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Original Research Article

Vibration and dynamic loads in external gear pumps

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ABSTRACT

This paper presents the model for simulation of vibration and dynamic loads in external gear pumps. The calculation has been carried out in Matlab/Simulink program. The vibrations of gears are excited due to the variable pressure forces and variable stiffness of the gearing. In this model the stiffness and damping coefficient of sliding bearings as well as the bending stiffness of gear wheels have been included. The influence of pressure and rotational speed on the dynamic forces in the bearings have been analyzed.

1. Introduction

External gear pumps are used in many applications, but their disadvantage is relatively high noise emission. As the causes of hydraulic noise generation are known, in particular, pressure and flow pulsation, trapped oil, pressure build-up and down and cavitation, most of these types of excitation can be largely compensated by various design measures [1,2]. Recent studies show that the pressure and flow pulsations are responsible for the noise and vibration generation in the hydraulic piping system, but not for the noise radiation of the pump itself. The intrinsic noise of the external gear pump is mainly dependent from the vibrations and dynamic loads on the gears [1]. These dynamic loads are transmitted through the bearings on the pump housing and cause vibrations and noise. In the paper the dynamical loads in an external gear pump have been evaluated by digital simulation of a dynamic model.

2. Vibration excitation forces on the gears

To analyze the dynamic processes in an external gear pump, the excitation forces, which are responsible for the vibration development of the gears, should be defined. Many papers are dealing with the definition of dynamic loads inside the sliding bearings of the gear pumps. The first approach has been published in 1990 [1], in which the excitation forces in the gear pump has been defined and the vibration model of the gearing has been described. The current papers regarding the dynamical behavior of gear pumps describe the flow and pressure distribution inside the gear pumps, excitations of gears due to the pressure and dynamical forces in the bearings are calculated [3]. These models include also the influence of the pressure ripples inside the pump, but as stated, this influence can be neglected at higher pressure.

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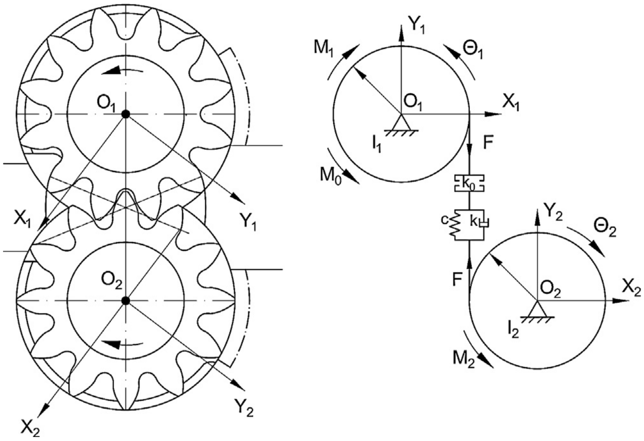

Fig. 1 – The dynamic model of the gearing [4–6].

In Fig. 1 the dynamic model of the gear pump shown in [4–6] has been described. This model includes the influence of the gearing on the reaction forces in the bearings.

2.1. *Torques on the gears due to the pressure*

The torque on the gears changes periodically depending on the position of the teeth contact point, which is sealing the delivery area from the suction area. In Fig. 2, two characteristic positions of this point have been shown at which the change of the torque as well as the radial forces on the gears occurs. In the range $0 < \varphi \le (\pi/z) \times (\varepsilon - 1)$ the trapped volume is connected with the pressure chamber. In the other phase of the double teeth contact it is connected to the suction chamber; thus, the contact point S (Fig. 2a) of the preceding tooth pair (at the angle position $\varphi + (2\pi/z)$) is separating the pressure area from the suction area.

The pressure torque on the both gears can be calculated from:

$$M_1(\varphi) = \frac{p \times b}{2}\left[r_k^2 - \rho_1^2\left(\varphi + \frac{2\pi}{z}\right)\right], \qquad M_2(\varphi) = \frac{p \times b}{2}\left[r_k^2 - \rho_2^2\left(\varphi + \frac{2\pi}{z}\right)\right] \quad (1)$$

The distances between the contact point S and the middle point of the gears can be established from:

$$\begin{aligned} \rho_1^2(\varphi) &= r_k^2 - 2 \times r_g \times \left(r_g \times \varphi - \frac{e}{2}\right) \times \sin\alpha_b + (r_g \times \varphi - e/2)^2 \\ \rho_2^2(\varphi) &= r_k^2 - 2 \times r_g \times \left(r_g \times \varphi - \frac{e}{2}\right) \times \sin\alpha_b + (r_g \times \varphi - e/2)^2 \end{aligned} \quad (2)$$

After connecting the trapped volume with the suction chamber the contact point P separates the pressure chamber from the suction chamber (Fig. 2b). The torques due to the pressure in the angle range $(\pi/z) \times (\varepsilon - 1) < \varphi \le (2\pi/z)$ can be determined from:

$$M_1(\varphi) = \frac{p \times b}{2}[r_k^2 - \rho_1^2(\varphi)], \qquad M_2(\varphi) = \frac{p \times b}{2}[r_k^2 - \rho_2^2(\varphi)] \quad (3)$$

In Fig. 2, the calculated courses of torques have been shown ($t = \varphi/\omega$). From Fig. 2, it follows that the change of the position of the sealing point causes respectively the change in torque on both gears. The frequency of this change is equal to the gearing frequency and can be established from:

$$f_z = n \times \frac{z}{60} \quad (4)$$

The profile of the tooth normal force caused by the torque on the driven wheel is determined from:

$$F_{zn}(\varphi) = \frac{M_2(\varphi)}{r_g} \quad (5)$$

2.2. *Radial pressure loads on the gears*

The radial loads on the gears follow from the pressure distribution on its circumference. The experimental investigations confirm pressure distribution p (φ) shown in Fig. 3. One can distinguish three areas:

- the suction area (φ_1),
- the pressure build-up area and (φ_2),
- the pressure range of constant and variable proportion (φ_3), (φ_3'), share.

The size of each area depends on the design of the pump. For pumps without extending radial clearance compensation, the pressure build-up area φ_2 is spread on several teeth, unlike

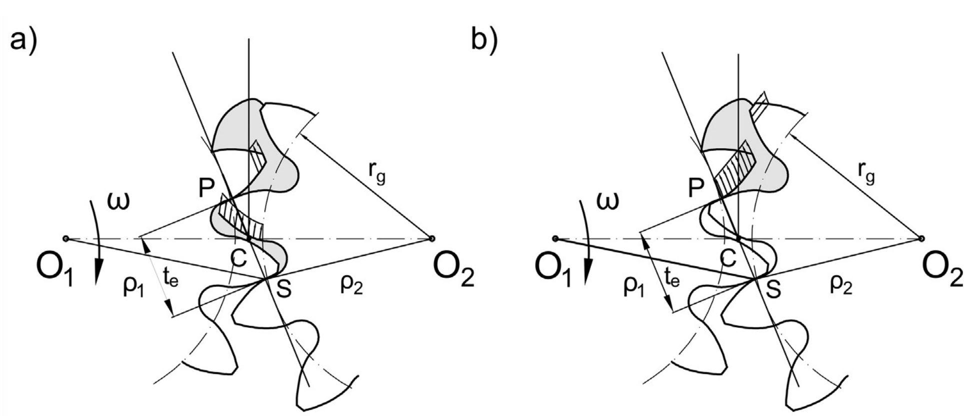

Fig. 2 – Scheme for the description of the torque changes.