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INFLUENCE OF DESIGN AND OPERATIONAL PARAMETERS ON THE DYNAMIC BEHAVIOUR OF GEAR PUMPS

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Abstract

This work concerns external gear pumps for automotive applications, which operate at high speed and low pressure and presents a sensitivity analysis about the influence of design and operational parameters on the pump dynamic behaviour. In previous work, a non-linear lumped-parameter kineto-elastodynamic model was developed, with the aim of including all the important effects, as well as to get a rather simple model. Two main sources of noise and vibration can be considered: pressure variation and gear meshing. The model has been validated by comparison with experimental vibration data, in a wide range of operational conditions and for different gear designs and several profile errors. This paper is focused on the analysis of the influence of the main design and operational parameters on the pump dynamic behaviour. In particular, the effect of operational pressure and speed, the influence of the clearance in the journal bearing and between tooth tip and pump case, and the effect of the dimension of the relief grooves in the bushes will be thoroughly discussed in the paper. Finally, the model could be a very useful and powerful tool in order to evaluate design improvements for noise and vibration reduction.

1. INTRODUCTION

External gear pumps are a popular power source well suited for handling viscous fluids such as fuel and lubricating oils. They are positive displacement devices simple and robust that can work on a wide range of pressures and rotational speeds providing at the same time high reliability. Their main applications can be found as lubrication pumps in machine tools, in fluid power transfer units or as oil pumps in engines. These features make them an interesting component in aerospace, industry, agricultural and automotive applications. On the other side, their main drawbacks are related with high noise and vibration levels and unavoidable output pressure ripple. This context increases the interest on modelling the dynamic behaviour of this mechanical system as a way to improve the initial design reducing testing efforts. Therefore, a good dynamic model could be an useful and powerful tool for the identification of noise and vibration sources and design improvement. For these reasons, the authors have developed a lumped-parameter kineto-elastodynamic model in order to study the dynamic behaviour of an external gear pump for automotive applications. Fluid pressure distribution around the gears, which is time-varying, is instantaneously computed and included as a resultant external force and torque acting on each gear [1], [2], [3]. Gear meshing phenomena have received particular attention [1], [4]: in particular the time-varying meshing stiffness and the tooth profile errors, the effects of the backlash between meshing teeth, the lubricant squeeze and the possibility of tooth contact on both contact lines have been included in the model [1][4]. One of the particular features of gear pump design is the use of hydrodynamic journal bearings for gear shaft support. The non-linear dynamics of this kind of bearings has been modelled using the theory of Childs called "finite impedance formulation"[5]. So, the model is highly non-linear. The dynamic equations of motion for each degree of freedom are obtained in [1], [4] and are numerically integrated in Simulink environment [6]. With the aim of reducing the integration time, the average positions of the shaft axes inside the journal bearings are estimated before the integration of the dynamic model. This estimation is carried out setting the periodically variable pressure and meshing forces of the model to a constant value equal to their mean values. Then the 'stationary' axis positions (also called orbit centroid) are computed as the solution of a non-linear system of algebraic equations obtained from the force balance of each gear. A non-conventional validation procedure [1], [7] has been developed in order to assess the dynamic model. The validation results can be considered rather satisfactory. Once the model is validated and its effectiveness is satisfactorily assessed, it can be used both in design optimization and diagnostics. In particular, since the model is able to predict the effects of variations of some parameters, a sensitivity analysis has been performed in this work: it will be shown hereafter the effect that a modification in the clearance into the journal bearing and between case and tooth tip has on the dynamic forces and on gear accelerations. Moreover, the effect of operational pressure and speed, and the effect of design parameter modifications (such as dimension of the relief grooves) will be thoroughly discussed.

The mechanical system under study is an external gear pump for vehicle steering. The most usual configuration has two twin gears (see Figure 1a), which are assembled by a couple of lateral floating bushes that act as seals for the lateral ends. Gear and floating bushes are jointly packed inside a case that encloses both the components and defines the isolated spaces that carry the fluid from the low to the high pressure chamber. This lateral floating bushes act as supports for gear shafts by means of two hydrodynamic bearings, which are hydraulically balanced in order to avoid misalignments between gear shaft and journal bearing. Power is applied to the shaft of one gear (gear 1), which transmits it to the driven gear (gear 2) through their meshing. Gear 1 is connected by an Oldham coupling with an electric drive. In the gear meshing area, when two tooth pairs come in contact, a trapped volume could arise and could undergo a sudden volume reduction and consequently a violent change in its pressure. To avoid this, the trapped volume is put in communication with the high or low pressure chambers. That is the role of the relief grooves milled in the internal face of the lateral bushes whose shape and dimension are therefore very important in the resulting dynamic behaviour. In more detail, we can refer as seal line the segment $\overline{CD}$ of the line of action (see Figure 1b),limited by the intersection with the relief groove edges. Three different situations can occur: the seal line can be equal, larger or shorter than the base pitch. If the seal line is larger than the base pitch, when the second tooth pair in contact enters into the seal line ($\overline{CD}$ in Figure 1b), the other meshing contact is still into the seal line and therefore during the gear rotation, the trapped volume will have a reduction and a consequent rise in pressure that can cause high vibration levels.

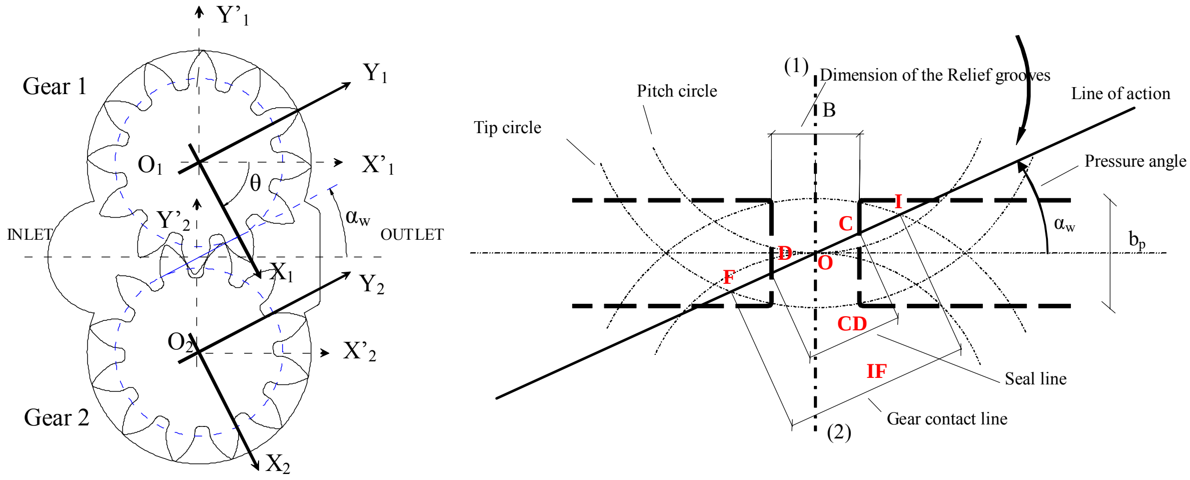

Figure 1. (a) Schematic of the gear pump and reference frames and (b) gear contact line and dimension of the relief grooves.

In the pump understudy, namely GENB, the dimension of the seal line is shorter than the base pitch and therefore when the second tooth pair enters into the seal line, the other tooth pair in contact is already beyond the D point (Figure 1b). In such a situation, there is contemporaneous communication between the inlet and outlet volume and consequently the volumetric efficiency of the pump decreases but the pressure rise in the trapped volume is reduced. In the GENB pump, the relief groove dimension is B=2.9 [mm] and the seal line length is :

$$\overline{CD} = \frac{B}{\cos(\alpha_w)} = \frac{2.9}{\cos(27.727)} = 3.276\ [mm] \tag{1}$$

where α_w is the pressure angle in working conditions. On the other hand the base pitch is :

$$P_b = \frac{2 \cdot \pi \cdot r_b}{z} = \frac{2 \cdot \pi \cdot 6.484}{12} = 3.395\ [mm] \tag{2}$$

where r_b is the base radius and z the teeth number. Therefore the difference between $\overline{CD}$ and P_b is 0.119 [mm] and in percentage of the base pitch becomes :

$$\%diff = \frac{\dfrac{B}{\cos(\alpha_w)} - P_b}{P_b} \cdot 100 = -3.5\% \tag{3}$$

So, the inlet and outlet chamber are in contemporaneous communication for 3.5/100 of the base pitch.
Two difference reference frames for each gear are used (see Figure 1a), both having their origins coincident with the centres of the gears; in reference frames $O_1X_1Y_1$ and $O_2X_2Y_2$, the X-axis is perpendicular to one of the lines of action and the Y-axis is parallel. On the other hand, in the reference frames $O_1X'_1Y'_1$ and $O_2X'_2Y'_2$, the Y'-axis is along the line connecting the centres of the gears and the X'-axis is orthogonal. In addition, Gear 1 is the driving gear and Gear 2 is the driven one.

2. INFLUENCE OF OPERATIONAL PARAMETERS: PRESSURE AND ROTATIONAL SPEED

Figure 2 and Figure 3a illustrate the influence of the output pressure on the stationary centre locations and on pressure distribution around the driving gear. The simulation results are obtained at rotational speed of 2000 rpm and increasing the output pressure from 20 to 90 bar; the circle in Figure 2 represents the maximum displacement of gear axes, allowed by the nominal clearance (h_{rn}) of 0.0245 mm between gears and pump case (see also Figure 1a). The results depicted in Figure 2 show that as the output pressure increases, the eccentricity modulus of both gears increases, in fact the markers in Figure 2 move closer to the maximum displacement circle. This behaviour can be interpreted as a consequence of the bearing reaction increase due to the enhancing of the gear pump output pressure. In fact, as the pressure forces increase, the new axis centre locations move reducing the actual clearance in the journal bearings (i.e. increasing the eccentricity modulus) and therefore the bearing reaction increases; this way, the new bearing reaction can balance the increased pressure forces. On the other hand, the pressure distribution in the tooth spaces, normalized to the output pressure value (Figure 3a), exhibits a sudden variation in the first tooth spaces; as depicted in Figure 2, due to the eccentricity direction, the minimum meatus height is located in the first tooth spaces. Moreover, the higher the output pressure, the more anticipated in the first isolated spaces this sudden variation is. This behaviour can be explained referring to the volumetric flow rate. In fact, it is worth noting that the volumetric flow rate directly depends on the pressure drop Δp_i between two consecutive vanes and on the meatus height h_i ($Q_i \propto h_i^3 \cdot \Delta p_i$, subscript i refers to the generic meatus) while the meatus height strictly depends on gear eccentricity. Considering the volumetric flow rate as approximately constant in all the meatus, the global pressure drop from outlet to inlet volumes is distributed among the meata as inversely proportional to h_i^3. Therefore, increasing the output pressure value, the actual clearance between tooth tip and pump case globally decreases, but this alters the ratios between two consecutives meatus heights h_i^3 in such a way that the sudden variation of the normalized pressure distribution is more anticipated in the first isolated spaces.

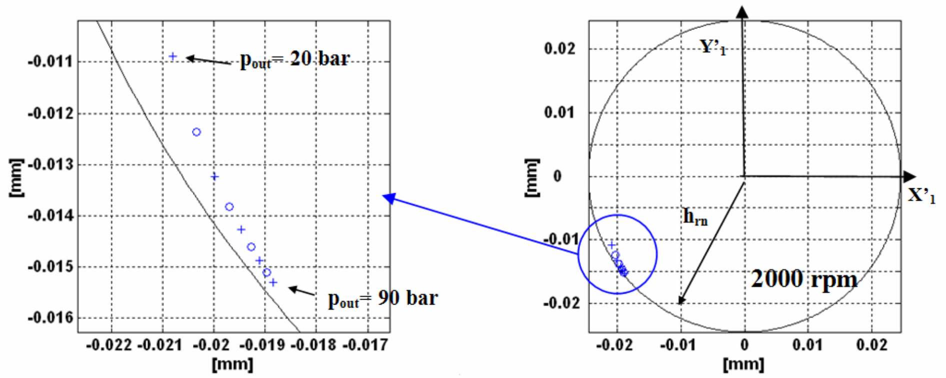

Figure 2. Stationary centre position for the driving gear with different output pressure values; the representation is in the reference frame of Figure 1; the circle represents the maximum displacement of gear axes, allowed by the nominal clearance h_{rn} = 0.0245 mm between gears and pump case.

Furthermore, Figure 3b shows the influence of the pressure variation on the pressure force concerning the driving gear in Y-direction, for one meshing period T, starting at the instant when the second tooth pair comes in contact. It can be noted that the pressure force is noticeably affected by an increase of output pressure due to their direct dependence on pressure distribution; the same behaviour occurs for the other directions and for the driven gear, not shown hereafter. On the other hand, for the gear accelerations, depicted in Figure 4a, the output pressure dependence is not as pronounced as for the pressure forces because the gear accelerations are the balance between all the dynamic forces (meshing forces, bearing reactions, pressure forces); anyway, considering the peaks in the accelerations at about 25% of the meshing period, the higher peaks are relative to the accelerations obtained at higher output pressure.

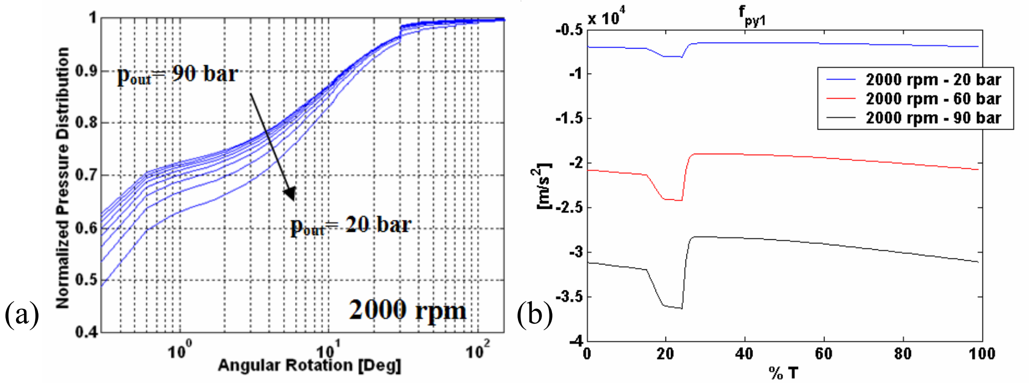

Figure 3. Normalized pressure distribution (a) and pressure force (divided by the gear mass) in Y_1-direction (b) in Gear 1 for different output pressure values.

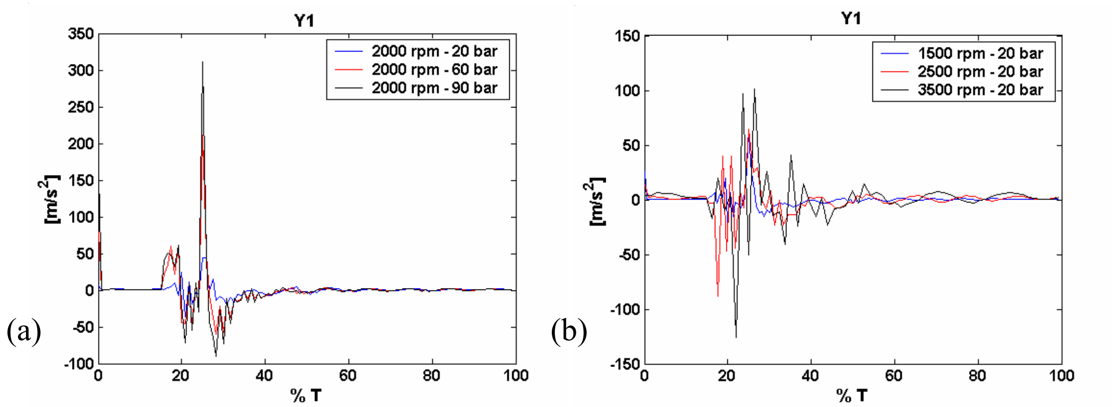

Figure 4. Acceleration for the driving gear in Y_1-direction for different (a) output pressure values and (b) rotational speed.

The influence of the rotational speed on the stationary centre position is similar to the influence of the pressure already described above; in particular as the shaft rotational speed decreases from 3500 to 1500 rpm, the eccentricity modulus value increases from 0.02284 to 0.02372 mm for gear 1 and from 0.02308 to 0.02384 mm for gear 2. In fact, as the rotational speed decreases, the bearing reaction tends to decrease too, due to the direct relation with the operational speed [1][2] and the new axis centre location will change, decreasing the actual clearance in the journal bearing (i.e. larger eccentricity modulus); this way, a new equilibrium between the dynamic forces is established. As a consequence, when the rotational speed decreases, the pressure distribution in the tooth spaces is altered in a similar way as occurs when the output increases (i.e. its sudden variation is more anticipated in the first isolated spaces), for analogous reasons. In addition, Figure 4b shows the influence of the operational speed variation on the acceleration in Y_1-direction for gear 1: it can be noted that an increase of the operational speed determines an increase of the gear acceleration.

3. EFFECT OF DESIGN PARAMETERS

3.1 Effect of clearance modifications

In the pump under testing, the nominal clearance h_{rn} between pump case and tooth tip has the same value as the nominal clearance C_r in the journal bearing, equal to 0.0245 mm. These dimensions, as shown below, have an important influence on pressure distribution, on gear accelerations as well as on the stationary centre position. In this section the influence of a modification of the nominal clearances C_r and h_{rn} will be shown. Reference [1] includes all the details about the clearances involved in the pump working and several clarifying figures. The simulation results presented in this section are obtained at the operational condition of 3350 rpm and 20 bar. Table 1 collecting the normalized eccentricity modulus and the eccentricity direction angle in the stationary centre position, shows the influence of a clearance alteration on the stationary centre position of gear 1; it can be observed that as the nominal clearance increases, the stationary centre position becomes closer to the maximum allowed displacement, in fact the eccentricity modulus increases too. Moreover, Table 1 collects the values of the actual minimum clearance $C_{r,\chi}$ in the journal bearings that obviously depends not only on the nominal clearances C_r but also on the eccentricity ($C_{r,\chi} = (1-\chi)\cdot C_r$). It can be noted that as the nominal clearance C_r increases, the bearing reaction tends to decrease; so, due to the dynamic force equilibrium, the actual minimum clearance decreases.

On the other hand, about the influence of these clearances on pressure distribution around the driving and driven gear, it can be noted that as the nominal clearance value increases, the pressure in the first tooth spaces reaches a higher pressure value earlier (as it occurs if the output pressure increases). This pressure distribution behaviour can be explained as a consequence of the increase of the eccentricity modulus (Table 1) which alters the ratios between two consecutives meatus heights (h_i^3) as explained in the previous section. The pressure forces depend on clearance alteration too, since they are obtained using the pressure distribution. Figure 5a shows the influence of the clearance value on pressure force evaluated over one meshing period, starting at the instant when the second tooth pair comes in contact. It can be noted that, as the clearance value increases, the pressure force amplitude also increases due to the different trends on the pressure distribution. Finally, Figure 5b shows the clearance influence on the acceleration acting on gear 1 at operational condition of 3350 rpm and 20 bar expressed as a function of meshing period. No important differences in accelerations between the three clearance conditions can be observed; in fact, even if pressure forces increase as the clearance value increases, the bearing reactions change in order to balance the pressure forces at best. For this reason, the accelerations, that are obviously the consequence of the balance between the pressure forces, meshing forces and the bearing reaction forces, do not change in an important way.

Table 1. Clearance influence on stationary centre position (in modulus and direction angle) at 3350 rpm and 20 bar for gear 1 and influence on the actual minimum clearance.

Nominal clearance [mm]	0.019	0.0245	0.0275
Eccentricity modulus χ (normalized)	0.900	0.935	0.946
Actual minimum clearance $C_{r,\chi}$ [mm]	0,0019	0,0016	0,0015
Eccentricity direction angle [deg] in the reference frame $O_1X'_1Y'_1$	194.6	201.8	204.7

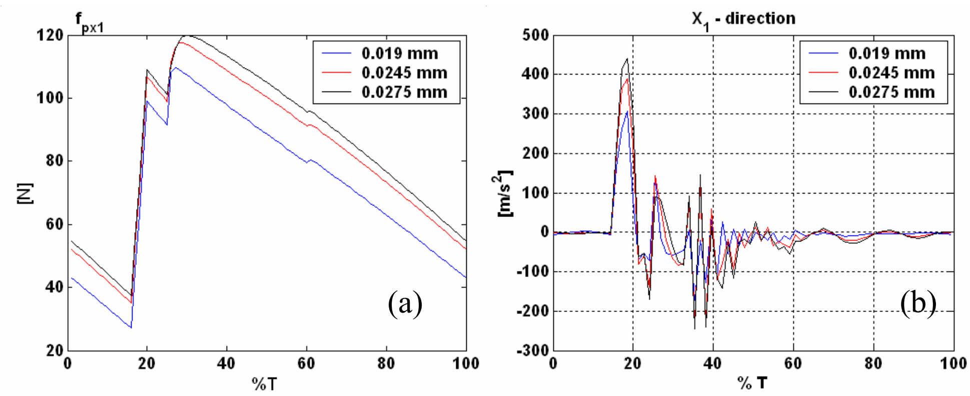

Figure 5. Clearance influence on (a) pressure force and (b) acceleration in X_1-direction for gear 1 over one meshing period T; operational conditions of 3350 rpm and 20 bar.

3.2 Effect of the relief groove dimension

Figure 6a shows the pressure force in X_1-direction for different kinds of relief grooves (length *B* of 2.4, 2.7 and 2.9 mm, see also Figure 1b); such a dimension has great influence on the performance of the gear pump, in fact it can be noted that with reduced length *B*, the discontinuities on the pressure forces is smoother due to the increase of the contemporaneous communication of the inlet and outlet chambers. In fact, remembering equation (3) the contemporaneous communication between the inlet and outlet volume is 3.5% of the meshing period with dimension *B* of 2.9 mm, while with *B*=2.7 mm it is 10% and finally with *B*=2.4 mm it is 20%.

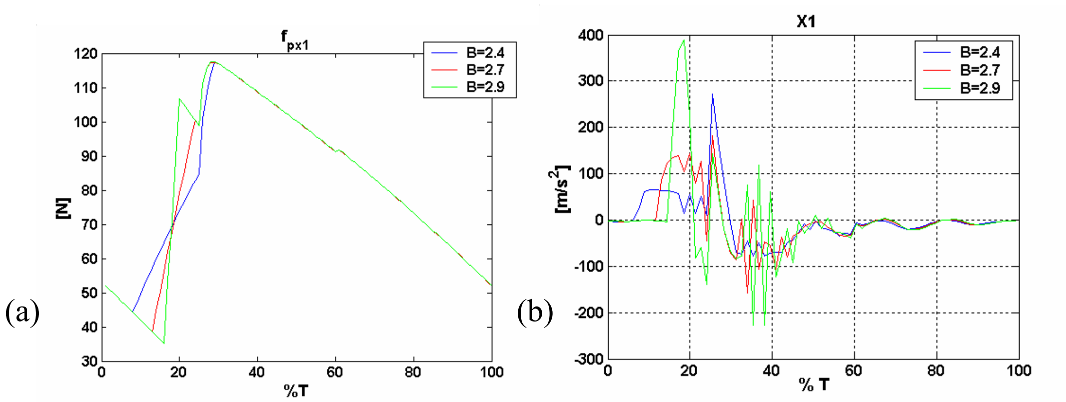

Figure 6. (a) Pressure force and (b) acceleration of gear 1 in X_1-direction over one meshing period with different dimensions of the relief grooves (length *B* of 2.4, 2.7 and 2.9 mm).

The results in terms of gear accelerations (Figure 6b) show that the relief grooves with length 2.9 mm determine larger oscillations than with lower relief groove lengths due to the decrease of the contemporaneous communication with the inlet-outlet volumes; moreover the smaller length *B*, the lower the pump efficiency. Therefore, the designer has to take into account that in order to reduce the gear vibrations, the relief groove length has to be reduced, but in order to increase the pump volumetric efficiency, the relief groove length has to be increased.

4. CONCLUSIONS

In this work an extensive sensitivity analysis by using a non linear kineto-elastodynamic model of an external gear pump has been performed. In particular, the model has been used in the design phase in order to evaluate possible design modifications (different lengths of the relief grooves and clearances) and in order to evaluate the influence of operational parameters such as output pressure and rotational speed.

The simulation results have demonstrated that the variation in the operational range of the rotational speed gives a lower effect on the gear accelerations than the output pressure variation. Moreover it has been shown that a pump with relief grooves of length 2.9 mm determines larger oscillations than a pump with lower lengths due to the decrease of the communication with the inlet-outlet volumes; moreover the smaller the relief groove length, the lower the pump efficiency. Therefore, the designer has to take into account that in order to reduce gear vibrations, the relief groove length has to be reduced, but in order to increase the pump volumetric efficiency, the relief groove length has to be increased.

Finally, this work can be useful in order to evaluate design directions and in order to foresee the influence of working conditions and design modifications on vibration and noise generation.

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