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WHICH ARE THE MOST IMPORTANT PRESSURE EFFECTS IN DYNAMIC ANALYSIS OF GEAR PUMPS?

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Abstract

In previous works, the authors have presented a numerical model of the dynamic behaviour of an external gear pump for automotive applications. The model takes the most important phenomena involved in the operation of this kind of machine into account. The simulation results have shown that the variable meshing stiffness has a notable contribution on the dynamic behaviour of the pump but this is not as important as the pressure phenomena. As a consequence, the original model was modified with the aim of improving the calculation of pressure forces and torques in order to achieve a result that was closer to measured values. The new pressure formulation includes several phenomena not considered in the previous one, such as the pressure variations at input and output ports, as well as an accurate description of the trapped volume and its connections with high and low pressure chambers. In particular in this paper the relative effect of each considered pressure phenomenon will be discussed in detail, highlighting which are the most important ones, in order to develop a simple but accurate dynamic gear pump model. So, the improved model could be used in order to analyse the dynamic behaviour of the pump and to identify noise and vibration sources.

1. INTRODUCTION

This work deals with external gear pumps for automotive applications, which operate at high speed and low pressure. In this situation gear pumps have two main noise sources both of them sharing the same fundamental period: fluid borne noise, as a consequence of the flow from the low to the high pressure chambers, and mechanical noise due gear meshing. With the aim of improving the design process and reduce the experimental task, a non-linear lumped-parameter kineto-elastodynamic model of an external gear pump was developed by the authors [1], [2], [3]. The model includes the main important phenomena involved on the pump operation as time-varying oil pressure distribution on gears, time-varying meshing stiffness and hydrodynamic journal bearing forces. Subsequently case wear, backlash as well as profile errors were added improving the original model features. This model was used to obtain a thorough knowledge about the interaction between both vibratory sources: gear meshing and pressure forces. On the basis of the simulations it was observed that the sudden pressure torque variation and in a secondary plane also the pressure forces, have a great effect in the dynamic behaviour of the pump while the variable meshing stiffness has a minor contribution. This scenario was also verified in experimental tests [3], [4]. Therefore, the validation results [4] show that the assumptions adopted in the pressure formulation provide a non realistic abrupt change in the pressure forces and as a consequence the original model should be improved.

Pressure distribution around the gears was obtained modelling a certain number of control volumes. The original model only considered the isolated tooth spaces “sealed” between teeth, lateral plates and case as control volumes. The new model combines both gears, adding new control volumes as the inlet and outlet chambers as well as the volume trapped between meshing teeth (called *trapped volume*). This trapped volume has an important role in the abrupt change of the pressure efforts, as suffers a sudden reduction leading to a violent pressure increase. This phenomenon is relieved by grooves milled on the lateral balancing plates. Therefore, a cautious design of this relief grooves is very important on the final noise and vibration behaviour. In the original model this pressure transient was simplified assuming a linear transition from the high to the low pressure value, taking into account that the relief groove dimensions provide a simultaneous connection of the trapped volume with the high and low pressure chambers. The new formulation takes into account the gear geometry obtained by means of the cutting tool, following the approach proposed by Litvin [5], as well as the shape of the relief grooves; then, a numeric procedure is used to obtain their intersection giving an accurate description of the trapped volume connecting areas with high and low pressure chambers [3], [6]. This way, it is possible to consider more complex relief groove designs. Furthermore, the case wear profile included in the original model on the basis of experimental measurements [2], [3], was defined on the basis of the run in process [8].

The aim of this work is to study the role of each design parameter involved in the resultant pressure distribution and their impact on the final dynamic behaviour of the pump.

2. THE MODEL

The operating principle of external gear pumps is very simple. The fluid that fills the space bounded by the case and two successive gear teeth becomes isolated at the input port and is carried out towards the output port. This way, the isolated volumes increase their pressure progressively up to the high pressure chamber. Modern gear pump designs are of hydraulically balance type composed by two gears (normally twin gears) supported by two hydrodynamic journal bearings contained in a couple of lateral floating bushes packed inside a case. In this work the interest is focused on hydraulically balance external gear pumps for automotive applications (as a power in steering systems), which c operate at high speed (from 1500 to 3400 rpm) and low pressure (from 3.5 to 100 bar). In this case, twin spur gears have 12 teeth, pressure angle of 20 degrees, module of 1.15 mm and width of 12.1 mm.

The dynamic model is of lumped parameter type with 6 degrees of freedom (two displacements and one rotation for each gear) and includes non linear effects as the parametric excitation due time-varying stiffness and profile errors, backlash between meshing teeth, lubricant squeeze and hydrodynamic journal bearings. Time varying meshing stiffness for each teeth pair is defined following the proposal of Kuang [9] adding a constant term to include the Hertzian stiffness. Damping is considered proportional to the stiffness when there is contact and otherwise as a lubricant squeeze model and the hydrodinamic journal forces are obtained using Childs’s formulation [10]. See [1], [2] and [3] for further details.

The control volumes considered were those isolated by the case and the inter-teeth of each gear, the inlet and outlet chambers and the trapped volume if exists. Figure 1 shows the volumes considered with a schematic about the fluxes between them.

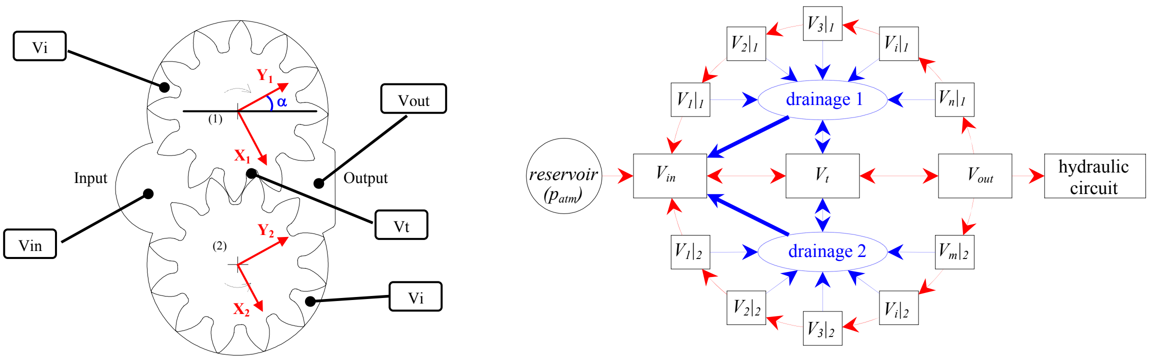

Figure 1. Control volumes and relating flows.

Applying to the opened thermodynamic system, the continuity equation and the steady-state fluid equation, assuming an adiabatic and isoentropic transformation the following relation is obtained [11], [12]:

$$\frac{dp}{dt} = \frac{B_{oil}}{V}\left(Q_{in} - Q_{out} - \frac{dV}{dt}\right) \tag{1}$$

Equation (1) allows the calculation of the pressure variation of a fluid (characterized by the oil bulk coefficient B_{oil}) contained in the control volume V caused by mass gain (Q_{in}-Q_{out}) and by the volume variation dV/dt. Each isolated volume V_i is connected with its adjacent volumes V_{i-1} and V_{i+1} through the clearance between case and tooth tip (h_i and h_{i+1} depending on case wear profile and gear centre position) but also through the clearance between gear lateral faces and floating bushings. They are also connected with the drainage circle through the lateral clearance and in axial sense with the high or low pressure through the clearances between the bushes and case. Each connection gives rise to a flow rate considered positive for incoming flows and otherwise negative. So, the considered flow rates are: tip flows $Q_{h,i}$, $Q_{h,i+1}$, lateral flows $Q_{f,i}$, $Q_{f,i+1}$, drainage flow $Q_{d,i}$ and axial flow $Q_{b,i}$. Tip and lateral flows will be composed by two terms: the first assumed as a laminar flow between two parallel plates, called Q_p, and another one due the drag flow consequence of the gear rotational movement, called Q_u. Furthermore, lateral, drainage and axial flows should be considered two times one for each side of the gear.

Inlet volume communicates with the reservoir, with the drainage circle through the lateral clearance, with the first isolated tooth space of both gears through the clearance at the tooth tip and the lateral face and finally with the trapped volume through the lateral clearance and the relief groove connection area that will be obtained numerically as a function of the gear position. Therefore, the flows considered for inlet volume will be: tip flows $Q_{h,1}|_1$, $Q_{h,1}|_2$, lateral flows with isolated tooth spaces $Q_{f,1}|_1$, $Q_{f,1}|_2$, and trapped volume $Q_{f,t,in}$; drainage flows coming from all the isolated volumes $\Sigma Q_{d,i}|_1$, $\Sigma Q_{d,i}|_2$, and from the trapped volume $Q_{d,t}$ (because the drainage circle is connected directly with the inlet chamber) and reservoir flow $Q_{T,in,atm}$ and trapped volume flow $Q_{T,t,in}$ when relief groove connection is open to the inlet chamber. The new formulation assumes reservoir and relief groove flows as steady-state turbulent efflux through a single orifice defined by their area and the corresponding discharge coefficient [3], [6]. $Q_{f,1}|_1$, $Q_{f,1}|_2$, $Q_{f,t,in}$, $Q_{T,t,in}$ are double flows, $Q_{d,t}$ is quadruple.

Outlet volume is connected with the last isolate volume of each gear, with the trapped volume through the lateral clearance and the relief groove connection area and with the pump outside through a regulating valve. The flows taken into account are: tip flows $Q_{h,n}|_1$, $Q_{h,m}|_2$, lateral flows $Q_{f,n}|_1$, $Q_{f,m}|_2$ and $Q_{f,t,out}$; trapped volume flow $Q_{T,t,out}$ and valve discharge flow $Q_{T,out,atm}$. Again trapped volume flow and discharge flow were taken as a turbulent efflux through an orifice. $Q_{f,n}|_1$, $Q_{f,m}|_2$, $Q_{f,t,out}$ and $Q_{T,t,out}$ are double flows.

Finally, trapped volume exist only when two tooth pairs are meshing and will be connected to the inlet, outlet volumes and drainage circle through the lateral face and further to the inlet and outlet volumes through the relief grooves areas giving the flows: $Q_{T,t,out}$, $Q_{T,t,in}$, $Q_{f,t,out}$, $Qf_{,t,in}$ and $Q_{d,t}$. They are all double flows, except $Q_{d,t}$ that is quadruple.

Then, an ordinary differential equation system should be solved obtaining the pressure in a vane for a whole turn of each gear. The system dimension will be variable, because the number of isolated volumes depends on the gear angular position, so six different integration intervals are defined. Furthermore, gear centre position should be known in order to define tip clearances and relief groove areas. The calculation of pressure efforts involves a high computational effort yielding to very tedious simulations if they were carried out simultaneously during the integration of the dynamic model. This scenario is solved estimating the average positions of the journal axes into the bearing under certain operating conditions considering the pressure forces and time-varying stiffness as constants equal to their mean values but taking into account the non linear behaviour of journal bearings. The force balance of each gear provides a non-linear system of algebraic equations that was solved numerically. Once the stationary position is known the time-varying pressure efforts were obtained and included in the dynamic model. This procedure has been assessed in [3].

3. FLOW DUE TO PRESSURE DROP VS ENTRAINED FLOW

As mentioned above, in the continuity equations there are some flow rates which are due to two contributions:

$$Q_{tot} = Q_p + Q_u \tag{2}$$

Q_p depends on the pressure drop between adjacent volumes and on the clearance h_i, while Q_u is directly proportional to the angular speed [3], [6]. Several simulations were carried out at different operational conditions, with the aim of evaluating the contribution of Q_p and Q_u to the overall flow rates Q_{tot}. Some flows are turbulent, in others the flow rate is due to the pressure drop contribution only, while in others Q_u can be simplified, so the flows to be study are only: $Q_{h,i}$ for each isolated tooth space and $Q_{f,t,in}$, $Q_{f,t,out}$ in the trapped volume. Table 1 summarizes the results obtained at 2000 rpm and 90 bar in percentage of the total flow rate. Each flow has been calculated averaging the values obtained in each integration interval.

	$Q_{h,i+1} - Q_{h,i}$	$Q_{f,t,in}$	$Q_{f,t,out}$
Q_p	29.27 %	52.98 %	34.35 %
Q_u	70.73 %	47.02 %	65.65 %

Table 1. Q_p and Q_u contribution at 2000 rpm and 90 bar.

Figure 2 presents Q_p in percentage of the global tip flow $Q_{h,tot}$ at different operational conditions. The difference between $Q_{h,tot}$ and Q_p is the flow rate Q_u, as it can be obtained from equation (2). At the maximum the contribution of Q_p is about 50%. A previous work [13] the effect of the output pressure on the center stationary locations was investigated and it has been noted that h_i decreases when pressure rises, while it increases when operational speed grows up. When the pressure increases and the clearance is reduced, the flow rate is almost constant because both effects tend to balance themselves. Figure 2 (a) shows this situation. On the contrary, when angular speed grows up, h_i increases and so the Q_p contribution becomes more important (Figure 2 (b)).

About the flow rates $Q_{f,t,in}$, $Q_{f,t,out}$, as the output pressure increases, the contribution of Q_p to the overall flow rate tends to increase, while the increase of the rotational speed does not have any influence on Q_p because the relative meatus height maintains a constant value in the experimental range.

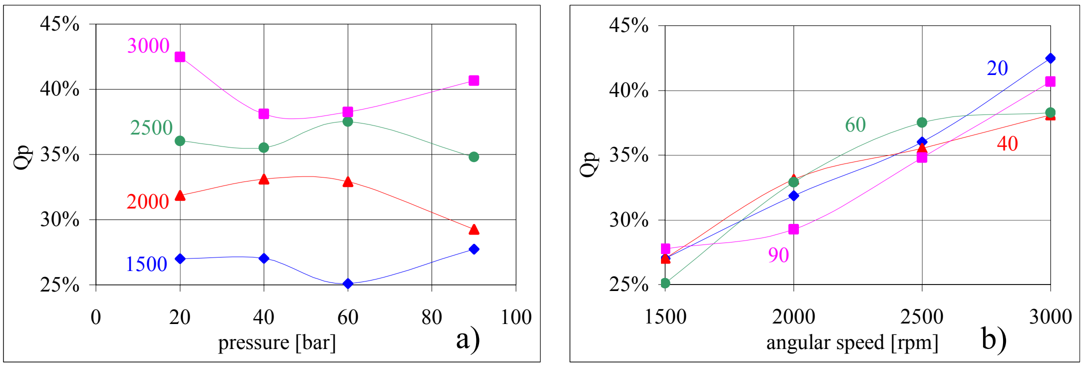

Figure 2. Q_p in percentage of the global tip flow $Q_{h,tot}$ at different angular speeds in rpm (a) and at different output pressures in bar (b).

4. MAIN, MINOR AND NEGLIGIBLE FLOWS IN CONTROL VOLUMES

Four different control volumes were selected with the aim of describing the contribution of each flow rate on the continuity equation. They are an isolated tooth space (the third in the travel from the low to the high pressure region for gear 1), the trapped volume, the inlet and the outlet volumes. Table 2 shows a list of all the flows involved in the pump working; in particular the table depicts the relative contribution of local flow rate to the global flow rate for a certain control volume. In the table, about the outlet volume, let us define Q_h and Q_f as the global tip flow rate and the global lateral flow exchanged with the last isolated tooth space, respectively, both concerning the two gears. The values in percentage shown in the table represent average values obtained at several operational conditions in the operational range. For example, about the selected isolated tooth space, the main contribution is due to $Q_{h,i+1} - Q_{h,i}$ (about 90% of the overall flow rate exchanged in such a control volume), while $Q_{d,i}$ has a minor contribution (about 10%) and $Q_{f,i+1}$ - $Q_{f,i}$ and $Q_{b,i}$ have a negligible role (less than 3%).

	Main flows	Minor flows (till 20%)	Negligible flows (less than 3% for all operational conditions)
Isolated tooth space	$Q_{h,i+1} - Q_{h,i}$ (90%)	$Q_{d,i}$	$Q_{f,i+1}$ - $Q_{f,i}$, $Q_{b,i}$
Trapped volume	$Q_{T,t,in}$ (90%)	$Q_{T,t,out}$	$Q_{d,t}$, $Q_{f,t,in}$, $Q_{f,t,out}$
Inlet volume	$Q_{T,in,atm}$ (>90%)	$Q_{T,t,in}$	$Q_{d,t}$, $\left.\sum_{i=1}^{n} Q_{d,i}\right\rvert_1$, $\left.\sum_{i=1}^{m} Q_{d,i}\right\rvert_2$, $Q_{f,t,in}$, $\left.Q_{h,1}\right\rvert_1$, $\left.Q_{h,1}\right\rvert_2$, $\left.Q_{f,1}\right\rvert_1$, $\left.Q_{f,1}\right\rvert_2$
Outlet volume	$Q_{T,out,atm}$ (55 %), Q_h (45%)	-	$Q_{f,t,out}$, $Q_{T,t,out}$, Q_f

Table 2. Flow rate contributions for the different control volumes.

In the trapped volume the contribution of $Q_{T,t,out}$ reaches values up to about 10% of the global flow rate for higher angular speeds and lower output pressures.

The leader flow rate in the inlet volume is $Q_{T,in,atm}$ with values always higher than 90% of the global flow rate at every operational conditions. The minor flow $Q_{T,t,in}$ is more noticeable (5% of the global flow rate) at higher pressures and lower angular speeds.

The outlet volume is the only control volume governed by two main flows (as shown in Table 2). Figure 3 plots the flow rates of the outlet volume in percentage of the global flow at 20 bar (left) and 60 bar (right) as a function of the operational speed. The trends are different, in fact at 20 bar the difference between Q_h and $Q_{T,out,atm}$ increases as the angular speed grows up, while at 60 bar the two curves matches at about 2500 rpm. In general, Q_h is bigger than $Q_{T,out,atm}$. In the range of 50-90 bar, at 2500 rpm, $Q_{T,out,atm}$ becomes higher than Q_h. Finally, the values of the two main flows of the outlet volume are similar at 20 and 60 bar.

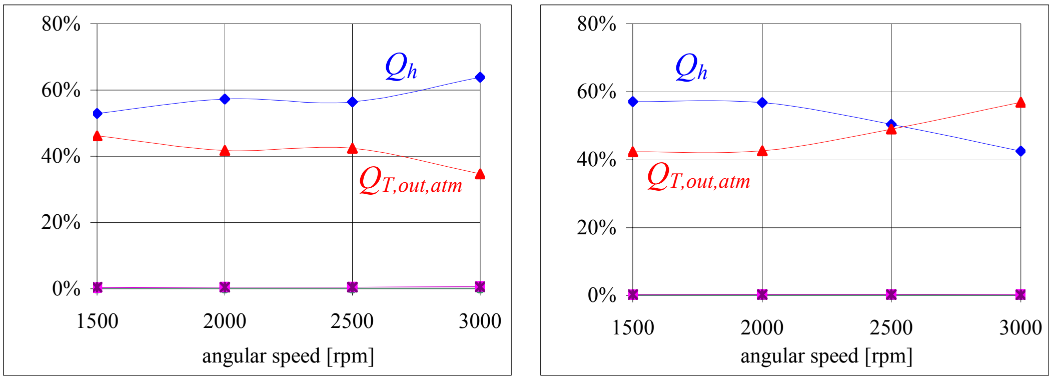

Figure 3. Flow rates for the outlet volume at 20 bar (left) and 60 bar (right).

5. FLOW RATE EFFECT ON THE PRESSURE DISTRIBUTION

In this section all the flow rates involved on the pressure distribution formulation have been annulled one-by-one in order to evaluate their effect in terms of pressure distribution trend and pressure forces. In particular, tests are carried out at 2000, 3000 rpm and 40, 90 bar. The flow rate $Q_{h,i,i}$ has not been deleted because it plays the main role in the travel from the input to the output chamber (as already shown in Table 2). So the flow rates that have been removed are: lateral flows, drainage flows, axial flows and the turbulent flow rates. For instance, if the contribution of the turbulent flow rates is annulled, this means that $Q_{T,t,in} = Q_{T,t,out} = 0$, and the continuity equation of the trapped volume, inlet and outlet chamber will be modified. Figure 4 shows the pressure distribution behaviour at 2000 rpm, 40 bar (a) and at 3000 rpm, 90 bar (b), while Figure 5 presents the pressure distribution neglecting the turbulent flow rates (a) and an enlargement of Figure 4 (b) around 150 degrees of gear angle. The black lines identified with *Q-all* are obtained considering all the flow contribution.

The resultant pressure forces acting on gear 1 at 2000 rpm, 40 bar are presented in Figure 6, using the reference frame of Figure 1, with the y-axis parallel to the line of action. Figure 4 and Figure 5 show that the turbulent flows are responsible of the most important difference with respect to the standard formulation case (*Q-all*). All the other flows give a minor contribution, localized at the beginning of the travel from the inlet to the outlet volume.

The peak in Figure 5 due to the turbulent flow annulment can be explained considering that the increase of the pressure in the trapped volume due to gear rotation can not be reduced excluding turbulent flows with the adjacent volumes.

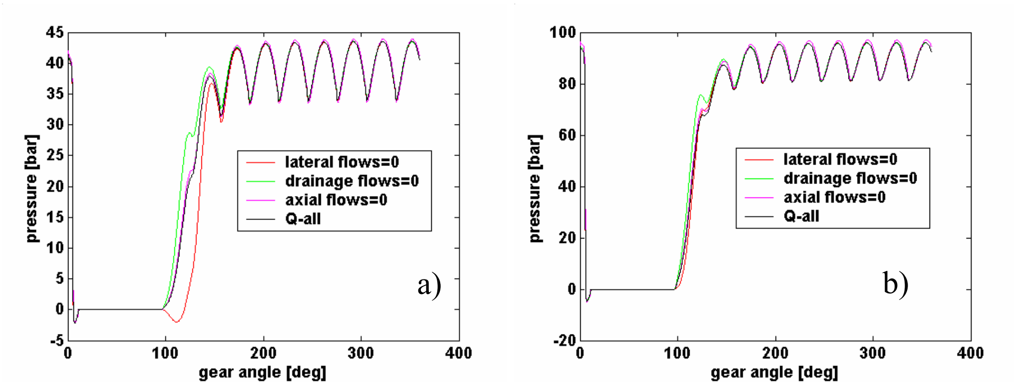

Figure 4. Pressure distribution comparison removing different flow rates at 2000 rpm, 40 bar (a) and 3000 rpm, 90 bar (b).

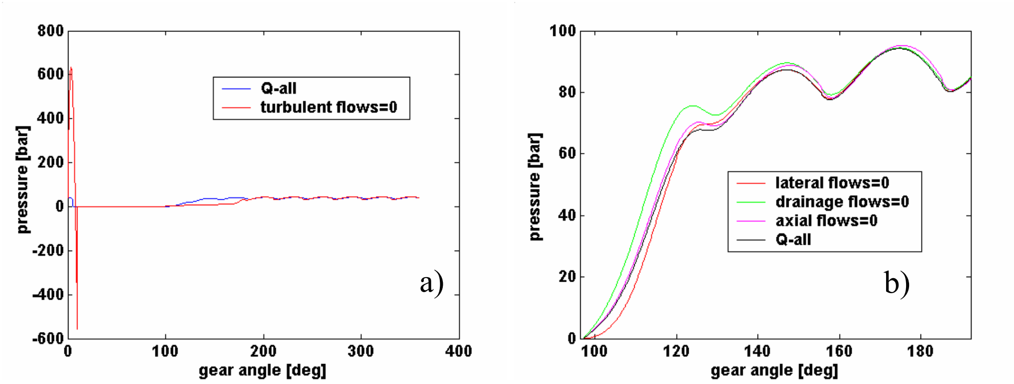

Figure 5. Pressure distribution at 2000 rpm and 40 bar (a) neglecting the turbulent flow rates and enlargement of Figure 4 (b) around 150 degrees of gear angle (b).

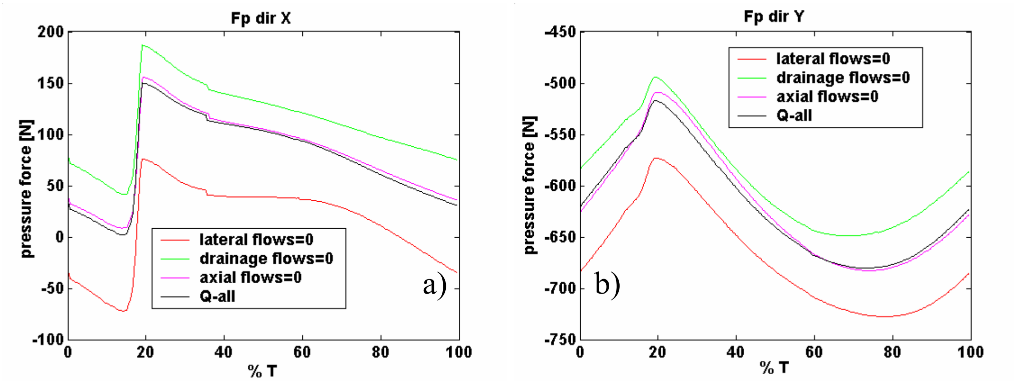

Figure 6. Pressure force acting on gear 1, over one meshing period in the reference frame of Figure 1 removing different flow rates at 2000 rpm, 40 bar.

It is worth nothing that the contribution of the axial flows on the pressure distribution and in the resultant forces is negligible in all the control volumes, with respect to the effects given by the lateral and the drainage flows. Figure 6 shows this behaviour in terms of resultant pressure forces: magenta and black lines are overlapped.

6. CONCLUSIONS

In this work a numerical model has been used in order to evaluate which are the most important pressure phenomena involved in the operation of an external gear pump for automotive applications. In particular, it has been shown the contribution of each flow rates exchanged in any control volume: tip flows and turbulent flows are phenomenon that have to be included in a pressure distribution formulation because they play a pivotal role, while axial flows have a minor contribution to the final results. Moreover, the contribution of the drag flow due to the gear rotation is dominant with respect to the contribution of the flow due to the pressure drop.

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