Investigation of dynamic pressure fluctuations in closed hydraulic circuits as a function of centrifugal pump speed gradients

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Abstract. Electrical consumers, such as pumps, contribute towards the stable operation of the electrical power grid due to active and reactive power consumption. If the consumers are indirectly connected to the grid via power electronic converters, the positive contribution is reduced or non-existent. Consequently, the ability to compensate imbalances within the power grid is greatly reduced. A research project founded by the European Union is focusing on investigations to utilize power converter technology of pump drives for grid stabilization. But in order to stabilize the fluctuating power grid frequency rapid adjustments of the pump speed are required, which can cause major troubles in the hydraulic system. For instance, rapid changes in pump speed are causing the emission of hydro acoustic pressure waves into the pipe system, which are exciting the structure of the hydraulic system and lead to undesirable vibrations and noise emissions. Most critical is the direct excitation of natural frequencies in the entire piping system based on the inherent pressure fluctuations caused by transient fluid flow and pump speed changes. Converter control for grid-adaptive speed adjustment is only possible as long as there is no significant increase in dynamic pressure fluctuations in the whole hydraulic system. For this purpose, a research pump is operated with different rotational speed gradients in a closed hydraulic circuit. Dynamic pressure sensors were used to record the resulting pressure fluctuations at various locations of the hydraulic system. Additionally, an outlook towards an active reduction of the dynamic pressure fluctuations by means of a specially developed hydrodynamic actuator is given. These results are an initial step but of particular interest towards the knowledge of permissible grid adaptive speed control regimes, which can be applied to a pump in a hydraulic piping system.

1. Introduction

The power consumption of pump drives is normally adopted to the requirements of the hydraulic system supplied by the pump. The reference variables of this control are mainly the head and the volume flow. Water pumps, in particular circulators, as used in heating and air conditioning systems, have been connected to the electrical power supply system mainly by means of frequency converters since the EU directive 2009/125/EC came into force. In many cases, permanent magnet synchronous motors are used as drives, which enables the pumps to be adopted to the requirements of the hydraulic system in a very efficient and energy-saving manner. Hereby, the electric drive of the pump is decoupled from the electric power supply. This means that the pump rotational speed no longer depends on the grid frequency,
instead it is controlled by the frequency converter. As a result, its power consumption is no longer influenced by the frequency of the electrical power grid. Yet, the European power grid has only very low storage capacities and thus the balance between generated and consumed electrical power is negatively affected in the long term. As described in [1], it is possible to adapt the power consumed by the converter, and thus the power of the electric drive and pump, based on the power available in the electric grid. The electrical grid frequency is used as the reference variable for this purpose as it represents the balance between generated and consumed electrical power in the power supply system. The electrical grid frequency is measured at the connection point of the converter and the difference $\Delta f'$ to the desired constant grid frequency of 50 [Hz] is determined. Deviations of $\Delta f'$ in the range of $\pm 2.5$[Hz] are investigated. The speed $n$ of the drive and therefore the electrical power consumed by the converter is adjusted in proportion to the frequency deviation by means of a rotational speed difference $\Delta n'$:

$$\Delta n' = k_{nf} \times \Delta f'$$  \hspace{1cm} (1)$$

The normalized values between 0.01 [pu/Hz] to 0.08 [pu/Hz] are assumed for the amplification factor $k_{nf}$ wherein the normalization factor pu (per unit) is based on the nominal speed of the pump. For a pump with a nominal speed of 1450 [rpm] this results in rotational speed difference $\Delta n'$ in the range of 36 to 290 [rpm]. Taking into account, that the grid frequency statistically changes with a frequency between 0.2 [Hz] and 1.5 [Hz] as shown in [2] and assuming a linear change of the rotational pump speed with a speed difference of 290 [rpm] results in speed gradients up to 1740 [rpm/s]. At this point the interaction between the change of the rotational speed and the fluid dynamic processes within the hydraulic system has not been considered, although it results in unsteady flows in the hydraulic system as it can be observed when the pump speed is rapidly increased from or decreased to rest. This causes pressure fluctuations which spread throughout the hydraulic system at the speed of sound. These pressure fluctuations can excite structural system resonances and lead to the radiation of airborne and structure-borne sound and may result in structural damages [3].

Pressure fluctuations are also generated by the inherent design of centrifugal pump impeller with a finite number of blades interacting with the tongue. The blade passing frequency (BPF or $f_b$) and its higher harmonics is a function of the number of blades $N$ and the speed $n$ of the pump [4] and have the highest shares in the power spectrum of the pressure pulsations.

$$f_b = N \times n/60$$  \hspace{1cm} (2)$$

In addition to the pressure fluctuation caused by the design of a radial centrifugal pump, steep rotational speed gradients result in additional pressure fluctuations as described in [8-13]. In the publications the transient behavior of centrifugal pumps in open and closed circuits was investigated. The pumps were started up and shut down with various rotational speed gradients. As a result, the volume flow and the head are evaluated but no conclusion about the pressure fluctuations have been drawn. To determine a gain factor $k_{nf}$ as a function of speed and frequency, this system-dependent variable must be known. To avoid structural damages of the hydraulic system and to reduce vibration amplitudes the dynamic pressure fluctuations caused by the pump operation must be known and may be actively or passively reduced. The aim of passive damping elements is to reduce the resulting pressure fluctuations near to its source. In [5] the principle is described that it is possible to dampen certain frequencies of pressure fluctuations with accumulators. Accumulators acting similar like low pass filter. In [6] a Helmholtz resonator is described which is lined with a urethane layer. Different combinations of urethane layers were investigated. It was shown that different low frequency bands could be damped depending on the combination of different urethane layers.

A changing operating point due to control processes or rapid speed changes lead to a shift in the frequency and amplitude of the dynamic pressure fluctuation. In order to be able to react to this change, an active control of the damping is necessary. In [7] a reduction of the pressure fluctuation according to the principle of active noise cancellation is described. For this purpose, a destructive pressure fluctuation is generated via a piezo valve as a function of a speed signal and coupled into the main flow via a T-
junction. It has been shown that this method in interference with the pulsation generated by the pump reduces the total pressure fluctuation very well. Currently, active noise cancellation (ANC) systems, which create adaptive interfering pressure fluctuations are not available. Within this paper a hydrodynamic actuator is presented and will be discussed towards its ANC applicability.

2. Objectives
Within this research work the influence of various pump speed gradients are analyzed. For this purpose, the dynamic pressure fluctuations are recorded and evaluated at different locations in a closed hydraulic circuit. Within each measurement three consecutive steps are conducted:
1. linear increase of pump speed from rest up to its nominal pump speed
2. constant operation at pump speed
3. linear decrease of pump speed from its nominal speed down to rest
The pump speed gradient in the 1st and 3rd step is equal. Various pump speed gradients are used in order to reveal their influence towards the pressure fluctuation amplitudes for the actual combination of pump and hydraulic circuit.
Additionally, the development of a highly innovative idea for an active reduction of pressure fluctuations using a hydrodynamic actuator is presented. The possibility of extending the control limits are discussed on the basis of already known research results from the field of active noise cancellation. Furthermore, first measurement results for pressure fluctuations generated by the hydrodynamic actuator are shown and compared to pressure fluctuations caused by the pump in operation.

3. Test Equipment and Methods

3.1. Experimental circuit with centrifugal pump
The closed hydraulic circuit of the anechoic chamber of the institute is used as the experimental system. It is a closed circuit operated with water, as it is found in typical pump applications. Multiple dynamic pressure sensors are applied at various locations of the hydraulic circuit in order to measure pressure fluctuations (cf. Figure 1). The radial pump used is a test pump which has a specific speed of \( n_q=13.4 \ [\text{rpm}] \) at a speed of \( n=1450 \ [\text{rpm}] \). The pump is driven by a servo motor with 17[kw] at 1500[rpm]. The pressure drop at the valve register is adjusted so that the pump delivers a volume flow of 64.5[m³/h] at its point of best efficiency. The closed circuit with test pump and valve register is shown in Figure 1. The servo motor is controlled by a SINAMICS S210 frequency converter. The analog input is used to control the converter. The electrical analog signal is generated by an NI-PXI system with a function generator card (NI PXI-5402). The output card is controlled by Labview script.

| Table 1. Parameter measurement campaign |
|----------------------------------------|
| Parameter                              | Value            |
| Measurement sample rate (3050-A-060)    | 16384 [Hz]       |
| Measuring time                         | 41 [s]           |
| Incremental disk                       | 70 [ppr]         |
| Switching frequency (BES00M4)          | 3500 [Hz]        |
| Dyn. pressure range (S112A22)          | ±345 [kPa]       |
| Frequency range (S112A22)              | 0.5 Hz-250 [kHz] |
| Static system pressure                 | 550 [kPa]        |
| Water temperature                      | 23 [°C]          |
| Flow rate at 1450 rpm                  | 64.5 [m³/h]      |
| Rotational speed range                 | 0 – 1450 [1/min] |
| Rotational speed gradient              | 100 – 3400 [rpm/s] |
| Step size rotational speed gradient    | 50 [rpm/s]       |
The dynamic pressure is measured in the suction pipe, in the pressure pipe and on the roof of the anechoic chamber by means of acceleration-compensated dynamic pressure sensors (PCB S112A22). The pressure sensors are radially integrated in the pipe wall. As can be seen in Figure 3, care was taken to ensure that the flow can pass the active sensor surface without disturbance. The positions of the measuring points are marked in Figure 1. The static pressure is measured with a pressure sensor (WIKA P-30). The angle of rotation of the pump shaft and the speed is measured by means of an incremental disc. The disc has two incremental ranges, one for detecting the zero position and one for detecting the angle of rotation. The increments are measured with two inductive sensors (BES 516-3044-G-E4-C-PU-05) as shown in Figure 2. The volume flow rate prevailing in the hydraulic circuit is recorded by a magnetic inductive flowmeter (KROHNE Optiflux 2300). The pressure signals, the speed signals and the volume flow rate are recorded with a Brüel & Kjaer LAN-XI measuring system with a measuring card (3050-A-060).

Figure 1: Hydraulic circuit of the anechoic chamber with pump and actuator

Figure 2. Pump shaft with incremental disc

Figure 3: Pressure sensor pressure pipe
3.2. Hydrodynamic actuator

A hydrodynamic actuator was developed to reduce the pressure fluctuation in the pressure pipe of the hydraulic circuit. The actuator is based on the principle of a piston radiator. The generated pressure fluctuation should enter the main flow as a plane wave front from the piston surface. The alternating pressure \( p \) or the sound pressure for a plane wave field can be calculated from the piston velocity \( v_0 \), the angular frequency \( \omega \) and the characteristic impedance \( \rho_0 c \) of the medium:

\[
p = \rho_0 c \cdot v_0 \cdot \sin \omega t
\]

The design is only justified if the excitation takes place below the cut-off frequency of the first transverse mode and if the pipe is acoustically narrow and has a reflection-free outlet. The cut-off frequency \( f_{1,0} \) for the first transverse mode can be calculated as follows:

\[
f_{1,0} = \frac{1}{2 \pi r_i} \cdot \frac{c}{f_1}
\]

\( r_i \) is the inner radius of the pipe. Assuming a sound velocity \( c \) of 1300[m/s] and an inner pipe radius of 33[mm], the cut-off frequency is about 11.5[kHz]. With a target frequency of the blade passing frequency of approx. 169.2[Hz] at 1450[rpm] and a blade number of seven, it is ensured that one is well below the cut-off frequency. The assumption that the outlet is free of reflections is not completely fulfilled. The actuator is driven by a piezo actuator (P-025.10P). The piezo is connected to the piston via a lever. The static pressure is compensated by a spring. The spring force is adjusted in such a way that the piezo receives the required preload. To minimize disturbance of the main flow, the actuator is connected to the pressure pipe via a T-junction with 30 degrees. The construction and the schematic control can be seen in the Figure 4 and 5.

![Figure 4. Schema of a hydrodynamic actuator with amplifier and control as well as the measuring system](image)

![Figure 5. Hydrodynamic actuator with amplifier](image)

4. Results

4.1. Evaluation of pump speed gradients

Figure 6 shows the speed curve and the resulting dynamic pressure in the pressure pipe at sensor PD_1. As shown in the figure, the speed is increased from 0 to 1450[rpm] within 0.725[s]. This corresponds to a rotational speed gradient of 2000[rpm/s]. This is followed by a phase in which the speed is kept constant and the dynamic pressure of the BPF dominates. From second 21 the rotational speed of the pump is reduced again with the same gradient.
During start-up as well as during shut-down it is obvious that there are peak values in the dynamic pressure. These dynamic pressure peaks will decay after a few oscillations. Whereby the decay time is not considered for the time being. However, it can be seen that the decay time at start-up is shorter due to the subsequent constant operation of the pump.

![Dynamic pressure / rotational speed ramp-up and down](image)

**Figure 6.** Speed curve and dynamic pressure during measurement with a gradient of 2000 rpm/s

This behavior has already been described by Ismaier and Schlücker [14] with regard to the damping of a pressure surge. The dynamic pressure fluctuations generated by the BPF are clearly visible in the constant speed range (cf. Figure 7).

![Dynamic pressure PD 1 constant speed (1450 rpm)](image)

**Figure 7.** Dynamic pressure by the BPF at constant speed in the pressure pipe

Measured dynamic pressure fluctuations are always related to the current quasi static time averaged pressure at the sensor location due to the characteristics of the electrical circuit of the used dynamic pressure sensor. Since the pressure fluctuations generated by the pump impeller are an inherent quantity, it can be assumed that significant problems will only occur if these amplitude levels are significantly exceeded. The positive amplitude of the dynamic pressure is particularly critical. Figure 8 shows the dynamic pressure peaks during start-up or shut-down in relation to the basic amplitude level due to BPF (cf. Figure 7). Here, the maximum peak amplitudes of the dynamic pressure fluctuation due to start-up and run-down are taken and marked as red and blue circles in Figure 6. The course of these peak values is plotted as a function of the pump speed gradient. The dynamic pressure at the sensor locations PS_1, PD_1 and PDach_Li has been evaluated in the same manner (cf. Figure 1). It is observed that there is a strong local dependency of the resulting pressure fluctuation based on the pump speed gradients.
The resulting pressure peaks strongly dependent on whether the pump speed is accelerated or decelerated. In the suction pipe (sensor PS_1), large maximum amplitudes of the dynamic pressure can be seen even at small pump speed gradients. Due to the kinetic energy of the fluid mass, a large peak occurs, similar to the event of a fast valve closing. In the pressure pipe (sensor PD_1), the maximum peak does not exceed the basic amplitude level of the BPF until a gradient of 2000 [rpm/s]. For even smaller gradients the dynamic pressure level is dominated by the BPF. This behavior cannot be transferred to the negative gradients. For negative speed gradients the passing of the basic amplitude level already occurs at -1000 [rpm/s] within pressure pipe. The pressure sensor (PDach_Li) shows a similar behavior like the sensor PD_1. However, the level is lower due to energy dissipation in pipe redirections and along the pipe length. It is assumed that the amplitude level of approx. 25[kPa] of the BPF in the pressure pipe does not lead to significant vibrations of the hydraulic system. This level is only exceeded when the rotational pump speed gradient is above 2000[rpm/s]. On the other hand, the maximum amplitude of 25[kPa] is exceeded at a negative gradient of approx. -400[rpm/s], especially in the suction pipe.

Figure 8. Maximum pressure peaks versus dynamic fluctuation due to BPF
4.2. Active Reduction of dynamic pressure fluctuations

In order to reduce the pressure fluctuation transferred into the system by the pump or quickly damp out strong transient pressure peaks by active noise cancelling (ANC) control processes, it has been experimentally proven that it is possible to reproduce the pressure fluctuation caused by the pump with a hydrodynamic actuator. In Figure 9 the dynamic pressure generated by the pump and the hydrodynamic actuator are compared to each other.

![Figure 9: Generated dynamic pressure by pump (top) and hydrodynamic actuator (bottom)](image)

The two pressure signals already show the same characteristic and consequently the same frequency. By adjusting the pressure amplitude and phase generated by the hydrodynamic actuator a destructive pressure interference field, as applied in active noise cancelation (ANC) systems, may be achieved as described by Pen in [7]. In Figure 10 first results towards an active reduction of the pressure fluctuations are shown. Within this test case a constant speed of 1000[rpm] is set, which corresponds to a BPF of 116.7[Hz], and a destructive pressure fluctuation generated by the hydrodynamic actuator is induced through a T-junction (cf. Figure 5). The generated pressure fluctuation is modelled as a sinusoidal signal with the corresponding frequency of 116.7[Hz] and the maximum accessible amplitude of the current actuator design. The phase of the sinusoidal signal has been manually adapted to minimize the measured pressure fluctuation right behind the T-junction at sensor location PD_2. Figure 10 (left) shows the amplitude response spectra with and without the actuator, which refers to “ANC on” or “ANC off” respectively. Figure 10 (right) shows the difference between the two amplitude spectra. Here, a reduction of the amplitude by approximately 8.3% of the first BPF can be achieved.

Currently, constructive measures are taken to increase the maximum dynamic pressure fluctuation amplitude of the hydrodynamic actuator. Furthermore, numerical and experimental research work is conducted to implement a control algorithm for phase and amplitude adaption to achieve an effective ANC system for arbitrary combinations of pumps and hydraulic systems and to address multiple harmonics of the BPF. Utilizing the hydrodynamic actuator with a control system would allow to extend the operating range for pump speed gradients as maximum peaks and inherent dynamic pressure fluctuations will be reduced.
5. Conclusion

In summary, it can be concluded that dynamic pressure fluctuations can be recognized for stationary operating conditions of a centrifugal pump and especially also for rapid speed changes. The investigation lead to the following conclusions:

- The amplitude of the pressure fluctuation due to a change in rotational speed depends on the rate at which the change is made, namely the pump speed gradient.
- The maximum and minimum peaks of pressure fluctuation depend on the observed location. The highest values are not only to be expected on the pressure side but also on the suction side of the pump.
- The maximum peak of the pressure fluctuation depends on whether the pump speed is increased or decreased. When the pump speed is increased critical dynamic pressure peaks occur at larger absolute speed gradient values as compared to the case of decreasing the pump speed.
- In order to determine a permissible amplification factor \( k_{nf} \) it is necessary to know the system-dependent relationship between rotational speed gradients, magnitude of the speed change and the resulting dynamic pressure fluctuations and their decay time.
- With a hydrodynamic actuator and fast control systems, it will be possible to generate pressure fluctuation which has a destructive influence on the pressure fluctuation created by the pump. If the dynamic pressure fluctuations can be reduced effectively, the importance of knowing the system-related variables could become obsolete. Therefore, further research is focused on the control and the increase of the performance of the hydrodynamic actuator.

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