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NOISE CONTROL OF FLUID POWER UNITS

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This paper describes the noise generation mechanisms and methods for noise reduction of fluid power units. The main noise sources can be identified with the sound intensity method. The results show, that the system consideration is required to achieve the noise reduction of power units. Higher noise radiation occurs mostly at the components of power units with larger sound radiating surfaces like electric motor or tank structure.

1. Introduction

Fluid power units determine often the noise development of many stationary industrial machines like hydraulic presses, machine tools, plastics moulding machines, etc. Similar situation is 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 control of machines. Noise generated by power units cannot be sufficiently decreased only 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 [1-6] have shown that noise emitted by pumps depends on their mounting, i.e., the application and the layout of the piping system. Practical experiences also show that a pump considered as silent in an anechoic room can cause high noise emission when built into a real system. These results show that the noise behavior of a pump is strongly system-dependent. The complete sound power of a hydraulic system is defined by the sum of the single sound powers of each component. The total sound power level is determined by the loudest source or sources of noise. Therefore, detection of the loudest noise sources in a power unit is most important. Successful noise reduction of hydraulic systems can only be achieved, when the design changes will be made to the loudest noise source or sources.

2. Noise development in a power unit

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 variable forces cause vibrations and dynamic loads on displacement elements, which are transmitted through the gaps filled with fluid and through the bearings. 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 reasons of noise development in a real system for structure- and fluid borne noise, cannot be separated in most cases. Investigations have shown that the noise behavior of a power unit with a horizontal mounted pump- motor group is quite different from a vertical power unit [6]. Noise behavior of a power unit depends on the pump, too. The relations between both causes of noise in a real system are relatively complex.

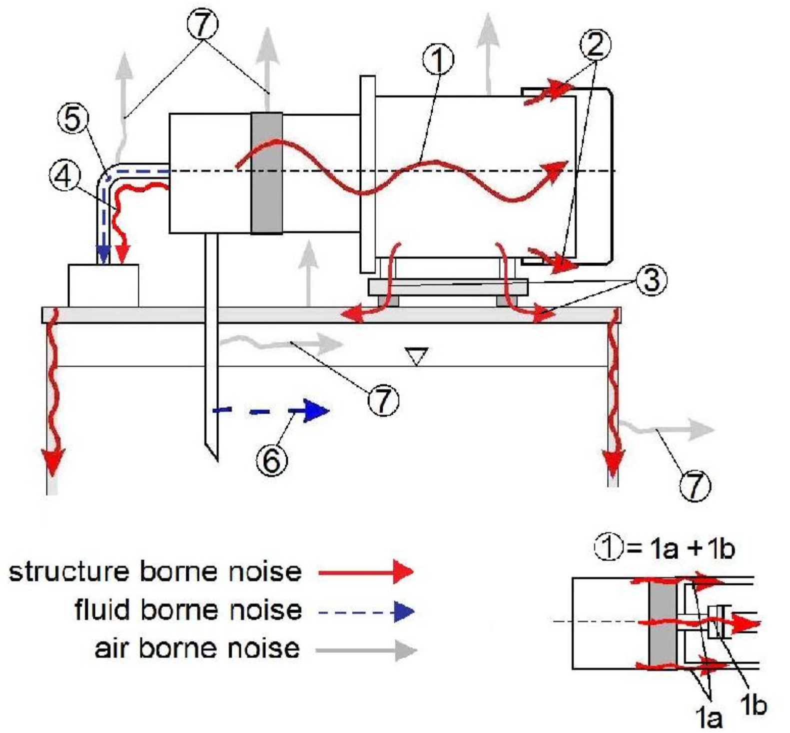

Figure 1: Noise transmission paths in fluid power unit.

Noise transmission paths in a standard fluid power unit are shown in Fig. 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 transmitted from the pump directly to the valve and the tank structure through pipes and hoses (path 4). With the pipes and hoses the fluid-borne noise from the pump is transmitted to valve and to the tank (path 5). Due to vibrations of the pump and the attached suction pipe, fluid-borne noise is excited into the oil volume in the tank (path 6). Depending on the acoustic behavior of the fluid volume in the tank some pump vibrational energy is transmitted directly to the tank structure. Airborne noise of the power unit comes from the pump, motor and the tank structure (path 7)with its attached components.

3. Localization of main noise sources in the power unit

Measurements were carried in an 400-L power unit with horizontally mounted pressure controlled vane pump with the flow rate Q= 63 l/min at 1500 rpm (Fig. 2).Sound intensity measurements has been done with the B&K Sound Intensity Analyzer 2144 and Sound Intensity Probe 3584.The distance between the probe and the surface of the power unit was 5cm.

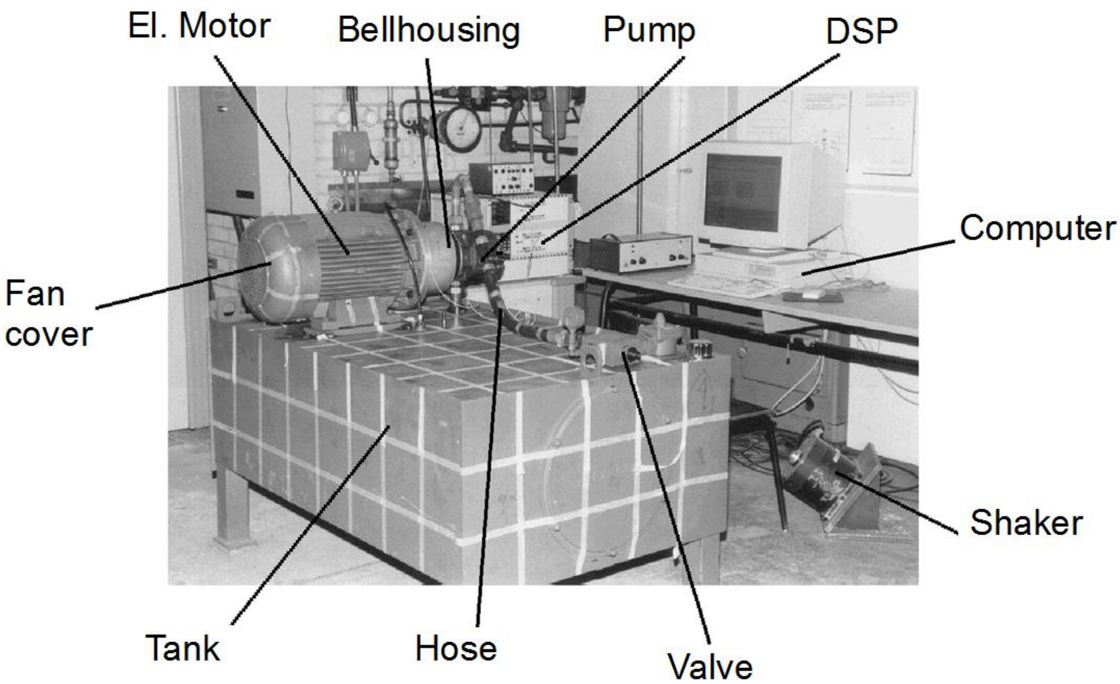

Figure 2:View of the investigated power unit.

Figure 3 illustrates the sound intensity distribution on the surface of a power unit for full flow operating condition. Active intensity in the frequency range between 100 and 8 000 Hz is displayed, which 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 noise especially at the fundamental pump frequency of 241 Hz and its harmonic components. This indicates that the vane pump is the main reason for the noise radiation of the electric motor.

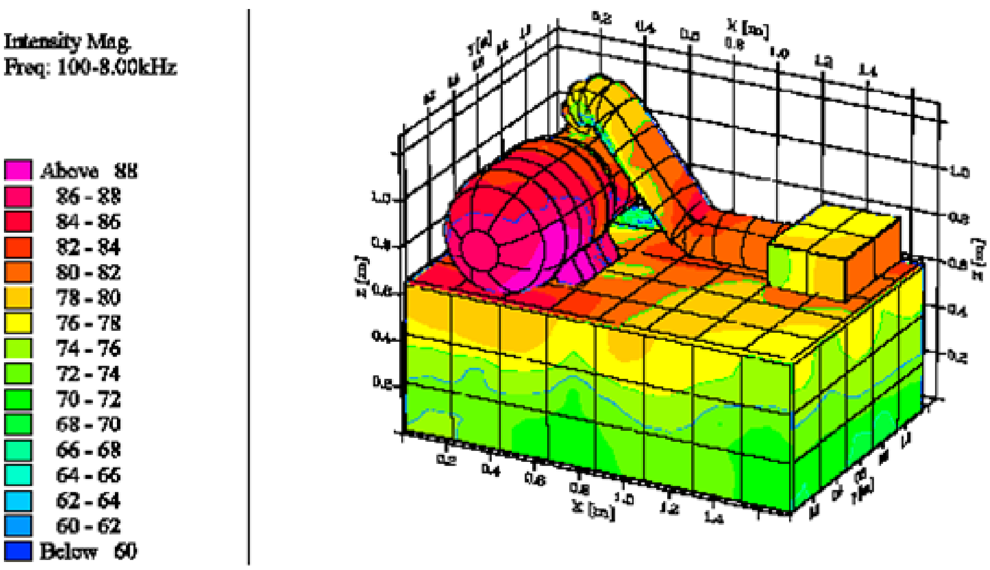

Figure 3:Sound intensity distribution on the surface of the power unit (p=10 MPa), frequency range: 100 Hz -8 000 Hz.

In Fig. 4 the ranking of noise sources in the power unit has been shown. It can be seen, that the hydraulic pump is on the third place regarding the sound power level. The difference of 7 dB(A) between the pump and motor means, that the sound power of the motor is 5 time higher than the sound power of the pump.

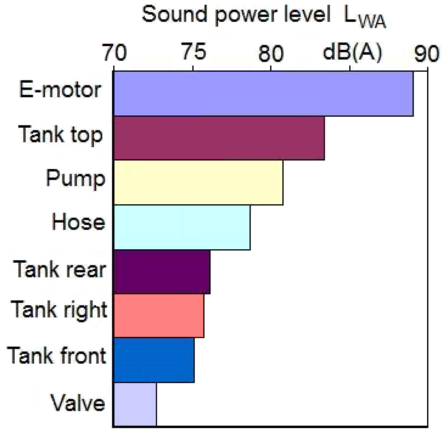

Figure 4: Ranking of noise sources in the power unit (Q=63/l/min, p = 10 MPa).

The noise sources of power units can be also identified with an acoustic camera and Beamforming Method [8].

4. Vibration measurements and simulation results

For further identification of the dynamic behaviour of the power unit Experimental Modal Analysis(EMA) and Operational Deflection Shape Analysis (ODSA)have been carried out. The measurement setup has been shown in Fig.5. Vibration has been excited with the shaker connected with power amplifier and control system. Measurement points has been defined on the pump-motor group and on the upper tank wall, as the main sources of the power unit.

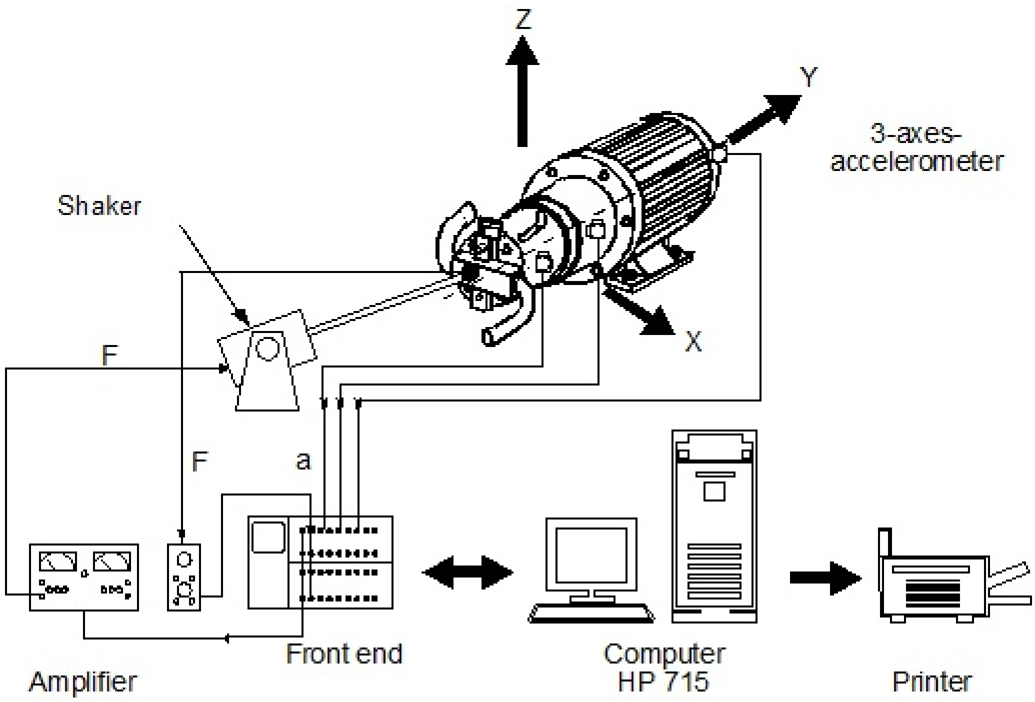

Figure 5: Measurement setup for Experimental Modal Analysis (EMA).

The vibration in the fluid power units result from the excitations due to pressure ripples and the transmission of the mechanical vibrations from the pump and other hydraulic components. The vibration amplitudes are strongly dependent on the resonance behaviour of the mechanical structure of the power unit. System consideration is therefore obligatory to establish the proper relationships by the vibration and noise development in the fluid power units[6].

In Fig. 6 the result of Operational Deflection Shape Analysis at the frequency of 1205 Hz (fourth harmonics) has been shown.

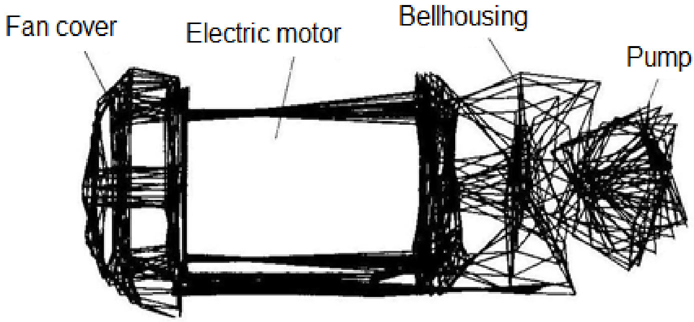

Figure 6: Operational Deflection Shape Analysis at 1205 Hz for the pump motor group (p = 25 MPa).

From Fig. 6 can be stated, that the pump and motor housing vibrate as rigid bodies while by bellhousing and fun cover the structural vibration can be found. In fluid power units the pump is usually connected with the motor through the bellhousing and rubber damping flange. The connection between electrical motor and tank structure is often made with rubber dampening elements. These vibration isolation elements are used to reduce the transmission of vibrations and noise emission. The results of experimental investigations [4] have shown, that in lower frequency range up 1,5 kHz, which is acoustically most important, the pump and motor vibrate as rigid body.

The natural frequencies and vibration shapes of the mechanical structure of fluid power unit can be measured and established theoretically with the Finite Element Method. The FE- Model can be simplified based on the results of the EMA- Experimental Modal Analysis and ODSA –Operational Deflection Shape Analysis. The simulation model of the power unit has been shown in Fig.7. The motor and pump has been modelled as rigid bodies, other elements like upper tank wall, fan cover, bellhousing, damping flanges have been modelled as elastic structures. The FE- model of fluid power unit, shown in Fig. 7consists from:

- tank structure, fan cover, motor feet and bellhousing are modelled with shell elements, because the thickness of these elements is small in comparison with other dimensions,
- damping flange and dampening rails represent 3D spring elements. The stiffness of spring elements has been established based on experimental investigations,
- the pump and motor body have been modelled as rigid masses.

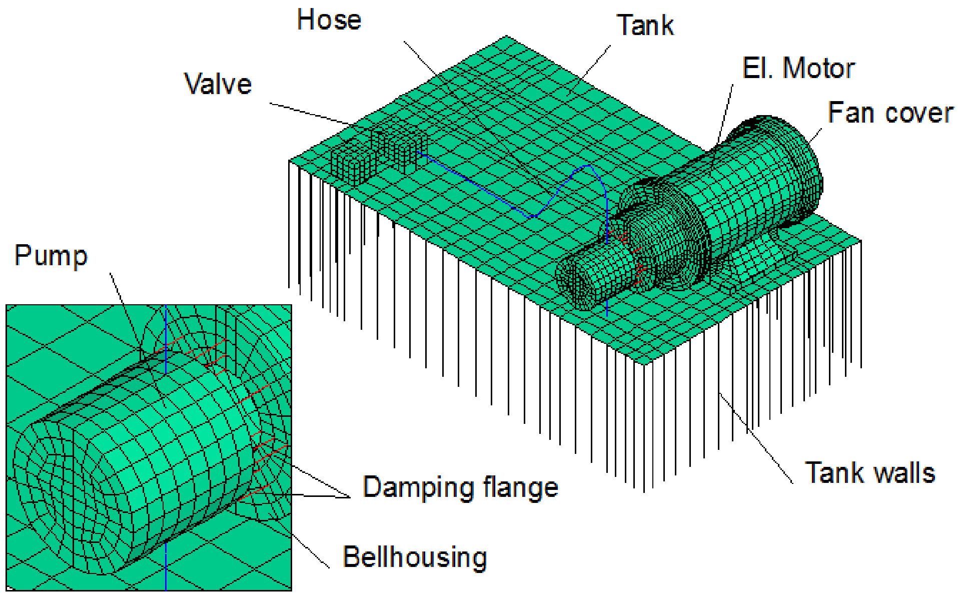

Figure 7: FE- Model of the power unit.

The comparison the results of FE- Simulation and Experimental Modal Analysis has been shown in Fig8.

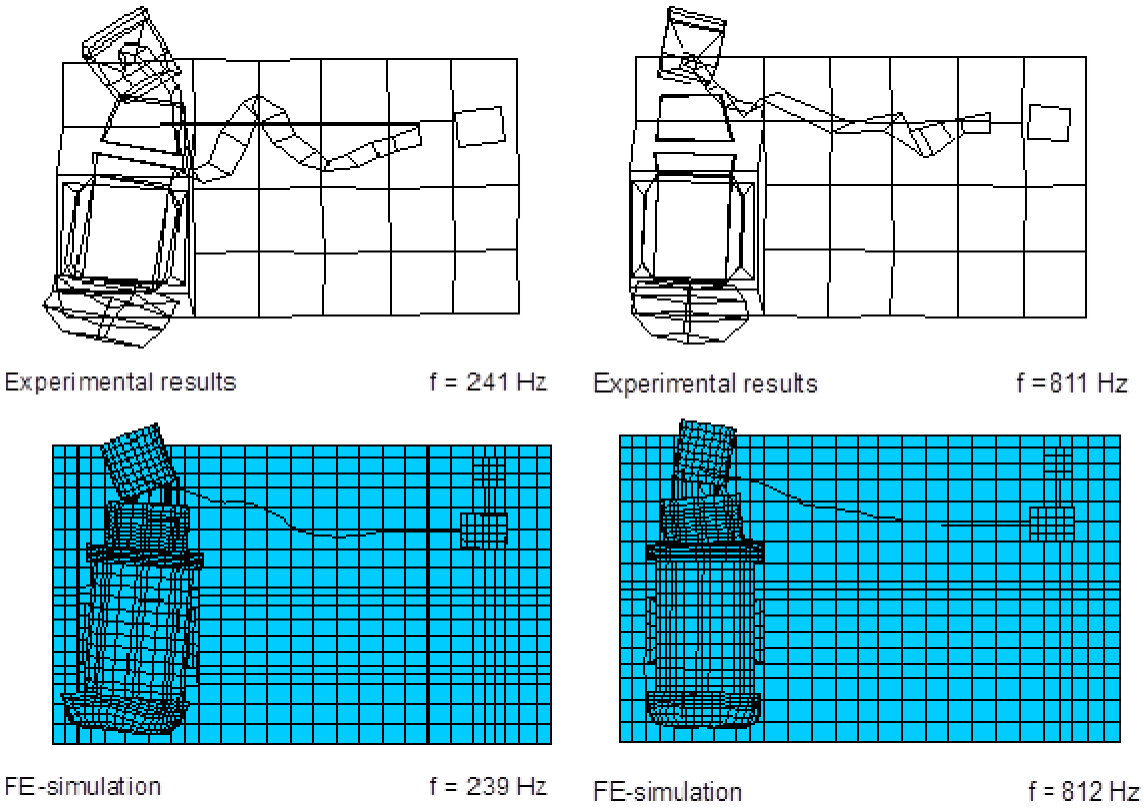

Figure 8:Comparison of experimental and simulation results.

The good coincidence between the calculated and measured natural frequencies and mode shapes has been established. The way to achieve the significant reduction of vibration amplitudes and noise is to decouple the natural frequencies of the mechanical structure of fluid power unit from the excitation frequencies, which are equal the basic pump frequency and harmonics. Based on that methodology the structure modifications for vibration and noise reduction has been obtained, which has been verified on many design of fluid power units [6].

The experimental investigations show, that the pump- motor group is mostly responsible for the noise development of power units. For vibration isolation of the pump – motor group the dampening rings, damping rails will be used. In Fig.9four different configurations of vibration isolation measures in the power has been shown.

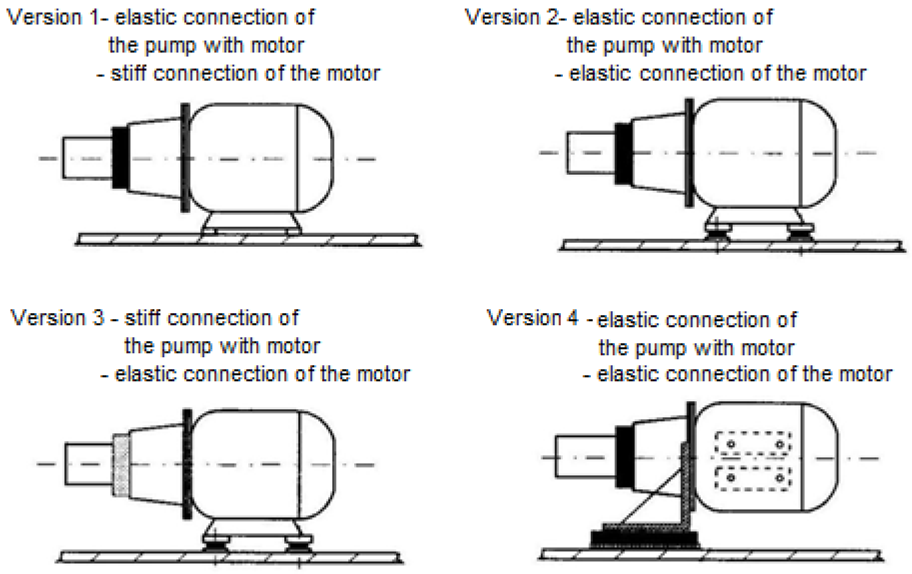

Figure 9: Investigated modifications of the pump motor group.

The modifications of standard power unit have been investigated with the pressure controlled vane pump. For each modification sound power measurements in a semi anechoic chamber according to ISO 9614 Part 2 [9] have been carried out. The results of these measurements have been shown in Fig. 9.

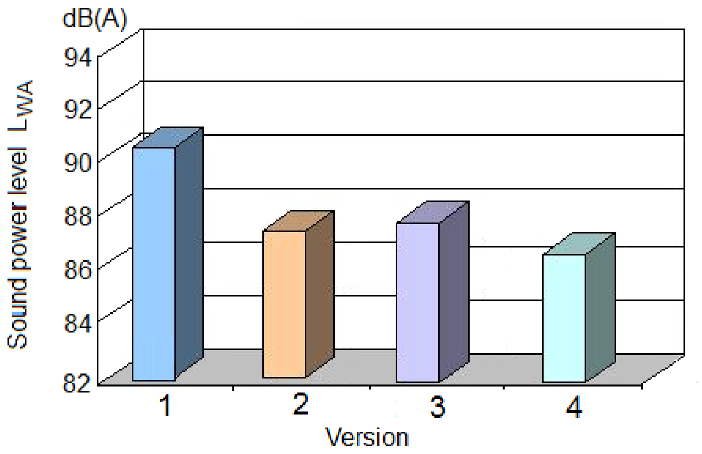

Figure 10: Comparison of sound power levels of power unit with vibration measures according Fig. 8.

Fig. 10 shows, that the highest sound power level occurs at the stiff connection between pump-motor group and upper tank wall. The damping rails for the connection to the upper tank wall lead to the reduction of sound power level of 4 dB(A). The difference in sound power levels between versions 2 and 3 with stiff and elastic connection of the pump to the motor are relatively small around 1 dB(A).

5. Conclusions

This paper discusses noise generation mechanisms and techniques for noise reduction of power units with horizontal mounted pump motor group. 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 may produce a major part of global noise radiation. The dampening rails reduce most efficiently the transmission of structure-borne noise from the pump motor group to the tank structure. They have also significant influence on noise radiation.

The effectiveness of measures for vibration isolation 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 power units can be achieved when the compliance of excitation frequencies and natural frequencies of the mechanical structure will be avoided. Avoiding of resonance phenomena for both fluid and structure borne vibrations is the most important potential for noise reduction.

The power unit is well optimized regarding noise when its overall sound power level (SPL) is close to the sound power level of the pump motor group. The potential for noise reduction of the power unit can be estimated by comparing a sound power level of the power unit with the SPL of the pump motor group.

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