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LEAKAGE LOCATION IN HIGH PRESSURE EXTERNAL GEAR PUMP

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Abstract: *This paper presents determination of flow ripple conducted by two pressures/two systems methods in accordance with ISO 10767-1: 2015. The tests were done for an external gear pump with an involute profile without backlash. The tested pump had a technological error in involute tooth profile. On the recorded flow ripple, a negative peak is visible - an internal leak which is caused by a short-time gap formed during gears meshing. Such an error in involute tooth profile leads to rapid wear of the gear teeth surface, which has been presented.*

Keywords: External gear pump, flow ripple, fluid-born noise.

1. Introduction

Flow ripple is the main factor of noise and vibration in hydraulic systems. The reduction of flow ripple is one of the leading directions for the development of external gear pumps. One of the parameters describing the size of the flow ripple is the flow nonuniformity coefficient (Osiński, 2013):

$$\delta = \frac{Q_{max} - Q_{min}}{Q_t} \tag{1}$$

Flow ripple is very difficult to measure directly, however, there are methods for experimental determination. There are two methods described in the literature that allow to determine the flow ripple by measuring the pressure ripple. The first is the secondary source method (Edge, 1990), which strongly complicates the measurement system and the second - two pressures/two systems (Kojima, 2000) on which the ISO 10767-1: 2015 standard is currently based. In the methods for determining the parameters of the source based on the pressure ripple in the system, the main problem is the correct determination of the speed of sound in the measurement system.

Measurement of the flow ripple can be used to determine the effect of various parameters, e.g. working fluid temperature (Ichiyanagi, 2017). Another application may be verification and improvement of pump parameters (Johansson, 2007).

2. Methods and experimental condition

The determination of the flow ripple Q_S of the external gear pump was conducted in accordance with ISO 10767-1: 2015. Figure 1 shows a simplified measuring system with specified elements having a significant influence on the measurement. On the rigid steel pipe with length L_r, two PCB M101 series dynamic pressure sensors (7, 8) were installed, from which a pressure signal was recorded on the Brüel & Kjær measuring device (16). The measurement was recorded twice for a given discharge pressure, loading the system in sequence by the first (11) and second (12) loading valves.

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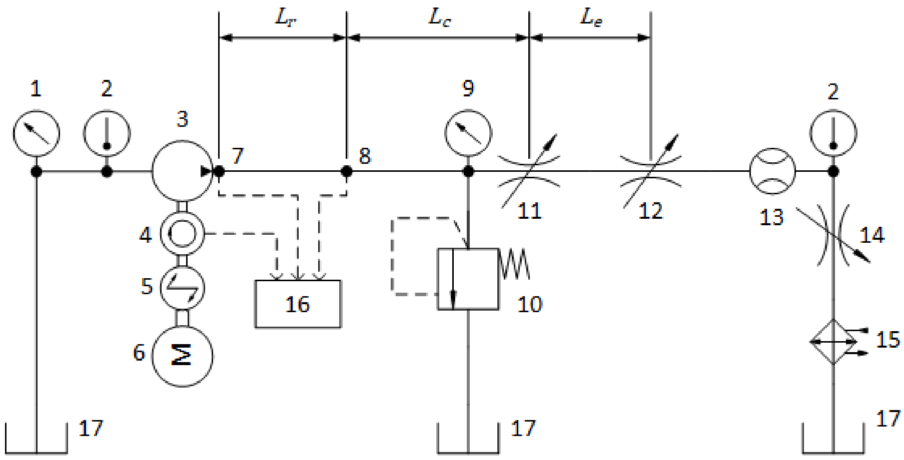

Fig. 1: The simplified measurement circuit diagram. 1 – pressure gauge, 2 – temperature indicator, 3 – tested pump, 4 – tachometer, 5 – torque sensor, 6 – electric motor, 7, 8 – dynamic pressure sensor, 9 – pressure gauge, 10 – direct operated relief valve, 11, 12, 14 – loading valves, 13 – flowmeter, 15 – cooler, 16 – amplifier and 24-bit recorder, 17 – tank

The flow ripple and the impedance of the source were determined according to the standard Norton model using the following:

$$Q_s = j\frac{1}{Z_c}\frac{P_0 P'_L + P'_0 P_L}{(P_0 - P'_0)\sin\left(\frac{\xi\omega L_r}{c}\right)} \tag{2}$$

$$Z_s = jZ_c\frac{(P_0 - P'_0)\sin\left(\frac{\xi\omega L_r}{c}\right)}{P_L - P'_L - (P_0 - P'_0)\cos\left(\frac{\xi\omega L_r}{c}\right)} \tag{3}$$

In the above formula, P_0, P_L represent the pressure in the frequency domain at the beginning and at the end of the reference pipe under load applied by the first valve (11). P'_0, P'_L represent the pressure when loading the system with the second valve (12). In addition, ω is the angular velocity, c the speed of sound, L_r the length of the reference pipe. The characteristic impedance Z_c and the complex coefficient of viscous friction ξ were determined as follows:

$$Z_c = \frac{\rho c \xi}{\pi r_0^2} \tag{4}$$

$$\xi \approx 1 + \sqrt{\frac{\nu}{2r_0^2\omega}} - j\left(\sqrt{\frac{\nu}{2r_0^2\omega}} + \frac{\nu}{r_0^2\omega}\right) \tag{5}$$

Where ν is the kinematic viscosity, ρ the fluid density, and r_0 the inner diameter of the reference pipe.

Knowing the source impedance Z_s and the source flow ripple Q_s, the modified flow ripple Q_s^* at the discharge port of the pump can be determined using the following equation:

$$Q_s^* = \frac{Z_s}{\sqrt{Z_s^2 - Z_c^2}} Q_s \tag{6}$$

The time history waveform $q(t)^*$ of the flow ripple Q_s^* can be obtained by the following formula:

$$q(t)^* = \sum_{i=1}^{10} |Q_{s.i}^*| \cos(2\pi f_i t + \angle Q_{s.i}^*) \tag{7}$$

The tested pump is an experimental low-noise high-pressure external gear pump, with displacement capacity of 24 cm^3. The gears have 12 teeth and do not have backlash. The reference pipe with inner diameter $r_0 = 19$ mm was connected to the flange type discharge port of the pump in which the first dynamic pressure sensor (7) was installed. At the distance $L_r = 98.2$ mm from the first sensor a second sensor (8) was placed. The working fluid was a mineral oil according to ISO VG 68. The temperature of the fluid during the measurement was approximately 45 $°C$. The measurements were conducted at a rotation speed of 800 rpm for discharge pressures 4, 12, 24 MPa, and at a back pressure of 1 MPa (Osiński, 2017).

3. Results

Figure 2 shows the theoretical and determined source flow ripple for different discharge pressures. The diagram shows a noticeable influence of the discharge pressure on the source flow ripple, while for a pump in good condition the influence of discharge pressure on the flow ripple is imperceptible. Particularly worth noting is the characteristic negative peak in performance in the initial phase of each ripple cycle. It is undoubtedly the result of an internal leak in the pump. This effect intensifies with the increase of the pressure difference between the suction and discharge area. This negative peak causes the characteristic is not smooth, but with pronounced oscillations of considerable amplitude. This effect unfortunately is the result of the influence of higher frequencies.

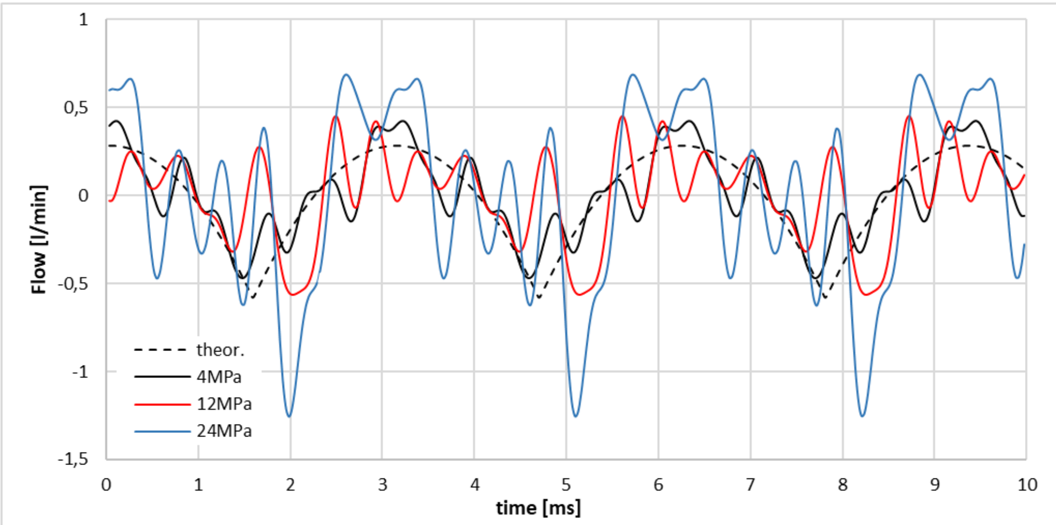

Fig. 2: Measured time history waveform of the source flow ripple at 800 rpm.

Table 1 presents the measured flow nonuniformity coefficient δ determined in accordance with the formula (1) and the volumetric efficiency η_v of the tested pump. It is worth noting that the volume efficiency of the pump drops to about 75 % at 24 MPa, which indicates the poor condition of the tested pump. The flow nonuniformity coefficient increases to 13.31 %, which is the result of both, an increase in the ripple amplitude ($Q_{\max} - Q_{\min}$) and a decrease in the mean flow Q_m. The increase in nonuniformity coefficient adversely affects the acoustic characteristic of the pump.

Tab. 1: Determined and theoretical flow nonuniformity coefficient and volume efficiency of the tested pump.

Parameter	**Theor.**	**Discharge pressure**		
		4 MPa	**12 MPa**	**24 MPa**
$Q_{\min}$ $\left[\frac{l}{\min}\right]$	-0,58	-0,47	-0,57	-1,26
$Q_{\max}$ $\left[\frac{l}{\min}\right]$	0,28	0,42	0,45	0,69
Q_{m} $\left[\frac{l}{\min}\right]$	19,3	18,7	17,6	14,6
δ [%]	4,47	4,74	5,77	13,31
η_{v} [%]	—	96,9	91,2	75,6

Visible cyclically occurring negative peak in flow indicates a temporary, noncontinuous occurrence of the gap. This type of leakage is clearly connected with the meshing process. The reason for the occurrence of this phenomenon may be: improperly made relief of the trapped volume, or an error in the gear tooth profile. Visual evaluation of the gears revealed the technological error in the involute profile. The effect of this error is a strong wear of the wheel in the region of the foot and tip, as shown in Figure 3.

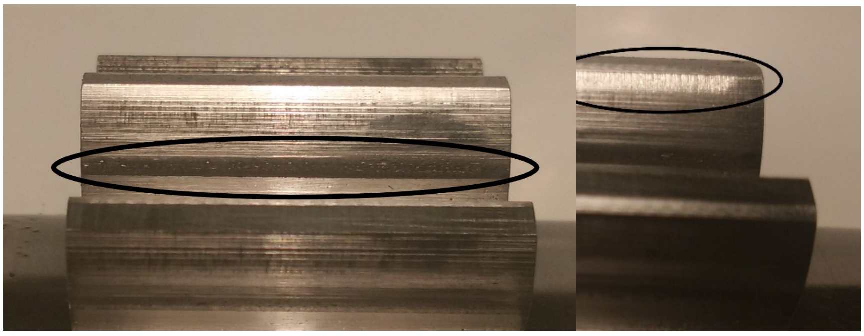

Fig. 3: Gear tooth with visible damage of foot and tip.

4. Conclusions

Conducted research indicate that the failure of the teeth in the gear causes a significant increase in the flow ripple. This increase brings increasment in pressure ripple. High values of pressure ripple amplitudes are the reason for higher torsional vibrations of gear wheels and additional sound-inducting vibrations of the casing. As a result, one should take into account the reduction of pump life and increased noise emission to the environment. Additionally, the measurements revealed a decrease in the efficiency parameters of the tested pump. For the highest considered discharge pressure, the volumetric efficiency dropped to almost 75%. An advantage of the presented test method is the ability to determine the timing of internal leak formation depending on the meshing stage without the need to register the rotation angle of the gears. The visual evaluation of wheel surface condition confirms the technological errors in the tooth profile.

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