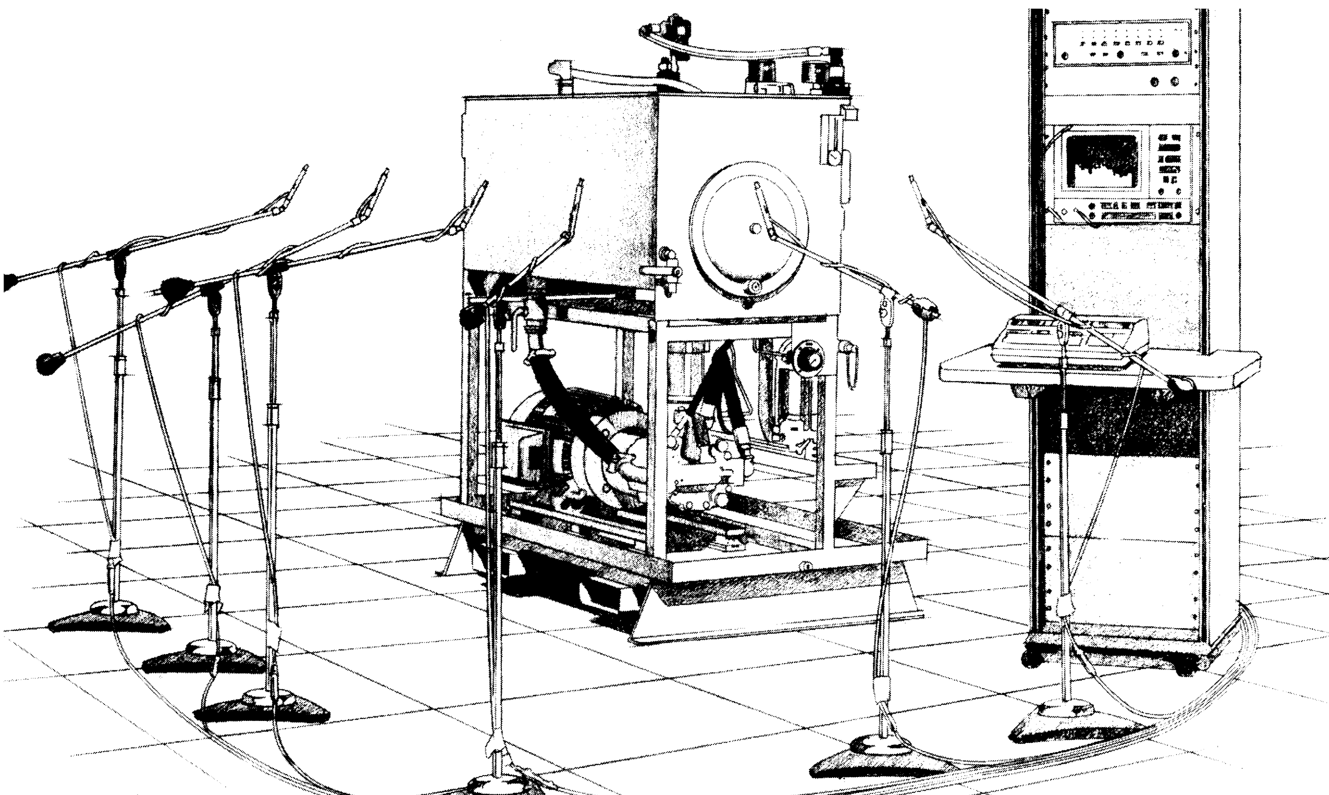
Noise Control in Hydraulic Systems



Introduction

Although a certain amount of noise control is required in hydraulic systems just to conform to government regulations, a conscientious noise control program actually provides a competitive edge. The combination of a quiet pump, well-engineered vibration and pulsation controls, and good, economical installation practices will result in a product with a distinct advantage in the marketplace.

The Vickers publication "More Sound Advice," issued in the 1970's, illustrated a variety of machine noise control methods. Because of the nearly infinite variety of hydraulic applications, it's not possible here to discuss the individual features of particular systems. There are, however, a number of installation techniques that can be applied to almost *all* hydraulic systems. When used correctly, these techniques can yield significant reductions in noise.

This publication describes the following:

1. Noise generation and noise control techniques.
2. Noise terms, definition and the use of the decibel.
3. Noise measurement procedures.

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INTRODUCTION

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SYSTEM NOISE CONTROL

Quiet Hydraulics – A Team Effort

A successful noise control program requires a team effort by individuals in several areas of expertise. A quiet hydraulic pump does not guarantee a quiet system. The choice of a quiet pump should be only one part of a multifaceted program that calls upon the talents of the system designer, fabricator, installer, and maintenance technicians. And if any member of the team fails to do their job, it can mean failure of the entire noise control program.

The pump and system designers play a key role in achieving successful noise control. They must evaluate every noise control technique available from the standpoints of both cost and practicality. Three of the basic approaches used in quieting hydraulic power systems are:

A) Internal and external pump pulsation control

B) Pump and structure isolation

C) Damping and/or stiffening

Noise Transmission and Generation

Noise is defined as the unwanted by-product of fluctuating forces in a component or system. In a hydraulic system, this noise can be transmitted in three ways: through the air, through the fluid, and through the system's physical structure. We generally think of noise as travelling only through the medium of the air, going directly from its source to some receiver (our ear). This is called *airborne noise*. That airborne noise, however, must have a source within some component of the hydraulic system. That component is normally the pump. Whether it's a piston, vane, or gear pump, the internal pumping and porting design can never be perfect. As a result, uneven flow characteristics and pressure waves are created and transmitted through the fluid. This is known as *fluidborne noise*. The pressure wave fluctuations of fluidborne noise in turn create corresponding force fluctuations. These result in vibration, also known as *structureborne noise*. This structureborne noise is transmitted not only through the pump body, but through attached structures as well. These structures then emit an audible sound.

The surrounding structures and surface areas in a hydraulic system tend to be much larger than the pump itself, and therefore radiate noise more efficiently. For this reason, while the pump design should minimize internal pulsations, it's also important to use proper isolation techniques to keep the remaining vibrations from reaching adjoining structures.

Design for Low Noise

An intelligent program of noise control should start at the source: the pump. A quiet pump is the responsibility of the pump manufacturer. The problem for the designer is that although a hydraulic pump is required to perform over a wide range of speeds and pressures, noise control can only be optimized for a relatively narrow portion of that range. The most common strategy is to use porting design to limit the pressure pulsations at the pump's rated speed and pressure. The pulsations are reduced as much as possible without creating a large amount of noise at lower speeds and pressures. Piston, vane, and gear pumps are similar in that their total output flow is the sum of the flows from the individual pumping elements or chambers. Fluid fills the chambers at the pump inlet, is compressed mechanically and/or hydraulically through orifices, and is then combined into a single discharge flow. Each pumping element in a piston pump delivers its fluid to the discharge port in a half-sine profile. The pump discharge is the total of the equally spaced half-sines added in phase. The result is an inherent flow ripple, as shown in Figure 1 for a nine-piston pump. This ripple is independent of any fluid compression, either through piston motion or any type of internal hydraulic metering. Vane pump flow ripple is more controllable. Cam contours can be designed to reduce mechanical compression effects. This is done by making pressure transitions in the dwell section, where there is controlled change in vane chamber volume. For this reason, vane pumps will normally generate less noise over a wider range of speeds and pressures than piston pumps.

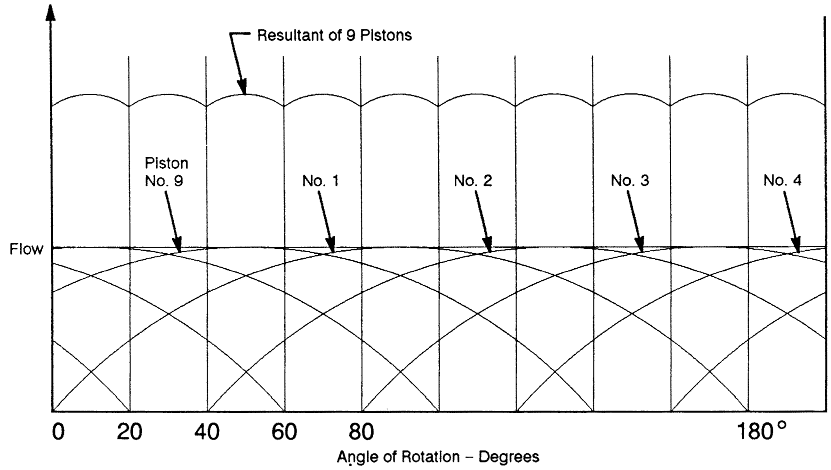


Figure 1. Inherent Piston Pump Discharge Ripple

Noise Frequencies

Pump noise energies are generated in several ways. Vibrational energy is created by an imbalance in the pump, drive motor, or couplings. It can also be produced by some undesired interaction in the assembly, but it's rare for any significant audible noise to be generated by these interactions. Nonetheless, care should be taken to minimize its effects on pump or motor life. Figure 2 shows the frequency spectrum of a ten-vane pump operating at 1800 rpm with shaft rotation frequency of 30 Hz. Any misalignments in the power train will produce noise components at twice and four times this frequency.

The strongest energy components occur at pumping frequency. This frequency equals the number of pumping chambers times the shaft frequency (300 Hz in Figure 2). Noise energy is also produced at multiples, or harmonics, of this frequency. 600 Hz and 900 Hz are the second and third harmonics seen in Figure 2. These harmonics have enough amplitude to produce significant noise. This noise comes not only from the pump itself, but from attached structures which are often more efficient at radiating the noise transmitted from the pump.

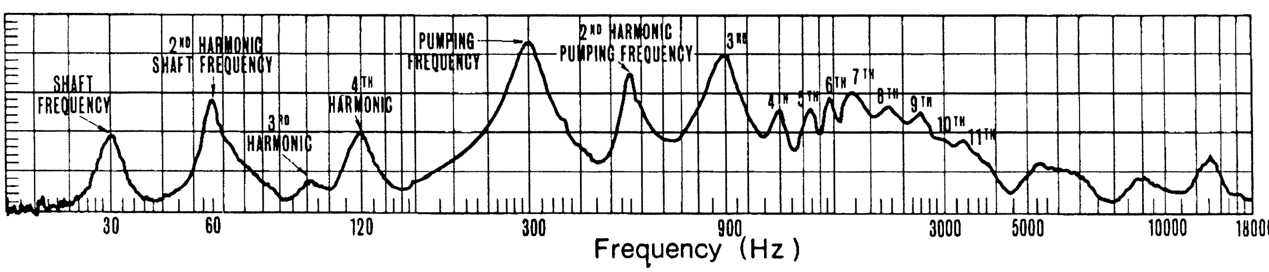


Figure 2. Structureborne or Fluidborne Spectrum Identifying Shaft and Pumping Frequency and Harmonics

Vibration Noise Control

Vibration control is used to prevent pulsation energy from the pump from being transmitted to machine structures. Most pump and drive motor assemblies are attached through a flexible coupling and mounted on a common base to maintain alignment. The common base is resiliently mounted to the support structure, as seen in Figure 3. An isolator should be selected that has a natural frequency approximately $^1/_2$ or less the pump's rotational frequency. For example, an isolator with a natural frequency of 10 Hz or less would be appropriate for a pump with a rotational or forcing frequency of 20 Hz at 1200 rpm, and would work even better for a pump with a frequency of 30 Hz at 1800 rpm. The higher the ratio between the forcing frequency and the natural frequency of the isolator system, the greater the amount of isolation (see Figure 4). A typical commercial isolator (which costs about $15 in moderate quantities) can reduce transmitted vibration energy by 10 dB at 1200 rpm and 15 dB at 1800 rpm (Figure 5).

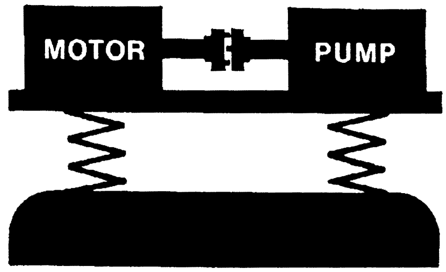
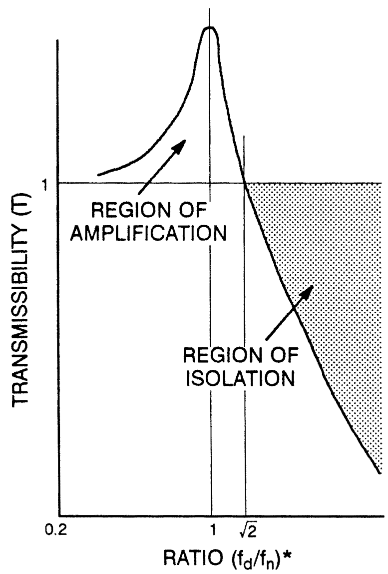


Figure 3. Pump and Motor on Subplate, Isolated from Stiff Foundation



*f_d = forcing frequency
f_n = natural frequency'

Figure 4. Typical Transmissibility Curve for an Isolated System

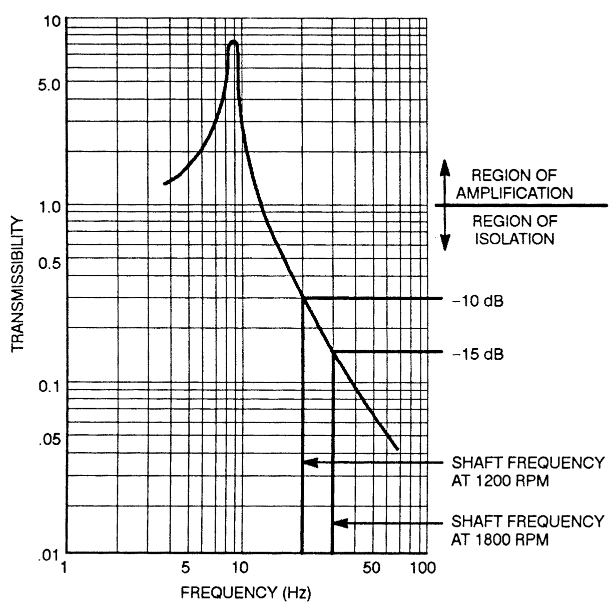


Figure 5. Transmitted Vibration Energy Reduction Using Typical Commercial Isolator

Isolators are classified by their load carrying capacity and related natural frequency. When pump, motor, and subplate assembly weights are known, the amount of evenly distributed weight on each of the isolators can be calculated. Isolators should be selected that will not be loaded above 60% to 70% of their capacity. This will allow a sufficient safety margin in the event of shock loading.

The same type of isolators can also be used on power units with overhead reservoirs. Eight isolators can be installed either under the reservoir feet (Figure 6), or under the upright leg structures supporting the reservoir. The isolators shown would be very effective because the ratio of forcing frequency to natural frequency is very high. For example, a nine-piston pump operating at 1200 rpm would have a pumping frequency of 9 x 20 rev/sec or 180 Hz. If a 10 Hz isolator system were used, there would be a very low level of vibration transmission, because the ratio between the two frequencies would be 18:1.

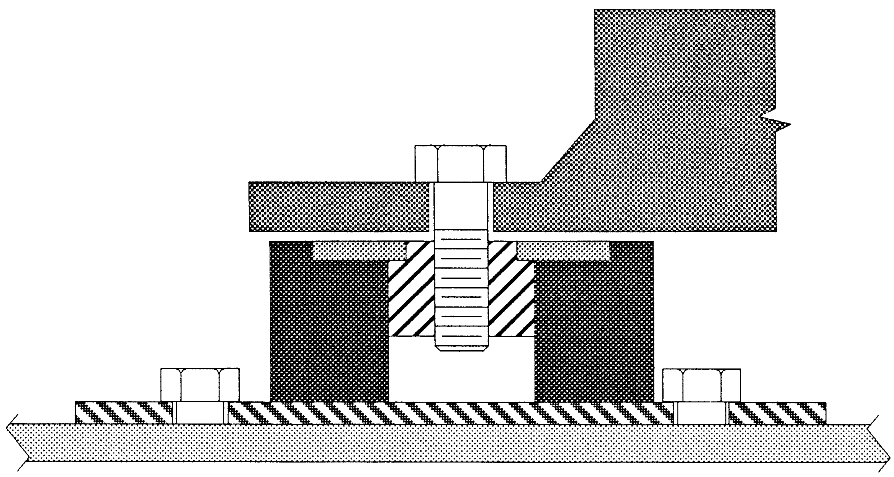


Figure 6. Reservoir Foot On Isolator

The chart on the right lists load ratings of typical isolators with carrying capacities of 60 to 4400 lbs. per isolator.

Load Series	Isolator Number	Max. Static Load Per Isolator (lbs.)
Light	L1 L2 L3 L4 L5	60 100 130 200 260
Medium	M1 M2 M3 M4	300 450 550 700
Heavy	H1 H2 H3	700 1000 1500
Extra Heavy	EH1 EH2 EH3 EH4	1500 2000 3000 4400

Table 1 on the following page provides two examples of proper isolator selection:

A) Pump, motor, and subplate weight	=	800 lbs.
Load on each of 6 isolators	=	133 lbs.
Maximum static load when loaded to 60 to 70% of capacity	=	190 to 222 lbs.
Selection: # L4 (from chart on page 4)		
B) Reservoir weight	=	550 lbs.
Attached accessories weight	=	150 lbs.
100 gallons oil weight (7 lbs./gal.)	=	700 lbs.
TOTAL	=	1400 lbs.
Load on each of 4 isolators	=	350 lbs.
Maximum static load when loaded to 60 to 70% of capacity	=	500 to 583 lbs.
Selection: # M3 (from chart on page 4)		

Table 1. Isolator Selection

Isolation With Hose

Rubber hose must be used to maintain the vibration isolation afforded when the pump and motor assembly are mounted on isolators. The isolation capabilities of hose will reduce the amount of vibration energy entering the system. Unfortunately, improper use of hose is probably the main cause of noise in many systems.

Although structureborne noise can be reduced by using long lengths of hose, the pressure pulsations from the pump will cause the hose to undergo cyclic radial expansion. One of the drawbacks of hose is that it acts as an efficient radiator in the frequency range where most of the energy is generated: the first few harmonics. Because of these two factors, long lines of hose are less effective for noise reduction than the use hose at either end of a solid line (Figure 7).

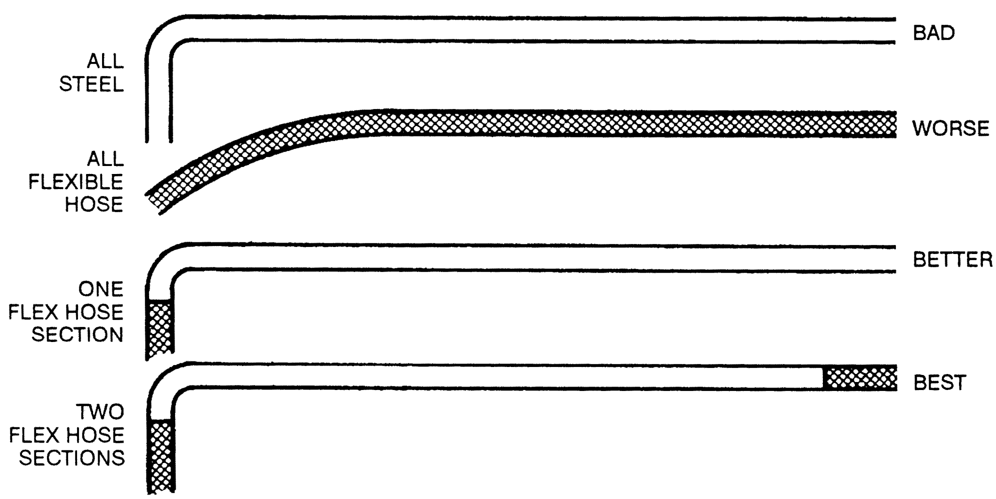


Figure 7. Long Hydraulic Line Configurations

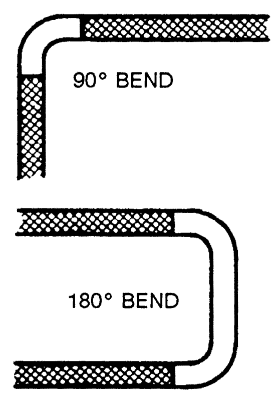


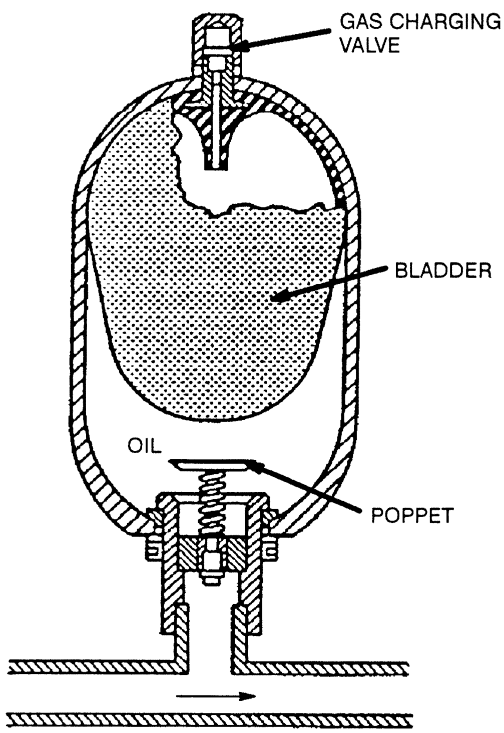
Figure 8. Preferred Short Hydraulic Line Configurations

Two other shortcomings of hose are that its length changes with pressure, and that when it's bent through a radius it acts like a Bourdon tube, trying to straighten out with increasing pressure. Both produce forces that act on connecting structures. The best way to maintain noise control while making bends with hose is to use a solid elbow with a hose section on each end. This eliminates problems caused by the Bourdon tube effect. Any change in the length of one hose is accommodated by bending in the other hose. Figure 8 illustrates the preferred configurations for 90° and 180° bends.

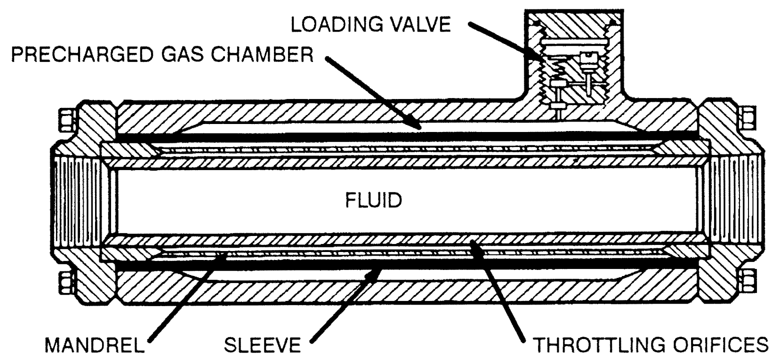
Fluidborne Noise Control

Fluidborne noise control begins with a pump's internal design. The ports should be configured so that the lowest practical pressure pulsations are generated. Additional external controls can be added to prevent as much pulsation energy as possible from being communicated to the system.

This is usually done by adding expansion volume at the outlet of the pump. These acoustic filters, as they are called, can take many forms. The two most common are gas charged side branch accumulators and flow-through type pulsation filters. Each has its advantages and disadvantages. The side branch type is cheaper, but limited to attenuating pulsations in only a narrow range of frequencies. As a result, it isn't totally effective throughout the first four pump harmonics (where most of the pulsation energy is generated). The flow-through device, although generally larger and more expensive, has a distinct performance advantage: pulsations throughout the spectrum are reduced, including those at the most significant harmonics. The filters (shown in Figure 9) have optimum effectiveness when gas charged to $^{1}/_{3}$ the maximum operating pressure of the hydraulic system.



SIDE BRANCH ACCUMULATOR



PASS-THROUGH GAS CHARGED FILTER

Figure 9. Acoustic Filters

Application of Techniques

Experimental evaluations were made on a typical automotive application: a power unit with a 150 gallon overhead reservoir supplying oil to a piston pump. The pump delivered 40 gpm at 1200 rpm at a pressure of 750 psi. The noise level, as received, was 88 dB(A). (Standard accepted noise measurement procedures are explained in the subsection entitled System Noise Evaluation.) Four different noise reduction techniques were applied to the system, starting with those that would have the greatest effect on noise levels. The results are shown in Table 2.

ITEM	DESCRIPTION	RESULTING NOISE LEVEL	CHANGE IN NOISE LEVEL
1	Changed radiused pressure hose to two pieces, separated by right angle fitting.	83 dB(A)	-5 dB(A)
2	Item 1, plus structure isolators under reservoir and upright supports (8 additional isolators).	79 dB(A)	-4 dB(A)
3	Items 1 and 2, plus valve plate designed for 1000 psi rather than 3600 psi. (Pulsations reduced from 200 psi to 140 psi.)	76 dB(A)	-3 dB(A)
4	Items 1, 2, and 3, plus flow-through pulsation filter. (Pulsations reduced from 140 psi to 35 psi.)	74 dB(A)	-2 dB(A)

Table 2. Noise Reduction Techniques

It's important to note that if the changes outlined above had been made in some other sequence, the amount of noise reduction at each step would have been different from that shown - particularly if items 3 and 4 had been evaluated before items 1 and 2. The final noise level of 74 dB(A) would be the same, but the first item tried wouldn't have yielded such a significant reduction. In the example shown, the areas of highest noise radiation were addressed first. This is essential because no appreciable noise reduction can be achieved unless the most significant noise source is identified and its level reduced first.

NOISE TERMINOLOGY

Noise Terms and Equations

Sound at a particular point in air is defined as the rapid variation in air pressure around a steady state value. Sound pressure is measured in the same units as atmospheric pressure. Since it's an alternating quantity, the term "sound pressure" is usually referred to by its root mean square (rms) value. At a frequency of 1000 Hz, a sound with an rms pressure of 2 x 10^{-4} microbars (ubar), or about 2 x 10^{-10} atmospheres, is just below the hearing threshold of someone with good ears. Expressed in more familiar terms, that level of sound pressure would be 2.9 x 10^{-9} psi. The fact that slightly greater pressures become audible shows the amazing sensitivity of the human ear. It can detect variations in atmospheric pressure as small as a few parts in 20,000,000,000.

In addition to this sensitivity, the human ear has an enormous dynamic range. Not only can it detect sounds as small as 2×10^{-4} ubar, it can accommodate sound pressures as high as 2000 ubar without being overloaded, i.e. causing pain. That's a dynamic range ratio from threshold to pain of 10,000,000:1 (Figure 10). Because this range is so large, it's more convenient to express ratios in powers of 10 (hence the use of the log scale). Sound pressure above the reference value of 2×10^{-4} ubar is referred to as sound pressure level (SPL) and expressed in decibels (dB).

$$\boxed{SPL = 20 \log \frac{P}{P_o}}$$

OR

$$\boxed{SPL = 10 \log \left(\frac{P}{P_o}\right)^2}$$

Where SPL = Sound pressure level (dB)
P = Sound pressure (bar)
P_o = Reference pressure (.0002 ubar)

From this equation, a pressure ratio of 10,000,000 (10^7) would result in the following noise level:

$$\begin{aligned} SPL &= 20 \log 10^7 \\ &= 7\ (20) \log 10 \\ &= 7\ (20)\ (1) \\ &= 140 \text{ dB (i.e., painful)} \end{aligned}$$

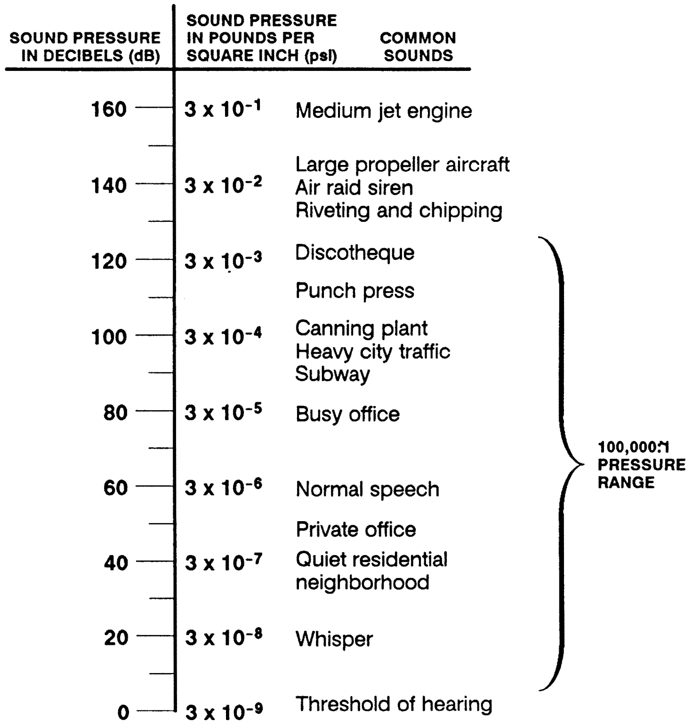


Figure 10. Dynamic Range of the Human Ear

Pressure ratios can also be calculated based on the changes in sound pressure levels (ΔSPL). For example, what is the pressure ratio (Rp) if the noise level changes by 3 dB?

$$Rp = 10^{\Delta SPL/20}$$

Where Rp = pressure ratio
= $10^{3/20}$
= 1.41

There are two important conclusions to note here: If the noise level increases from 82 to 85 dB, there's actually a 41% increase in noise; if the noise level decreases from 85 to 82 dB, there's a 29% decrease in noise ($^1/_{1.41}$ or 71% of the original level).

Table 3 lists the pressure ratios for changes in SPL from +10 to -10 dB with the previous example in bold:

Change In SPL	Press. Ratio	Change In SPL	Press. Ratio
1	1.12	-1	.89
2	1.26	-2	.79
3	**1.41**	**-3**	**.71**
4	1.59	-4	.63
5	1.78	-5	.56
6	2	-6	.5
7	2.24	-7	.45
8	2.51	-8	.40
9	2.82	-9	.35
10	3.16	-10	.32

Table 3. Pressure Ratios

(A chart of typical noise levels is shown in Figure 10.)

Human Response to Noise – The "A" Scale

A microphone measures actual sound pressures emitted from a noise source, but the human ear doesn't treat equal levels with equal tolerance over the audible frequency range of up to 12,000 Hz. The ear is more sensitive to noise above 1000 Hz. This sensitivity is simulated by using the "A" scale filtering system in the signal processing of measured noise. This internationally standardized system gauges the ear's response to noise through the use of a passive frequency related filter placed between the microphone and output (Figure 11). The resulting energy, when summed in the respective weighted frequency bands, is then expressed in "A" scale, "A" weighted, or dB(A) levels.

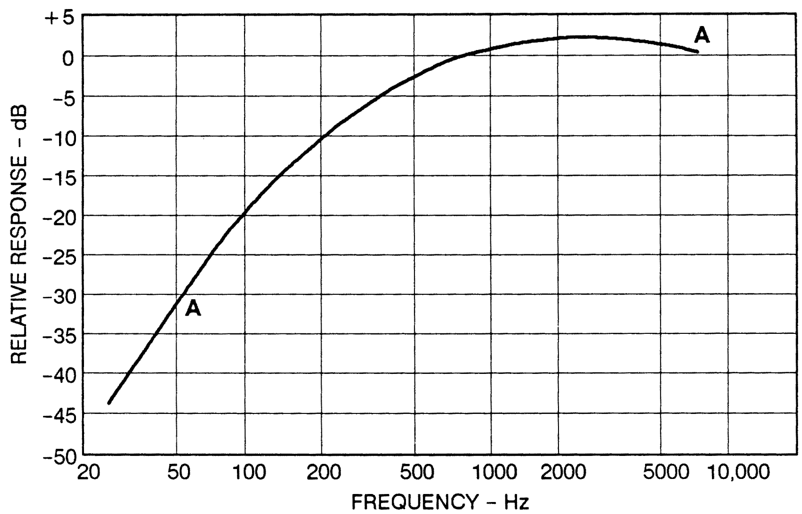


Figure 11. Response Characteristics of Standard A Filter

Sound Power

Sound pressure levels, in dB(A), are one measure of the noise of a source, but sound can also be expressed in terms of sound power level (PWL), also in dB(A). Sound power is the actual acoustic radiation (though not measurable), expressed in watts. This sound power remains constant, whereas sound *pressure* decays as the distance from the source increases. A good analogy would be light bulbs, which are classified in terms of watts (power) - not illumination level, which falls off with increasing distance.

Sound power levels can be calculated with the following formula:

$$PWL = 10 \log \frac{W}{W_o}$$

Where PWL = sound power level in dB or dB(A)
W = acoustic radiation of source in watts
W_o = reference radiation in watts (10^{-12})

As in the expression of pressure levels, the power level is a logarithmic expression of ratio. Applying the same 3 dB change in power level as was previously calculated for pressure level results in:

$$R_w = 10^{\Delta PWL/10}$$

Where R_W = power ratio
$= 10^{3/10}$
$= 2.0$

In this example, if the power level increases from 82 to 85 dB, the power increases by a factor of 2.0; if the power level decreases from 85 to 82 dB, there's a 50% decrease in power ($^1/_2$ or 50% of the original level).

Table 4 lists the power ratios for changes in PWL from +10 to -10 dB with the above example in bold:

Change In PWL	Power Ratio	Change In PWL	Power Ratio
1	1.26	-1	.79
2	1.59	-2	.63
3	**2**	**-3**	**.5**
4	2.51	-4	.40
5	3.16	-5	.32
6	4	-6	.25
7	5	-7	.2
8	6.31	-8	.16
9	7.94	-9	.13
10	10	-10	.1

Table 4. Power Ratios

Relationship Between Sound Pressure and Sound Power

The correlation between sound pressure and sound power can be seen by comparing the two ratio tables shown above. For an equal change in decibel level the ratios are different. For example:

Change In Db	Pressure Ratio	Power Ratio
3	1.41	2.0

The relationship is such that ***power is proportional to the pressure squared.***

Effect of Distance on Noise Levels

In an environment where noise is radiated from a source into a reflection-free space, called a free field, the sound pressure level will vary according to the following formula:

$$\Delta\, SPL = 20 \log \frac{d_1}{d_2}$$

OR

$$\Delta\, SPL = 10 \log \left(\frac{d_1}{d_2}\right)^2$$

Where d_1 = initial distance from sound source (noise standards specify either 3 feet or 1 meter)
d_2 = distance of observer (greater than d_1)

This forms the basis for the *inverse square law*. If the distance of observer is doubled, the noise level is decreased by 6 dB.

For example:

$$d_1 = 3 \text{ feet}$$
$$d_2 = 6 \text{ feet}$$

then

$$\Delta SPL = 20 \log .5 = -6 \text{ dB}$$

OR

$$\Delta SPL = 10 \log (.5)^2 = -6 \text{ dB}$$

Noise Addition

Noises can only be added or subtracted on the basis of acoustic **power** - *not pressure*. Noise sources of equal sound pressure levels are combined as follows:

$$SPL_2 = SPL_1 + 10 \log X$$

SPL_2 = noise level of all sources
SPL_1 = noise level of 1 source
X = number of sources

For example:

$$SPL_1 = 80$$
$$X = 3 \text{ sources}$$
$$SPL_2 = 80 + 10 \log 3 = 80 + 4.8 = 84.8 \text{ dB}$$

A chart showing this relationship appears in Figure 12.

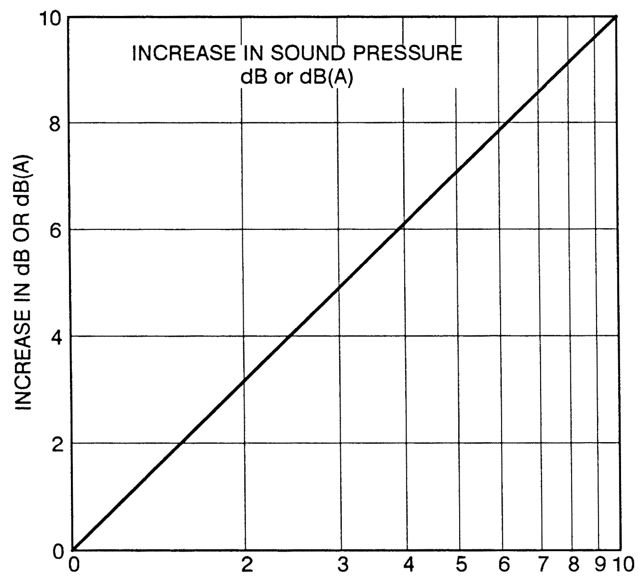


Figure 12. Addition of Equal Sounds

Unequal sound pressure levels can also be added. (See Figure 13.) The difference in the pressure levels of two sounds is used to determine how much their combined level will exceed the higher of the two. To add a third level, use the same process to combine it with the total from the first two levels.

The following example shows the two steps needed to find the total noise from sources of 70, 74, and 77 dB:

A) 74 - 70 = 4 dB (X Axis)
dB to add to larger = 1.5 (Y Axis)
TOTAL = 75.5 dB

B) 77 - 75.5 = 1.5 dB (X Axis)
dB to add to larger = 2.3 (Y Axis)
TOTAL = 77 + 2.3
= 79.3 dB

The total can also be calculated from the equation:

$$SPL = 10 \log \Sigma\, 10^{SPL/10} = 10 \log (10^{7.4} + 10^{7.0} + 10^{7.7}) = 79.3 \text{ dB}$$

NOISE MEASUREMENTS

Component Evaluation and Rating

To assist machine tool builders in their selection of components on the basis of noise, the National Fluid Power Association (NFPA) developed a standard that assures uniformity in the measurement and reporting of sound levels. This standard, T3.9.12, contains guidelines for obtaining standardized sound ratings. It is concerned only with the radiated noise of components, primarily pumps.

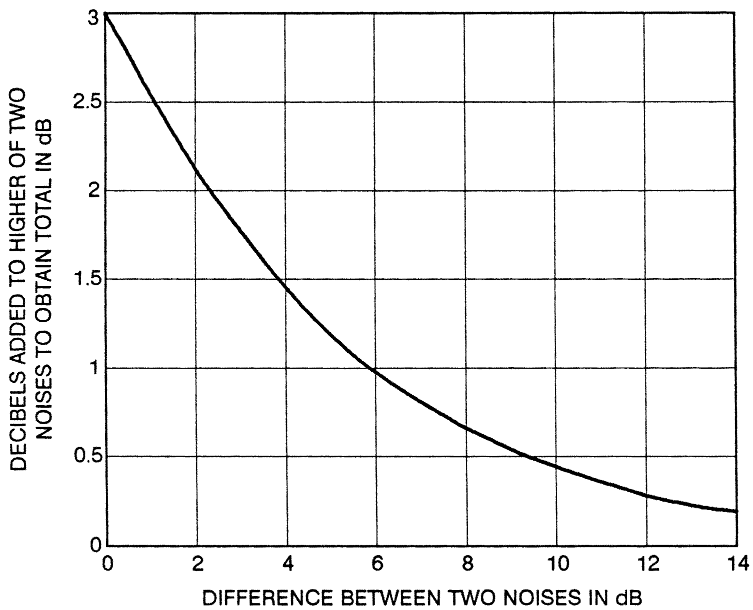


Figure 13. Addition of Unequal Sounds

The rating is usually expressed in dB(A) at a distance three feet in a free field above a reflecting plane (semi anechoic). This is a computed figure derived from a mathematical model. This model assumes that all the sound power from a pump is radiated from a single point located in the center of a hypothetical test hemisphere (Figure 14). The standard allows for masking all parts of the circuit that might contribute to noise. This includes wrapping hydraulic lines and enclosing any load valves. ***All radiated noise must be attributable to the pump, with no corrections for background.***

Microphones are positioned on spatial coordinates, each of which is located at the centroid of equal areas of the hemisphere surface. The rule of thumb is to use one microphone for each square meter of area. With the area of the hemisphere at $2\pi r^2$, and r equal to approximately 1 meter, six microphones are sufficient.

Power is defined as follows:

$$F = P \times A$$

where F = power
P = sound pressure
A = area acted on by P ($2\pi r^2$)

In terms of decibels:

$$PWL = \overline{SPL} + 10 \log 2\pi r^2$$

where PWL = sound power level
$\overline{SPL}$ = average sound pressure level of 6 microphones
r = radius in meters (3 feet = .914 meters)

Therefore:

$$PWL = \overline{SPL} + 10 \log 2\pi + 20 \log .914$$
$$= \overline{SPL} + 8 - .8$$
$$= \overline{SPL} + 7.2$$
(for 3 feet)

OR

$$PWL = \overline{SPL} + 8.0$$
(for 1 meter)

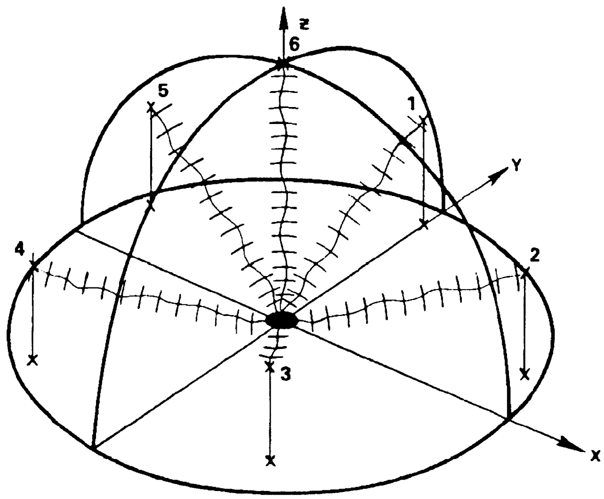


Figure 14. Microphone Positions for Measuring Pump Noise

System Noise Evaluation

The procedure for measuring system noise is different from the one used for components. Power unit systems are normally located in areas where background acoustics cannot be controlled. Guidelines for measurement in such environments are included in the National Machine Tool Builders Association's (NMTBA) "Noise Measurement Techniques" booklet.

Microphones are positioned ***1 meter from the perimeter of the machine and 1.5 meters above floor level***, as shown in Figure 15. It's extremely important that these distances be measured accurately. This insures uniformity of measurement and comparison, and compliance to customer noise level specifications. A 4-inch position error at a nominal 1 meter distance can result in a 1 dB error in measurement accuracy.

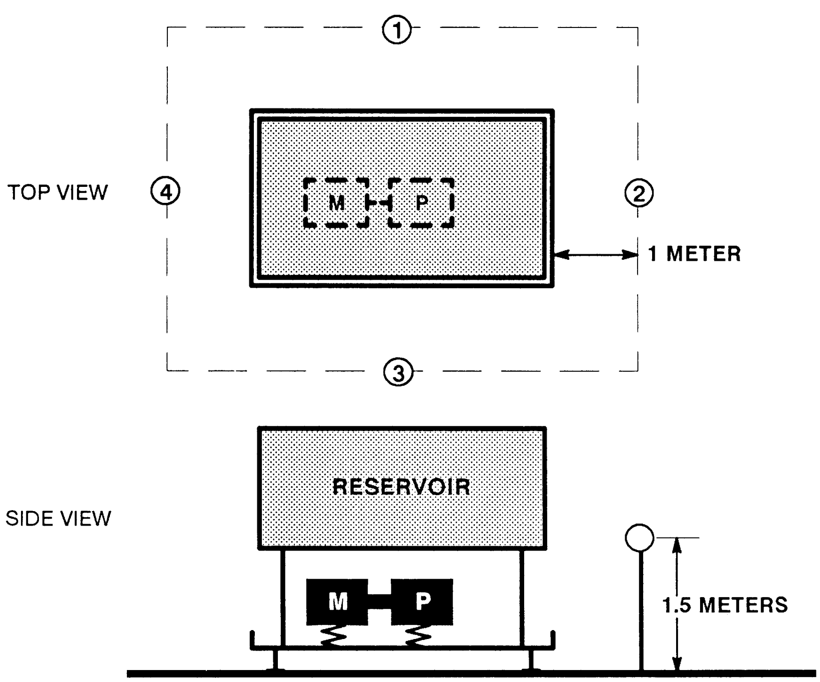


Figure 15. Microphone Positions for Measuring System Noise

At the very least, measurements should be taken on all four sides of the machine. It may also be necessary to measure at other locations around the envelope if highly directional noise levels are evident in other spots. Documentation of conditions is also needed to create some reference for comparison to other installations, acoustic environments, or pump types. The following conditions should be recorded:

- motor/pump speed
- pump type
- pump delivery
- operating pressure
- fluid type and temperature
- load valve location (if used)

Correction for Background Noise

Accurate system noise measurement may involve correcting for the noise of the surrounding area. When ambient sound levels are within 10 dB(A) of the levels when the machine is operating, correction factors may be applied. This is done in accordance with Table 5, derived from the NMTBA booklet.

Increase In Sound Level Due To Machine Operation (dB(A) above ambient)	**Correction Factor To Be Subtracted From Measured Sound Level** (dB(A))
3 or less	3
3 to 6	2
6 to 9	1
10 or more	0

Table 5. Background Correction Factors

One advantage of this chart is that it allows the use of whole numbers for dB(A). It's actually a "rounding off" of the curve shown in Figure 16 for the subtraction of sound levels. The example shown in the figure can also be expressed as:

$$\text{machine noise} = 10 \log [10^{6.0} - 10^{5.3}] \text{ dB} = 59 \text{ dB}$$

Documentation of noise measurements made using four microphone positions might look like that shown in Table 6.

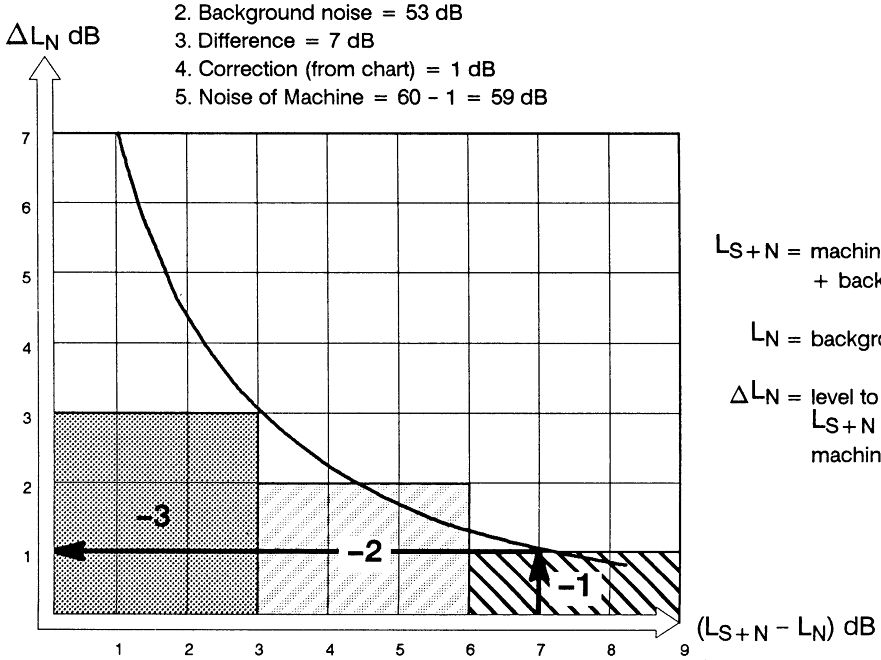
Noise level should be expressed as the maximum level measured (in this example, 78 dB in position 3) or possibly as the average of the four levels:

$$\overline{SPL} = 10 \log \left[\frac{10^{7.4} + 10^{7.0} + 10^{7.8} + 10^{7.5}}{4}\right] = 75 \text{ dB(A)}$$

Measuring Position	1	2	3	4
	dB(A)			
Total noise	76	73	79	75
Background noise	71	70	70	65
Background correction	-2	-3	-1	0
Machine noise	74	70	78	75

Table 6. Noise Measurement Documentation

Example (dark arrows):
1. Total noise = 60 dB
2. Background noise = 53 dB
3. Difference = 7 dB
4. Correction (from chart) = 1 dB
5. Noise of Machine = 60 - 1 = 59 dB



L_{S+N} = machine noise level + background noise level

L_N = background noise level

ΔL_N = level to be subtracted from L_{S+N} to obtain corrected machine noise level

Figure 16. Subtracting Sound Levels

Eaton
14615 Lone Oak Road
Eden Prairie, MN 55344
USA
Tel: 952 937-9800
Fax: 952 974-7722
www.hydraulics.eaton.com

Eaton
20 Rosamond Road
Footscray
Victoria 3011
Australia
Tel: (61) 3 9319 8222
Fax: (61) 3 9318 5714

Eaton
46 New Lane, Havant
Hampshire PO9 2NB
England
Tel: (44) 23 92 486 451
Fax: (44) 23 92 487 110

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