

VIBRATIONS

Vibrations from machinery and other equipment can have serious consequences. Knowledge of the phenomena involved, together with design rules for pump installations, are necessary to reduce these unwanted effects. Although vibrations cannot be entirely eliminated, pump installations can be designed to limit their effects.

Introduction

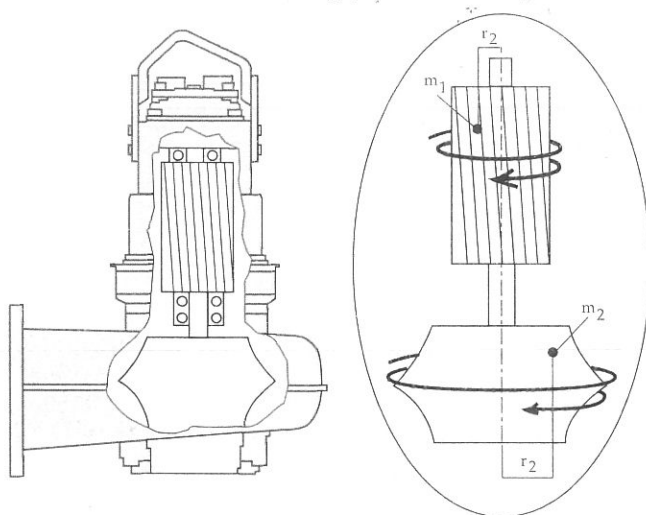
Whenever machinery with rotating parts are used, vibrations occur. The designer's goal is to keep these vibrations within acceptable levels, particularly at points where they can affect the performance of the equipment. Consequences to be avoided include fatigue failures, noise and wear.

Disturbances

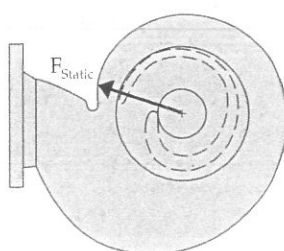
The main disturbances that are caused by submersible pumps are listed below, together with their frequencies. They are listed in the order of their frequency, with the lowest frequency first. The second number in the ratio 1:N is the multiple of the rotational speed.

- I. 1:1. Corresponds to the rotating speed of the pump, i.e. the rpm frequency. Normally it is the dominant one, and it comes from A, B and C.

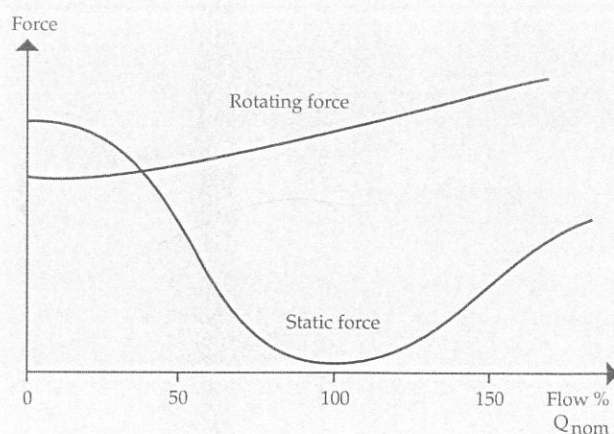
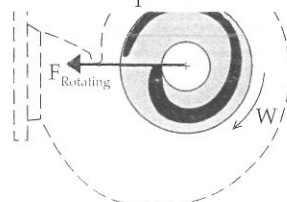
- A. Imbalance in the rotating parts.



- B. Hydraulic forces caused by the volute.



- C. Hydraulic forces caused by a single-vane impeller.



The magnitude of the forces is dependent on the duty point

- II. 1: number of vanes. The rpm frequency multiplied by the number of impeller vanes.

- A. Hydraulic forces that occur when a vane passes through zones of uneven pressure in the volute.

- III. 1: number of vanes multiplied by the number of guide vanes.

- A. Pressure impulses caused as the impeller vanes pass guide vanes.

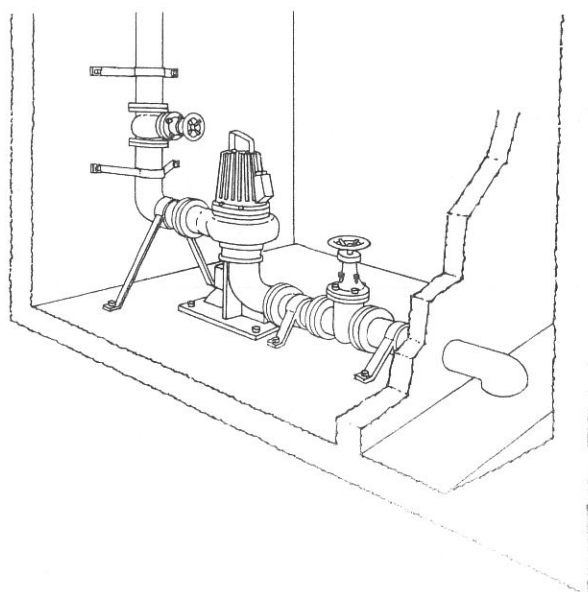
- IV. 1: higher numbers.

- A. Higher frequencies normally come from the electrical motor; they may cause noise, but are normally harmless in their impact on the structure involved.

Installation

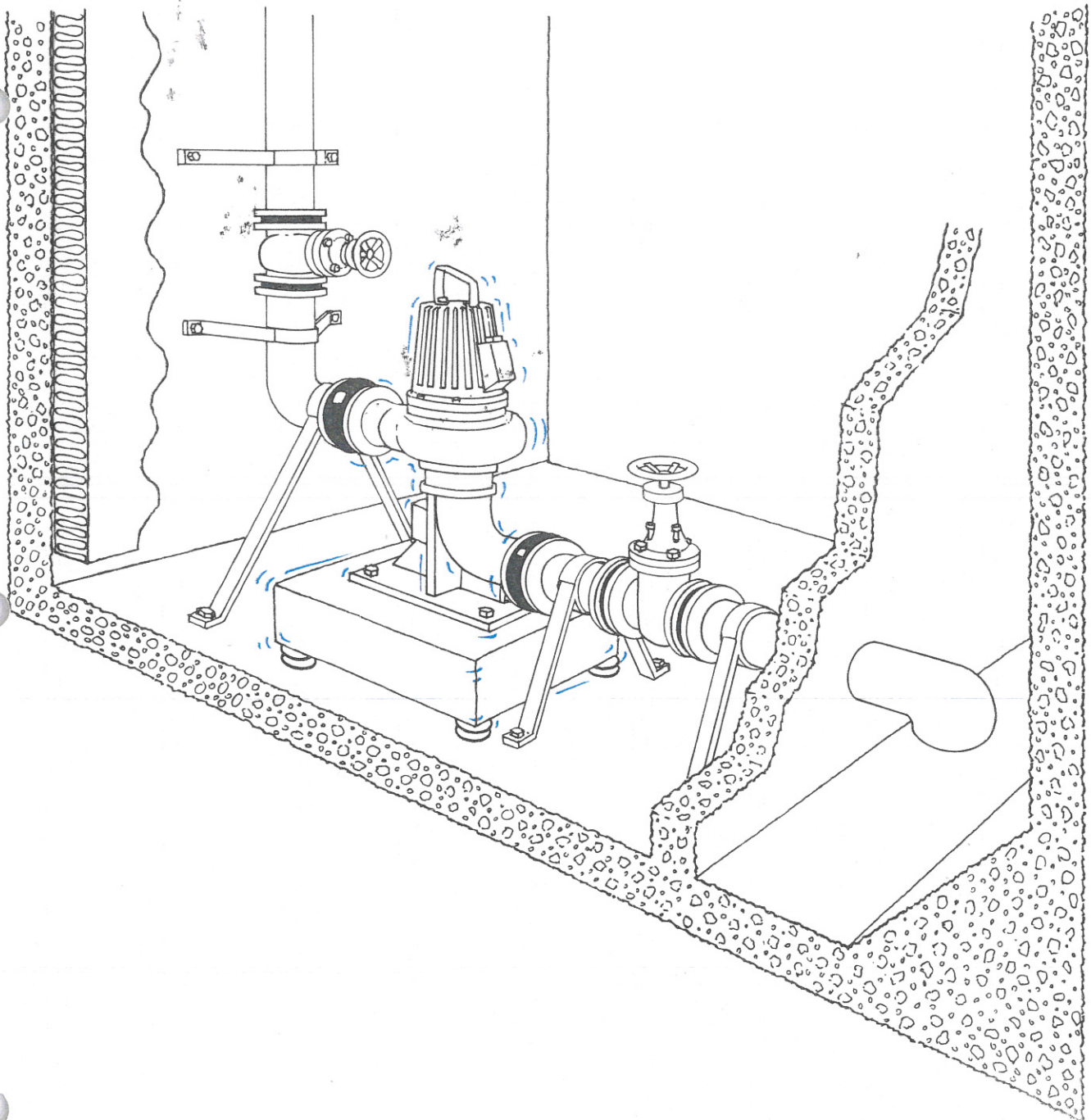
If a number of general rules are followed, the vibrations can be kept to acceptable levels.

- I. All parts of a system should be sufficiently stiff and firmly anchored so that the primary disturbances have frequencies below the lowest natural frequency of the system. The entire system can be involved, including piping, valves, pump, support, etc.

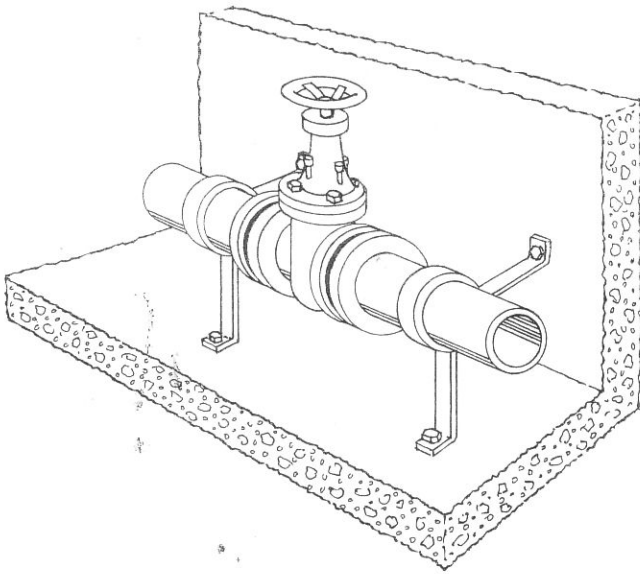


II. If the pump is to be insulated from the structure, the normal procedures are as follows:

- A. Include a base of adequate weight, at least two times heavier than the rotating parts.
- B. Anchor the pump firmly to the base.
- C. Use a flexible support, e.g. machine feet or a rubber carpet, between the base and the floor or ground.
- D. Use flexible joints for the pipes.
- E. Anchor the pipes to the floor or to another solid structure.



- III. Because vibrations are independent of gravity, horizontal supports should be provided since they are as essential as vertical supports.



- IV. Pipes should have a support at a distance of $1/3$ of the critical pipe length from the pump (this is the length at which the first natural frequency occurs). See appendix.
- V. Pipe length should be less than 70% of critical pipe length.
- VI. Heavy parts of the piping, like valves, must be properly supported.
- VII. The piping must be anchored if bellows are used. Bellows between the pump and the pipe can transform pressure fluctuations into disturbances, causing severe vibrations in the piping.

Example:

A pump station is equipped with a pump that has a three channel impeller and that is running at 1475 rpm. This implies that the main disturbances will act at a frequency of:

$\omega_1 = 1475 \cdot \pi / 30 = 154.5 \text{ rad/s}$ (imbalance in the rotating parts)

$\omega_2 = 3 \cdot 1475 \cdot \pi / 30 = 463.4 \text{ rad/s}$ (hydraulic forces that occur when a vane passes through zones of uneven pressure in the volute).

The pump is connected to a discharge pipe in steel with :

- | | |
|----------------------------|--|
| - a Young's module of | $E = 1.96 \cdot 10^{11} \text{ N/m}^2$ |
| - a density of | $\rho = 7900 \text{ kg/m}^3$ |
| - an inner diameter of | $D_i = 150 \text{ mm}$ |
| - and an outer diameter of | $D_o = 154 \text{ mm}$ |

The critical pipe length for the pipe filled with water ($\rho = 1000 \text{ kg/m}^3$) and the k factor is set to 2.25 (first mode of order and fixed in both ends see equation A9 in appendix) can be calculated according to the equation A10 in the appendix.

The critical pipe lengths are thus:

4.6 m for the disturbances generated by imbalance (154.5 rad/s) and

2.6 m for the disturbances generated by the three channel impeller (463.4 rad/s).

In order to avoid problems with vibration and noise, the intervals of 2.6 and 4.6 m between the pipes should be avoided. The pipe should be supported every 1.8 m, i.e. 70% of 2.6 m).

Variable Frequency Drive

The use of a variable frequency drive can excite a broader range of frequencies. If problems with high levels of vibration occur, one should scan the frequency range involved to see if any peaks occur. If so, the installation has natural frequencies that are being excited and one should alter the installation or avoid certain frequency ranges.

Conclusions

Unwanted vibrations in installations of machinery like pumps can be avoided or reduced to acceptable levels if certain practices are followed. Any improvements must be based on an understanding of the phenomena involved. Proper anchoring at correct positions is vital. With flexible installations the mass ratio, total to rotating mass, must be high enough to ensure low vibration levels. If a variable frequency drive is used, it might be necessary to avoid certain frequency ranges.

NOISE FROM PUMPING INSTALLATIONS

Noise in pumping installations is affected by the hydraulic, mechanical and acoustical properties of the pump and the installation surrounding it. The pump is clearly not the sole source of noise, the connected components and structure also radiate noise. Proper engineering of the installation, with help from some basic design rules, is required to keep the noise to acceptable levels.

Introduction

The noise emitted at a pumping installation arises from vibrations of the pump structure and the various components connected to it. The noise emission depends on a variety of factors: the motor and impeller that are used, duty point of the pump, pumped media and of course the mechanical and acoustical properties of the whole system. The first steps towards controlling the noise level are to identify its sources and to understand the mechanism of noise generation. The most effective place to control noise is at its point of origin. If this approach is not feasible, the noise from the machine can be redirected or absorbed. However, some noise is inherent in any mechanical process.

The dB scale

Sound is the size or amplitude of the pressure fluctuation. A healthy human ear can detect amplitude ranges from 20 millionth of a Pascal (Pa) up to 200 Pa. Thus if we measured sound in Pa, we would end up with quite large numbers. To avoid this the decibel scale is used.

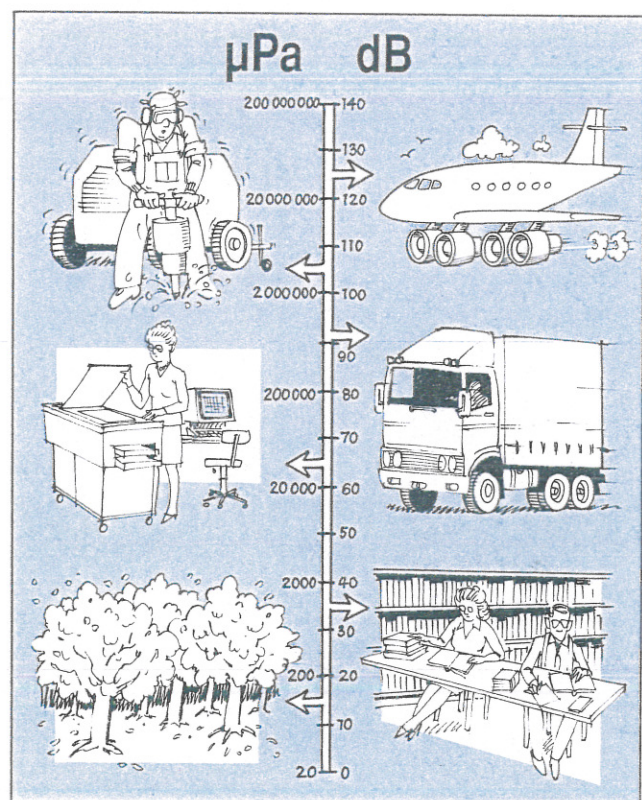
The decibel is not an absolute unit of measurement. It is a ratio between a measured quantity and an agreed reference level. The dB scale is logarithmic and uses the hearing threshold of 20 μ Pa as the reference level. This is defined as 0 dB. When we multiply the sound pressure in Pa by 10, we add 20 dB to the dB level. So 200 μ Pa corresponds to 20 dB (re 20 μ Pa), 2 000 μ Pa to 40 dB and so on. Thus, the dB scale compresses a range of a million into a range of only 120 dB. The sound pressure levels (SPL) in dB and Pa of various familiar sounds are shown in the figure.

Sources of noise

The many sources that can create noise in a pump and station are discussed in the following paragraph. Low frequency noise is here referred to as noise with frequencies below the impeller blade passing frequency. High frequency noise is noise with frequencies above that frequency.

Sources of noise in a pump.

- The passage of the impeller blade by the volute tongue creates not only alternating forces to act on the impeller and volute but also pressure pulses in the water. The excitation occurs at a frequency that is the product of the pump rotation frequency and the number of vanes; if guide vanes are used the number of guide vanes should also be included. This excitation may be strong in amplitude so that it causes vibration and noise. The excitation is especially strong for single channel impellers. The frequency of the noise is low, like a rumble and it is often emitted from the structure connected to the pump.
- Cavitation noise is due to bubbles of water vapour imploding sometimes on the impeller and volute structure and more often somewhere within the liquid. The frequency of cavitation noise is high and it covers a broadband. It can vary in time and produce a sound that pulsates.
- The electric motor causes noise because the electromagnetic fields of the rotor and stator produce alternating forces on the stator. The stator vibrates and radiates noise. The noise is high-pitched in character and produces a whining sound. Variable frequency drive (VFD) of the motor can increase the level of noise.



Some sound pressure levels (SPL) in dB and Pa of various familiar sounds.

Noise due to wear and flaws

- Imbalance of the rotating parts of the pump causes the pump to vibrate and radiate noise. The frequency of the noise is low and like a rumble.
- Clogging of the impeller disturbs the flow and can create a kind of imbalance that produces vibration and noise.
- Recirculation at the wear rings can produce a noise that is high pitched and can vary with time.
- Bearings makes noise whenever they lack proper lubrication or have become worn. The frequency of the noise is high.

Noise due to the installation

- Pre-rotation of the fluid entering the volute can affect the pumping action and cause both vibration and noise. Its frequency is low, again like a rumble.
- Turbulence at valves can cause the valves to radiate noise. The noise is high-pitched.
- The formation of a vortex provides a channel that leads hydraulic noise in the volute up to the surface, noise that would otherwise be muffled by the water.
- The interaction of liquid and piping causes the pipes to vibrate and excite the surrounding air. The more turbulent the flow, the greater the excitation and the louder the noise.
- Hydraulic resonance can occur if pump induced pressure fluctuations are reflected by the piping system so that they add in phase with outgoing pulses. The levels of vibration and noise can become quite high.

Sound is mechanically transmitted to components connected to the pump. These components may be better emitters of noise than the pump itself. Reduction of the mechanically transmitted noise can be achieved by employing vibration isolators described above.

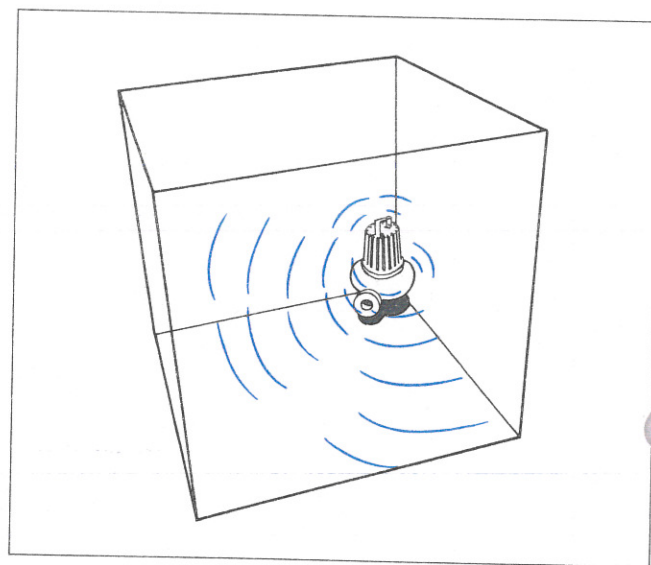
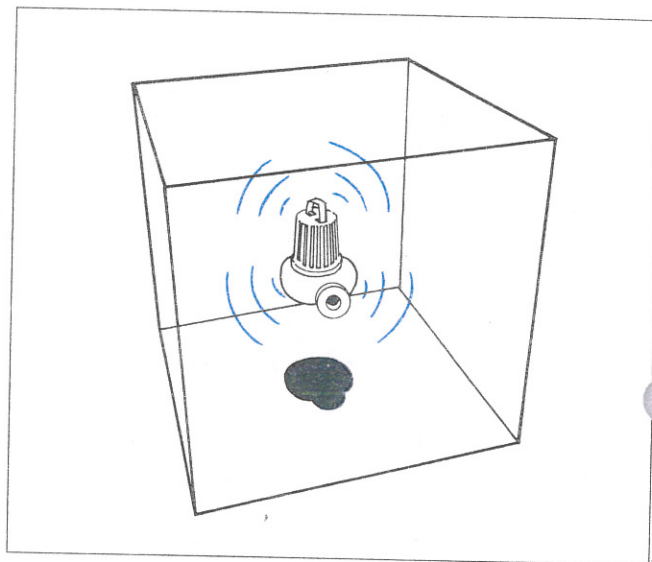
Acoustical properties of pump installations

Sound may be airborne, structure borne, or liquid borne. The noise from the pump may propagate over some distance with little attenuation before it becomes airborne and audible. For a permanently wet-installed pump most of the direct sound from the pump and adjacent piping network is radiated into the surrounding water. Because the acoustical transmission between water and air is poor, most of the sound energy stays in the water. In a permanently dry installation, the pump radiates sound directly into the air. However, mechanically transmitted sound still occurs in both types of installations.

Most pumping stations are made of concrete, a substance that absorbs little sound energy. Acoustic

dampings therefore low at a typical pump station and a substantial amount of reverberation takes place. The reverberation adds significantly to the sound pressure level in the station.

Predicting sound levels in a dry pump installation



The direct sound of a "free hang source" is eight times lower compared to one installed in a corner

Sound pressure levels in a station that are caused by a pump can be approximated from sound power data for the pump and the acoustical properties of the station.

The sound pressure level at a point in the enclosure relative to the pump can be approximated for an arbitrary enclosure, from the simple formula,

$$L_P = L_W + 10 \log \left[\frac{D}{4\pi r^2} + \frac{4}{R} \right] \quad (6)$$

in which L_W is the sound power level of the machine or pump; r is the distance between pump and measurement point, and R is an acoustic constant of the room that depends on the volume and acoustic absorption in the room. D is a directivity factor that depends on the position of the pump in the room, (e.g. a pump in a corner position has a directivity factor of 8, in the junction between two walls 4, above the floor 2); This equation is derived using statistical energy methods and is valid as an approximation for high frequencies.

The sound pressure level at a point due to several incoherent sources, as for n pumps, can be calculated from their individual contributions at that point from

$$L_p = 10 \log \left(\sum_{i=1}^n 10^{L_{pi}/10} \right) \quad (7)$$

Thus addition of one extra pump, with the same contribution in sound pressure level causes a 3 dB increase ($10 \log 2$).

The sound absorption of an enclosure can be estimated from measurements of the time required for the sound pressure level to decay 60 dB after a sound source is turned off.

T_{60} is the reverberation time of the room and is frequently

$$T_{60} = \frac{0.161 V}{S \bar{\alpha}} \quad (8)$$

$$R = \frac{S \bar{\alpha}}{1 - \bar{\alpha}} \quad (9)$$

ency dependent; V is the volume of the room and S is its surface area; $\bar{\alpha}$ is the averaged acoustic absorption coefficient of the walls in the room. The reverberation time can be measured or estimated from the absorptivity of the different materials in the station.

Example. The sound pressure level is measured at a distance of 2 m from a pump situated in a corner position in a concrete walled pump station that is 5 m by 6 m in plan and 2 m high. The acoustic absorption coefficient is 0.05 for concrete at 250–500 Hz and 0.4 for a 25 mm thick rockwool blanket. From equation 8, one can show that adding a 25 mm rockwool blanket to the ceiling would result in a decrease in sound pressure level of more than 4 dB for frequencies in the range 250–500 Hz. Further away from the pump, the decrease would be greater. Equation 8 indicates that the decrease is independent of the power output of the sound source.

Near the pump, typically within one meter, the direct sound field is dominant over the reverberant field, and the addition of acoustic absorption has little effect. The near field and far field meet at the

point where the contributions from the direct sound field equal the contributions from the reverberant field. In a highly reverberant environment, like a pump station, the direct sound field distance is small, consequently, the addition of absorption has a positive effect at most locations.

Installation

Some general rules should be followed if a pump installation is to be made as quiet as possible.

- The duty point of the pump should always be as close as possible to the design point of the pump. Requirements of net positive suction head must be met.
- The pump sump and piping network should be well designed from hydraulic point of view.
- The structural dynamics of the installation of the pump must be adequate. One should follow the guidelines for installation in the section about vibration above.
- Mechanically transmitted sound from the pump can be reduced by isolating the pump base from the ground using flexible supports and the pipes from the pump using bellows.
- Liquid borne sound transmitted from the pump along the piping system can be reduced by employing pressure pulsation dampers.
- Fluid-structure induced vibrations of the pipes can be reduced by avoiding unnecessary sharp bends of pipes near the pump. Noise radiation from the pipes can be decreased by the addition of acoustic damping and absorptive material to the pipes.
- Some of the noise from the valves can be avoided by placing the valves far away from bends to avoid excessive turbulence and uneven pressures.
- The amplification of noise in a pump station can be reduced by increasing its acoustic damping.

Conclusions

Noise and vibration are good indicators of the functioning and condition of a pump and its installation. Good design, for the hydraulic and mechanical elements of pump and installation is required to keep the noise to an acceptable level. The acoustical properties determine the radiation and amplification of the noise.

APPENDIX

Vibration Theory

The level of vibration at a given frequency ω can be measured in terms of its displacement or the corresponding velocity or acceleration. For any harmonic disturbance, the velocity is the product of displacement and ω . Also, the acceleration is the product of velocity and ω . The level of vibration is a function of the disturbance and of the stiffness and mass of the structure.

Disturbances due to imbalance of rotating parts or to other periodic forces can be expressed as a series of harmonic forces of different frequencies ω in the form:

$$F_i = \hat{F} \sin(\omega_i \cdot t + \Phi_i) \quad (A1)$$

The natural frequencies of vibrating systems are determined from their stiffness and mass. High stiffness leads to high natural frequencies, and high mass leads to low natural frequencies. For a simple spring with a mass m and stiffness K , the natural frequency ω_e is $\sqrt{K/m}$.

The stiffness, or the relationship between deformation and force, gives the "static" deformation. In order to obtain the dynamic deformation, a response function is needed. The response function of a system depends on the natural frequencies of the structure together with the damping factor η in the following way:

$$\Phi_{i,n} = \frac{1}{\sqrt{\left(1 - \frac{\omega_i^2}{\omega_{e,n}^2}\right)^2 + \eta^2}} \quad (A2)$$

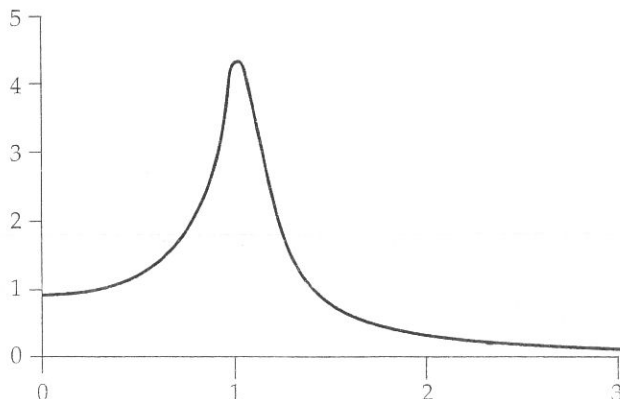


Figure 1 The response function Φ versus ω_i/ω_e (from equation (A2)) for $\omega_e=1$ and $\eta=10\%$.

This relationship shows how much the deformation is amplified for different frequencies. At low frequencies, lower than the lowest natural frequency, the response function has a value close to 1; at high frequencies its value is less than 1. The damping factor, as used in this expression, normally has values in the range of 1% to 5% for steel and concrete structures. This type of damping is called material damping. Another form of damping is viscous damping, which is caused by the resistance of a moving body in a fluid. The reaction force is a factor c times the velocity. The following relationship is used to connect the two forms of damping:

$$\eta = 2 \cdot \zeta \cdot \frac{\omega}{\omega_e} \quad (A3)$$

$$2 \cdot \zeta = c / \sqrt{K \cdot m} \quad (A4)$$

The dynamic deformation in its simplest form can be expressed as:

$$u = \frac{F}{K} \cdot \Phi \quad (A5)$$

All structures have several natural frequencies and mode shapes. Hence, the overall pattern of a vibration can be quite complex. The form of the mode corresponding to one natural frequency is denoted by X , with an index to represent different positions. X is normalised by dividing by the maximum translation, that is the maximum value of all X_i is 1. The stiffness values K differ depending on where the force acts and where the deformation is measured. If K_i is the value where F_i acts and the deformation is measured at the same spot, the following formula can be used to determine the deformation at position j due to the force i :

$$u_j = \sum_n \frac{F_i}{K_i} \cdot X_{i,n} \cdot X_{j,n} \cdot \Phi_{i,n} \quad (A6)$$

In order to change a specific natural frequency, one must first know the mode shape for that frequency. One reason is that the addition of mass or stiffness at a position with small movements (a node) is not effective, as shown by equation A6.

Impact loads contain all frequencies, and they can therefore cause responses that include all the natural frequencies. Repeated impacts may therefore cause vibrations at several frequencies. Furthermore, damping and non-linearities will always transfer energy to frequencies that are not directly excited. Consequently, responses will occur at other frequencies.

Flexible installation

A common way to reduce the vibrations of machines is to use flexible supports with high damping ($\eta \approx 0.2$ – 2) between the machine and its foundation. Then the force transmitted to the surrounding structure is low. The force can be expressed as a rotating imbalance of mass m_1 times the eccentricity e . For a system with the total mass m and the stiffness of the support K , the natural frequency is $\sqrt{K/m}$

If this is substituted into equation A5 one gets:

$$u = e \cdot \sin(\omega t) \cdot \left(\frac{m_1}{m} \right) \cdot \frac{\omega^2}{\omega_e^2} \cdot \Phi \quad (\text{A7})$$

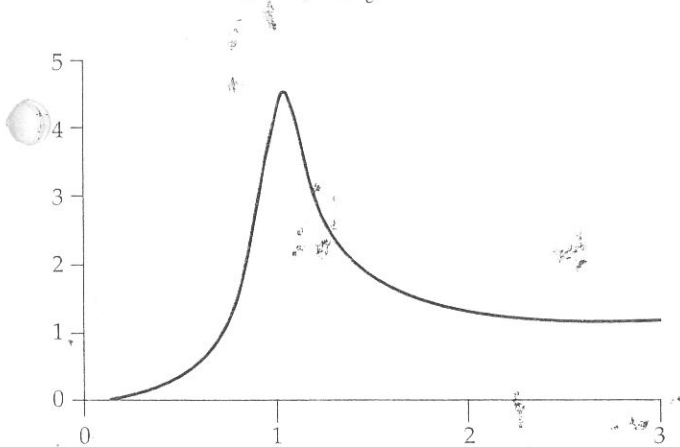


Figure 2. Deflection u versus ω (from equation A7 for $e=m_1=\omega_e=1$ and $\eta=20\%$)

The difference between figure 1 and 2 is significant, although they look similar. If the supports that are used have a relatively low natural frequency, then the last part of equation A7, the frequency ratio and response function, approaches 1 and the displacement depends only on the mass ratio and the eccentricity. Hence, the higher the mass is, the lower the displacement, and furthermore, the lower the natural frequency. This flexible installation can, however, lead to other difficulties. One is that other degrees of freedom may be introduced into the structure because the supports are flexible. Another is that all other connections to the machine also have to be flexible. The third is that during start-up and stop the natural frequency will be reached, with maybe large deflections as a result, and then passed.

The transmissibility, the ratio between the force transmitted to the ground and the disturbance force, is also important if other machinery has to be protected against vibrations. This ratio follows the response function given in equation A2. Consequently, the displacement of the working machinery may be high if the support is flexible, but the surrounding would be better protected. Moreover, a machine can also be

protected against vibrations that come from its surroundings if their frequencies are sufficiently high.

Calculation of natural frequencies

Analyses of vibrating systems must be based on knowledge of their natural frequencies. This type of calculation is often the only one performed. Several formulas are available for different types of simple situations. For more complicated ones, finite element analyses may be required. The most common components that can have natural frequencies of importance in installations with pumps are beams and pipes. The natural bending frequencies of a thin beam can be expressed as:

$$\omega = \frac{k \cdot \pi^2}{L^2} \cdot \sqrt{\frac{E \cdot I}{\rho \cdot A}} \quad (\text{A8})$$

$$k = (n + x)^2 \quad (\text{A9})$$

ω = natural frequency [rad/s]

E = Young's modulus [N/m²]

I = moment of inertia [m⁴]

L = length [m]

ρ = density [kg/m³]

A = section area [m²]

n = mode or order, 1 for 1st mode

x = 0 if simply supported ends,

0.25 if one end fixed,

0.5 if both ends fixed,

-0.5 if cantilever beam

For a circular conduct containing a mass without stiffness that follows the movements of the conduct, the appropriate form equivalent to equation A8 is:

$$\omega = \frac{k \cdot \pi^2}{4 \cdot L^2} \cdot \sqrt{\frac{E \cdot (Dy^4 - Di^4)}{\rho \cdot [Dy^2 - Di^2 \cdot \left(1 - \frac{\rho_m}{\rho}\right)]}} \quad (\text{A10})$$

Dy = outer diameter [m]

Di = inner diameter [m]

ρ_m = density of added mass [kg/m³]

The natural bending frequency of a pipe filled with fluid (added mass) can be estimated from equation A10. It can be used even if the pipe has several supports. In this case, with L being the distance between supports, the results are slightly conservative.

Measurement of noise emission

The sound pressure level at any point in a sound field is the time average of the acoustic pressure perturbations p_{dist} , and it is defined as:

$$L_p = 10 \log \left(\frac{\bar{p}_{dist}^2}{p_{ref}^2} \right) \quad (A11)$$

The reference sound pressure is $p_{ref} = 20 \mu\text{Pa}$. The human ear is more sensitive to high frequency sound than low frequency sound. The sound pressure level is therefore weighted at different frequencies to match the dynamics of the ear. The most common weighting is the A-weighting, indicated with an A, e.g. dB(A). The noise emission of a machine is stated as the sound power level of the machine. This level is defined as:

$$L_W = 10 \log \left(\frac{W}{W_{ref}} \right) \quad (A12)$$

Here W is the sound power output of the source and the reference sound power W_{ref} is 10-11 W. The sound power level of the Flygt pumps is measured according to the ISO 9614-2 standard using sound intensity scanning. The pump is installed on a foot and fixed with a sliding bracket. The water level is kept at the minimum allowable level so as to assure the maximum amount of airborne acoustical emission. The corresponding sound pressure level at a distance of 1 m from the pump for free-field conditions is obtained from straightforward calculations, in accordance with ISO 11203.