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NOISE CONTROL IN OIL HYDRAULIC SYSTEM

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Abstract: *This paper reviews the techniques for analyzing the Noise Control in Hydraulic System components. Hydraulic power transmission was quieter than the gears, cams, connecting rods, and cables that replaced. However, as economics led to higher pressures and speeds, it also led to less beefy components. Power levels of machines also grew rapidly. All of these factors increased noise. Hydraulic machines generally have a number of noise sources. This paper explains how Pumps, Motors and Valves generate airborne, structure-borne, and fluid-borne noises. It identifies hydraulic parameters that influence noise and guide planning programs for designing and developing of some quiet hydraulic components. Economics requires that all quieting efforts be focused on identifying and controlling the key noise generator.*

Keywords: *Cavitation, fluidborne, noise, pump, valve*

1. Introduction

Sources of noise in hydraulic systems are the pump and its driver, the distribution lines, control elements and actuators. Of these, the pump is normally the main source of noise, but this can usually be reduced to moderate levels by suitable acoustic treatment. The main aim at this end of the system should be to minimize pump-generated noise and vibration and any associated resonance. At the same time it is highly desirable to isolate the distribution line(s) so that vibration generated by the pump unit is not transmitted through the pipe work, with the possibility of resonance occurring at other connecting points. This does not, however, eliminates the possibility of pressure pulsations being transmitted by the fluid to the pipe work system and attached components, which phenomenon may need separate treatment.

Noise in fluid power systems is generally caused by pressure waves in the fluid stream. The pumps in general use are positive displacement types and employ pistons, vanes, or gear teeth to move the fluid. Since the fluid moves in sequential packets, pressure waves are set up in the fluid stream. These pulses of fluid are a cause of noise and system instability.

All pump and motor noise standards require that only the sound radiated from the test unit be measured. This is difficult and a whole technology has evolved for excluding unwanted noises from support brackets, bed plates, drive shafts, and hydraulic lines. Valves present a different problem. Their noise is due to a cavitation plume streaming downstream from the valve. For this reason the sound radiated by the first few feet of line as well as that of the valve must be included in the measurement.

Cavitation is an undesirable phenomenon in fluid power systems, which not only changes the fluid characteristics like density and compressibility, but also induces vibration, noise and erosion. In oil hydraulic systems, cavitation most frequently occurs in valves, pumps and actuators, inside which the pressure plays an important role in formation and development of cavitation. It is clear that when the local hydrostatic pressure drops below the fluid vapor pressure at the actual temperature, cavitation occurs.

Cavitation inception was investigated a lot in various cavitation models and by various ways, and several engineering methods were proposed. As in pump cases, detection of the net positive suction head (*NPSH*) and paint erosion were usually used as direct index for cavitation inception judgment. In some cases, the measurement of static pressures on the volute-casing wall was used as evidences for cavitation distributions. Sound and vibration were mostly used for cavitations' detection. Acoustical signals are used to judge the thresholds of cavitation inception. The interferences from environmental noise can be easily eliminated based on the spectrum method. The cavitation distribution states play an important role on the cavitation noise [1].

2. Types of noise

Vibrations are called structure borne noise and fluid pulsations are called fluid borne noise. Pump commonly generate as much as 1000 times more energy in the form of structureborne or fluidborne noise than they do in the form of airborne noise. These forms act on other machine elements and frequently end up generating more airborne noise than that coming directly from the pump. Because of this, it is important to control all three forms of noise. Controls for the three types noise are different [2].

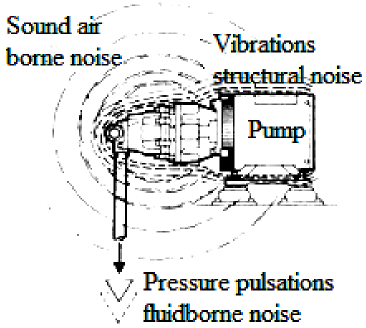

Fig. 1. Three forms of noise

2.1 Fluidborne Noise

Fluidborne noise is that it has as much as a thousand times the energy of pump airborne noise. It therefore poses the risk of exciting other machine components to produce high airborne noise levels. Even when the conversion is relatively inefficient, it can produce louder than the pump itself. Pulsations generate vibrations as well as sound. They are responsible for hydraulic line vibrations that cause metal lines to fail in fatigue and flexible lines to fail by chaffing. Often, these fluidborne noise-induced vibrations, in turn, also generate sound. Fluidborne noise is generally more important in mobile machinery than in industrial. There is a new and growing concern over fluidborne noise. Pulsations can interfere with the electronic controls that are finding wider use in hydraulic systems. Most of these depend on pressure oscillations generally require more sophisticated electronics for signal processing and may also limit the system’s sensitivity.

<u>Fluidborne Noise Mechanics</u>

It is periodic fluid flow perturbations or pulsations. When these encounter flow resistance, they result in pressure pulsations as well. For this reason it generally considers fluidborne noise to be pressure pulsations or waves. The major difference in airborne and fluidborne noise is the medium in which they exist. Because fluids have higher stiffness or bulk moduli, fluid waves travel faster than air waves. The sound velocity in a fluid is:

$$C=\sqrt{\frac{\beta}{\rho}}$$, where β = bulk modulus, ρ = mass density

It follows that the length of fluidborne waves, the distance between like points on successive waves, is
Wavelength, λ = C / f , where c = sonic velocity , f = frequency

It is often necessary to deal with fractions of a wavelength. In this we consider wavelength in terms of angles, with one wavelength equal to 2π rad. Angular fractions of a wavelength are found by multiplying distances by a coefficient equal to:

$$\beta = \frac{2\pi}{\lambda}, rad\ /m$$

This quantity is called the phase shift constant because the phase difference in pressures at two points is determined by multiplying it by the distance between the points.
It may be necessary to analyze the various possible sources of noise in the complete hydraulic system in detail in order to arrive at satisfactory noise treatment. In this case the possible sources of noise generation are, in decreasing order of significance:

(i) Pump Noise
(ii) Appliance noise
(iii) Control element noise
(iv) Water hammer
(v) Chatter
(vi) Cavitation
(vii) Resonance
(viii) Pipework noise
(ix) Thermal effects

Pump Noise and Vibration can be minimized by mounting by the pump and motor on a common base (or Mounting the motor integral with the pump) and isolating the complete unit on a resilient mount. A general recommendation is that the natural frequency of the isolated mount should not exceed on-quarter of the shaft speed (frequency), although it may be permissible to approach one-third of the shaft speed if a stiffer mount is required. If further acoustic treatment is required the whole pump/motor unit can be fitted with a suitable enclosure. The majority of hydraulic pumps are driven by electric motors, so no special problems are involved other than ensuring an adequate airflow for cooling the electric motor. If necessary a forced draught ventilating system can be used with a completely sealed enclosure, employing dust silencers of the absorptive type.

3. Measuring Pump Fluidborne Noise

The pump pressure pulsations are influenced by discharge line parameters. Because the discharge line has this much influence, there is a problem in finding a measurement that accurately scales a pump’s intrinsic fluidborne noise without being affected by the test circuit.

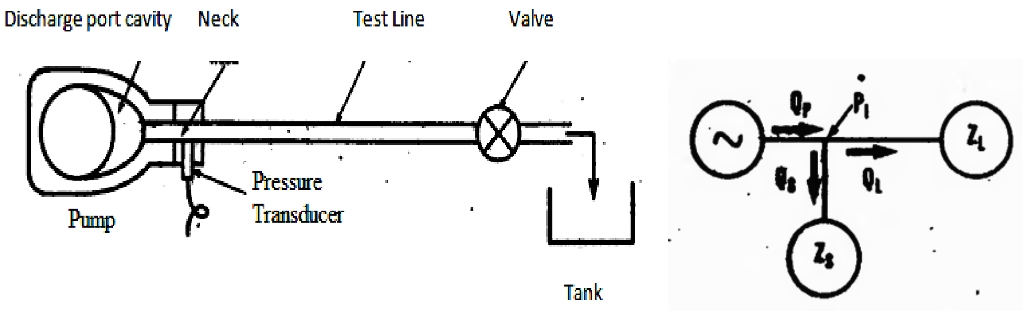

Fig. 2. Schematic diagram of pump circuit

A cavity in the pump represents the fluid volume in the discharge passages. This volume is the primary factor in determining the pump impedance.
The periodic flow generated by the pump, at each frequency, is divided between the pump and test circuit impedances inverse proportion to their magnitudes.
The flow then is:

$$Q_P = P_1\left[\frac{1}{Z_L} + \frac{1}{Z_S}\right]$$

Where P_1 = measured pressure at interface, Z_s = pump internal impedance , Z_L = test circuit impedance at interface

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Solving this for the pressure measured at the interface

$$P_1 = \frac{Q_P Z_S}{1 + {}^{Z_S}/_{Z_L}}$$

The numerator consists of two inherent pump parameters. Their product is called the blocked pressure because it is the pressure pulsation that would be generated if the pump only produced its noise flow and its outlet was blocked. It is an inherent property of the pump that is analogous to the open-circuit voltage of an electrical generator.
Although pump flow perturbations are the basic cause of fluidborne noise, it is generally agreed that the blocked pressure is the best measure of this noise. The principal reason is that dynamic pressures are easily measured, whereas dynamic flows are very difficult to measure.
The locked pressure is measured by the transducer at the pump interface if the test circuit impedance is much higher than the pump impedance.
The impedance of the test circuit is:

$$Z_L = Z_o \frac{Z_T \cos\beta 1 + jZ_o \sin\omega 1}{Z_o \cos\beta 1 + jZ_T \sin\beta 1}$$

Where Z_T = load valve impedance, Z_O = test line characteristic impedance
B = phase shift constant, l =test-line length

In most pumps, the compressibility of the fluid in the discharge passages and the pumping chambers in communication with the discharge port account for most of the pump impedance.

4. Hydraulic Components

4.1 Decoupling of pump vibrations

The flexible pump carrier is used to connect the hydraulic pump to the drive motor, whereby the transfer of component vibrations and oscillations is avoided to a large extent. The pump vibrations are isolated and damped by a temperature and fluid stable rubber ring which transfers all the forces. By using a rotary flexible coupling there is no metallic connection between the pump and motor. The noise level within a hydraulic system may be reduced considerably by this means.

Fig. 3. Pump carrier with damping of vibrations (Flexible pump carrier)

The possible reduction in the noise level depends on many factors (type of pump, operating pressure, type of pipes, construction, etc).Hence exact values cannot be provided. In general noise levels may be reduced by up to 6 dB (A). The damping materials used in the pump combinations.

Figure shows the measuring arrangement and typical noise reduction for an flexible pump carrier in comparison with a rigid pump carrier [3].

4.2 Valves

Noise due to the operation of valves, regulators and control elements is transient and related to the degree of turbulence or cavitation produced, although in specific designs and certain circumstances individual elements may be subject to vibration and generate a continuous noise. So much depends on the design and finish of the flow passages involved that no general analysis can be attempted. The noise level of such devices is dependent on the design and localized flow velocities produced and also on the response time, where applicable.

The latter effect can be minimized by arranging that the response time is shorter than that required by the system, This will result in minimum 'hammer', 'Water hammer', in fact depends on the switching velocity of the valve-i.e. on the spool-switching velocity in the case of spool valves. Valves operated by dry solenoids have in fact; uncontrolled response and so often produce 'hammer'. Wet solenoids are cushioned by the hydraulic fluid so move more smoothly and open the valve passages more gradually.

Many of the valves used to control hydraulic systems are constrained with steel coil springs. The valves are free to move within the valve housing and are biased on one or both ends with springs. The valve and spring effectively act as a spring-mass system. The valve may then oscillate at its own natural frequency and is also subject to the forcing function provided by the pressure waves in the fluid stream. Pressure waves in hydraulic systems can cause control valves to become unstable during operation and also contribute to vibration and noise. Therefore, it becomes desirable to filter out or at least reduce the magnitude of the pulses, in order to optimize the performance of fluid power systems and their controls. Reduction in pressure wave amplitude also reduces wear and damage to system parts. The fluid pump is usually the primary source of pressure pulsations. These waves travel throughout the fluid system. Therefore, it becomes advantageous to reduce the amplitude of the pressure waves as close to the source as possible.

Cavitation in valves

Cavitation is a breakdown inflow caused by the localized fluid falling below the vapour pressure of the fluid. Consequently, vapour bubbles are formed resulting in irregular and noisy flow. Such a reduction in pressure can occur in regions of localized high flow velocities, such as are caused by restrictions to the flow path. Accurate prediction of cavitation conditions is most difficult, and usually impossible, in the design of valves and fittings, and problems have to be tackled on empirical lines. If the flow rate is sufficiently restricted cavitation and noisy flow can be expected.

Thus partially closed tap or valve is nearly always noisier than when fully opened; also quiet a small change in position, and thus flow rate can cause a change from cavitating to non cavitating flow. It is also a characteristic of many valves, that for flow rates (valve openings) below that which produces cavitation, cavitation noise increases with increasing frequency: whilst for higher flow rates, where flow is non-cavitating, cavitation noise does not vary greatly with frequency.

Cavitation is by far the leading noise generating mechanisms in valves. It is quite common in pressure-regulating and pressure relief valves. When these valves cavitate for a long time, there may be more of an erosion problem than a noise problem. When the erosion occurs, it not only reduces valve life but pollutes the fluid within metal particles, which cause pump damage. Valves control fluid flow by constructing the flow path. This causes the fluid to speed up in passing through the construction. Since friction losses in turbulent flow are proportional to the square of velocity, the needed energy loss is then achieved in a relatively short distance.

In the case of high pressure systems, or valves subject to high pressure drops, it is desirable to utilize flow paths designed to eliminate cavitation as this can cause physical damage to the valve components as well as excessive noise. The problem, basically, is one of preventing the pressure in the valve throat from falling below the fluid vapour pressure in order to prevent cavitation occurring.

4.3 Cavitation Noise

Cavitation itself does not cause noise. It is the collapse of the cavities that causes noise. With valves this occurs when the jet dissipates into a more normal flow and pressure recovery takes place. Often this happens in the discharge line, outside the valve. Bubble collapse releases a surprising amount of energy. When it occurs at a solid surface, it is capable of causing surface fatigue failures, pitting, in all but the hardest materials. The energy also causes structural vibration that can end up as a loud noise. In addition, the reaction with the rest of the fluid results in high levels of fluidborne noise.

Cavitation generated fluidborne noise is confined to the downstream side of the valve orifice. While the impedance of the orifice is sufficiently different from that of the downstream passage, to reflect much of this noise downstream, it is believed that the cavitation bubbles do most of the reflecting. Cavitation noise is random, like the bubble collapse that causes it. Fluidborne noise exists at all frequencies but is strongest in the range 4 to 8 kHz. Because of its strong high frequencies, it is efficiently radiated as airborne noise. It is generally described as a hissing sound.

4.4 Cavitation Control

A good way to avoid cavitation is to design the throttling device so that it has laminar flow. The advantage of this type of flow is that It uses viscous friction to achieve fairly good energy loss with low velocities and it does not generate vortices. This type of flow occurs when the Reynolds number of the flow is below about 2000.
The Reynolds number is: N =2500vR /v
where v = fluid velocity, R= hydraulic radius, v =kinematic viscosity
The hydraulic radius, in this equation, depends on the shape of the flow cross section. It is defined as R = cross-sectional area / perimeter.
In case of passages that are round or whose cross-sectional dimensions remain proportional, their areas decrease as the square of their size while their allowable velocity changes only linearity. It would seem desirable to keep the passages as large as possible to keep from having to provide large numbers to handle a given flow. However, as velocities are reduced, the passage length must also be increased, to achieve a given pressure drop. It can be shown that pressure drop for laminar flow.

$\Delta P = k\frac{lV^3}{N_R}$, where k = a constant, l= path length

V = average flow velocity, N_R = Reynolds number
From this it can be seen that the real advantage is in reducing passage size and increasing velocity, because this rapidly reduces the length needed for a given pressure drop. Porous materials are sometimes used to achieve laminar flow. Both compacted stainless steel wool and sintered powdered metal have been used. These not only provide small pore sizes, they also offer paths with many turnings which are good for energy loss. However, they erode easily and shed debris if velocities are too high or if some cavitation occurs.
A difficulty with using this material is in making the valve adjustable. Figure 4 and 6, shows the general idea of an adjustment scheme. With this configuration, either the porous plug of the outer port sleeve can be moved back and forth to change the flow path length. A similar throttling mechanism utilizes flow paths etched in the faces of washers. Multiple paths are provided by stacking many washers together to form a porous sleeve.
Since cavitation is caused by low pressure, it seems reasonable to except that it could be suppressed by increasing a valve’s back pressure. Increasing back pressure increases sound power. It has been observed that increasing back pressure shifts the bubble collapse zone upstream. From this it is concluded that the sound increase is due to having more bubbles collapse near the solid valve surfaces. The maximum occurs when the collapse zone reaches the valve. The fact that this maximum noise occurs at back pressures that increase with flow appears to

support this theory. Noise reductions occurring when back pressure is increased above the maximum noise pressure are probably due to reduced cavitation.

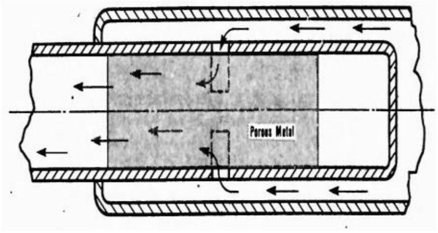

Fig. 4. Porous metal valve concept

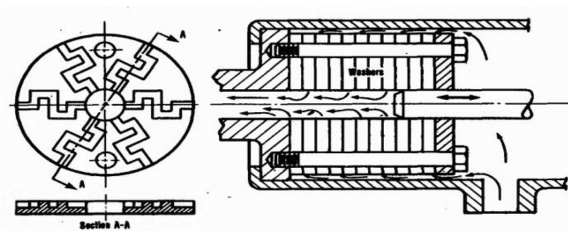

Fig. 5. Etched washer valve concept

Some noise reduction is achieved by using a series of pressure drops, with the downstream ones providing back pressure to suppress cavitation in their upstream counterparts. A pilot operated valve was built with two poppet valves in series with the control, dividing the pressure drop equally between the two stages. This valve produced less noise than that of a comparable valve of conventional design.
Reductions of sound levels have been made by recontouring the throttling device and valve passages. The objective of these modifications is to move cavitation collapse farther from solid surfaces and to smooth discharge flow to reduce vortex formation. It is believed that noise reductions achieved by contouring the spool valve shown in Fig is due to this factor. Throttling element modification can produce larger noise reductions. Test results shows that , there was a definite increase in noise as the poppet angle was increased. The quietest seat was the straight sided orifice shown. Convergent and divergent orifices had little effect on noise levels for the low-angle poppets. It was found that part of the reason for the high noise levels with the large angle poppets was due to the flow restriction that they caused. Reducing the poppet diameter to provide a more generous downstream passage reduced the noise level.

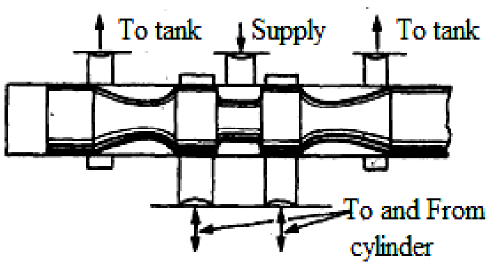

Fig. 6. DCV spool contoured for low noise

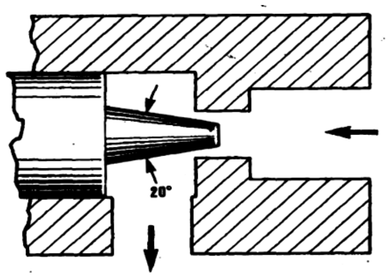

Fig. 7. Quietest poppet and seat assembly

4.5 Valve Oscillation

This noise is a single-frequency or pure tone sound, generally described as a squeal or whistle. Single-stage poppet valves are the most liable to be unstable. This can be due either ability to react more quickly or because they have less damping than that of spool or pilot operated valves.
The fact that valve instability produces a pure tone shows that a resonance is involved. Research has shown that valves are capable of generating self-sustaining oscillations. One source of these oscillations is a flow instability called hydraulic jet flip.
At lower pressure drops, jets tend to cling to a wall. At higher pressures it takes on a more independent form. There is a transient pressure range where it exists in either form and may flip back and forth from one to the other. Each time it flips, the flow is perturbed and the valve begins to oscillate. Squeal from such sources occurs at some operating conditions and not others.
Flow and acceleration forces that tend to open the valve further than called for provide another motivation for valve oscillation. The onset of vibration is related to the strength of these forces in comparison to other parameters, such as stiffness, damping, and how much the ports open for a

given valve movement. These forces increase with pressure drop through the valve, which explains why some valves provide good service at some pressures but begin to squeal at higher pressures.
The system in which the valve operates may have natural frequencies that are excited by transients and cause the valve to sustain the oscillations. This occurs with valves having just a little more than border line stability. Valves with even greater inherent stability succumb if their natural frequency nearly matches that of the circuit. These valves operates satisfactorily in some circuits and squeal in others. Sometimes, changing the line connected to a valve makes this difference.
Long lines increase the likelihood of an unfavorable match with a valve because these have a larger number of resonant frequencies in the range of valve natural frequencies. For that reason instabilities are sometimes avoided by using shorter lines. Valve inner chamber volume has an effect similar to adding line length, so reducing this volume is another option. Another way of discouraging this type of instability is to use lines composed of two different diameters so that reflections from the change in diameters interferes with the organ pipeline resonances.
Analyzing valves and their circuits to determine what factors must be changed to avoid instability requires considerable effort. Literature on this process universally prescribes replacing a squealing valve with one of a different design. Where the offender is a single-stage valve, it is best replaced with a pilot operated valve.
Pilot operated valves have instabilities, although these are relatively rare. When it occurs after operating without trouble for a long time, it is usually cured by replacing worn seats or poppets. One unstable poppet valve problem was cured by installing guides that prevented sideways poppet motion. Such motion will produce flow perturbations as well as axial motions. This suggests that some valve instabilities are due to lateral vibration, which does not appear to have been considered in past research [4].

4.6 Reservoir

Reservoirs should be designed to avoid the entrainment of air in the fluid, and the recommendations given in the following paragraphs are intended to assist in this. Return lines should enter and suction lines should leave the reservoir well below the surface of the fluid when at its lowest permissible level. Return lines should be fitted with a pepper pot and suction lines with a bell mouth entry or a suction strainer where they enter the reservoir. A baffle should be fitted between the suction and return lines. Variable capacity systems require special attention. The reservoir should be fitted with a bubble separator. This may be a single 60 mesh wire gauze set between the return and suction tank openings. Maximum effect is obtained when the gauze lies at an angle of 30^{o} to the horizontal.
Where a reservoir is pressurized with air, there should be a separator at the oil-air interface. The pressure drop across the entire suction line should not exceed 0.3 bar regardless of reservoir pressure, nor should suction pressure at the pump inlet be less than 0.2 bar. Suction line velocities should not exceed 1.5 m/s. Working and return line velocities should not exceed 4.5 m/s. Reservoirs should be flexibly mounted and isolated from surrounding structures.

4.7 Pipelines

A simple method of isolating or decoupling the pump from the delivery line is by a flexible hose connection. Isolation can be further improved by suing two such hose lengths in close proximity and mounted at 90^{o} to each other. Ideally, isolation by flexible pipe should include bends in two mutually perpendicular directions with equal distances between bends. In general, isolation of the pump and motor from the tank by suitable mountings and decoupling from the pipework will free the rest of the system from the transmission of mechanical vibrations and the consequent possibility that these would be amplified. Care must be taken to ensure that there is no short-circuiting of the isolation or decoupling employed.
Noise produced in hydraulic lines may be pump generated (changes in power and pressure, or varying amplitudes of pressure pulsations) or fluid generated (flow instability, turbulence or simple fluid friction). Fluid generated noise in small bore pipes with low to moderate flow rates is generally

negligible, unless pressure pulsations are present. Thus pipe vibration, and consequent radiation of airborne noise, is usually due to the higher level of noise generated by fittings; pipe resonance is due to mechanical vibration or resonant noise generated in supporting systems.

4.8 Suction Line

The suction line is a first suspect in a hydraulic installation which proves noisy, and where the noise cannot be directly attributed to pump or components. Suction lines can generate noise if there is an excessive pressure drop when the pump is sucking at sub-atmospheric pressure and drawing air out of solution. The resulting formation of air bubbles, and their subsequent collapse, can cause ‘mechanical’ noise which is often erroneously diagnosed as pump noise. Suction line noise can also be caused by a partially blocked or undersized suction filter, poor placement of the outlet pipe in the reservoir or entrained air.

4.9 Delivery Lines

Delivery lines can carry mechanical vibrations to distant parts of the circuit. These vibrations may be amplified at local points by the resonance of supporting structures or components directly connected to the pipework. Resonance can be eliminated by decoupling connections. All pipework installations should be designed on the basis of avoiding abrupt changes of section which could lead to large flow velocity changes and generation of turbulence.

5. Components to Reduce Noise

Hydraulic systems include components which agitate fluid and air and these effects influence each other. In order to reduce noise in hydraulic power units various measures are available. Due to the large surfaces and the thin metal walls used, oil tanks are very good resonators. By using materials which damp vibrations it is possible to decouple noise from the tank. Measures which help in this are: Placing pump on an anti vibration element , Installing a vibration damping pump carrier, Using pipe ducts made of rubber, Fixing lines with noise damping fixing clamps. Vibrations occur in fluids especially when pressure pulses are present. Measures which help in this are the use of accumulators which remove pressure pulses and creation of opposing pulses, which neutralize the pulses within the complete system. Decoupling of air vibrations is only possible by using an acoustic absorption cover over the hydraulic unit.
The vibration and noise damping characteristics of these clamps are the most important, as transfer of component vibrations to the complete system may be avoided in this way. Components for the decoupling of vibrations in fluids. Accumulators are mainly used to decouple vibrations in fluids. It is only possible to decouple the noise travelling in air in hydraulic systems by using noise absorbing covers.
Sufficient damping for pipes is usually provided by suitable supports, or pipe clips spaced at regular intervals, the supports having resilient linings so that vibration in the pipe is not transmitted directly to the surface to which the supports are fixed.

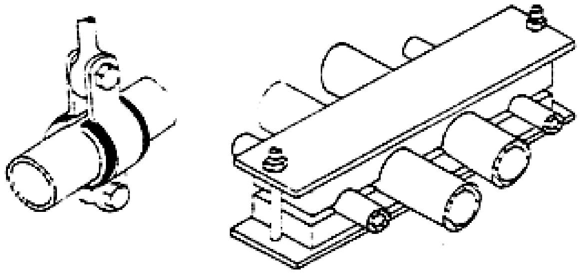

Fig. 8. Vibrating and isolating pipe clip and Pipe clamp block with vibrating isolating material

Shock Preventers
Shock preventers are pulsation dampers (or accumulators) characterized by having very large flow inlet apertures which are partially closed off by liquid trying to flow back out of them. They are

not shock absorbers, as they prevent shock or surge occurring. For the same reason , they do not attenuate shock.

Shock Removers
These are sensitive hydro devices which prevent a standing wave from passing farther down a system or from bouncing back through them. They are normally of tubular or sleeve form with a flexible membrane.

Acoustic Filters
Acoustic filters can be fitted to systems where pressure ripple is high. These are essentially tuned silencers which are critical in design and are usually effective over only very narrow frequency bands, although the attenuation achieved can be quite high. Untuned silencers simply comprise an expansion chamber with broader coverage but reduced attenuation. An accumulator is, in effect, an unturned hydraulic acoustic silencer and is most effective at lower frequencies.

Conclusion

The cheapest quiet machines are those that initially were planned, designed and developed to be quiet. Many noise controls can be incorporated into a new design for little cost but are expensive when added to an existing machine. It is important to set these goals for machines while they are still in their planning stage. Having a clear noise-level goal saves time in making other planning decisions. If it indicates the need for noise controls such as isolators or enclosures, other planning factors can be adjusted to accommodate these without losing time.
A machine can have almost any noise level, from a very high one to a very low one, depending on how much noise control is built into it. In planning a new machine, estimate its noise level for some known degree of noise control. Operating parameters such as size, speed, and pressure all affect noise, so selecting the optimum combination of these three parameters is the first step in finding a quiet pump. Since speed has the greatest influence. This is because, as speed increases, more of the strong, lower pumping harmonics move into the frequency range where they radiated efficiently.
Noise increases equally with both pressure and pump size, so there is no advantage of trading off one against the other. For quietness, the optimum operating parameters are the slowest practical speed and any combination of pressure and displacement that provides the needed hydraulic power. Unit selection has to be on the basis of airborne noise ratings. There is not enough data on structure and fluidborne noise for making comparisons. It is generally assumed that steps that reduced audible noise of a pump also reduced the other two noises as well. This is a reasonable assumption when comparing pumps and motors of the same type but may not be valid when comparing different types.
The leading noise generator provides a basis for estimating a proposals machine's noise level. Usually, it is a pump, but it could just as well be a hydraulic motor or, in rare cases, a valve. The noise source's sound pressure rating is useful because it approximates numerically the noise level that the unit will generate in a machine that utilizes normal machine design practices and does not have any special noise control features. Since pumps, motors and valves are the leading noise sources, it follows that selecting the quietest ones is the first step in producing a quiet machine. To generalize, noise levels of industrial and mobile machines should not exceed 85 dB.

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