Location of Noise Sources in Fluid Power Machines

Article *in* International journal of occupational safety and ergonomics: JOSE · February 2007

Impact Factor: 0.35 · DOI: 10.1080/10803548.2007.11105102 · Source: PubMed

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International Journal of Occupational Safety and Ergonomics (*JOSE*) 2007, Vol. 13, No. 4, 441–450

Location of Noise Sources in Fluid Power Machines

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This paper discusses noise generation mechanisms and techniques for noise reduction in fluid power units. Major noise sources in fluid power units can be identified with a sound intensity method. It has been proved that components of power units with larger sound radiating surfaces such as an electric motor and an oil reservoir produce a major part of global noise radiation.

fluid power units noise generation noise sources

1. INTRODUCTION

Annoyance caused by noise emitted by different machines is often determined by power units. This can be observed not only in static applications but also in the field of mobile hydraulics. Noise reduction in fluid power drives is increasingly important in view of their increasing power and stricter requirements concerning noise emitted by machines.

Noise generated by power units cannot be sufficiently decreased with primary approaches towards single components. There are guidelines for the design of power units regarding noise reduction. However, transferring empirical results from one type of power unit to another is rarely successful.

Several investigations [1, 2, 3] have shown that noise emitted by pumps depends on their mounting, i.e., the application and the layout of the piping. Practical experience also shows that a pump considered silent in an anechoic room can cause high noise emission when built into a real system. These results show that the noise behaviour of a pump is strongly system-dependent.

The complete sound power of a hydraulic system is defined by the sum of the single powers of each component. The total sound power level is given by the loudest source or sources of noise. Successful noise reduction of hydraulic systems can only be achieved when the design at the loudest noise sources is changed. Therefore, detection of the loudest noise sources in a power unit is most important. This is possible with a sound intensity method this paper presents.

2. NOISE SOURCES IN POWER UNITS

The variable forces inside the pump mainly depend on variations of pressure within the pump. They change periodically with the change of the load at the displacement elements. The excitation forces from flow ripple of the pump are important especially in the lower frequency range. Additional variable forces caused by mechanical shocks have to be taken into consideration as well. Both kinds of noise in a real system, i.e., structure- and fluid-borne noise, cannot be separated in most cases.

Whether noise produced by a hydraulic system is fluid- or structure-borne from internal excitation in the pump depends on the structure of the system. Investigations have shown that the noise behaviour of a power unit with a horizontal mounted motor pump group is quite different from a vertical

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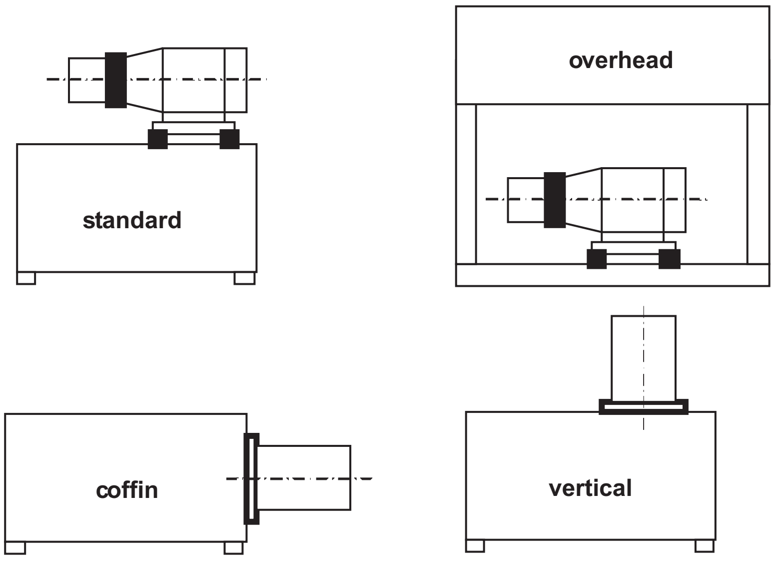


Figure 1. Types of fluid power units.

power unit [1]. Noise behaviour of a power unit depends on the pump, too. The relations between both causes of noise in a real system are very complex.

The four basic types of fluid power units are illustrated in Figure 1.

2.1. Standard Power Units

In standard power units, the motor pump group is mounted horizontally on the top side of the tank. Noise emitted by the pump is usually much lower than that emitted by the motor [1]. However, the pump is responsible for the vibrations and noise radiation of the motor. The damping flanges can be used to reduce transmission of vibration from the pump to the motor. Dampening rods are the most effective supporting elements of the motor. Some additional measures on the motor, e.g., a fan cover with damping material, and sound absorbing material and/or damping material on the bellhousing, lead also to noise reduction.

2.2. Overhead Power Units

In overhead power units there is no significant noise radiation from the tank structure [1]. The tank can be isolated from the motor pump group. The dampening flange between the pump and the motor as well as the dampening rods for motor support are effective in reducing noise transmission from the pump to the motor, and from the motor, to the tank structure. It is an advantage that suction pressure is higher than in other types of power units. The suction hose should be isolated from the pump to avoid transmission of vibrations from the pump to the hose. Measures for reducing noise emitted by the motor and bellhousing should be considered in that type of power unit, too.

2.3. Coffin Power Units

Coffin power units are similar to the vertical ones. Using quiet motors in this type of power units is very important. To reduce the transmission of vibrations between the pump and the motor a dampening flange should be used. Noise emitted by a pump submerged in a reservoir will be transmitted to the tank structure like in the vertical power units.

Noise transmission paths in a vertical fluid power unit are shown in Figure 1. Structure-borne noise is transmitted from the pump to the electric motor along path 1. Transmission occurs through damping flange and bellhousing 1a as well as through coupling 1b. From the motor housing structure-borne noise is transmitted to the fan cover (path 2) and to the top plate of the tank (path 3). Some structure-borne noise is

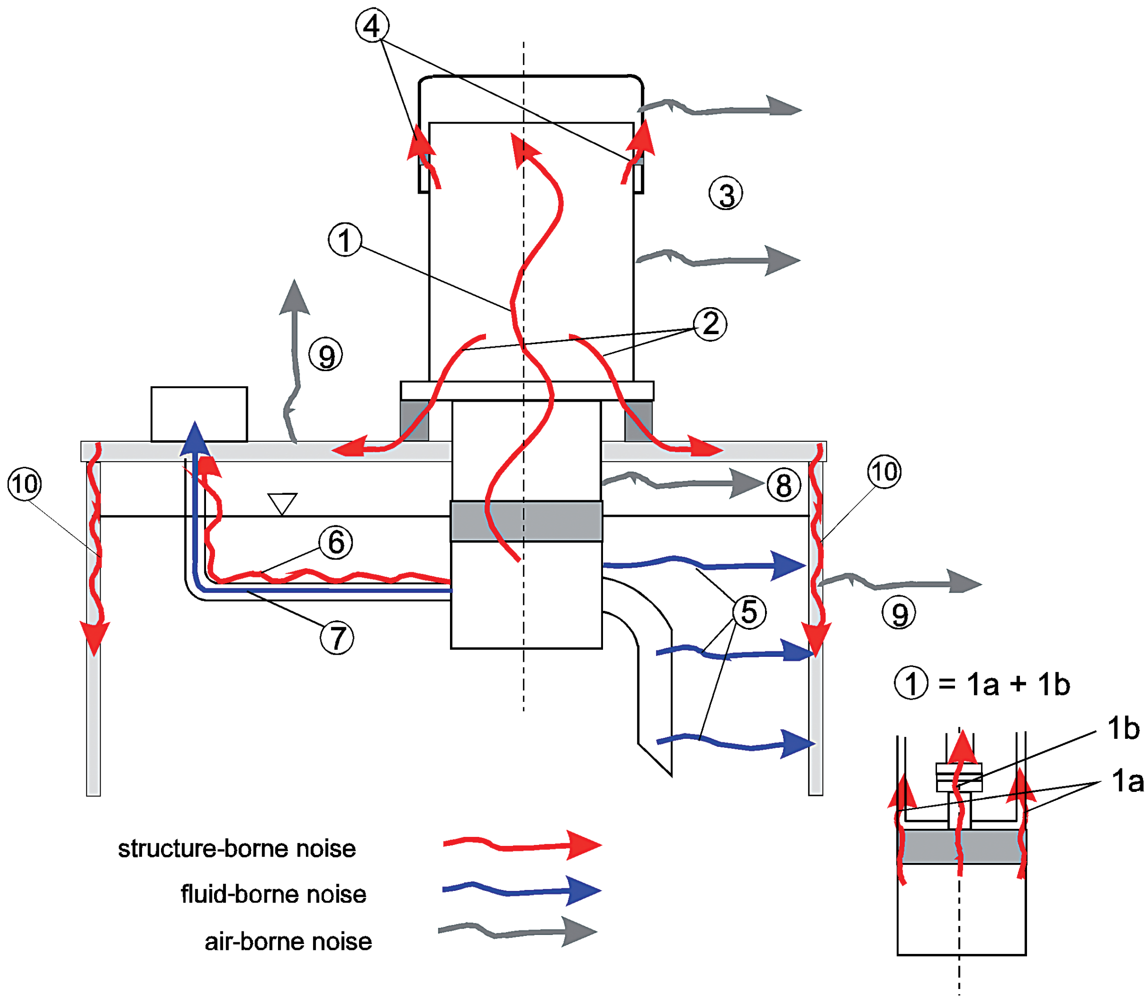


Figure 2. Noise transmission in a vertical power unit. *Notes.* 1–10—transmission paths.

transmitted from the pump directly to the valve and the tank structure through pipes and hoses (path 4). With the pipes and hoses fluid-borne noise from the pump is transmitted, too (path 5). Due to vibrations of the pump and the attached components fluid-borne noise is excited into the oil volume in the tank. Depending on the acoustic behaviour of the fluid volume in the tank some pump vibrational energy is transmitted directly to the tank structure (path 6). Airborne noise of the power unit comes from the motor and the tank structure with its attached components. Airborne noise from the bellhousing transmitted through the tank walls is shown in Figure 1 as path 7.

3. IDENTIFICATION AND RANKING OF NOISE SOURCES

Measurement of sound pressure level (SPL), based on Standard No. ISO 4412-1:1991 [4], depends upon the characteristics of the room where the machine is mounted. After taking SPL measurements, the next step is to identify the major sources of noise, which is very difficult using SPL data. This is where sound intensity measurements are needed. The main advantages of the sound intensity method are as follows:

- single noise sources can be quantified;
- stationary ambient noise has little influence on the measured values;
- the influence of room conditions and other nearby noise sources is negligible; and
- measurements close to the object are possible.

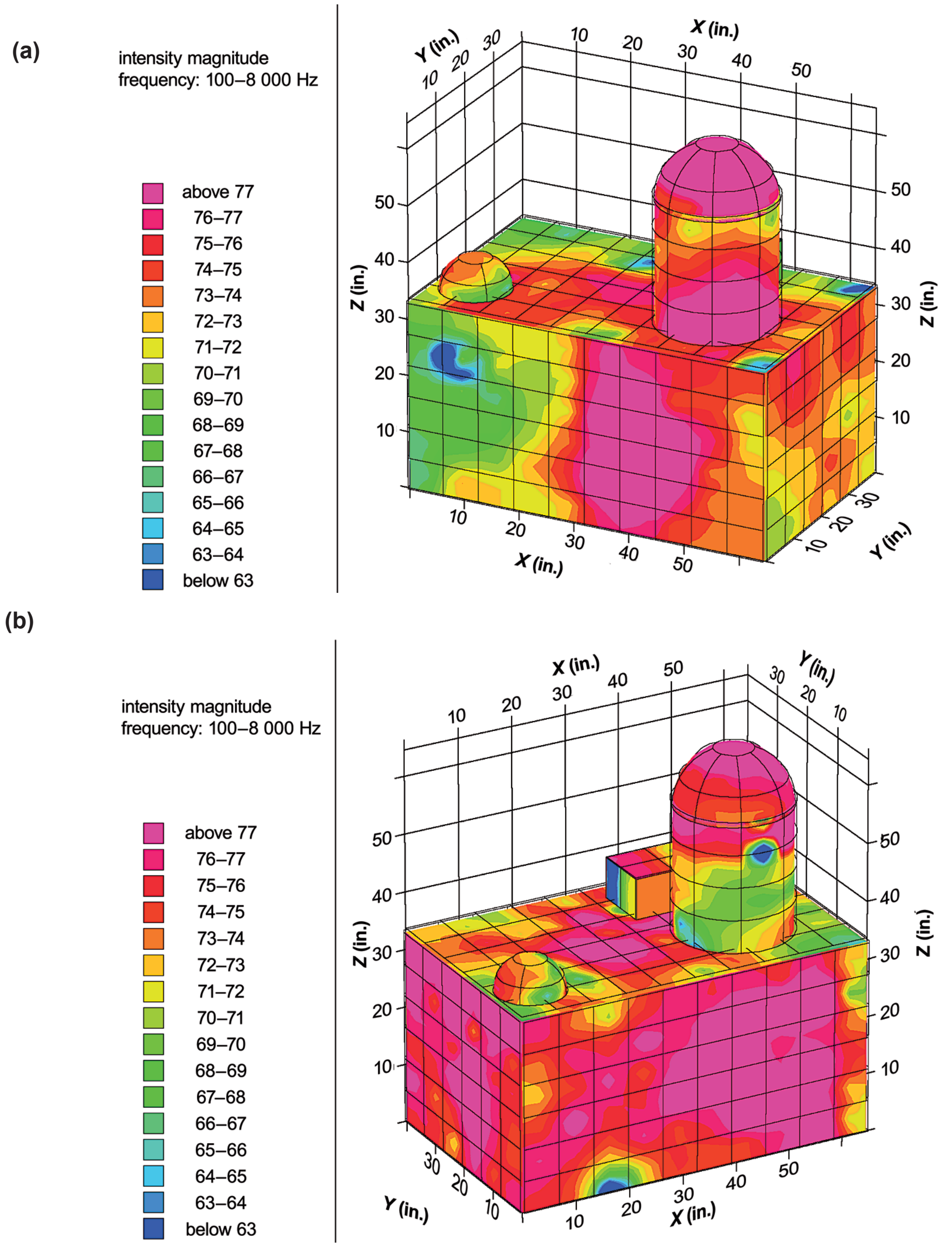


Figure 3. Distribution of sound intensity on the surface of an 800-L power unit (1 200 min^{-1}, 120 bar), for (a) full flow and (b) deadhead, frequency range 100–8 000 Hz. *Notes*. 1 in. = 2.54 cm.

Measurements were carried in an 800-L vertical power unit with various pressures. Sound intensity was measured to identify noise sources close to the surface of the power unit, at a distance of 5 cm.

Figure 3 illustrates the distribution of sound intensity on the surface of a power unit for full flow and deadhead operating conditions. Active intensity in the frequency range between 100 and 8 000 Hz is displayed; it is responsible for the transmission of sound energy from the radiating surface to the surrounding air. During full flow the electric motor is the major source of noise. The motor radiates high noise especially at the fundamental pump frequency of 216 Hz[1] and its harmonic components (432, 648 Hz, etc., see

[1] An rpm of 1 180 was measured during the motor load. 1 180 rpm/60 s • 11 vanes in the pump = 216 Hz (cycles/s).

Figure 4). This indicates that the pump is the main reason why the electric motor generates noise.

For deadhead, the sound intensity levels on the tank walls (Figure 3b) are higher than the levels measured on the motor and on the fan cover. Noise radiation of the tank is especially high in the frequency range from 2 000 to 8 000 Hz. Sound intensity on the side walls of the tank is distributed evenly in this frequency range.

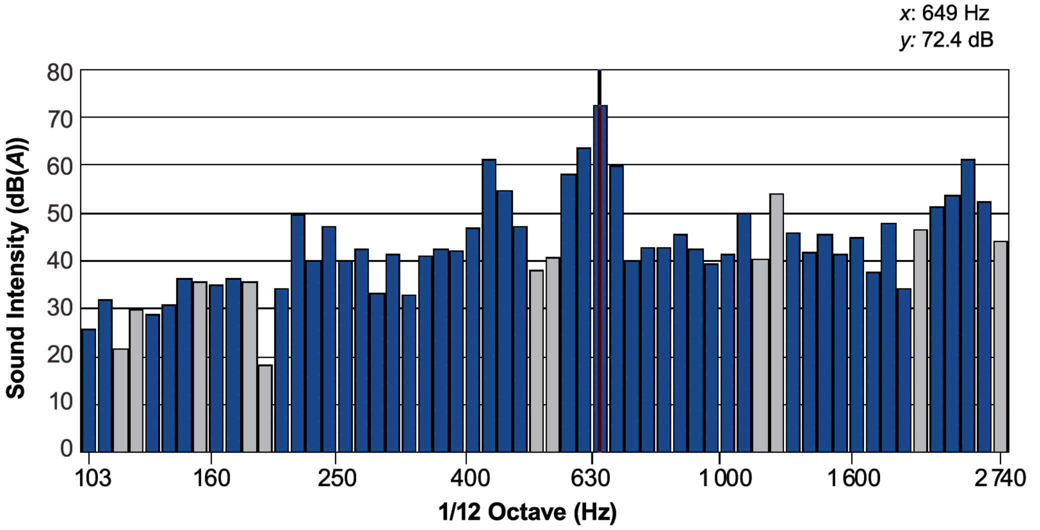


Figure 4. Sound intensity spectrum for one measurement point at the motor (1 180 min^{-1}, 120 bar, reference value 1.00 pW/m^2, deadhead). *Notes.* blue—active sound intensity, grey—passive sound intensity.

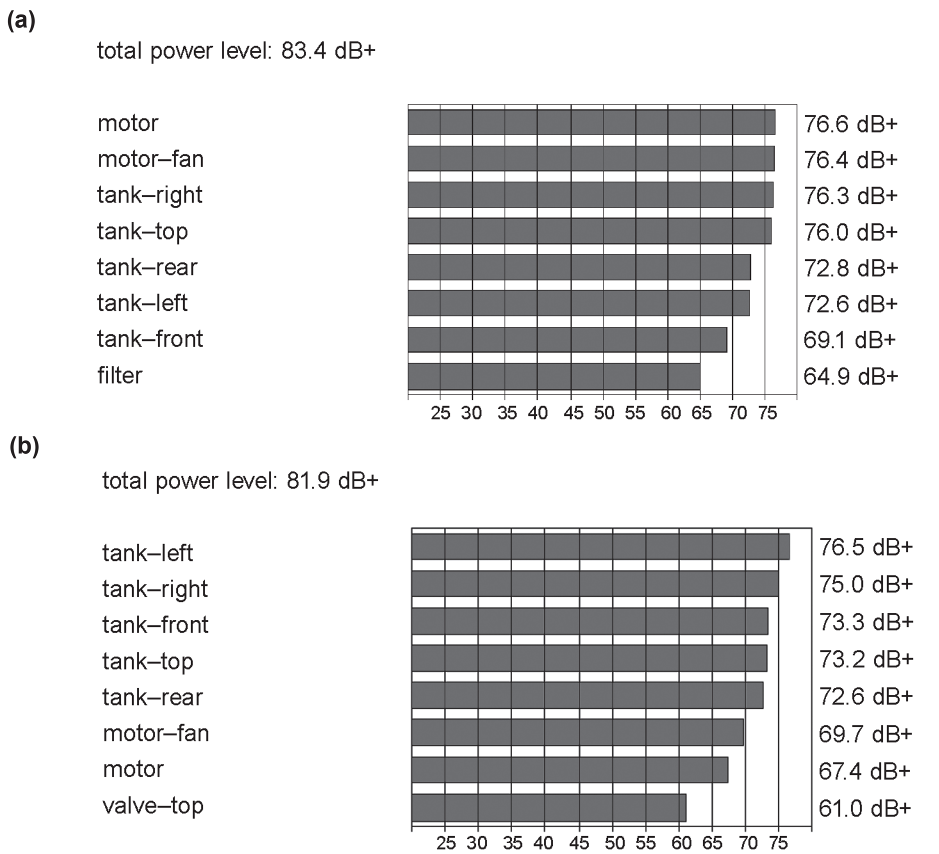


Figure 5. Ranking of noise sources in a power unit; (a) full flow, (b) deadhead. *Notes.* 1 180 min^{-1}, 120 bar, A-weighted, frequency:100–8 000 Hz.

Figure 5 shows the ranking of noise sources in the power unit for full flow and deadhead conditions. During full flow, not only motor noise, but also noise from the top side and one of the longer sides of the tank has to be reduced to achieve significant noise reduction in the power unit. In deadhead conditions, noise radiation from the walls of the tank is more significant than motor noise. As already mentioned only measures on the major sources of noise lead to significant noise reduction.

4. VIBRATION MEASUREMENTS AND SIMULATION

Vibration measurements were carried out on major noise sources in order to further identify the dynamic behavior of the 800-L power unit. The operational deflection shape analysis (ODSA) showed that the electrical motor (Figure 6a) vibrated like a rigid body in the frequency range of up to 1 000 Hz. Those vibrations depended mainly on the support of the motor; the dampening ring in the investigated case. ODSA on the tank structure (Figure 6b) showed that higher amplitudes of vibrations of the tank occured mainly in frequency ranges above 2 000 Hz. That correlates with the results of sound intensity measurements, which showed higher noise radiation of the tank structure in a frequency range above 2 000 Hz. Table 1 lists characteristic frequencies, which were identified in the noise and vibration measurements.

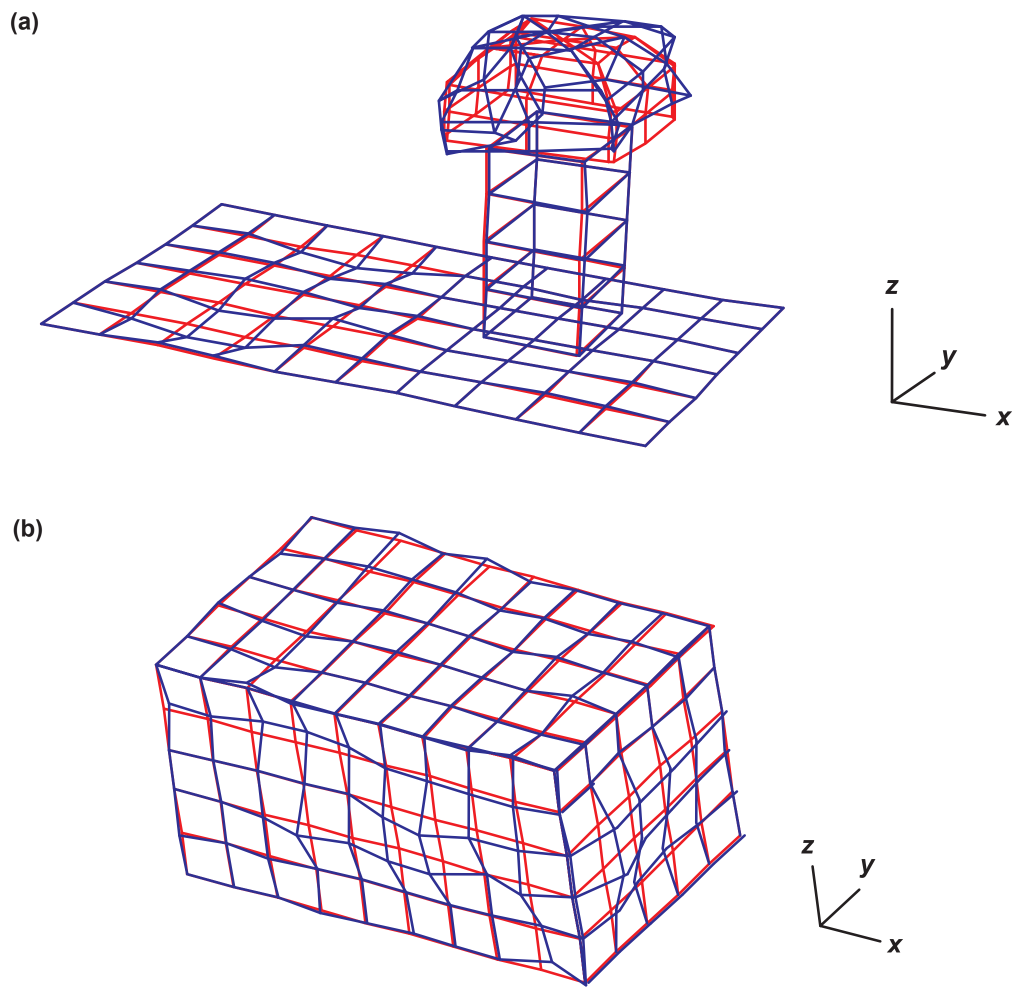


Figure 6. Vibration shapes from operational deflection shape analysis; (a) for 432 Hz: motor and top side of the tank, (b) for 2 390 Hz: tank structure.

TABLE 1. Identified Frequencies During Investigations of an 800-L Power Unit

Frequency (Hz)	Description
216	1st harmonic vane pump
275	1st harmonic gear pump
292	motor (5th harmonic of operational frequency)
432	2nd harmonic pump
596	motor (10th harmonic of operational frequency)
648	3rd harmonic pump
876	4th harmonic pump
948	motor (magnetic force oscillations)
1 080	5th harmonic pump
1 296	6th harmonic pump
1 512	7th harmonic pump
1 728	8th harmonic pump
1 944	9th harmonic pump
2 163	10th harmonic pump
2 376	11th harmonic pump
2 595	12th harmonic pump
2 812	13th harmonic pump

Notes. Pump vibration frequency: $fk = k \bullet n \bullet z/60$, where k—number of harmonic, n = 1 180 rpm, z = 11, $f1$ = 216 Hz. Motor vibration frequency: $fM = 2 \bullet fL \bullet R \bullet (1 - s)/P + 2 \bullet fL$, where fL = 60 Hz—operational frequency, R = 42—number of rotor conductor bars, $s = 1 - ns/n$—loaded motor speed slip s = 1 – 1 180/1 200 = 0.0166, P = 6—number of motor magnetic poles, fM = 945 Hz.

Figure 7 is an illustration of a mode shape of the electrical motor (calculated with the final element method) at the natural frequency of 716 Hz. In the final element model the pump and the electrical motor are considered to be a rigid mass. The belhousing and the motor fan cover are modelled with elastic shell elements. The connection between the pump and the motor is modelled with three-dimensional spring elements. Simulation is a very helpful tool in interpreting results of an experiment and in identifying quickly the influence of the model parameters on the natural frequencies of the motor pump group.

In vertical power units the motor is supported at the top of the tank. With this interface (path 2 in Figure 2), structure-borne noise is transmitted from the motor pump group to the tank structure. A dampening ring is used to reduce the transmission of structure-borne noise from the motor pump group to the tank. The dampening ring has two effects: (a) motor vibrations are isolated from the top plate, and (b) the amplitude of the vibrations of the motor body (including the fan cover) increase due to the elastic support. Investigations have shown that there is no significant noise reduction in SPL with or without the dampening ring. The

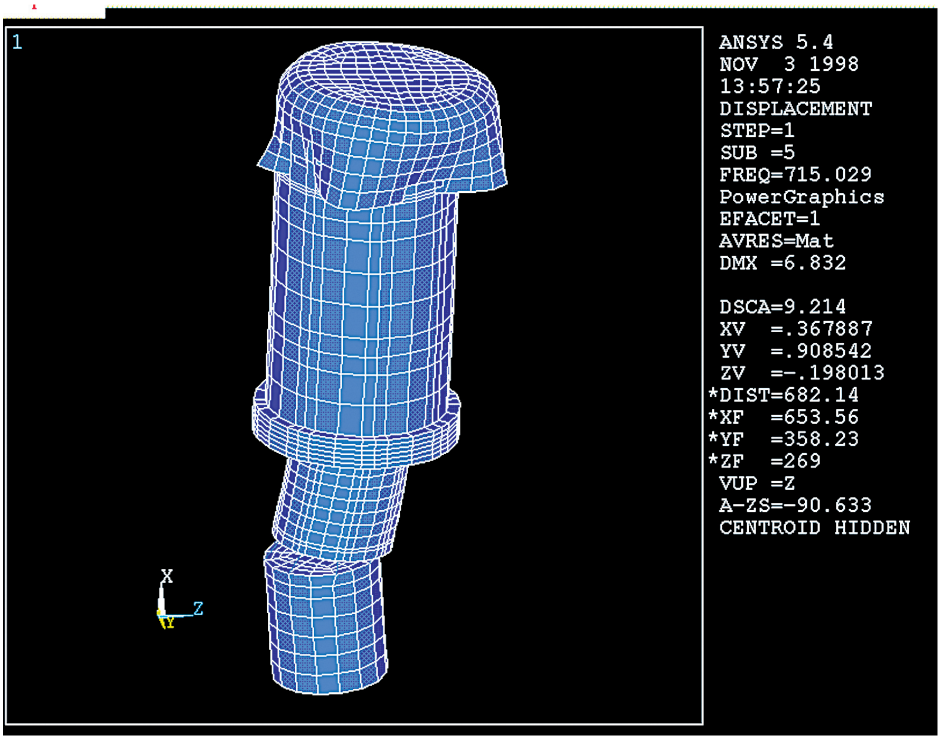


Figure 7. The mode shape for the natural frequency of a motor pump group at 716 Hz (final element method simulation).

measurements have been carried for 100-, 400- and 800-L power units.

5. TRANSMISSION LOSS (TL) INSIDE THE TANK

The oil-submerged pump in a vertical hydraulic power unit is in a kind of enclosure. Noise emitted by the pump is absorbed by the oil as well as the tank structure. TL in the tank was measured with a special loudspeaker, which was mounted in a sealed enclosure and put in several positions in the tank. The harmonic source signal of various frequencies was supplied to the loudspeaker from an FFT analyzer (Zonic, USA). Figure 8 shows average SPL for the loudspeaker alone (without the tank) and average SPL of the loudspeaker mounted in the tank with oil. In the frequency range up to 2000 Hz, TL was 30 dB(*A*). For frequencies over 2600 and 3300 Hz, TL was 20 and 10 dB(*A*), respectively. For frequencies above 4000 Hz, TL was up to 20 dB(*A*).

Transmission of noise emitted by the pump through the oil and the walls of the tank is effective especially in the frequency range between 2200 and 3500 Hz.

Higher SPL was measured for the positions of the noise source closer to the side walls of the tank. Therefore in vertical power units the pump should be positioned as far as possible from the side walls of the tank. The critical distance was in the investigated case about 15 cm.

Due to mechanical vibration of the pump and the attached components (suction and delivery pipe/hose, bellhousing, etc.) fluid-borne noise in the oil volume in the tank is excited. These small pressure changes in the fluid (like in the air) are transmitted to the tank structure, whose vibrations they excite. Up to now fluid-borne noise generated by the vibration of the pump could not be separated from noise generated directly by the pump.

Up to 20% of the volume of the tank is not filled with oil. In that volume airborne noise from the bellhousing and other vibrating components in the tank is transmitted through the walls of the tank into the environment of the power unit (path 8). TL of the tank structure only is not high. Model investigations with the noise source were carried out to determine what it was.

Average SPL of the noise source (loudspeaker) without the tank and after putting it in the tank (but without oil) is compared in Figure 9. TL inside the tank structure in the frequency range 2200–2800 Hz and 3400–4000 Hz is under 5 dB(*A*). That means airborne noise in the tank will be easily transmitted for several frequencies into the environment of the power unit.

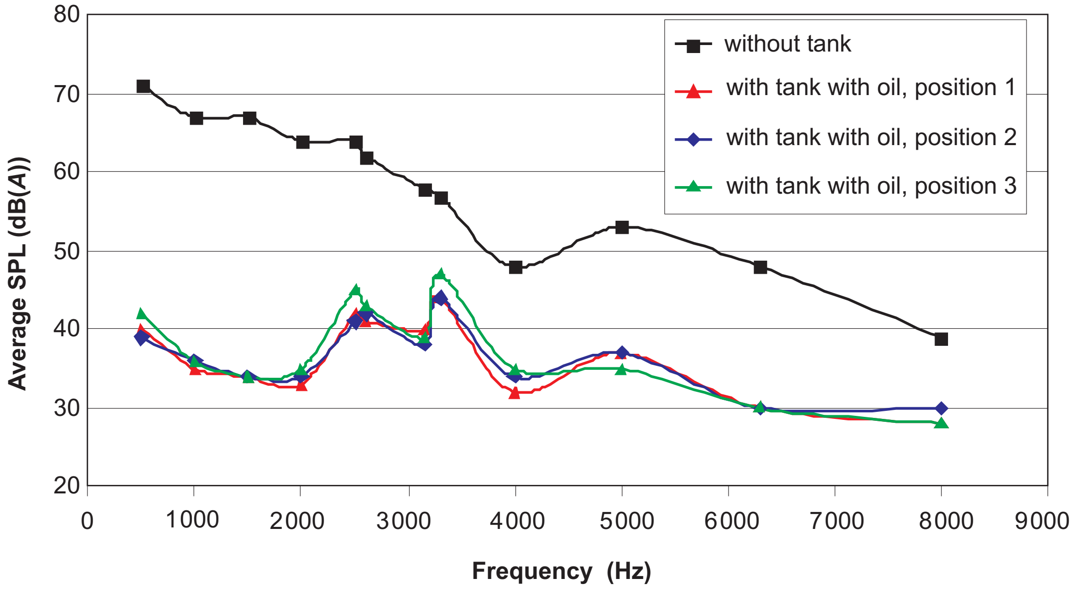


Figure 8. Sound transmission in a tank of an 800-L power unit. *Notes.* SPL—sound pressure level.

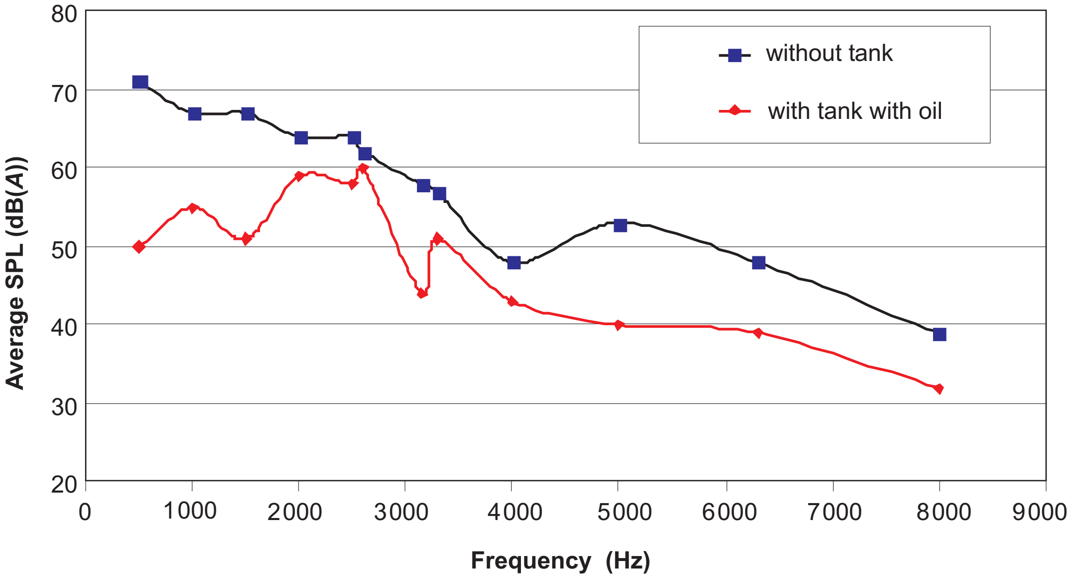
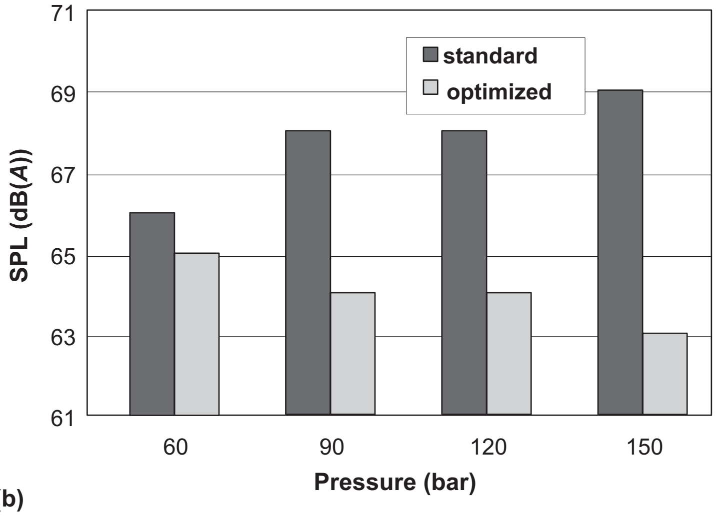


Figure 9. Transmission losses of airborne noise inside a tank of an 800-L power unit. *Notes*. SPL—sound pressure level.

(a)



(b)

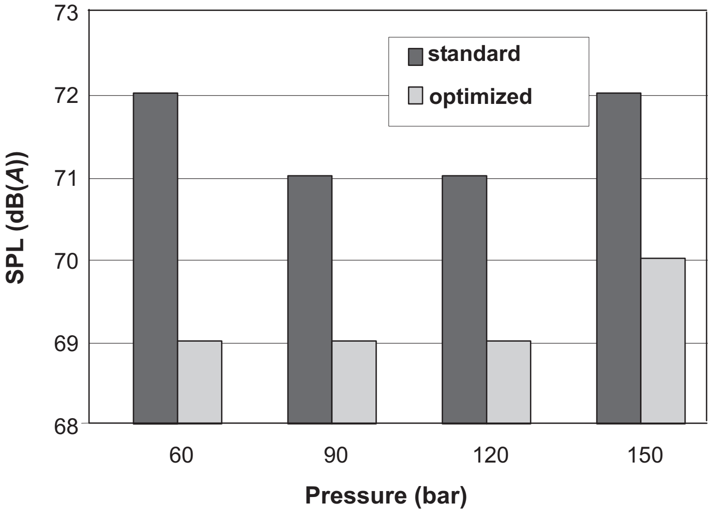


Figure 10. Comparison of average sound pressure levels versus pressure for a standard and optimized 800-L power unit: (a) full flow, (b) deadhead. *Notes*. SPL—sound pressure level.

As already mentioned, tank noise is usually above 2 000 Hz. If acoustic absorption material is used in the tank, especially high frequency noise of the pump will be significantly reduced. The tank is much quieter also because direct transmission of fluid-borne noise from the pump (due to its vibration) to the side walls of the tank is reduced.

Figure 10 illustrates noise reduction in the 800-L power unit. The standard version of the unit and the version without the dampening ring or with acoustic absorption material in the tank have been compared. The differences between standard and optimized versions of the 800-L power unit for deadhead increase with pressure (Figure 10). For pressure of 1 200 psi there is noise reduction of 6 dB(*A*).

The effectiveness of isolating vibrations in the described transmission paths depends on the importance of the isolated noise source in the ranking of all noise sources. Significant noise reduction in vertical power units can be achieved by reducing noise transmitted from the pump to the tank structure. The use of acoustic absorption materials in the tank resulted in noise reduction of up to 2 dB(*A*) for full flow and up to 6 dB(*A*) for deadhead (for 1200 min^{-1} and 150 bar) conditions.

The power unit is well optimized regarding noise when its overall SPL is close to the SPL of the motor pump group. The potential for noise reduction in the power unit can be estimated by comparing SPL of the power unit and of the motor pump group.

6. FINAL REMARKS

This paper discusses noise generation mechanisms and techniques for noise reduction in vertical power units. Major noise sources in fluid power units can be identified with the sound intensity method. Components of power units with larger sound radiating surfaces such as the electric motor and the oil reservoir produce a major part of global noise radiation. The dampening flange reduces transmission of structure-borne noise from the pump to the bellhousing and the motor. Noise reduction of 2 dB(*A*) for full flow and 1 dB(*A*) for deadhead is achieved with the dampening flange in an 800-L power unit. Dampening rings in the investigated power units do not have a significant influence on noise radiation.

REFERENCES

1. Fiebig W, Wernz Ch. Untersuchung des Geräusch- und Schwingungsverhaltens von hydraulischen Systemen. o+p Ölhydraulik und Pneumatik. 1997;(5):368–72.
2. Fiebig W, Zirkelbach T. Simulation des Körperschallverhaltens von hydraulischen Systemen. Teil 2. Rehnergestützte Struktur-optimierung. o+p Ölhydraulik und Pneumatik. 1997;(8):610–4.
3. Dahm M. Wirksame Schallreduzierung von Hydraulikaggregaten. o+p Ölhydraulik und Pneumatik. 1997;(2):210–4.
4. International Organization for Standard-ization (ISO). Hydraulic fluid power—test code for determination of airborne noise levels—part 1: pumps (Standard No. ISO 4412-1:1991). Geneva, Switzerland: ISO; 1991.