SECTION 8.4
PUMP NOISE

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The major concern regarding pump noise falls into two categories:

1. Noise levels that do not meet applicable environmental criteria. Examples range from personnel noise exposure criteria to overside noise criteria for submarines.
2. Noise signatures that can be used to diagnose faulty pump operation or incipient failure.

The proliferation of industrial noise regulations in recent years has taken much of the guesswork out of allowable noise levels insofar as personnel and community exposure is concerned, and various noise standards have specified noise measurement techniques. Several organizations have developed test procedures and codes for machinery-generated noise levels.[1,2] The Hydraulic Institute code was specifically developed for the measurement of airborne sound generated by pumps (Reference 3).

The most common approach for controlling airborne noise levels from pumps is to interrupt the paths by which noise reaches the listener. When noise is an indicator of abnormal pump operation, modification of pump internals or operating conditions is normally required.

The measurement of noise for diagnostic purposes is not well prescribed, either for instrumentation or for interpretation. Even a well-designed and properly operated pump will of course produce noise. Variations in noise amplitude and frequency that result from malfunction or improper operating conditions will depend upon the type and design of the pump and the type of problem causing the noise. Measurement and analysis techniques for interpreting these signatures will depend upon whether the noise is solid-, liquid-, or airborne and upon the nature of coexisting noise from other sources.

Determining the source and cause of noise is the first step in evaluating whether noise is normal or an indicator of possible problems. Noise in pumping systems can be generated both by the mechanical motion of pump components and by the liquid motion in the pump and piping systems. Liquid noise sources can result from vortex formation in high-velocity (shear) flow, from pulsating flow, and from cavitation and flashing.

Noise from internal mechanical and liquid sources can be propagated to the environment by several paths, including the pump and support structure, attached piping, the liquid in the piping, and ultimately the surrounding air itself.

This section discusses various pump noise-generating mechanisms (sources) and common noise conduction paths as a basis for both effective diagnostics and treatment.

SOURCES OF PUMP NOISE

Effective control of pump noise requires knowledge of the liquid and mechanical noise-generation mechanisms and the paths by which noise can be transmitted to a listener.

Mechanical Noise Sources Mechanical sources are vibrating components or surfaces which produce acoustic pressure fluctuations in an adjacent medium. Examples are pistons, rotating unbalance vibrations, and vibrating pipe walls.

In positive displacement pumps, noise is generally associated with the speed of the pump and the number of pump plungers. Liquid pulsations are the primary mechanically induced noise, and these in turn can excite mechanical vibrations in components of both pump and piping system. Incorrect crankshaft counterweights will also cause shaking at running speed, which may loosen anchor bolts and produce rattling of the foundation or skid. Other mechanical noises are associated with worn hearings on the connecting rods, worn wrist pins, or slapping of the pistons or plungers.

In centrifugal machines, improper installation of couplings often causes mechanical noise at twice pump speed (misalignment). If pump speed is near or passes through the lateral critical speed, noise can be generated by high vibrations resulting from imbalance or by the rubbing of bearings, seals, or impellers. If rubbing occurs, it may be characterized by a high-pitched squeal. Windage noises may be generated by motor fans, shaft keys, and coupling bolts.

Liquid Noise Sources When pressure fluctuations are produced directly by liquid motion, the sources are fluid dynamic in character. Potential fluid dynamic sources include turbulence, flow separation (vorticity), cavitation, waterhammer, flashing, and impeller interaction with the pump cutwater. The resulting pressure and flow pulsations may be either periodic or broad-band in frequency and generally excite either the piping or the pump itself into mechanical vibration. These mechanical vibrations can then radiate acoustic noise into their environment.

In general, pulsation sources are of four types in liquid pumps:

1. Discrete-frequency components generated by the pump impeller or plungers
2. Broad-band turbulent energy resulting from high flow velocities
3. Impact noise consisting of intermittent bursts of broad-band noise caused by cavitation, flashing, and waterhammer
4. Flow-induced pulsations caused by periodic vortex formation when flow is past obstructions and side branches in the piping system

A variety of secondary flow patterns that produce pressure fluctuations are possible in centrifugal pumps, as shown in Figure1, particularly for operation at off-design flow. The numbers shown in the flow stream are the locations of the following flow mechanisms:

1. Stall
2. Recirculation (secondary flow)
3. Circulation
4. Leakage

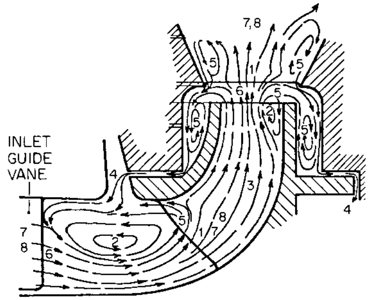


FIGURE 1 Secondary flows in and around a pump impeller stage (Reference 11)

5. Unsteady flow fluctuations
6. Wake (vortices)
7. Turbulence
8. Cavitation

Most of these unstable flow patterns produce vortices by boundary layer interaction between a high-velocity and low-velocity region in a fluid field, for example, by flow around obstructions or past deadwater regions or by bidirectional flow. The vortices, or eddies, are converted to pressure perturbations as they impinge on the sidewall and may result in localized vibration excitation of the piping or pump components. The acoustic response of the piping system can strongly influence the frequency and amplitude of this vortex shedding. Experimental work has shown that vortex flow is most severe when a system acoustic resonance coincides with the natural or preferred generation frequency of the source. This source frequency has been found to correspond to a Strouhal number (S_n) from 0.2 to 0.5, where

$$S_n = \frac{f_e D}{V}$$

where f_e = vortex frequency, Hz
D = a characteristic dimension of the generation source, ft (m)
V = flow velocity in the pipe, ft/s (m/s)

For flow past tubes, D is the tube diameter, and for branch piping excitation, D is the diameter of the branch pipe. The basic Strouhal equation is further defined in Table 1, items 4A and B. As an example, flow at 100 ft/s (30 m/s) past a 12-in diameter (0.3-m) stub line would exhibit instability tendencies at a frequency of approximately 50 Hz. If the stub is acoustically resonant at a frequency near 50 Hz, rather large pulsation amplitudes can result.

When a centrifugal pump is operated at flows less than or greater than best efficiency capacity, noise is usually heard around the pump casing. The magnitude and frequency of this noise vary from pump to pump and are dependent on the magnitude of the pump head being generated, the ratio of *NPSH* required to *NPSH* available, and the amount by which pump flow deviates from ideal flow. Noise is often generated when the vane angles of the inlet guide, impeller, and casing (or diffuser) are incorrect for the actual flow rate. Another major source contributing to this noise is referred to as *recirculation.*[4,5]

Before the pressure of the liquid flowing through a centrifugal pump is increased, the liquid must pass through a region where its pressure is less than that existing in the suction pipe. This is due in part to acceleration of the liquid into the eye of the impeller. It is also due to flow separation from the impeller inlet vanes. If flow is in excess of design and

TABLE 1 Piping vibration excitation forces

Generation mechanism	Excitation frequency f_e, Hz
1. Reciprocating compressors	nf
2. Reciprocating pumps	nf, nPf
3. Centrifugal compressors and pumps	nf, nBf, nvf
4. Flow excitation	
A. Flow through restrictions	$\frac{0.2V}{D}$ to $\frac{0.5V}{D}$
B. Flow past stubs	$\frac{0.5V}{D}$
C. Flow turbulence due to quasi-steady flow	0–30 Hz (typically)
D. Cavitation and flashing	Broad band

$n = 1, 2, 3, \ldots$
f = running speed, Hz
P = number of pump plungers
B = number of blades
v = number of volutes or diffuser vanes
V = flow velocity, ft/s (m/s)
D = restriction diameter, ft (m)

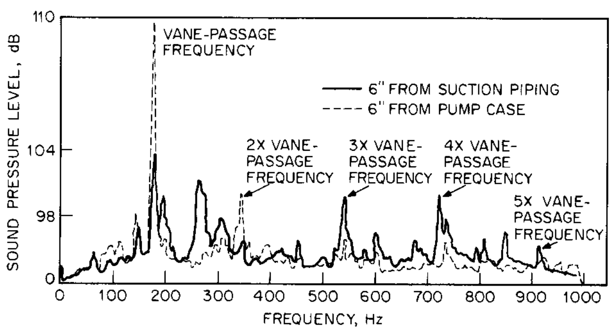


FIGURE 2 Noise spectra of cavitation and vane passage on a centrifugal pump

the incident vane angle is incorrect, high-velocity, low-pressure eddies will form. If the liquid pressure is reduced to the vaporization pressure, the liquid will flash. Later in the flow path the pressure will increase. The implosion that follows causes what is usually referred to as *cavitation* noise. The collapse of the vapor pockets, usually on the nonpressure side of the impeller blades, causes severe damage (blade erosion) in addition to noise.

Sound levels measured at the casing of an 8000-hp (5970-kW) pump and near the suction piping during cavitation are shown in Figure 2. The cavitation produced a wide-band shock that excited many frequencies; however, in this case, the vane passing frequency (number of impeller blades times revolutions per second) and multiples of it predominated. Cavitation noise of this type usually produces very-high-frequency noise, best described as "crackling."

Cavitation-like noise can also be heard at flows less than design, even when available inlet *NPSH* is in excess of pump required *NPSH*, and this has been a puzzling problem. An explanation offered by Fraser[4,5] suggests that noise of a very low random frequency but very high intensity results from backflow at the impeller eye or at the impeller discharge, or both, and every centrifugal pump has this recirculation under certain conditions of flow reduction. Operation in a recirculating condition can be damaging to the pressure side of the inlet or discharge impeller blades (and also to casing vanes). Recirculation is evidenced by an increase in loudness of a banging type, random noise, and an increase in suction or discharge pressure pulsations as flow is decreased. Refer to Subsections 2.3.1 and 2.3.2 for further information.

Pressure regulators or flow control valves may produce noise associated with both turbulence and flow separation. These valves, when operating with a severe pressure drop, have high flow velocities that generate significant turbulence. Although the generated noise spectrum is very broad-band, it is characteristically centered around a frequency corresponding to a Strouhal number of approximately 0.2.

CAVITATION AND FLASHING For many liquid pump piping systems, it is common to have some degree of flashing and cavitation associated with the pump or with the pressure control valves in the piping system. High flow rates produce more severe cavitation because of greater flow losses through restrictions.

In the suction piping of positive displacement pumps, high-amplitude pulsations can be generated by the plungers and amplified by the system acoustics and cause the dynamic pressure to periodically reach the vapor pressure of the liquid even though the suction static pressure may be above this pressure. As the cyclic pressure increases, the vapor bubbles collapse, producing noise and shock to the system, and this can result in erosion as well as undesirable noise. (See Figure 9, Section 3.4.)

Flashing is particularly common in hot water systems (feedwater pump systems) when the hot, pressurized water experiences a decrease in pressure through a restriction (for example, flow control valve). This reduction of pressure allows the liquid to suddenly vaporize, or flash, which results in a noise similar to cavitation. To avoid flashing after a restriction, sufficient back pressure should be provided. Alternately, the restriction could be located at the end of the line so the flashing energy can dissipate into a larger volume. Refer to Subsec. 2.3.4 for additional information.

NOISE CONTROL TECHNIQUES

Environmental noise usually does not emanate directly from the energy source; rather, it is transmitted along mechanical or liquid paths before it finally radiates from some vibrating surface into the surrounding environment.

The approaches to treating pump noise generally include the following:

1. **Source modification** Modify the basic pump design or operating condition to minimize the generation of acoustic energy.
2. **Interruption of transmission** Prevent sources from generating airborne (or overside) noise by interrupting the path between the energy source and the listener. This approach may range from isolation mounts at the source to physically removing the listener.

Source Modification Equipment modification to eliminate the noise source is usually quite specific to each particular situation; therefore, only general guidelines can be given. Many technical papers relate pump configurational modifications to the degree of noise reduction achieved; however, noise reduction depends on many parameters, and hence a particular modification may or may not help in a specific case. On the other hand, if the noise results from a resonant condition in the pump system, almost any reasonably conceived modification can destroy the resonance with varying degrees of improvement.

Some of the source modification approaches for pump applications are:

1. Increase or decrease pump speed to avoid system resonances of the mechanical or liquid systems.
2. Increase liquid pressures (*NPSH*, and so on) to avoid cavitation or flashing; decrease suction lift.

3. Balance rotating or oscillating components.
4. Change drive system to eliminate noisy components.
5. Correct acoustic resonance to minimize liquid-borne energy.
6. Modify centrifugal pump casing vanes so clearance between impeller diameter and casing cutwater (tongue) or diffuser vanes is increased.
7. Modify centrifugal pump impeller discharge blade configuration by pointing, slanting, grooving, adding holes, or staggering one-half pitch (if double suction).
8. Modify centrifugal pump casing cutwater (tongue) by slanting or adding holes.
9. Replace pump with different model or type to permit operation at reduced speed and the least number required.
10. If noise is due to operation of a centrifugal pump at flows less than design and recirculation is the problem, install minimum flow recirculation system bypass to increase total pump flow; if several pumps are operating in parallel, operate all pumps at the same speed and the least number required.
11. Use heavier bearing lubricant or increase number of bearing rolling elements.
12. Inject small quantity of air into the suction of a centrifugal pump to reduce cavitation noises.

The degree of improvement that can be achieved by any of the source modification approaches obviously depends upon the particulars of each installation, that is, the basic causes of excess noise. Modification of the pump internals, for example, is extremely difficult in existing installations and can produce undesirable side effects unless the pump was poorly designed or selected initially.

As described earlier, the major sources of internal noise in reciprocating pumps are usually associated with piston-induced pulsations, piston mechanical reactions, turbulence, vortex formation from separated flow around obstructions, and cavitation. In centrifugal pumps, in addition to recirculation noise, interaction of the impeller flow with the pump case (especially the cutwater), high-velocity and pressure gradients at the impeller blade tip, and flow separation can make significant contributions to pulsation levels and noise. Internal modifications to the pump can ameliorate any or all of these conditions if they are severe initially. The techniques are well known to most pump designers: Use adequate valve sizes, avoid high velocities and obstructed flows, keep pressures above the vapor pressure of the fluid being pumped, degasify the fluid, provide adequate pulsation control equipment, and maintain proper angles of attack in centrifugal machines.

Many references give examples of how changing pump design parameters affects noise.[6–10] Although such examples are valuable, they are of interest more in suggesting approaches than in predicting the degree of noise reduction that can be achieved in other pump applications.

Sudo, Komatsu, and Kondo[9] investigated pressure pulsations (and noise) generated in a centrifugal pump as a result of interference between the impeller discharge vanes and the receiving spiral casing single cutwater vane. The geometry of the pump (Figure 3) defines the gap G between the impeller outer diameter D_2 and the casing cutwater. How varying the gap G/D_2 and the skew ratio (inclination of the cutwater or impeller vanes) affected discharge pressure pulsations is shown in Figure 4. Increasing G and the skew ratio decreases pulsation and noise amplitudes. Some investigators suggest that the ratio of cutwater diameter to impeller outer diameter be as large as 2:1 for optimum operation. When the cutwater diameter (or gap) is unknown, a rule of thumb suggested to optimize impeller diameter selection is not to use an impeller larger than 85% of maximum diameter.

To reduce the effect of impeller/casing vane passing pulsations, and consequently noise, double-suction impellers should have staggered vanes; i.e., the discharge vane tips should be shifted one-half pitch. This allows the fluctuation of the flow from each half of the impeller to interfere and thus reduces pressure pulsations at the pump discharge.

Florjancic, Schöffler, and Zogg[10] have reported that centrifugal pump impeller blade and casing tongue configuration can affect sound pressure levels and alter pump head and efficiency (Figure 5).

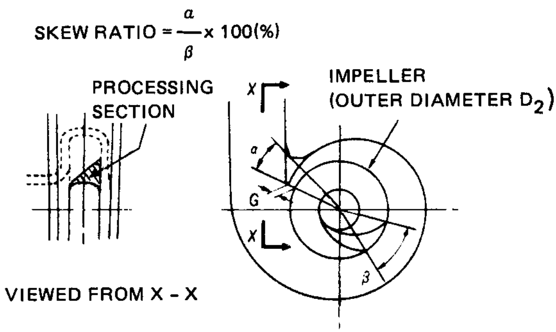


FIGURE 3 Cutwater of a spiral casing and impeller (Reference 9)

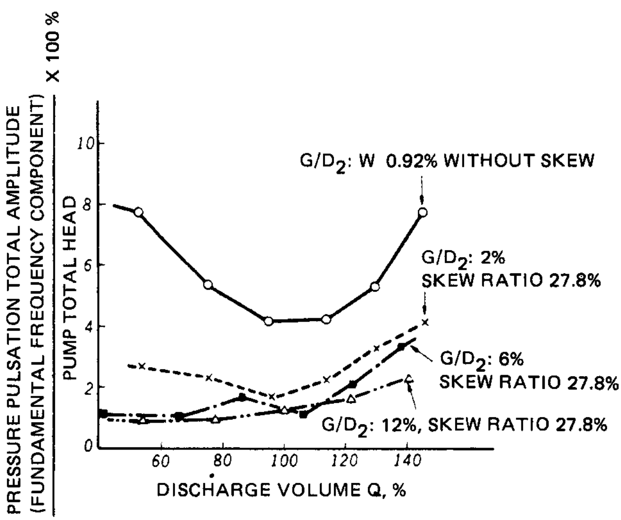


FIGURE 4 Effects of gap width between cutwater of a spiral casing and trailing edge of the impeller blade (Reference 9)

Control of Noise Paths These approaches consist of system modifications and treatments to disrupt the conduction of sound, whether borne by the structure, liquid, or air. They include approaches other than those that directly affect the originating source of oscillating energy (Figure 6).

LIQUID PATH Noise generated in the region of the pump is conducted both upstream and downstream by the liquid in the piping system. Such paths are most effectively reduced by acoustic (pulsation) filters or other pulsation control equipment, such as side branch accumulators (see Section 3.6).

STRUCTURE-BORNE NOISE Obvious paths for solid-borne noise are the attached piping and pump support systems. Oscillatory energy generated near the pump by pulsation, cavitation, turbulence, and so on, can be conducted as solid-borne noise for substantial distances

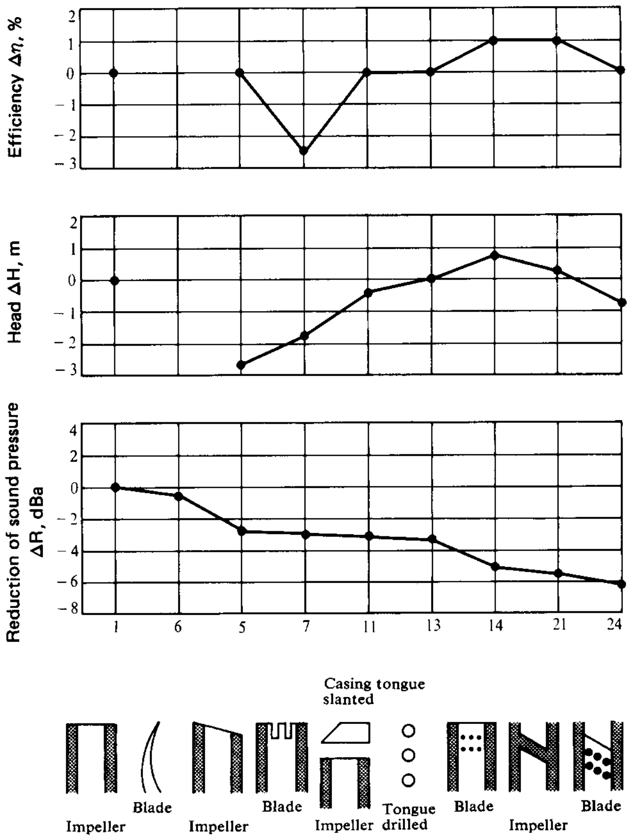
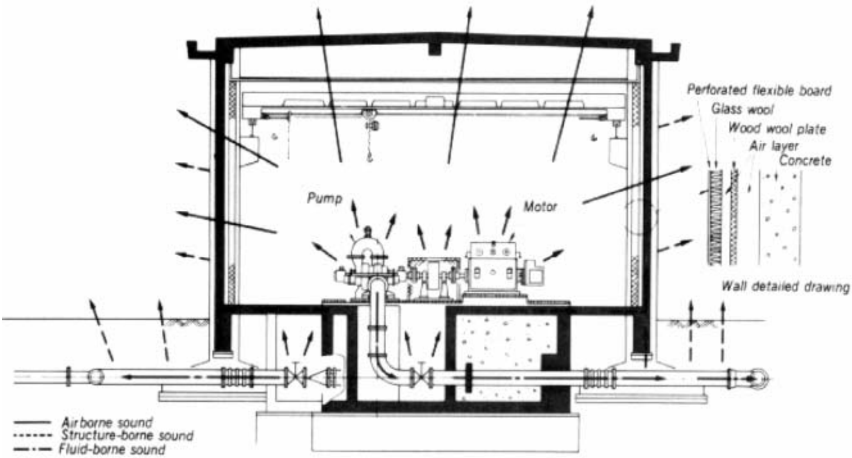


FIGURE 5 Reduction in sound pressure level with influences on head and efficiency at 100% flow for various impeller blade and casing tongue configurations (Reference 10)

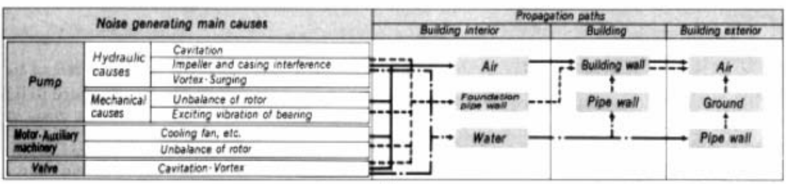
before it is radiated as acoustic noise into the atmosphere. Some success in controlling solid-borne noise propagation can be achieved by the use of flexible couplings in the piping systems, mechanical isolation (vibration mounts and so on) for the pump and drive systems, and resilient pipe hangers and supports.

Techniques for the vibration isolation of mechanical equipment are well known and available from many suppliers of vibration isolators or mounts. These may consist of resilient supports at each mounting point of the machine, although it is often necessary to mount the pump and drive system on a single rigid skid (to assure alignment) and then isolate the skid from its support system. For best isolation, the lowest resonant frequency of the supported system should be well below the minimum operating frequency, and none of the higher resonant modes should be coincident with running speed or multiples thereof.

The application of elastomeric coatings to the exterior surface of the pump or piping to damp pipe wall vibrations is normally ineffective except on very thin conduit. However, such coatings may have a small acoustic effect in confining or absorbing high-frequency noise that would otherwise be radiated by the pipe wall vibrations. (See the following discussion on pipe wraps.)



(a)



(b)

FIGURE 6 Noise propagation paths in a pumping plant (Reference 9)

Airborne Noise Although acoustic energy can be generated in the pumped liquid by purely fluid processes (turbulences and so on), most noise radiated to the surrounding air is the direct result of mechanical vibrations in the pump case, the pipe wall, or other structures to which the pump system is coupled by liquid or mechanical attachment. (The exceptions to purely mechanical sources are windage noise produced by the rotating coupling, by cooling air in the drive motors, and so on).

When excessive noise is encountered, the source or sources can sometimes be identified by making sound measurements at points in a grid around the suspect equipment and plotting sound level contours. A sound level contour of a typical flow control valve installation is shown in Figure 7. Octave band analyses or spectral analyses and contours can also provide clues by identifying the source of various frequency components of the noise. These components can also be compared with the information in Table 1 to aid in determining the source and location of the noise-generating mechanism.

The Hydraulic Institute standard[3] gives specific details of the procedures for noise measurement around pumps, including microphone locations (Figure 8), measurement procedures, and a data sheet (Figure 9).

The approaches for reducing noise from pumps and piping systems after it is airborne generally consist of either interrupting the transmission path (barriers) or controlling the reverberation characteristics of the pump room.

A highly reflective (reverberant) pump room or enclosure can increase pump noise levels several decibels by reflections of the noise back and forth in the enclosure. The maximum reduction that can be achieved by the application of acoustic absorption material to the interior surfaces is normally about 10 dB for a highly reverberant enclosure. At most, such a treatment can reduce noise levels to those that would exist if the pump were operating in the

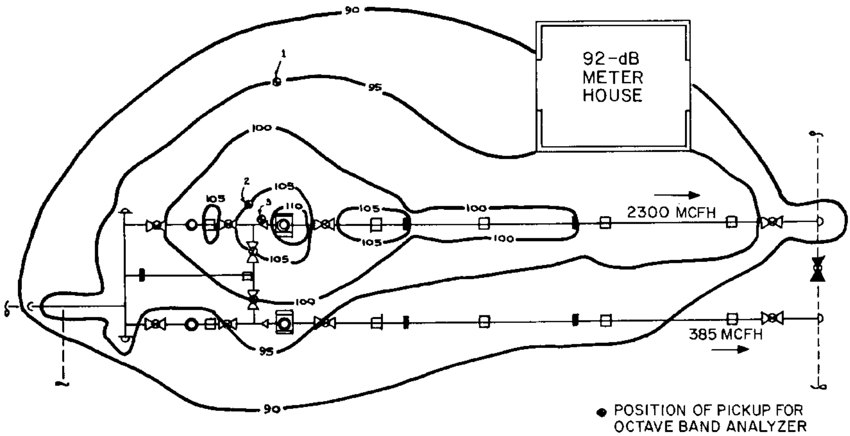


FIGURE 7 Contours of equal sound level (decibels) (Reference 12)

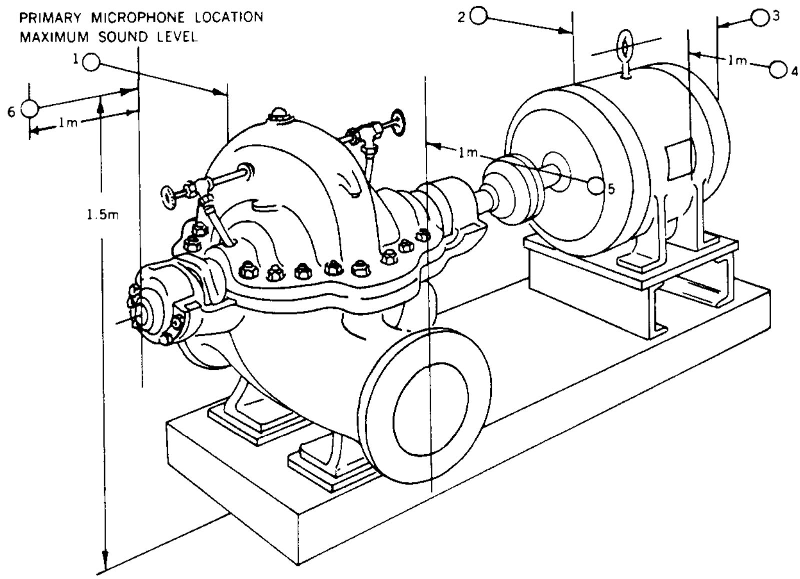


FIGURE 8 Placement of microphones on a horizontally split centrifugal pump (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3)

open, totally free of reflecting surfaces. Quantitatively, the noise reduction *NR* in decibels that can be achieved is

$$NR = 10 \log \frac{\overline{\alpha}_a}{\overline{\alpha}_b}$$

where $\overline{\alpha}_a$ = average absorption coefficient of the surfaces after treatment

$\overline{\alpha}_b$ = average absorption coefficient of the surfaces before treatment

AIRBORNE SOUND LEVEL TEST REPORT FOR PUMPING EQUIPMENT REPORT FORM

SUBJECT:

Model:________ Manufacturer:________ Serial:________
Rated Pump Speed:________ Capacity:________ Total Head:________
Type of Driver:________ Speed:________
Auxiliaries such as Gears:________
Applicable Figure No:________
Description:________

TEST CONDITIONS:

Distance from Subject to Microphone:________ Meters
Operating Speed as Tested:________
Height of Microphone Above Reflecting Plane:________ Meters
Reflecting Plane Composition:________
Primary Microphone Location No:________
Remarks:________

INSTRUMENTATION:

Microphone:________ No.________
Sound Level Meter:________ No.________
Octave Band Analyzer________ No.________
Calibrator:________ No.________
Other:________ No.________

DATA:

	db Ref. 2 × 10· N/m²	BACKGROUND	LOCATION * 1	2	3	4	5	6	7	8	9	10	AV.
MIDBAND FREQ. Hz	dB A												
	63												
	125												
	250												
	500												
	1k												
	2k												
	4k												
	8k												

*Corrected for background sound. Readings having 3 dB corrections must be reported in brackets. Only octave bands of interest as defined on page 299 need be reported.

TESTED BY:________ DATE:________

REPORTED BY:________ DATE:________

FIGURE 9 Hydraulic Institute data sheet for measurement of airborne sound from pumping equipment (Hydraulic Institute ANSI/HI 2000 Edition Pump Standards, Reference 3)

The average absorption coefficient is defined as follows:

$$\bar{\alpha} = \frac{\alpha_1 A_1 + \alpha_2 A_2 + \alpha_3 A_3 + \cdots \alpha_n A_n}{A_1 + A_2 + A_3 + \cdots A_n}$$

where $\alpha_1, \alpha_2, \alpha_3, \ldots$ are absorption coefficients of various surface areas within the enclosure and $A_1, A_2, A_3, \ldots$ are the corresponding surface areas.

Absorption coefficients for typical building materials are given in Table 2. Note that the absorption coefficients vary with frequency, and hence calculated values of *NR* can be quite frequency-sensitive. Absorption data for various absorbing materials can best be obtained from the suppliers. One of the most common and most effective of such materials is fiberglass matting or the more rigid fiberglass board.

The noise reduction that can typically be attained with fiberglass piping insulation is shown in Figure 10. Note that noise reduction varies with noise frequency and with the thickness and density of the fiberglass matting. Densities of 5 to 6 lb/ft^3 (80 to 96 kg/m^3), found in the rigid fiberglass board, are normally more effective than the 1 to 2 lb/ft^3 (16 to 32 kg/m^3) found in rolled matting.

Normally, the average absorption in a room must be increased by a factor of at least three before noise improvement is discernible to the ear. Unless the absorption coefficient of the untreated room is less than about 0.3 in the frequency range of maximum noise, the

TABLE 2 Sound absorption coefficients of common construction materials

Material	Frequency, Hz 125	250	500	1000	2000	4000
Brick						
Unglazed	0.03	0.03	0.03	0.04	0.04	0.05
Painted	0.01	0.01	0.02	0.02	0.02	0.02
Concrete block, painted	0.10	0.05	0.06	0.07	0.09	0.08
Concrete	0.01	0.01	0.015	0.02	0.02	0.02
Wood	0.15	0.11	0.10	0.07	0.06	0.07
Glass	0.35	0.25	0.18	0.12	0.08	0.04
Gypsum board	0.29	0.10	0.05	0.04	0.07	0.09
Plywood	0.28	0.22	0.17	0.09	0.10	0.11
Soundblox concrete block						
Type A (slotted), 6 in (15 cm)	0.62	0.84	0.36	0.43	0.27	0.50
Type B, 6 in (15 cm)	0.31	0.97	0.56	0.47	0.51	0.53
Carpet	0.02	0.06	0.14	0.37	0.60	0.65
Fiberglass typically 4 lb/ft^3, (64 kg/m^3), hard backing						
1 in (2.5 cm) thick	0.07	0.23	0.48	0.83	0.88	0.80
2 in (5 cm) thick	0.20	0.55	0.89	0.97	0.83	0.79
4 in (10 cm) thick	0.39	0.91	0.99	0.97	0.94	0.89
Polyurethane foam, open-cell						
$\frac{1}{4}$ in (0.6 cm) thick	0.05	0.07	0.10	0.20	0.45	0.81
$\frac{1}{2}$ in (1.3 cm) thick	0.05	0.12	0.25	0.57	0.89	0.98
1 in (2.5 cm) thick	0.14	0.30	0.63	0.91	0.98	0.91
2 in (5 cm) thick	0.35	0.51	0.82	0.98	0.97	0.95
Hairfelt						
$\frac{1}{2}$ in (1.3 cm) thick	0.05	0.07	0.29	0.63	0.83	0.87
1 in (2.5 cm) thick	0.06	0.31	0.80	0.88	0.87	0.87

[a]For specific grades, see manufacturer's data; note that the term *NCR,* when used, is a single-term rating that is the arithmetic average of the absorption coefficients at 250, 500, 1000, and 2000 Hz.

Source: Reference 8.

addition of absorbing material to the walls or ceiling will likely be ineffective. Nevertheless, the noise reduction could amount to 5 dB and thus could mean the difference between compliance and noncompliance with existing criteria.

The use of barriers near or enclosing the pump can be much more effective than environmental absorption treatments, particularly if the barriers totally confine the noise-producing components (Figure 11). Acoustic barriers simply interrupt the airborne path from the source to the listener. Such barriers work best (have the highest transmission loss) when

1. Their area density is high.
2. They are total enclosures and all joints are well sealed against air leaks.
3. They are not mechanically tied to the vibrating surfaces of the pump (that is, they are mechanically isolated).

For total enclosures, the barrier area density controls transmission loss for noise frequencies above the lowest mechanical flexural frequency of the barrier panels. Loosely sprung but well-sealed and heavy panels therefore serve this function well. Treating the interior of the enclosure with sound-absorbing material tends to reduce the apparent source strength by preventing reverberant buildups in the enclosure. Note that the use of a total enclosure around a pump may cause other operational problems that must be dealt

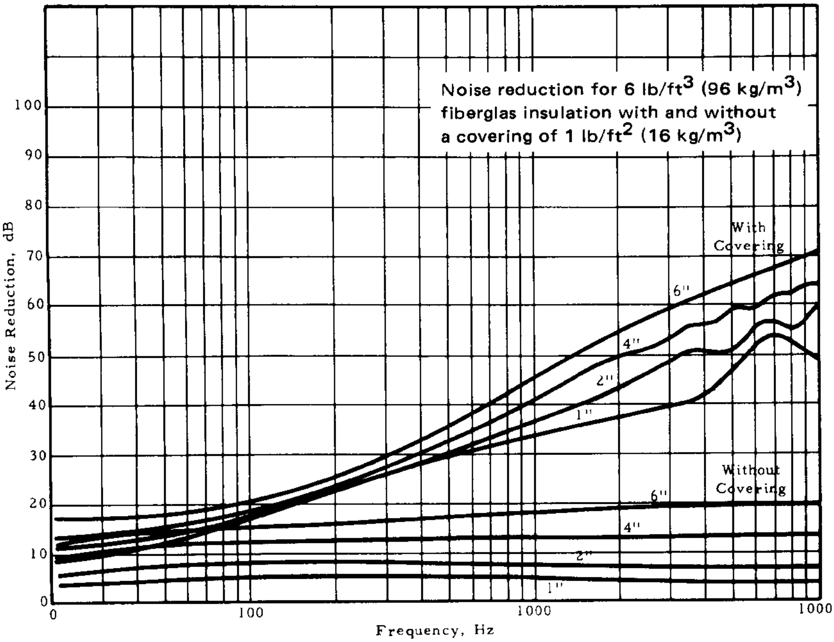


FIGURE 10 Noise reduction for piping with fiberglass insulation (1 in = 2.54 cm)

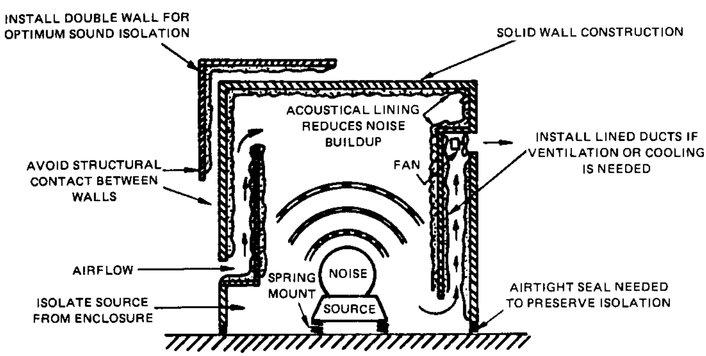


FIGURE 11 Design of an effective sound-isolating enclosure requiring air circulation for cooling (Reference 8)

with, for example, overheating, formation of explosive mixtures, or difficulty in getting to the pump for maintenance.

Although partial barriers around a pump usually cannot provide the degree of noise reduction afforded by total enclosures, they may be quite useful. Such barriers normally redirect the noise away from the listener and work best when they are located very near either the source or the listener. They also work better in an open or acoustically dead room than in a highly reverberant one. Their performance can often be enhanced by treating the interior (pump-side) surface with absorbing material or by reflecting the noise into an absorbing surface. The transmission loss of partial barriers is usually not critical as long as it is 20 dB or greater, as reverberation and flanking noise paths will normally limit overall reduction to less than 20 dB.

Exterior control techniques for pipe noise usually consist of pipe wrapping materials in addition to the resilient pipe supports and hangers already mentioned. The function of such materials is identical to that of total enclosures; hence, pipe wraps should be

1. Massive (high area density)
2. Mechanically decoupled from the pipe (loosely sprung)
3. Airtight

For all the very-low-frequency noise, decoupling between a heavy, airtight outer coating and the vibrating pipe wall can be achieved by a compliant acoustic absorbing material, such as preformed fiberglass pipe insulation. This material provides some acoustic absorption of noise as it "passes through," but, more important, it provides vibratory decoupling between the vibrating pipe wall and the external pipe wrap shell or layer.

Multiple-layered panels or pipe wrap normally provide more attenuation than single homogenous layers, as long as the layers are mechanically decoupled. Examples of several pipe wrap configurations are given in Figure 12. Alternative lead-free materials are replacing leaded vinyl.

NOISE MEASUREMENT

In very general terms, sound consists of cyclic modulations of ambient pressure at frequencies that are detectable to the human ear, normally between 30 Hz and 17 kHz. For our purposes, noise is simply unwanted sound. The complications in measuring noise or in assessing its effects on personnel arise chiefly because of the remarkable complexity of the human ear and its very nonlinear acoustic response with both frequency and amplitude.

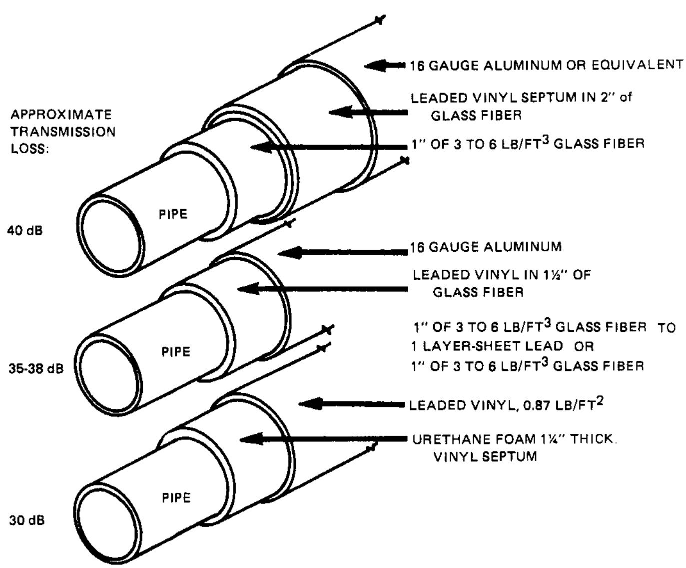


FIGURE 12 Three possible pipe wrap configurations (1 to = 2.54 cm; 1 lb/ft³ = 16 kg/m³) (Reference 12)

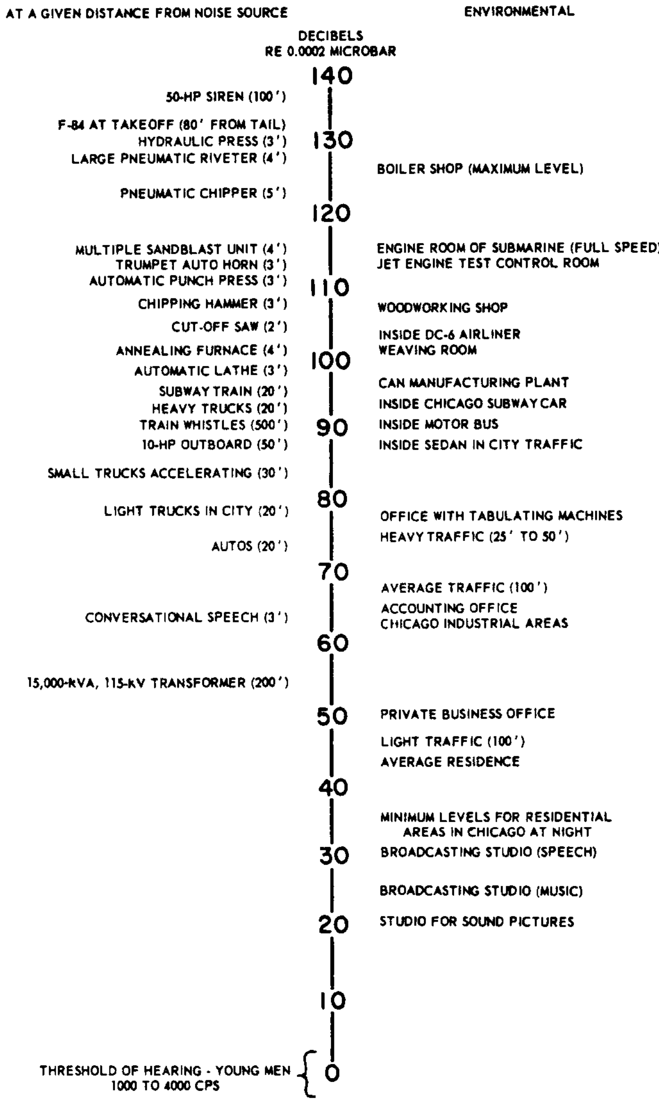


FIGURE 13 Typical overall sound levels (1 ft = 0.3048 m) (Reference 13)

The dynamic range of the human ear is 160 dB, from a minimum detectable level of about 3.7×10^{-9} lb/in^2 to about 0.37 lb/in^2 (2.6×10^{-10} to 0.026 bar*). In order to compensate for the ear's dynamic range, the decibel scale was adapted to roughly simulate the ear's nonlinear sensitivity and reduce numerical sound level values to a manageable range. The strength of familiar sounds relative to their typical decibel level is given in Figure 13.

*1 bar = 10^5 Pa.

The decibel is a unit of sound measurement relating the dynamic pressure variations produced by the source of interest to a reference sound pressure (0.0002 μbar). The sound pressure level in decibels is equal to 20 times the common logarithm of this ratio:

$$SPL = 20 \log\left(\frac{P}{0.0002}\right)$$

where P is the rms sound pressure in microbars.

For example, if one sound pressure level is twice another, the measured noise level will be 6 dB greater.

The basic instrument used to measure sound intensity is the sound level meter, which provides a numerical reading of decibel values by integrating sound throughout the audible frequency spectrum. Because the frequency response of the ear varies with sound intensity, the sound level meter has three internal filters that can be used to approximate the ear's frequency response at 45, 75, and 95 dB. These are called the A, B, and C filters, respectively. In recent years, use of the C weighing filter has become popular in prescribing noise criteria. The dBC scale has been thus adapted, partially because it is easy to use and partially because it attenuates very-low-frequency noise, which is not as potentially injurious to the ear.

CRITERIA

In order to decide whether noise treatment is necessary, the sound levels must be measured and compared with applicable criteria. Most protective noise criteria have been developed to protect a specific statistical sample of people from prescribed typical noise conditions and exposure patterns. Although such criteria are admittedly approximate, they are usually sufficiently conservative to protect a large majority of those who may be exposed, whether they are written to prevent hearing loss, community annoyance, speech and communication masking, or any of the other physiological or psychological effects of noise.

When noise criteria are legislated into noise codes or regulations, they normally specify noise measurement locations, instruments, and sampling times as well as allowable levels and exposure durations. Thus they have taken much of the guesswork out of evaluating machinery noise. The main criterion for pump and other machinery installations can be obtained from the U.S. Department of Labor, OSHA Bulletin 334. OSHA guidelines for allowable daily noise exposure as a function of time and sound pressure level for personnel near the installations are given in Table 3. The allowable exposure time depends upon the maximum sound level in the work space. When the noise exposure for personnel is intermittent, the allowable exposure may be calculated from Table 3 by computing the fractions of actual exposure time at a given noise level to the allowable exposure time at that level. If the sum of these fractions is less than 1, the noise level can be considered safe. It is important to know one underlying implication in this criterion. If a reduction of the noise level is not economically feasible, it may be reasonable to schedule operators' work so the criterion levels will not be exceeded. This rescheduling may permit a wider latitude in the final solution of a plant noise problem.

MODEL TESTING

It is possible to conduct tests on reduced-size pumps and pumping systems in order to predict and adjust pressure pulsations and noise from the pump and piping. This has been successfully done by Sudo, Komatsu, and Kondo[9] using a model variable-speed, single-stage, double-suction pump in addition to mathematical computer confirmation. A comparison of performance with nonstaggered and staggered impeller vanes revealed the good pulsation suppression characteristics of the latter impeller configuration (Figure 14).

These investigators, in their modeling, used the following relationships to provide the required similarity between model and prototype:

TABLE 3 OSHA noise exposure limits

Sound level, dBA	Allowable exposure time: Minutes	Allowable exposure time: Hours
90	480	8.00
91	420	7.00
92	360	6.00
93	320	5.33
94	280	4.67
95	240	4.00
96	210	3.50
97	180	3.00
98	160	2.67
99	140	2.33
100	120	2.00
101	105	1.75
102	90	1.50
103	80	1.33
104	70	1.16
105	60	1.00
106	54	0.90
107	48	0.80
108	42	0.70
109	36	0.60
110	30	0.50
111	27	0.45
112	24	0.40
113	21	0.35
114	18	0.30
115 (maximum allowable level)	15	0.25

1. Equal ratios of pipe length to wavelength
2. Equal ratios of pipe cross-sectional area
3. Model pump speed adjusted to make the product of pulsation wavelength and frequency equal to the speed of sound in water

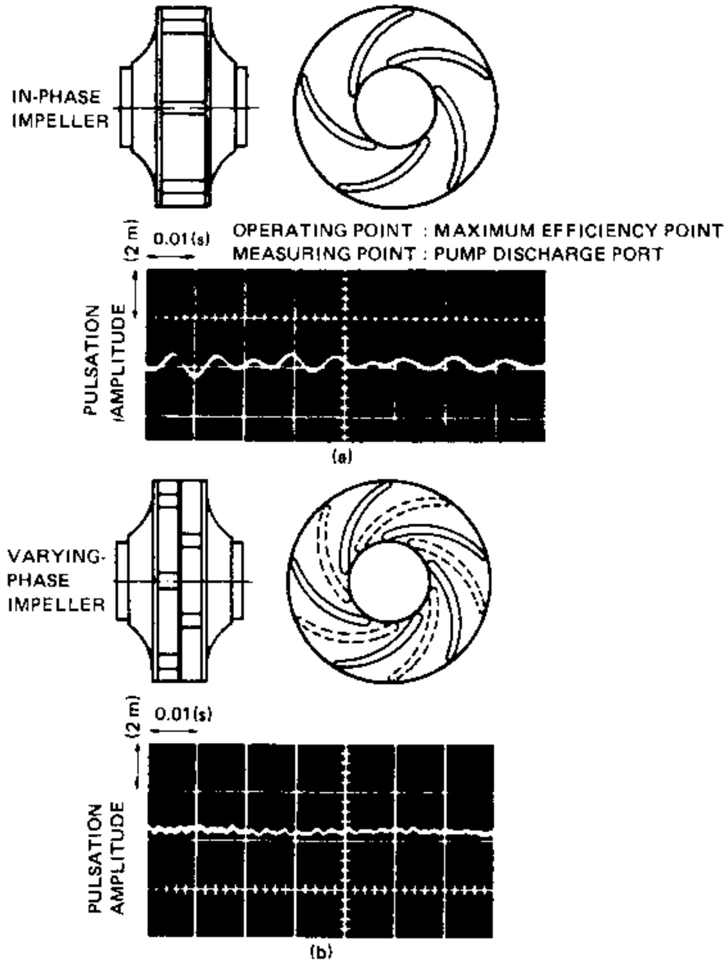


FIGURE 14 Comparison of pressure pulsation waveforms: (a) a double-suction in-phase centrifugal pump impeller and (b) a varying-phase (staggered) centrifugal pump impeller (Reference 9)

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