Investigation of the Dynamic System Behaviour of Pump Cases

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Abstract

Hydraulic pumps convert mechanical power into hydraulic fluid power. During this process, alternating forces in hydraulic pumps incite vibrations in the pump case and are responsible for the structure-borne noise. Mainly dynamic forces in bearings are the source causing structure-borne noise which generates finally air-borne noise. For this reason, those dynamic forces have been investigated based on a dynamical model. The structure borne noise has been measured on the pump case surface. The natural vibrations of the pump case have been determined by the Finite Element Method for several boundary conditions. The comparison between calculated and measured natural vibrations indicates that the coupling of the pump parts influences the dynamic behaviour of the case.

Keywords: *Machine Tool, hydraulic pump, vibrations*

Introduction

The optimization of pumps concerning noise emission is done by influencing the excitation, i.e. by optimizing the pressure changes inside the pump. This important method is limited by the design of the pumps and efficiency ratio and further noise reduction is not possible in many cases. The experimental results show, that noise generation is mainly influenced by internal forces in the pump and the response of the case [1,2,3]. Here, the transmission of vibrations inside the pump as well as the behaviour of the whole structure of the case is regarded. With the discribed investigations, methods for the minimization of structure-borne noise and air-borne noise are developed for selected pumps by changing the dynamic system behaviour. Problems of structure-borne noise based on external gear pump are discribed in this paper.

Excitations of Gear Wheels

To simulate the dynamic processes, which occur within an external gear pump, excitations caused by the action of the gear wheel pump have to be regarded. Variable forces are working on the gear wheels of external gear pumps. These forces caused by the conveying pressure and interaction of the gear wheels.

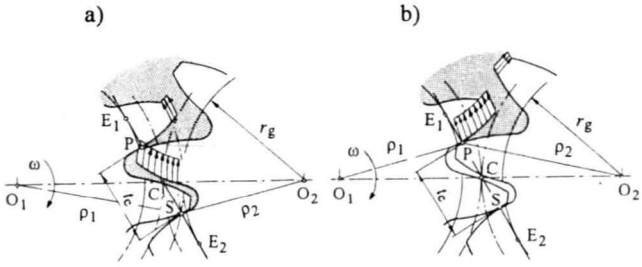

Fig. 1: Scheme for the determination of load alterations in the meshing zone

The torque and the radial forces occuring in connection with the conveying pressure change periodically and depend on the position of the meshing point seperating the pressure side from the suction side. Fig.1 shows two characteristic positions of this point, where both, torques and radial forces, change. The suction area is connected to the pressure area by releasing grooves, as long as its volume is decreasing. During the remaining phase of the double mesh it is connected to the suction area. That prerequisite is tolerable in case of releasing grooves, which are optimally designed [4].

The area of trapped volume in Fig. 1a is connected to the pressure area. Thus, the meshing point S of the previous tooth pair seperates the pressure area from the suction area. According to the connection of trapped volume to the suction area, the meshing point P seperates the pressure area from the suction area (Fig. 1b). The radial loads of the gear wheels result of the pressure distribution at their circumference.The experimental tests of the pressure distribution at the circumference of the gear wheels confirm the behaviour as shown in Fig.2. Three areas are to distinguish:

- the suction area Φ_1
- the build up pressure area Φ_2 and
- the pressure area with constant proportion (Φ_3) and variable proportion (Φ_3')

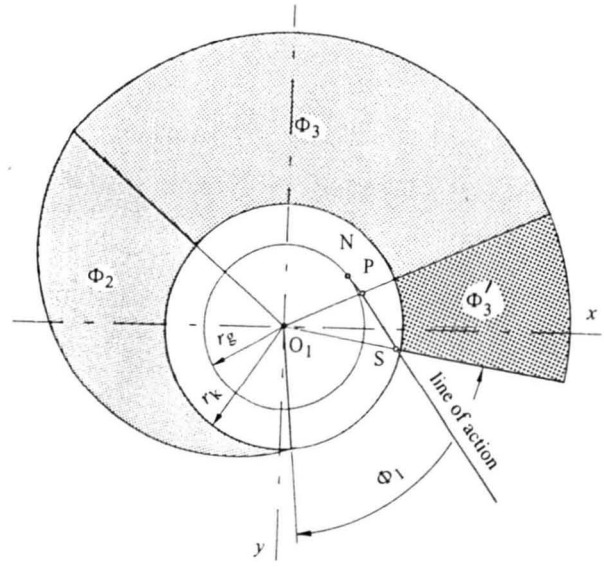

Fig. 2: Pressure distribution at the gear wheel circumference

The values of the different areas depend on the construction of the pump. In comparison to pumps with compensation of the radial clearance, pumps with no compensation have an pressure area composition Φ_2 expanding to several teeth. The range of this area depends also on the operating parameters of the pump

[2]. Calculating the radial forces, the range of the areas Φ_1 and Φ_2 can be determined based on the measured pressure distribution at the gear wheel circumference. Alternations of the sealing points position cause periodical alterations of the pressure area around Φ_3' . If the trapping area is connected with the suction area, the sealing point turns from point S to point P. This causes sudden changes of the radial forces. The resulting radial forces are found by integration of the pressure over the circumference. The way how to calculate the torques and radial forces due to the conveying pressure is explained in [5].

The method for calculating the pressure ripple of external gear pumps is described in [6]. The torques and the radial forces caused by the conveying pressure were calculated with and without pressure ripple. As it was noticed, the influence of the pressure ripple on time dependent torques and radial forces inside the pump decreases by increasing the conveying pressure. Concerning higher static pressure (p > 5MPa) alterations depending on changes of the sealing point location are most important. The influence of the pressure ripple, however, is meaningless.

Dynamic model of gear pumps

As mentioned above, the alternating torques and the radial forces, influence on translational and rotational vibrations of both gear wheels. These vibrations are coupled together by the tooth system. Thus, the whole system is a coupled dynamic bending-torsion-system. The dynamic model of the gear pump is described in Fig. 3.

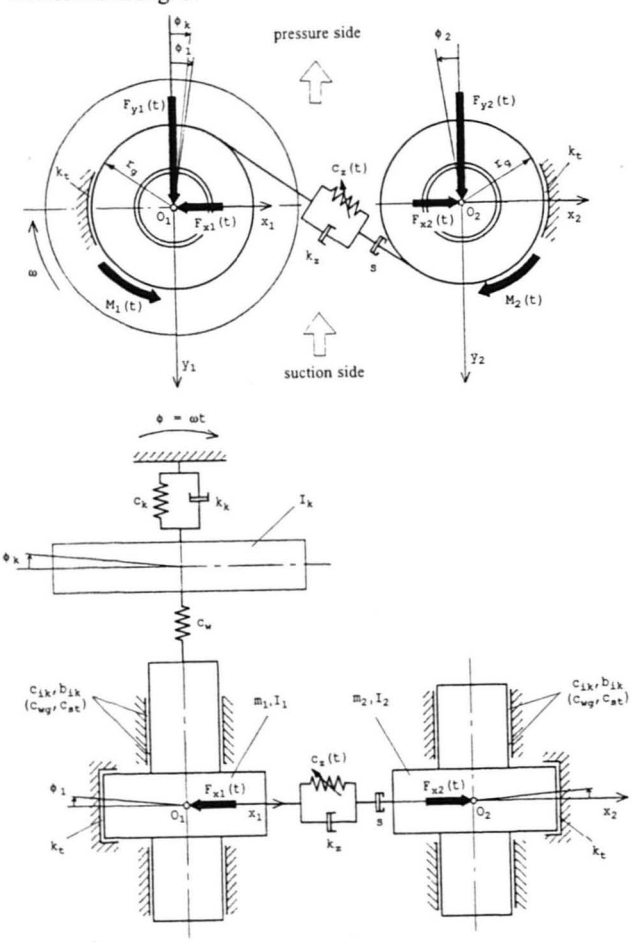

Fig. 3: Dynamic model of gear pump

The torsional damping is considered by the help of factor k_t and results from frictional effects of the fluid film between the gaps of the pump. It is assumed to be equal for both gear wheels. Further it has a character of external damping and causes the torque loss at the gear wheels.

The gear wheels are displaced excentrically by the static conveying pressure to the suction chamber. Those shiftings depend on the bearing clearance, on the flexibility of the bearings and on the bending of the shafts. Fig.4 shematically shows the displacements of the shaft center x_w, y_w basing on the position of static balance.

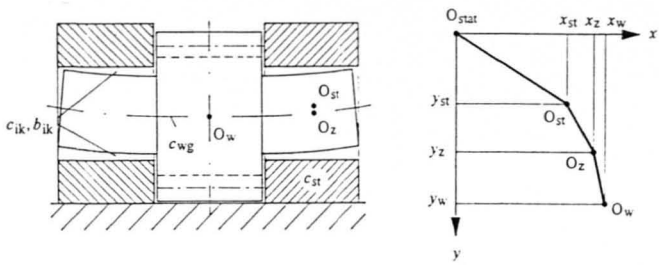

Fig. 4: Displacement of the gear shaft center

The absolute displacement $O_{stat}O_w$ of the shaft center is divided in the displacements $O_{stat}O_{st}$, $O_{st}O_z$ and O_zO_w resulting from the elasticity of the bearing bush, from the elasticity of the oil film and the bending elasticity of the shaft, respectively.

In consideration of added superposed vibrations the properties of the slide bearings are linearized around the position of static balance on condition of small oscillations. They are determined by four stiffness coefficients and four damping coefficients c_{ik}, b_{ik} (i,k = 1,2) , respectively. This prerequisite is permissible because of high static loads of the gear wheels. The stiffness and damping coefficients are mainly dependent on the load and the geometry of the bearing, its clearance and the oil viscosity [7]. They were calculated by the help of a computer program with the data of the sliding bearings of a pump.

The following equations describe the components of the dynamic bearing forces [7]:

$$F_{Lxi} = c_{11}(x_{zi} - x_{st}) + c_{12}(y_{zi} - y_{st}) + b_{11}(\dot{x}_{zi} - \dot{x}_{st}) + ;b_{12}\left(\dot{y}_{zi} - \dot{y}_{st}\right)$$

$$F_{Lyi} = c_{21}(x_{zi} - x_{st}) + c_{22}(y_{zi} - y_{st}) + b_{21}(\dot{x}_{zi} - \dot{x}_{st}) + ;b_{22}\left(\dot{y}_{zi} - \dot{y}_{st}\right) \qquad (i = 1,2)$$

The dynamic bearing forces are transmitted to the pump case and are therefore cause directly pump case oscillations.

The differential equations of movement were solved by a digital simulation. The data of an external gear pump with an axial clearance compensation were provided for the calculations. A part of the required model parameters as the mass inertia moments $J_k, ;J_1, ;J_2$, the torsionial stiffness c_w , the bending stiffness c_{wg} of the pump shaft and the time dependent tooth system stiffness $c_z(t)$ were calculated. The residual parameters as the torsional stiffness and damping factors of the coupling c_k, k_k , respectively, the torsional damping factor k_t and the damping

factor of the tooth system k_z were determined by experimental tests.

As it can be concluded from the courses of the dynamic bearing force components (Fig. 5), the bending oscillations with the high natural frequencies are stimulated by the changing pressure force in the meshing zone. The causes of the high amplitude of the excitation are slightly damped bending oscillations of the shafts. The differences between the components in x- and y-direction for the driven and the driving gear depends on the speed and is caused by phase relations of oscillations and by different reduced masses of both gear wheels. The amplitude of the oscillations mainly depends on harmonics of the exciting forces, on the bending stiffness of the shafts and on the stiffness and the damping coefficients of the sliding bearings.

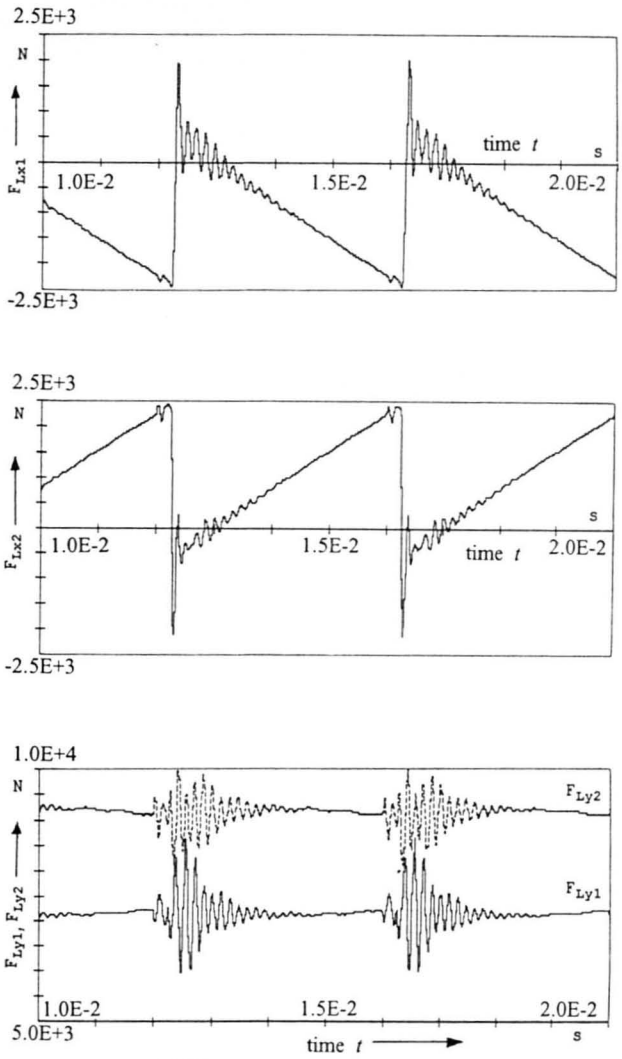

Fig. 5: Calculated dynamic bearing forces for both gears in x- and y-direction

Fig. 6 shows the results of structure-borne noise measurements at the pump case, in x- and y-direction for the driving gear. It can be seen, that the higher frequency oscillations of the pump case are excited by the period $T = 1/f_z$ (f_z - meshing frequency). The time dependent courses of the velocity at the pump case are thus equivalent to the courses of the calculated dynamic bearing forces. That qualitative comparison clearly shows the validity of the presumptions concerning the excitation and the model parameter.

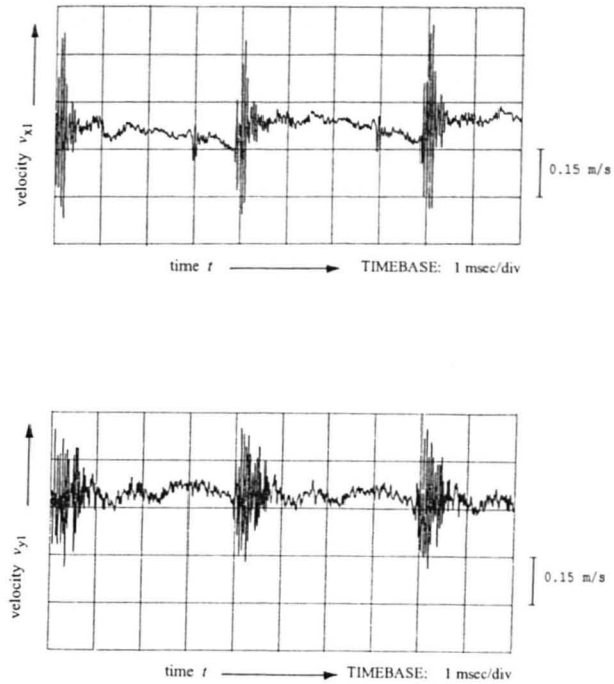

Fig.6: Measured velocity of structure-borne-noise at the pump case in x- and y-direction

Dynamic system behaviour of pump case

The transmission of the vibration energy from the gears to the case is influenced by its natural vibrations. With the Finite Element Method, the natural frequencies and shapes of the case can be calculated. On the supposition that the mounting flange is infinitely rigid, the pump case carry out the bending vibrations by the first and second natural frequencies and torsional vibration by third natural frequency. In Fig. 7, the first mode shape is presented existing at 2000 Hz.

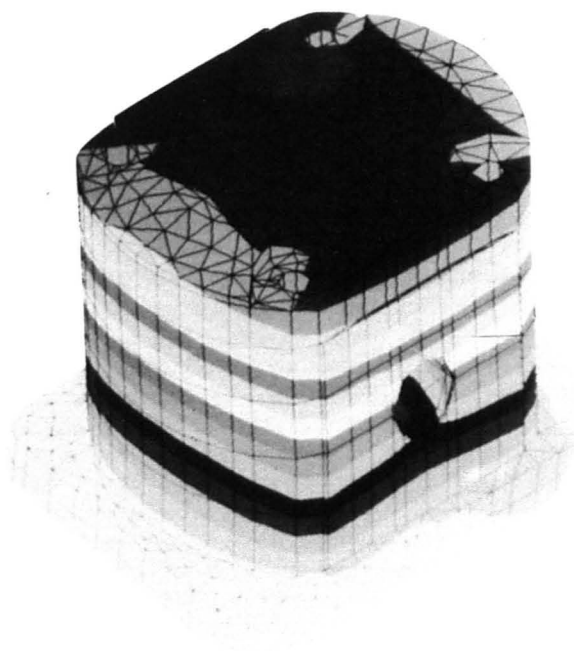

Fig. 7: First mode shape of the gear pump case at 2000 Hz

The results of calculations are strongly dependent on boundary conditions, e.g. mounting, connection of the single case parts. For this reason the bounduary condition have to be set exactly to achieve results comparable with the measurements. The mounting flange oscillate together with the mounted pump case. The vibration behaviour of the flange with the mounted flange have to be identified by measurements and FEM analysis.

The contact conditions between the pump case parts are of great importance. The range of the contact areas were found by using the Finite Element Method. The contact analysis was performed with regard to the static conveying pressure and the static bearing force. The obtained results were verified by mesurements using pressure measuring films, and they were taken into account for the modelling of the connection points of the pump case parts. These boundary conditions are required for the dynamic calculations.

The natural frequencies and mode shapes can be measured by modal analysis. By an in-process-analysis of vibration the shapes and the frequencies of the case during operation can be shown and compared with the excitation of the case. Those ranges of frequencies, which are responsible for the noise emission, have to determined. Fig. 8 shows spektra of sound levels and acceleration levels of case vibrations. For these measurements in an acoustically dead room the accelerating transducer and the measuring microphon were placed at the suction side. The conveying pressure was kept constant while the number of revolutions was varied continously for the Tracing Analysis.

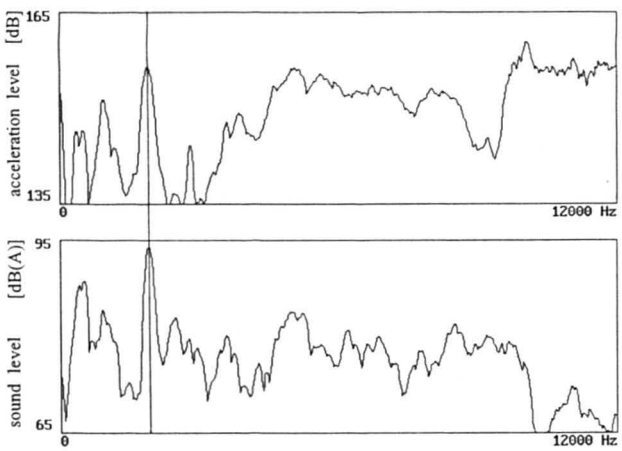

Fig. 8: Spektra of sound levels and acceleration levels of case vibrations.

As it can be noticed in Fig. 8, the highest peak of sound level lies at 1800 Hz. The acceleration level shows a peak at the same frequency, too. These results were confirmed by an Order Analysis of air-borne noise and structure-borne noise. Therefore, natural frequencies being mainly responsible for air-borne noise can be determined and then can be taken into account for further considerations.

The purpose of further investigations is the simulation of the forced vibrations of the pump case. These can be determined e.g. by the harmonic-response method. By doing so, quantitative values of the forced vibration amplitudes of the case are achieved and then can be compared with the results of the mesurements. Therefore, due to the vibrations at the external case surface, the air-borne noise can be determined theoretically. Actually, the Finite Element Method programs for solving acoustic problems can be used. The calculated amplitude peaks of sound levels then can be compared with the measured peaks.

Summary

The investigation presents a procedure to analyze the dynamic behaviour of external gear pumps. A dynamic model of the external gear pump has been presented. The digital simulation of the model makes possible specific investigations of the particular influences of the pump operating and construction parameters on the internal dynamic loads. The analysis of the excitation shows, that the alterations of the gear wheel loads caused by the conveying pressure are the most important excitations. Those excitations are caused by sudden changes of the sealing point position between pressure area and suction area. The amplitudes of the exciting forces directly depend on conveying pressure. The transmission of the gear wheel oscillations to the pump case causing of structure borne noise is determinate by damping-elastic properties of the sliding bearings. The higher frequency oscillations are excited by slightly damped bending oscillations of the gear wheels. This leads to strong deviations of the dynamic bearing forces and the generation of structure-borne noise in the case. The experimental tests confirm the usefulness of the explained dynamical model for the analysis of the dynamic behaviour. Actually, the natural frequencies and mode shapes of the pump case are investigated by the Finite Elements Method as well as by measurments. The frequency range including the natural frequencies of the case will be compared with the excitation spectra. Changing the case design , e.g. mass distribution, wall thickness and ribbings, an adequate distance between excitation frequencies and natural frequencies can be achived to avoid resonances. Finally, the pump case is optimized referring to noise emission. In general, this method of optimization can be applied to other types of pumps.

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