turbines, motors, transformers, gears and reciprocating machines.

No useful mechanical process can occur without generating vibration. Small unsteadinesses associated with work-producing forces and torques produce vibrations which are ultimately transmitted to radiating surfaces and are therefore sources of noise. We are concerned here with the production of these vibrations. Although most data available on machinery noise are for airborne noise levels, the same functional relations may be expected to apply for underwater sound.

10.1 Mechanical Unbalances

Motions in machines are either rotating or reciprocating. Each generates a different type of mechanical unbalance.

Rotational Unbalances

All rotational systems have slight amounts of static and dynamic mechanical unbalance due to imperfection of materials or construction, load and thermal distortions and/or bearing misalignments. As discussed by Klyukin (1961), *static unbalance* can be represented by a displacement of the center of gravity of a rotor from the center of rotation, and *dynamic unbalance* can be represented by two unbalanced masses lying in separate transverse planes. The resultant fluctuating force and moment are both proportional to the square of the *angular speed*, $\omega = 2\pi n$, where n is rotational speed in rps. Unbalance forces and moments are transmitted through bearings to the frame and foundation. Since vibratory velocities are proportional to the forces causing them and acoustic pressures are proportional to vibratory velocities of radiating surfaces, it follows that for a given machine the sound power radiated from mechanical unbalances increases as the fourth power of rotational speed. Thus,

$$W_{ac} \sim p^2 \sim (\Delta F)^2 \sim (m\omega^2)^2 \sim m^2 n^4 \quad . \qquad (10.1)$$

Since the useful mechanical power of a rotational machine increases as the cube of its rotational speed, it follows that for a given machine

$$\eta_{ac} \equiv \frac{W_{ac}}{W_{mech}} \sim \frac{n^4}{n^3} \sim n \quad . \qquad (10.2)$$

This result can also be expressed by

$$PWL = A + 13.3 \log hp \quad , \tag{10.3}$$

where the constant A depends on type of machine, amount of unbalance, foundation system, degree of isolation and nature of radiating surface, and hp is the horsepower which the machine is capable of delivering at the speed at which it is being operated. Although derived for unbalance forces of a single machine, Eq. 10.3 has been found to be generally applicable for centrifugal pumps and other types of rotating machinery. Individual machines vary considerably from one another due to differences in degree of balance.

Radiated spectra attributable to rotational unbalances are dominated by single tones at rotational frequencies, the bandwidths of which are determined by stability of the power source and of the rotational speed. Slight motions within the bearings distort the normal sine wave and cause strengthening of alternate cycles, thus producing weak second harmonics as well as tones at subharmonics. In addition, rotational unbalances may modulate other spectral components such as blade-rate tonals of fans and pumps and contact frequencies of gears. The result is radiation of high-order harmonics of shaft frequency centered around these other components.

Reciprocating Unbalances

Machines characterized by reciprocating motions of pistons in cylinders that transmit forces to crank shafts through connecting rods generally create large unbalance forces and moments. Entire texts such as that of Biezeno and Grammel (1954) are devoted to calculation of the forces and moments produced by these machines and to methods of reducing such unbalances. Reciprocating unbalance forces and moments occur at low-order harmonics of crank rotational speed. As noted by Lewis (1961), different physical arrangements and cylinder phase angles produce different strengths of the various harmonics, and some of the stronger components may be balanced by judicious placement of counterweights.

Figure 10.1 depicts a single cylinder of a reciprocating machine showing attachment of a piston to a crank shaft by means of a connecting rod. Variations of cylinder gas pressure, of

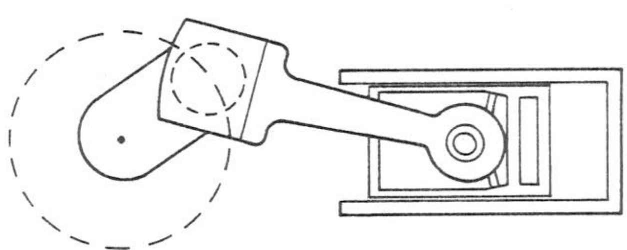

Fig. 10.1. Single Cylinder of Reciprocating Machine

inertial forces of the moving parts and of the crank angle cause cyclical fluctuations of torques and forces. These can be smoothed by using multiple cylinders with their connection points to the crank shaft equally spaced around a circle. However, their longitudinal spatial separation results in net fluctuating moments applied to the foundation.

A number of different arrangements have been developed for multiple cylinder units, including in-line, vee, opposed-piston and radial. Each of these has advantages and disadvantages; the radial arrangement is probably best from the point of view of static and dynamic balance. However, although dynamic unbalance is often an important source of annoying low-frequency vibrations, it is seldom a dominant noise source of reciprocating machinery.

10.2 Electromagnetic Force Fluctuations

Most modern electrical equipment and machinery used on ships are relatively quiet compared to other types of machinery. Electric motors are seldom as noisy as the pumps, compressors and blowers they drive, though they may occasionally be notable noise sources. In the 1930's, however, motor noise was often severe, and extensive research on this subject was carried out at that time. The present brief discussion is included because failure to recognize and understand electromagnetic force fluctuations could result in return to the earlier situation.

Magnetostriction

When most materials are magnetized, they change dimensions slightly due to re-alignment of elementary magnets. The iron cores of a.c. magnetic systems experience such dimensional changes during each half cycle of voltage. Consequently, their surfaces vibrate at twice power-line frequency. Due to non-linear and hysteresis effects vibration is not a pure sinusoid but contains higher-order harmonics. The resultant spectra consist of several harmonics of twice the power-line frequency; the fundamental is usually strongest, producing *transformer hum*. If a resonant frequency of the housing should coincide with one of the harmonics, then that frequency will also radiate strongly.

Articles on the causes and cures of transformer magnetostrictive vibrations and noise have been written by Churcher and King (1940), Swaffield (1942), King (1957, 1965), Thompson (1963) and Wilkins (1966). They agree that noise increases with flux density and weight of iron and that the best material is 6% silicon iron in hot-rolled laminations. Thompson has also found that deviations from uniform lamination thickness increase noise. Although magnetostrictive vibrations also occur in rotating a.c. machinery, Alger (1954), Fehr and Muster (1957) and Campbell (1963) have found that this effect is small compared to vibrations at the same frequencies caused by magnetic force variations.

Magnetic Force Variations

Two distinct types of magnetic force fluctuation occur in a.c. motors. Low-frequency noise similar to transformer hum occurs at twice the line frequency and is independent of rotational speed. High-frequency noise occurs at frequencies related to rotor speed and to the number of armature segments, or teeth.

Low-frequency vibrations and noise arise from fluctuations of the radial attractive force between stator and rotor. This force is proportional to the square of the instantaneous flux density and so goes through a complete zero-maximum-zero cycle twice during each voltage cycle. In this respect it is similar to magnetostrictive hum. Formulas for the noise produced at twice power-line frequency have been developed by Alger (1954), who found that noise decreases significantly with increasing number of poles. Since the speeds of a.c. machines are also reduced by using large numbers of poles, slow-speed motors are much quieter than those rotating at close to power-line frequency. Although fundamental low-frequency vibration of a.c. motors occurs at twice the line frequency, Robinson (1963) has shown that motion of the shaft in the bearings permits a side-to-side subharmonic vibration at half this frequency, i.e., at line frequency itself. Summers (1955) has found that because of slight asymmetries the hum component of two-pole a.c. induction motors is modulated at slip frequency, which is the frequency differential between line and rotational frequencies. For this reason, and because hum noise may be as much as 20 dB higher in these units, two-pole a.c. machines should be avoided in situations where motor noise might be significant.

Rotor slot noise occurs in d.c. as well as in a.c. machines and is due to the small flux variations that occur as the relative positions of the rotor teeth vary with respect to stator poles. The fundamental frequency of this type of noise is the number of rotor slots times the actual rotational frequency. In a.c. machines, two other strong components occur at the slot frequency plus and minus twice the line frequency. Thus,

$$f = Rn \overset{+}{\underset{0}{-}} 2\tilde{f} \tag{10.4}$$

where R is the number of armature teeth, or rotor slots, n is the rotational speed in rps, $\tilde{f}$ is line frequency and the symbol $\overset{+}{\underset{0}{-}}$ means plus, minus and zero. Muster and Wolfert (1956) have termed these high-order components *dissymmetry harmonics*, since their strengths are determined by stator as well as rotor dissymmetries. As noted by Lübcke (1956), structural resonances influence the relative strengths of the components. It seems clear that rotor slot noise can be minimized by the use of heavy, highly damped frames and by precision in manufacture. Skewing of slots to produce smoother flux transitions is also effective in reducing these components, especially in d.c. machines.

10.3 Impact Sounds

A number of machinery noise sources such as gears, engine valves and chain drives are characterized by repeated impacts of metal parts against other metal surfaces. Before considering specific sources, we will examine the general theory of impact sounds as developed and applied by Cremer (1950, 1953), Heckl and Rathe (1963) and Ungar and Ross (1965).

Impact Vibratory Relations

Impacts are characterized by short impulsive excitations of structures, the period between successive pulses being long compared to the duration of each pulse. As depicted in Fig. 10.2, each

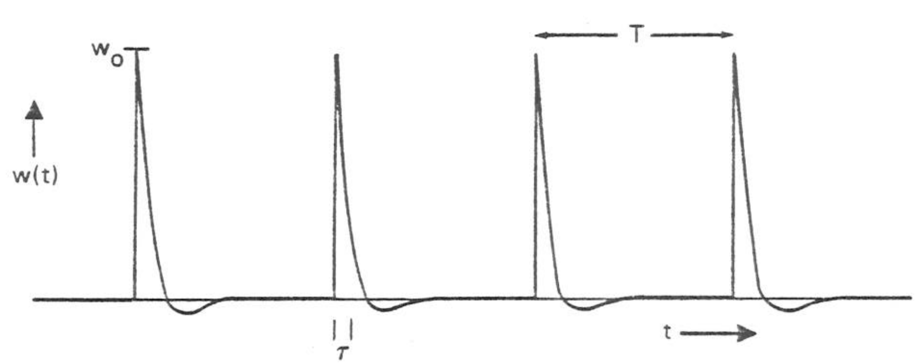

Fig. 10.2. Repeated Impacts

impact causes a rapid acceleration to an initial velocity, w_o, followed by a rapid decay prior to the next impact. The instantaneous velocity of each pulse may be represented by

$$w(t) = w_o \, e^{-t/\tau} \, . \tag{10.5}$$

Cremer assumed that the impacted structure takes on the velocity of the mass that strikes it, which represents an upper limit. The time constant, τ, in which the velocity decays to e^{-1} of its initial value is a function of the mass of the object relative to the impedance of the structure and also of the damping of the structure. In terms of τ, the mean-square velocity is given by

$$\overline{w^2} = \frac{1}{T} \int_0^T w_o^2 \, e^{-2t/\tau} \, dt \doteq \frac{w_o^2 \tau}{2T} \, , \tag{10.6}$$

provided $T >> \tau$. From Eq. 5.124, the power transferred from the impacting body to the structure is given by

$$W_{vibr} = R_i \overline{w^2} = \frac{w_o^2 \tau}{2T} R_i \, , \tag{10.7}$$

where R_i is the real part of the input impedance of the structure. Cremer equated this power to the kinetic energy carried by the impacting mass per unit time, namely,

$$W_{Kin} = \frac{E_{Kin}}{T} = \frac{m w_o^2}{2T} \, , \tag{10.8}$$

thereby finding

$$\tau \doteq \frac{m}{R_i} \, . \tag{10.9}$$

The spectrum of repeated pulses consists of a large number of tonals separated in frequency by the reciprocal of the repetition period, i.e.,

$$f_j = j \, \frac{1}{T} \, . \tag{10.10}$$

The amplitudes of these harmonics can be found from Fourier analysis, using Eq. 1.64, from which

$$w(\omega) \sim \frac{1}{T} \int_0^T w_o \, e^{-t/\tau} \, e^{-i\omega t} \, dt \doteq \frac{\tau w_o}{\sqrt{1 + (\omega\tau)^2}} \, . \tag{10.11}$$

The impact spectrum is therefore flat up to $\omega \doteq (3\tau)^{-1}$ and decreases at 6 dB/octave for $(\omega\tau > 3)$, as shown in Fig. 10.3. The actual vibration spectrum of a structure is this input spectrum multi-

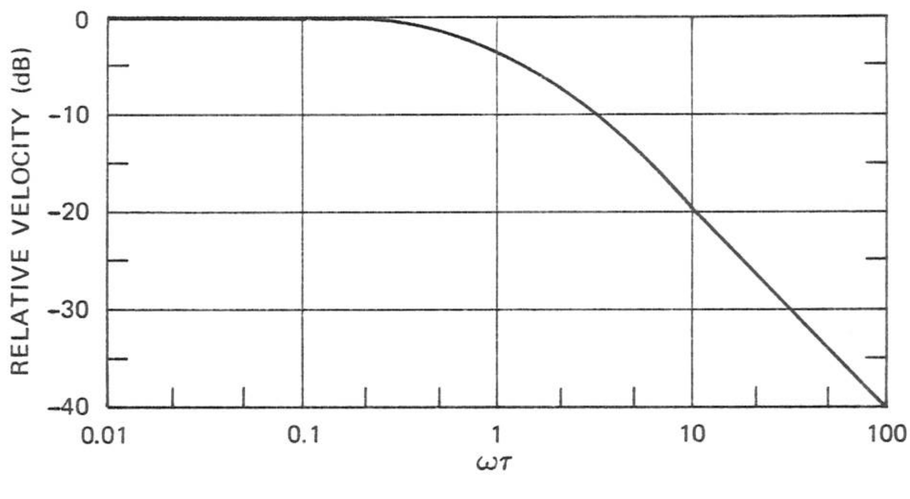

Fig. 10.3. Envelope of Spectrum for Repeated Impacts

plied by the structure's response spectrum. Therefore, highest vibratory and acoustic levels can be expected when harmonics of the impact frequency coincide with structural resonances. Because such coincidences are virtually unavoidable, use of structural damping to reduce resonant response is especially important in reducing impact noise.

Gear Noise

Gears are often important sources of machine noise. The two major causes of gear noise are tooth impacts and hobbing error. *Tooth impacts* produce tones at multiples of the tooth contact frequency. Usually the fundamental is strongest. Rosen (1961) measured noise spectra of two *planetary gear systems* composed of gears having straight teeth, which are called *spur gears*. He recorded as many as six harmonics, with the second harmonic strongest. Gear noise depends on the shape of the teeth as well as the accuracy with which they are machined. Except for resonance effects caused when tooth contact frequencies excite specific mechanical resonances of the web or casing, Rosen found that the noise is dependent only on the mechanical power being transmitted. From his tests, we can conclude that $\eta_{ac} \doteq 3 \times 10^{-6}$ for spur gears.

Helical gears are generally about 10 dB quieter than spur gears for the same power transmission. Attia (1969, 1970 and 1971) found that contact-frequency noise of helical gears decreases with increased pitch, but increases markedly if the number of teeth in contact at any one time is a whole number. He also found that *Novikov gears*, which use circular-arc tooth profiles, are 6 to 8 dB noisier than *involute helical gears*. Nakamura (1967) reported that gear noise increases with load, with peripheral speed for a given load and with use of thinner webs. He also found that under light loading conditions subharmonics of the contact frequency may be produced from abnormal meshing of every second or third tooth. Moeller (1957) and Attia (1969) have noted that proper lubrication is required to avoid excessive noise caused by friction between the teeth when under heavy load.

Hobbing error occurs because gears are cut on a machine that is itself driven by gears. If the gear drive of the cutting machine is not smooth, it will produce a wavy outer surface of the gear being cut, which waviness acts like a second set of gear teeth. Hobbing tones can be eliminated by honing or polishing the teeth after cutting, as discussed by Klyukin (1961). Klyukin also recommended use of heavy and/or damped rims and webs as well as operation at moderate load factors.

10.4 Piston-Slap Noise in Reciprocating Machinery

Piston Slap

Piston slap refers to impact of a piston against a cylinder wall as a result of sidewise motion of the piston across the cylinder clearance space due to reversal of the direction of the cross-force component of connecting-rod force. As shown in Fig. 10.1, the connecting rod of a typical reciprocating machine moves from side to side relative to the primary line of piston motion. The direction of the cross-force component changes when this occurs. The piston, which has been riding against one cylinder surface, moves to the other side where it strikes the cylinder wall and causes an impact-type vibration. Piston slap also occurs when the connecting-rod force changes sign. Figure 10.4 shows an oscillogram taken from an accelerometer mounted on an engine cylinder. Each individual impact excites high-frequency resonant vibrations which decay prior to the next impact.

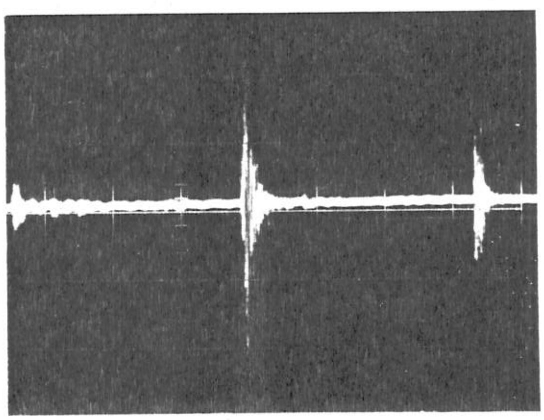

Fig. 10.4. Oscillographic Record of Piston Slap

Several piston slaps occur during each crank shaft revolution. Because they are not equally spaced, the fundamental repetition frequency is the crank rotational frequency. The resultant spectrum from each cylinder consists of numerous harmonically-spaced tonals whose envelope is described roughly according to Eq. 10.11 and Fig. 10.3. In a multiple-cylinder engine, the angular connections to the crank shaft are staggered to counter dynamic unbalances, as discussed in Section 10.1. As a result, harmonics that are multiples of the number of in-line cylinders are accentuated. These are called *firing-rate* tonals and occur in compressors as well as in diesels. The spectrum of a typical engine therefore consists of as many as 120 harmonics of the fundamental rotational frequency; those harmonics that are multiples of the firing rate and those that excite structural resonances are strongest.

Piston slap exists in most, though not all, reciprocating machines. It is sometimes eliminated by tight cylinder clearances and/or use of offset crankshafts and wristpins. Large, slow-speed (under 250 rpm) marine reciprocating and diesel propulsion engines usually incorporate articulated connecting rods, thereby eliminating piston cross forces and virtually eliminating piston slap. Rhombic drive mechanisms, such as that used in the Stirling engine, also eliminate cross forces.

Significance of Piston Slap

It is generally recognized today that piston slap is a major mechanical noise source of most reciprocating compressors and of medium- and high-speed marine diesel engines and, furthermore, that it is a major cause of pitting erosion of cylinder liners, as noted by Joyner (1957). However, the importance of piston slap as an underwater sound noise source has been recognized only within the past 20 years. Prior to about 1955, the general assumption was that the dominant sources of diesel engine structural vibrations were mechanical unbalances and pulses associated with high cylinder pressures during combustion. In the U.S. in the late 1940's, *pancake diesels* having radial cylinder arrangements were selected for submarine installations, rather than the more usual in-line or vee arrangements, because it was thought that elimination of low-order unbalances would reduce underwater diesel noise. Modification of firing cycles was also investigated in order to reduce pressure pulses associated with combustion. Neither of these projects was successful; in fact, the pancake engines were found to be somewhat noisier than the engines they replaced.

Evidence of the dominance of another source was revealed by extensive noise and vibration tests carried out under Mercy (1955) at the Brooklyn Naval Shipyard. Mercy compared the noise and vibration of an engine when motorized with that when fueled and found very little change of either spectrum shape or amplitude. He then gradually dismantled the engine, finding the biggest decrease in noise and vibration when the pistons were removed. Similar results have been reported by Hobson (1960) and Griffiths and Skorecki (1964).

Mercy's finding that a motorized diesel produces almost the same spectrum as a fueled engine and the knowledge that spectra of reciprocating compressors are similar to those of naval diesels led Ross and Ungar (1965) to the conclusion that piston-slap noise described and analyzed by Zinchenko (1957) is the dominant source of noise of medium- and high-speed marine engines. Other studies such as those of Griffiths and Skorecki (1964) and Haddad and Pullen (1974) confirmed this conclusion. Measurements of diesel noise as a function of load, speed, piston clearance and type of connecting rod all conform to expectations based on piston impact analysis.

Piston Impact Velocity

Detailed analyses of lateral piston motions in reciprocating machines have been made by Zinchenko (1957), Crane (1959), Griffiths and Skorecki (1964), Alvarez (1964) and Ungar and Ross (1965). These analyses, which differ only in small details, all consider inertia and gas pressure forces acting on a piston and calculate the cross component as a function of time. When the cross component passes through zero, the piston accelerates as it crosses the gap and arrives at the far wall with impact velocity, w_o.

Figure 10.5 depicts the geometry and forces involved in piston impact analyses. The various dimensions and angles shown in the figure are:

D	piston diameter,
L	connecting-rod length,
R	crank radius,

S piston stroke (= 2R),
x distance of lateral motion,
y piston position relative to center of stroke,
δ piston clearance,
β connecting-rod angle, and
θ crank angle.

The instantaneous forces experienced by the piston ~~when it is free to move~~ are:

f_{CR} connection-rod force,
f_i inertia force of the reciprocating mass in direction of piston motion,
f_p pressure force, and
f_x lateral force.

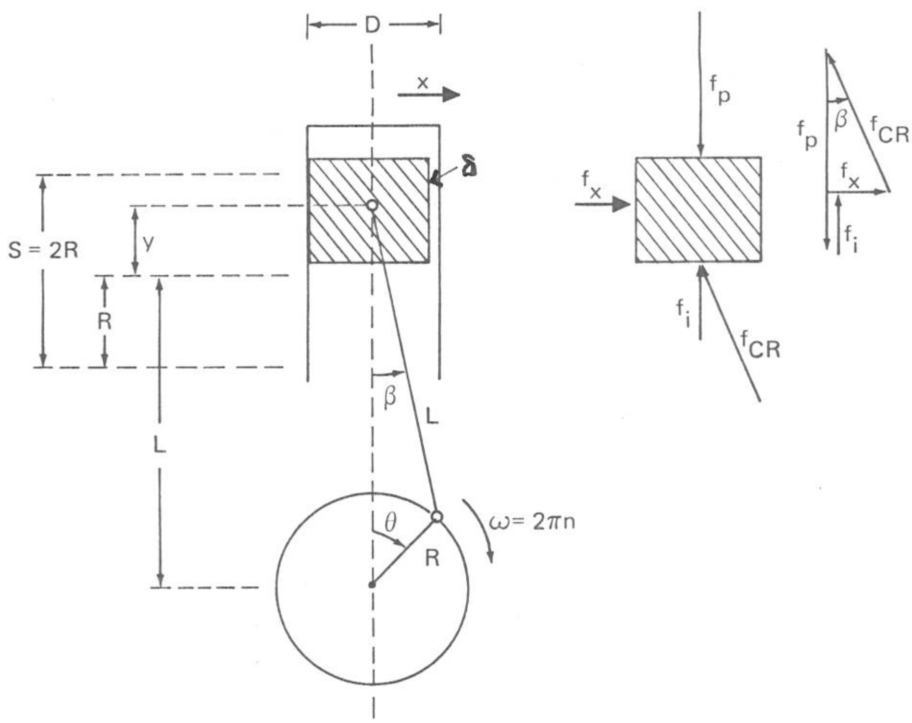

Fig. 10.5. Geometric and Force Diagrams for Piston Impact Analyses

When the piston rubs against a surface of the cylinder, the cylinder-wall normal force balances the lateral component of the connecting-rod force. But when the latter changes sign, either due to a change in the angle of the connecting rod or to a change in the sign of the connecting-rod force, the piston moves over to the other side of the cylinder. We are concerned here with the impulse of the resultant impact and the attendant noise and vibration.

From the force diagram of Fig. 10.5 we may write two scalar equations balancing the y and x force components:

$$f_p - f_i - f_{CR} \cos\beta = 0 \tag{10.12}$$

and

$$f_x = f_{CR} \sin\beta \ . \tag{10.13}$$

Solving for f_{CR} from Eq. 10.12, Eq. 10.13 becomes

$$f_x = (f_p - f_i) \tan\beta \ . \tag{10.14}$$

The angle β is related to the crank angle θ by

$$\sin\beta = \frac{R}{L} \sin\theta \ , \tag{10.15}$$

from which it follows that for small angles

$$\tan\beta = \frac{\sin\beta}{\sqrt{1 - \sin^2\beta}} \doteq \frac{R}{L} \sin\theta \left(1 + \frac{1}{2}\left(\frac{R}{L}\right)^2 \sin^2\theta\right) \ . \tag{10.16}$$

The piston position measured from mid stroke is given by

$$y = R\cos\theta - L(1 - \cos\beta) \ . \tag{10.17}$$

Assuming $R^2 << L^2$,

$$\frac{y}{R} \doteq \cos\theta - \frac{R}{4L}(1 - \cos 2\theta) \ . \tag{10.18}$$

For constant crank rotational speed, ω, the inertia force associated with piston reciprocating motion is given by

$$f_i = -m_p\ddot{y} \doteq m_p R\omega^2(\cos\theta + \frac{R}{L}\cos 2\theta) \ , \tag{10.19}$$

where m_p is the effective mass of the piston and includes that fraction of the mass of the connecting rod that can be considered to move as part of the piston.

The cylinder pressure follows a cycle similar to that shown in Fig. 10.6. It is generally difficult to express the pressure by a meaningful analytical function; however, at times of piston lateral motion the piston pressure is either close to its maximum value or close to zero. We may therefore distinguish two kinds of impact: those in which cylinder pressure is important and $p \doteq p_m$, i.e., its value close to the peak of the pressure cycle, and those in which the cylinder pressure is virtually zero and inertia forces dominate.

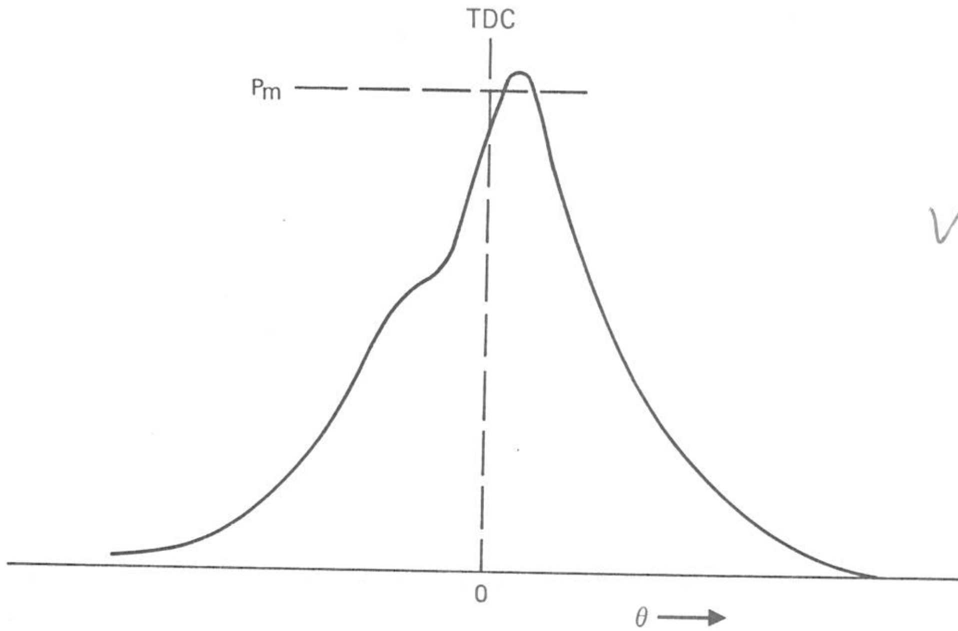

Fig. 10.6. Cylinder Pressure as Function of Crank Angle

For *pressure-controlled impacts*, which occur near the top-dead-center (TDC) position, the pressure force is

$$f_p \doteq \frac{\pi}{4} D^2 p_m = A_p p_m \tag{10.20}$$

and the inertia force can be approximated by

$$f_i \doteq m_p R \omega^2 \left(1 + \frac{R}{L}\right) \left(1 - \frac{L + 4R}{2L + 2R} \theta^2\right) . \tag{10.21}$$

Since θ is small, the inertia force is virtually constant during a pressure-controlled impact. Combining Eqs. 10.14, 10.16, 10.20 and 10.21 and assuming $\theta^2 << 1$, the cross force during a pressure-controlled impact near TDC is given by

$$f_x = \left(f_p - f_i\right) \tan \beta \doteq \left[A_p p_m - m_p R \omega^2 \left(1 + \frac{R}{L}\right)\right] \frac{R}{L} \theta . \tag{10.22}$$

Neglect of θ^2 terms can be shown to introduce no more than a 5% error in the calculations for the range of parameters found in practice.

In those parts of the cycle for which the pressure forces are negligible, such as at bottom dead center (BDC) and during the scavenging stroke of a four-stroke engine, the corss force is controlled by the inertia force. Combining Eqs. 10.14, 10.16 and 10.19, the cross force during *inertia-controlled impacts* is

$$f_x = - f_i \tan\beta = - m_p R\omega^2 \frac{R}{L} \sin\theta \left(1 + \frac{1}{2}\left(\frac{R}{L}\right)^2 \sin^2\theta\right) \times$$
$$\left(\cos\theta + \frac{R}{L}\cos 2\theta\right) . \qquad (10.23)$$

Two types of sign reversal can occur in this equation. One type occurs at the top and bottom of the stroke when $\sin\theta$ changes sign by going through zero; the other occurs near 90° when the cosine term changes its sign. Because the two resultant motions cause similar impacts, we may estimate the force by calculating it at the point of reversal of the sign of $\sin\theta$. For this type, and neglecting terms involving the square of the angle relative to unity, the normal force is approximately

$$f_x \doteq \mp\, m_p R\omega^2 \frac{R}{L}\left(1 \pm \frac{R}{L}\right)\theta , \qquad (10.24)$$

where the top sign refers to impacts at the top of the stroke and the lower sign to those at the bottom. Neglect of higher powers of θ causes somewhat more error for inertia-dominated impacts than for impacts controlled by pressure. However, the author has carried out calculations retaining these extra terms and has shown them not to be significant.

The impact speed of the piston is calculated by integrating Newton's second law for motion across the gap. For inertia-controlled impacts at TDC or BDC,

$$m_p\ddot{x} \doteq \mp\, m_p R\omega^2 \frac{R}{L}\left(1 \pm \frac{R}{L}\right)\theta . \qquad (10.25)$$

It follows that

$$\dot{x} \doteq \mp\, R\omega \frac{R}{L}\left(1 \pm \frac{R}{L}\right)\frac{\theta^2}{2} \qquad (10.26)$$

and

$$x \doteq \mp \frac{R^2}{L}\left(1 \pm \frac{R}{L}\right)\frac{\theta^3}{6} . \qquad (10.27)$$

At the far wall $x = \delta$, the cylinder clearance, and the resultant angle, θ_i, is

$$\theta_i \doteq \sqrt[3]{\frac{6\delta L}{R^2(1 \pm R/L)}} . \qquad (10.28)$$

The velocity at impact is thus

$$w_{o_i} \doteq R\omega \sqrt[3]{\frac{9}{2}\left(\frac{\delta}{R}\right)^2 \frac{R}{L}\left(1 \pm \frac{R}{L}\right)} . \qquad (10.29)$$

By a similar analysis, the velocity of the piston for a *pressure-controlled impact* is

$$w_{o_p} \doteq R\omega \sqrt[3]{\frac{9}{2}\left(\frac{\delta}{R}\right)^2 \frac{R}{L}\left(\frac{1-\alpha}{\alpha}\right)} \quad , \tag{10.30}$$

where

$$\alpha \equiv \frac{m_p R\omega^2}{A_p p_m} \tag{10.31}$$

is the ratio of inertia to pressure forces during a pressure-controlled impact. As noted by Ungar and Ross (1965), this quantity is also the ratio to the peak piston pressure of the centrifugal force per unit area that the piston would exert if it were attached at the crank radius. This ratio is the critical parameter controlling the nature of the dominant impacts.

The parameter α can be shown to be a function of piston material, *linear piston speed*, $s \equiv 2nS$, and peak piston pressure, p_m. Thus, expressing the piston mass, m_p, by

$$m_p = \rho_p A_p H \quad , \tag{10.32}$$

it follows that

$$\alpha = \frac{\pi^2}{2}\frac{\rho_p}{p_m}\frac{H}{S}s^2 \quad , \tag{10.33}$$

where H is the effective piston depth including a contribution from the connecting rod. Assuming H to be about one third of the stroke,

$$\alpha \doteq \frac{5}{3}\frac{\rho_p s^2}{p_m} \quad . \tag{10.34}$$

Values of α for diesels generally lie between about 0.06 and 0.25.

The average ratio of the velocities for the two types of impact is

$$\gamma \equiv \frac{w_{o_p}}{w_{o_i}} \doteq \sqrt[3]{\frac{1-\alpha}{\alpha}} \quad , \tag{10.35}$$

which is also the ratio of crank angle travel during an impact controlled by inertia to crank angle travel during an impact controlled by pressure.

Cylinder Wall Vibrations

From our previous discussion of impact noise it follows that the energy transferred to the cylinder wall by each piston impact can be estimated from Eq. 10.8 using Eqs. 10.29 and 10.30 for impact velocities. The total power transferred is the product of the number of cylinders, j, the rotational frequency, n, and the vibrational energy associated with each cylinder per revolution. Distinction must be made between 2-stroke-cycle engines that fire every revolution and 4-stroke-cycle engines that intersperse scavenging strokes between power strokes and fire only every second revolution. From Eqs. 10.8, 10.29, 10.30 and 10.35,

$$W_{vibr} \doteq j \frac{2n}{spc} \frac{m_p R^2 \omega^2}{2} \left(\frac{9}{2}\right)^{2/3} \left(\frac{\delta}{R}\right)^{4/3} \left(\frac{R}{L}\right)^{2/3} \left(\gamma^2 + N_i\right) , \qquad (10.36)$$

where *spc* stands for *number of strokes per cycle* and N_i is number of inertia-controlled impacts per complete cycle. The number of inertia-controlled impacts depends on the balance between inertia and gas forces. For 2-stroke-cycle engines there is always one impact at BDC where the sign of β changes and sometimes two additional impacts at other angles. Four-stroke-cycle diesels have two impacts at BDC, one at TDC and either two or four additional impacts due to force reversals, as shown in Fig. 10.7. Thus, on average, $N_i \doteq 1.4\, spc$ and the vibratory power is approximately

$$W_{vibr} \doteq 1.7 j \alpha s p_m A_p \left(\frac{\delta}{S}\right)^{4/3} \left(\frac{R}{L}\right)^{2/3} \left(\frac{\gamma^2}{spc} + 1.4\right) . \qquad (10.37)$$

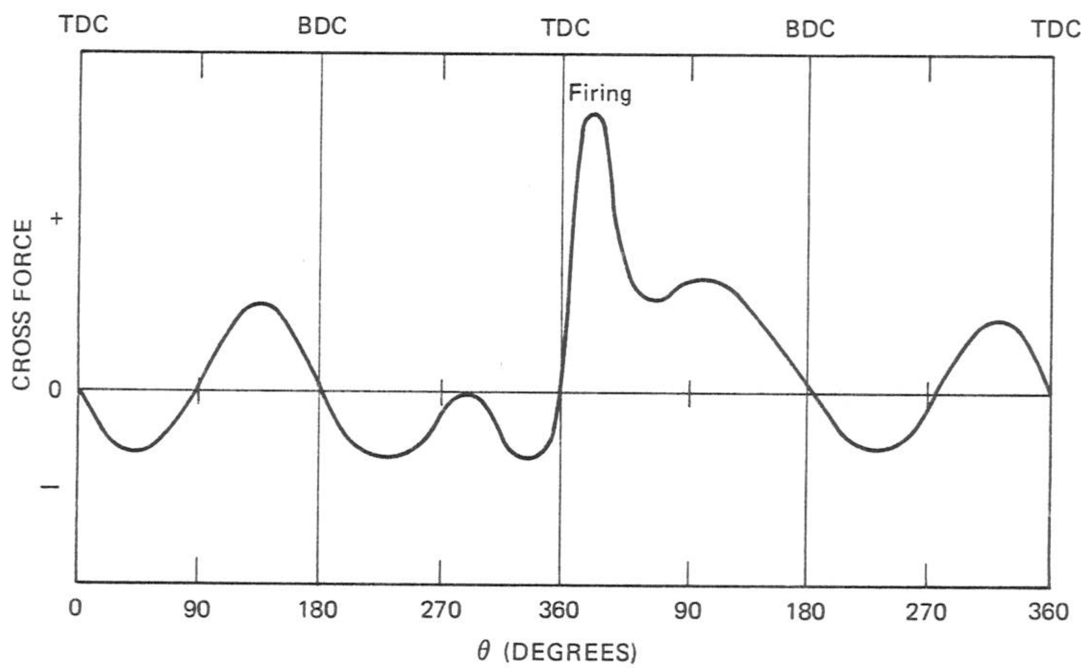

Fig. 10.7. Piston Cross Force during Full Cycle of a Four-Stroke Engine, after Zinchenko (1957)

Vibratory power due to inertia-controlled impacts for a given engine increases as the cube of piston speed; vibratory power due to pressure-controlled impacts increases approximately as the 5/3 power. It is now clear why piston-slap noise of a motorized diesel is almost as great as that of a fueled engine. When an engine is motorized, α is large and γ^2 small; this elimination of γ^2 from Eq. 10.37 generally decreases the vibratory power by less than a factor of two, i.e., by less than 3 dB.

The efficiency of conversion of mechanical power into vibratory power can be estimated by dividing Eq. 10.37 by an expression for the mechanical power of a fueled diesel. This power is given by

$$W_{mech} = \frac{2jn}{spc} A_p S\bar{p} = \frac{js}{spc} A_p \bar{p} \ , \qquad (10.38)$$

where $\bar{p}$ is the *mean effective pressure*. Ungar and Ross (1965) have noted that under normal operating conditions $\bar{p}$ is usually close to one third of the peak cylinder pressure, p_m. It follows that the vibration conversion efficiency is approximately

$$\eta_{vibr} \doteq 5\alpha \left(\frac{\delta}{S}\right)^{4/3} \left(\frac{R}{L}\right)^{2/3} \left(\gamma^2 + N_i\right) \ . \qquad (10.39)$$

Ungar and Ross found from examination of data on a number of marine engines that δ/S is of the order of 3×10^{-3} and R/L is usually about 0.25, from which $\eta_{vibr} \approx 10^{-3}$ for typical values of the parameters.

Ungar and Ross used the methodology of Section 6.4 and estimated that the radiation efficiency of airborne sound from typical engines is of the order to 10^{-4}. Multiplying by the vibration conversion efficiency, they concluded that a typical 1000 hp engine would produce an airborne acoustic power level, *PWL*, of about 110 dB re 10^{-12} W. Variations of piston clearance, strokes per cycle, piston speed and engine material can cause deviations of at least ±10 dB about this mean value.

Experimental Verification

Oscillograms of the type shown in Fig. 10.4 and observation of multi-harmonic spectra are two indications that an impact phenomenon produces the dominant vibrations in most reciprocating machines. Evidence in support of piston slap as that phenomenon is found in experiments in which a single engine parameter is varied. Equation 10.37 for vibratory power is the basis for predictions of these variations. Substituting for α in Eq. 10.37 from Eq. 10.34, one obtains

$$W_{vibr} \doteq 18 j \rho_p n^3 S^2 D^3 \left(\frac{\delta}{D}\right)^{4/3} \left(\frac{R}{L}\right)^{2/3} \left(\frac{\gamma^2}{spc} + 1.4\right) \ . \qquad (10.40)$$

This expression predicts noise dependence on number of cylinders, rotational speed, stroke, diameter, piston clearance, load, strokes per cycle and ratio of crank radius to length of connecting rod.

Zinchenko (1956, 1957) measured the airborne engine noise of over 60 Soviet diesels including some that differed only in number of cylinders. He verified the proportionality of noise to j predicted by Eq. 10.40. The dependency of cylinder-wall vibration on piston clearance expected from Eq. 10.40 was confirmed by measurements made by Alvarez (1964). However, as shown in Fig. 10.8, Skobtsov et al (1962) found that airborne noise levels of diesels increase with piston clearance at about half the predicted rate.

Nowhere in the analysis is dependence of noise on load specifically included. Any changes due to load must therefore be secondary. Load can affect γ slightly and differences of cylinder heating

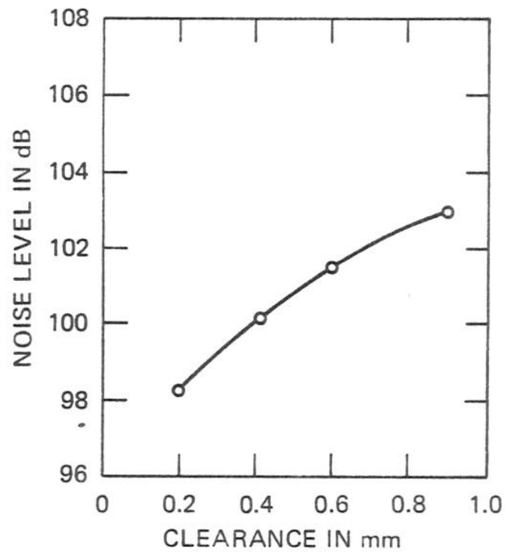

Fig. 10.8. Airborne Noise of a Diesel Engine as Function of Piston Clearance, after Skobtsov et al (1962)

due to load may affect piston clearances. As expected, data from six diesels reproduced in Fig. 10.9 show no consistent trend of noise with load.

It is well known that diesels are noisier when they first start up than after 15 or 20 minutes' operation. This can be explained by initial lack of an oil film and by the larger clearances

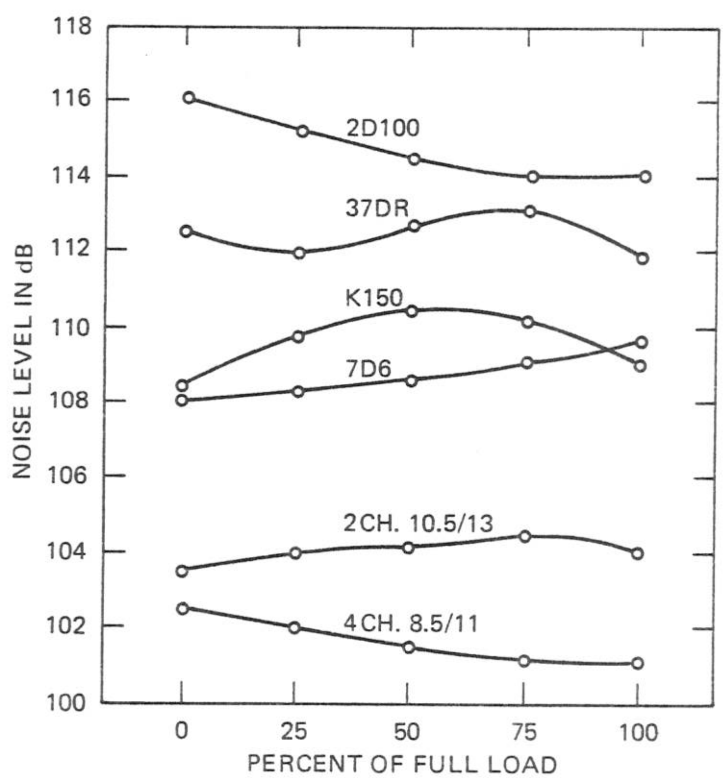

Fig. 10.9. Dependence of Noise on Load of Several Russian Diesels, after Zinchenko (1957)

characterizing a cold engine compared to a warm one. Both these factors cause more severe piston slap in a cold engine.

Equation 10.40 predicts only a slight dependence of piston-slap vibratory power on number of strokes per cycle. However, Eq. 10.38 indicates that mechanical power is reduced when *spc* is increased from two to four. Statistical analysis of Zinchenko's noise data shows that for the same horsepower rating four-stroke engines are indeed 2.5 to 3 dB noisier than two-stroke engines.

At very low speeds, for which inertia impacts produce less vibration than pressure impacts, Eq. 10.40 predicts that vibrations and noise will increase approximately as the square of the rotational speed (6 dB/ds). On the other hand, in the limit at high speeds when inertia impacts dominate, the dependence should be as the cube of speed (9 dB/ds). Most engines operate between these limits, though closer to the upper one. We would therefore expect that, on average, noise of a given engine would increase at a rate of 8 dB per double speed. Figure 10.10 shows data on seven Russian diesels confirming this expectation based on piston-slap analysis. Similar results were reported by Brammer and Muster (1975).

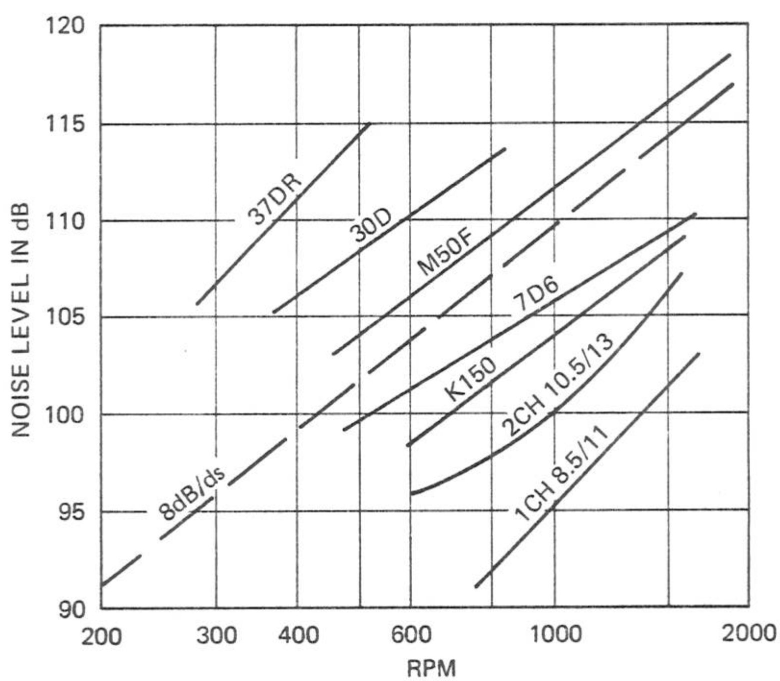

Fig. 10.10. Diesel Noise as a Function of Speed, as reported by Zinchenko (1957)

The ratio of crank radius to connecting-rod length determines the maximum angle of connecting-rod relative to direction of piston motion and therefore the fraction of connecting-rod force that contributes to cross force. Equation 10.40 indicates that this angle should play a role in piston-slap noise. Although this has not been studied directly, it is interesting to note that the radial *pancake* engines mentioned previously have exceptionally short connecting rods and experience severe cylinder liner pitting from piston impacts. They also produce exceptionally high noise levels.

Equation 10.40 predicts that vibratory power is proportional to density of piston material.

However, radiation analyses indicate that, other things being equal, radiation efficiency decreases with increasing material density. Ross and Ungar (1965) concluded that aluminum block engines should be somewhat noisier than those made of steel. In two cases in which aluminum and steel block models of the same engine were tested, Zinchenko (1956) found the aluminum engines to be 3 to 4 dB noisier.

In another confirmation of the dominance of piston impacts, Zinchenko measured the noise from two engines differing from the others only in having articulated connecting rods. He found engine noise in these cases to be 7 to 10 dB lower than for similar engines with single connecting rods.

Empirical Noise Formulas

A number of empirical formulas for airborne noise levels of diesel engines have been developed from the noise data published by Zinchenko (1956, 1957). Since Zinchenko's sound pressure measurements were made relatively close to the engines, i.e., at 0.5 m, they do not actually reveal the fifth power dependence on linear dimensions predicted by the analysis, but rather seem to fit third to fourth power relations. Ross and Ungar (1965) found

$$SPL \doteq 94 + 10 \log j \left(\frac{\rho_{steel}}{\rho_{block}} \right) n^3 D^2 S^2 \quad . \tag{10.41}$$

An equally good fit to the data is given by

$$SPL = 91 + 10 \log j + 28 \log nD \quad , \tag{10.42}$$

which is shown in Fig. 10.11. Zinchenko's data can also be correlated with engine horsepower and rpm, as plotted in Fig. 10.12. A formula which represents these data is

$$SPL \doteq 86 + 10 \log \left(hp \, \frac{spc}{2} \right) + 18 \log \frac{N}{1000} \quad , \tag{10.43}$$

where N is rotational speed in rpm. This formula indicates clearly that for the same rated power high-speed engines are generally noisier than slow-speed engines.

Underwater Noise Implications

Although underwater noise measurements of marine diesels have not been published, Zinchenko's airborne noise measurements and the foregoing analyses of the basic mechanisms involved make certain conclusions clear. First, piston slap is a dominant source of underwater noise from marine reciprocating machines. Second, the selection of machines with low rpm's and low linear piston speeds will result in relatively low noise levels, as indicated by Eqs. 10.40 and 10.43 and Fig. 10.12. Third, if piston clearances are kept as small as is consistent with reasonable frictional wear, lower noise levels will be achieved. Fourth, tall machines with relatively long connecting rods are quieter than more compact ones incorporating short rods. Finally, slow-speed marine diesels and old-fashioned steam reciprocating engines are especially quiet since they not only operate at low speeds but also incorporate articulated connecting rods. In this connection, we note a recent trend in merchant ships in which medium-speed geared diesels are being used in some installations in place of slow-speed direct-drive engines. These medium-speed engines may be

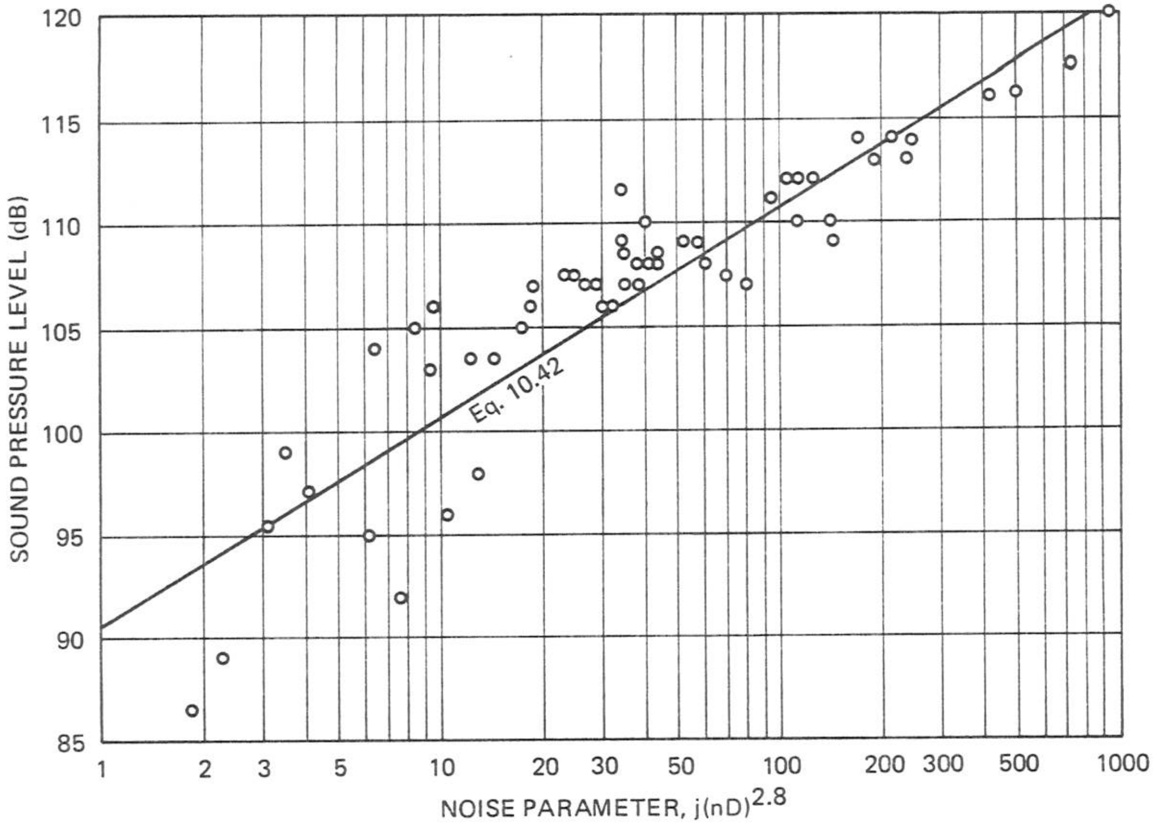

Fig. 10.11. Airborne Noise of Diesel Engines, as measured by Zinchenko (1956, 1957)

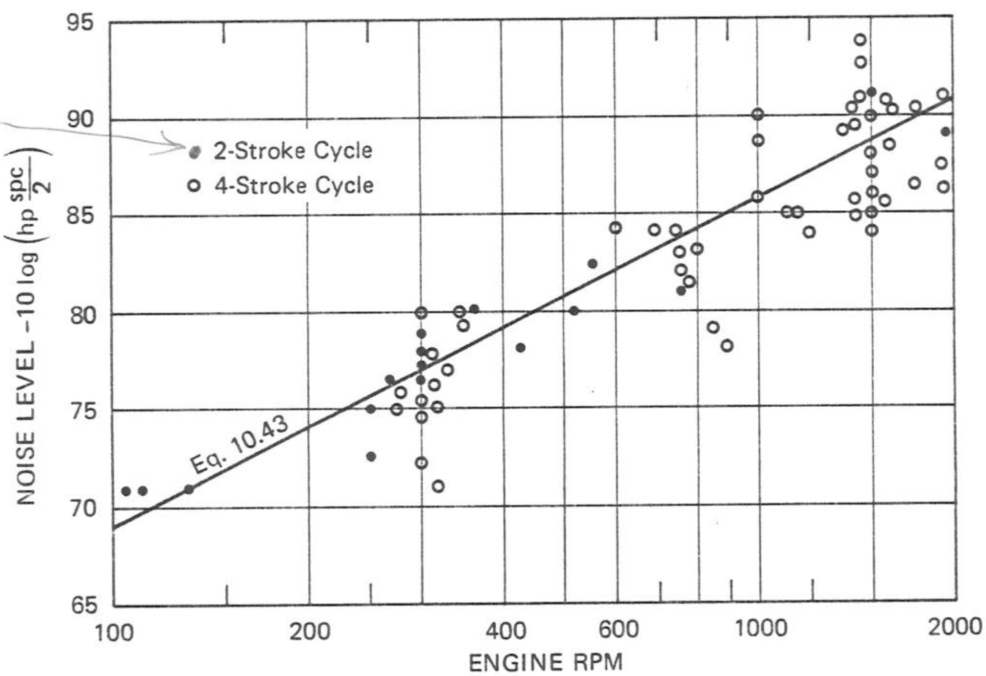

Fig. 10.12. Diesel Noise as Function of Engine Power and Speed, as measured by Zinchenko (1956, 1957)

expected to product as much as 20 dB more piston-slap noise since the speed factor implies about 10 to 12 dB higher levels and the replacement of the articulated connecting rod could add another 8 to 10 dB.

Equations 10.39 and 10.43 imply that for equal power two-stroke engines are quieter than four-stroke engines. However, this is true for the piston-slap component only. As discussed in Section 9.4, the fact that two-cycle engines use positive displacement scavenging blowers, which are quite noisy, may offset the 2 to 3 dB piston-slap advantage. In this connection, Hempel (1966, 1968) reported that turbocharging an engine has little effect on piston-slap noise, but that the turbocharger itself adds strong high-frequency components to the overall noise.

There are other noise sources that sometimes contribute significantly to the spectra of reciprocating machinery. Thus, combustion applies a sudden pulse to the cylinder structure similar to an impact. This source, which is dominant in many automotive engines, has been studied extensively by Goswami and Skorecki (1963), Austen and Priede (1965), Priede and Grover (1967) and Brammer and Muster (1975). These investigations have all reported that the rate of rise of cylinder pressure is a factor controlling combustion noise spectra above 500 Hz. Priede (1967) has also measured noise from fuel-injection equipment, finding that rapid changes of fluid pressure in fuel-injection pump elements excite pump camshaft vibrations in the 500 to 800 Hz range and that the injectors themselves produce noise above 2 kHz. Priede concluded that injector noise would only be noticeable at low engine speeds or on small engines. Finally, impact sounds associated with diesel valve mechanisms have been shown by Fielding and Skorecki (1966) to be occasional noise sources.

10.5 Bearing Noise

Bearings in rotating machinery both transmit and generate sonic vibrations. Of the two common types, *sliding bearings* more commonly transmit noise than generate it. Occasionally, when they become damaged or are poorly lubricated, sliding bearings generate high-pitched resonant vibrations that are both loud and annoying. More usually, noise generated by the oil pumps required to lubricate sliding bearings is much greater than that attributable to the bearings themselves.

Ball bearings, on the other hand, are often sources of tones that are related to both rotational speeds and resonances of the outer ring. Igarashi (1960, 1962 and 1964) found that the frequencies of outer ring resonances can be estimated from

$$f_m \doteq \frac{1}{2\pi} \frac{m(m^2 - 1)\kappa c_\ell}{\sqrt{1 + m^2 R^2}}, \qquad (10.44)$$

where κ is the radius of gyration, R the radius of the ring and m is an integer. As speed changes, the distribution of intensity among these resonances changes, but the frequencies remain constant. Ring resonances are excited by impacts of the balls. Igarashi and Ruffini (1963) reported that such impacts are reduced significantly by axial preloading, and this has become common practice in situations where ball-bearing noise is deemed to be important.

Ball bearings sometimes produce subharmonic vibrations. As explained by Tamura and Taniguchi (1961), when the number of balls is small, a bearing may act like a spring whose spring constant varies cyclically with motion of the balls. The result is excitation of the retaining ring at a frequency equal to half of the product of the retainer speed times the number of balls.

Twenty years ago, bearings were considered to be important noise sources of electric motors. Today, with the development of precision manufacturing methods and axial preloading, bearings are only noisy when they are close to being worn out or when they are improperly installed.

REFERENCES

Alger, P.L., The magnetic noise of polyphase induction motors, *Trans. Amer. Inst. Elect. Engin., 73* (III-A), 118-125, 1954.

Alger, P.L. and Erdelyi, E., Calculation of the magnetic noise of polyphase induction motors, *J.A.S.A., 28*, 1063-1067, 1956.

Alvarez, F.F., Noise and Vibration Problems of Reciprocating Machinery, *M.I.T. Dept. of Naval Arch. and Mar. Engin.*, M.S. Thesis, May 1964; results exerpted in *Bolt Beranek and Newman Rept. 1139*, July 1964.

Attia, A.Y., Noise of involute helical gears, *J. Engin. for Industry, 91B*, 165-171, 1969.

Attia, A.Y., Noise of gears of circular-arc tooth profile, *J. Sound and Vibr., 11*, 383-397, 1970.

Attia, A.Y., Effect of change of pitch on gear noise, *J. Sound and Vibr., 18*, 129-137, 1971.

Austen, A.E.W. and Priede, T., Noise of Automotive Diesel Engines: its Causes and Reduction, *Paper 1000A, Soc. of Automotive Engin.*, Jan. 1965.

Biezeno, C.B. and Grammel, R., *Engineering Dynamics: Vol. 4, Internal-Combustion Engines*, Blackie and Son, London, 1954.

Brammer, A.J. and Muster, D.F., Noise radiated by internal-combustion engines, *J.A.S.A., 58*, 11-21, 1975.

Campbell, J., Electric-motor noise, *Machine Design, 35*, 19, 139-158, Aug. 1963.

Churcher, B.G. and King, A.J., The limitation of transformer noise, *J. Inst. Elect. Engin., 87*, 539-569, 1940.

Crane, P.H.G., The Initiation of Diesel Engine Cylinder Liner Vibrations by Piston Transverse Motion, *G.B. Admiralty Res. Lab. Rept. R2/B10*, Aug. 1959.

Cremer, L., The Propagation of Structure-Borne Sound, *Dept. of Scientific and Indust. Res. (London) Sponsored Res. Rept. Ser. B.*, No. 1, 1950.

Cremer, L., Calculation of sound propagation in structures, *Acustica, 3*, 317-335, 1953.

Cremer, L., Heckl, M. and Ungar, E.E., *Structure-Borne Sound*, Springer-Verlag, New York, 1973 (Chapter 4).

Fearon, W., Waterside attack of diesel-engine cylinder liners, *J. Royal Naval Scientific Service, 19*, 22, 1964.

Fehr, R.O. and Muster, D.F., "Electric Motor and Generator Noise," Chapter 30 of *Handbook of Noise Control*, C.M. Harris (Ed.), McGraw-Hill, New York, 1957.

Fielding, B.J. and Skorecki, J., Identification of mechanical sources of noise in a diesel engine: sound emitted from the valve mechanism, *Proc. Inst. Mech. Engin., 181*, 437-451, 1966.

Goswami, D. and Skorecki, J., Some Vibration Experiments on a Single-Cylinder Diesel Engine, *A.S.M.E. Paper 63-WA-144*, Nov. 1963.

Griffiths, W.J. and Skorecki, J., Some aspects of vibration of a single cylinder diesel engine, *J. Sound and Vibr., 1*, 345-364, 1964.

Haddad, S.D. and Pullen, H.L., Piston slap as a source of noise and vibration in diesel engines, *J. Sound and Vibr., 34*, 249-260, 1964.

Heckl, M. and Rathe, E.J., Relationship between the transmission loss and the impact-noise isolation of floor structures, *J.A.S.A., 35*, 1825-1830, 1963.

Hempel, W., A contribution to knowledge of piston side movement, *Motortechnische Zeits., 27*, 1-6, 1966 (in German).

Hempel, W., Does Turbocharging Increase Diesel Engine Noise? - Observations on the Generation, Emission and Reduction of Diesel Engine Noise, *Paper 680406, Soc. of Automotive Engin.*, May 1968.

Hobson, A., Interim Report of Various Measurements of the Vibrations of Diesel Engines, *G.B. Admiralty Res. Lab. Rept. R4/93.20/D*, May 1960.

Igarashi, T., Noise of ball bearings, *Bul. Japan Soc. Mech. Engin., 3*(10), 220-227, 1960; *5*(17), 184-194, 1962; and *7*(25), 200-208, 1964.

Joyner, J.A., Reduction of pitting of diesel engine cylinder liners. *Trans. S.A.E., 65*, 337-348, 1957.

King, A.J., "Reduction of the Noise of Iron-Core Transformers and Chokes," Chapter 29 of *Handbook of Noise Control*, C.M. Harris (Ed.), McGraw-Hill, New York, 1957.

King, A.J., *The Measurement and Suppression of Noise*, Chapman and Hall, London, 1965 (Chapter 8).

Klyukin, I.I., *Control of Noise and Sonic Vibration in Ships*, Sudpromgiz, Leningrad, 1961; translation, U.S. Dept. of Commerce, Washington, 1963 (JPRS: 18,177,OTS63-21340) (Chapters 16-19).

Kohler, H.K., Pratt, A. and Thompson, A.M., "Dynamics and Noise of Parallel-Axis Gearing," Paper 14 of *Gearing in 1970, Proc. Inst. Mech. Engin., 184, 30*, 111-121, 1970.

Lewis, F.M., "Torsional Vibration in Reciprocating Machines." Chapter 38 of *Shock and Vibration Handbook*, C.M. Harris and C.E. Crede (Ed.), McGraw-Hill, New York, 1961.

Lübcke, E., Vibration problems in electrical machinery, *Acustica, 6*, 109-114, 1956 (in German).

Mercy, K.R., Analysis of the basic noise sources in the diesel engine, *A.S.M.E. Paper 55-OGP-4*, March 1955; also, *N.Y. Naval Shipyard, Materials Lab. Rept. NS 713-212*, April 1961.

Moeller, K.G.F., "Gear Noise" and "Bearing Noise," Chapters 23 and 24 of *Handbook of Noise Control*, C.M. Harris (Ed.), McGraw-Hill, New York, 1957.

Muster, D.F., "Balancing of Rotating Machinery," Chapter 39 of *Shock and Vibration Handbook*, C.M. Harris and C.E. Crede (Ed.), McGraw-Hill, New York, 1961.

Nakamura, K., Experimental studies about the effects of dynamic loads upon gear noise, *Bul. Japan Soc. Mech. Engin., 10*(37), 180-188, 1967; and *10*(41), 846-854, 1967.

Priede, T. and Grover, E.C., "Noise of Industrial Diesel Engines." Paper 2 of *Noise from Power Plant Equipment, Proc. Inst. Mech. Engin., 181, 3C*, 73-89, 1966.

Priede, T., Noise of diesel engine fuel injection equipment, *J. Sound and Vibr., 6*, 443-459, 1967.

Robinson, R.C., Line-frequency magnetic vibration of a.c. machines, *Trans. Amer. Inst. Elect. Engin., 81*(3), 675-679, 1963.

Rosen, M.W., The noises of two spur-gear transmissions, *Noise Control, 7*, 6, 11-19, 1961; also, *U.S. Naval Ord. Test Station Repts. NAVORD 6569 and 6594*, 1959.

Ross, D. and Ungar, E.E., On Piston Slap as a Source of Engine Noise, *A.S.M.E. Paper 65-OGP-10*, April 1965; also, *Bolt Beranek and Newman, Inc. Rept. 1139*, July 1964.

Ruffini, A.J., Bearing noise, *Machine Design, 35*, 11, 232-235; and *35*, 12, 158-166, 1963.

Skobtsov, Ye. A., Izotov, A.D. and Tuzov, L.V., *Methods of Reducing the Vibration and Noise of Diesel Engines*, Moscow, 1962 (in Russian).

Summers, E.W., Vibration in 2-pole induction motors related to slip frequency, *Trans. Amer. Inst. Elect. Engin., 74*, 69-72, 1955.

Swaffield, J., The causes and characteristics of transformer noise, *J. Inst. Elect. Engin., 89*, 212-224, 1942.

Taggart, R., Noise in reduction gears, *J. Amer. Soc. Naval Engin., 66*, 829-849, 1954; also, *BuShips Rept. 371-N-4*, May 1951.

Tamura, A. and Taniguchi, O., On the subharmonic vibration of order one-half caused by passing balls in a ball bearing, *Bul. Japan Soc. Mech. Engin., 4*(14), 193-200, 1961, and *4*(15), 482-488, 1961.

Thompson, J.E., Magnetostriction and transformer noise, *J. Inst. Elect. Engin., 9*, 72-74, 1963.

Ungar, E.E. and Ross, D., Vibrations and noise due to piston-slap in reciprocating machinery, *J. Sound and Vibr., 2*, 132-146, 1965; also, *Bolt Beranek and Newman, Inc. Rept. 1106*, April 1964.

Wilkins, J.T., "Noise in Transformers," Paper 3 of *Noise from Power Plant Equipment, Proc. Inst. Mech. Engin., 181, 3C*, 131-137, 1966.

Zinchenko, V.I., Effect of constructional factors on the noise of ships' engines, *Vestnik Mashinostroyeniya, 4*, 13-17, 1956 (translated).

Zinchenko, V.I., *Noise of Marine Diesel Engines*, Sudpromgiz, Leningrad, 1957; summarized in *Bolt Beranek and Newman, Inc. T.I.R. 61*, July 1962.

PAGE 348 IS A BLANK PAGE.