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Elasto-dynamic analysis of a gear pump–Part IV: Improvement in the pressure distribution modelling

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ABSTRACT

This work concerns external gear pumps for automotive applications, which operate at high speed and low pressure. In previous works of the authors (Part I and II, [1,2]), a non-linear lumped-parameter kineto-elastodynamic model for the prediction of the dynamic behaviour of external gear pumps was presented. It takes into account the most important phenomena involved in the operation of this kind of machine. The two main sources of noise and vibration are considered: pressure pulsation and gear meshing. The model has been used in order to foresee the influence of working conditions and design modifications on vibration generation. The model experimental validation is a difficult task. Thus, Part III proposes a novel methodology for the validation carried out by the comparison of simulations and experimental results concerning forces and moments: it deals with the external and inertial components acting on the gears, estimated by the model, and the reactions and inertial components on the pump casing and the test plate, obtained by measurements. The validation is carried out by comparing the level of the time synchronous average in the time domain and the waterfall maps in the frequency domain, with particular attention to identify system resonances. The validation results are satisfactory global, but discrepancies are still present. Moreover, the assessed model has been properly modified for the application to a new virtual pump prototype with helical gears in order to foresee gear accelerations and dynamic forces. Part IV is focused on improvements in the modelling and analysis of the phenomena bound to the pressure distribution around the gears in order to achieve results closer to the measured values. As a matter of fact, the simulation results have shown that a 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 at improving the calculation of pressure forces and torques. The improved pressure formulation includes several phenomena not considered in the previous one, such as the variable pressure evolution at input and output ports, as well as an accurate description of the trapped volume and its connections with high and low pressure chambers. The importance of these improvements are highlighted by comparison with experimental results, showing satisfactory matching.

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Nomenclature

Latin symbols

A	orifice section
B	length between the delivery sides of the relief grooves (see Fig. 4a)
b_b	bearing block width
b_k	face width of gear k
B_{oil}	oil bulk modulus
$h_{b,i}$	radial clearance between the casing and the bearing block in correspondence of tooth space i
h_f	lateral clearance between bearing block and the lateral flank of the tooth
h_i	radial clearance between tooth tip i and the casing
l_t	tooth tip thickness
p	pressure in the generic control volume
p_{atm}	atmospheric pressure
p_d	pressure in the drainage circle (equal to p_{atm})
p_i	pressure in the tooth space volume i
p_{in}	pressure in the inlet volume
p_{out}	pressure in the outlet volume
p_t	pressure in the trapped volume
r_c	internal radius of the casing
r_d	radius of the drainage circle
$r_{ext,k}$	outside radius of gears
r_m	radius at half height of the tooth
Q	volumetric flow rate
$Q_{b,i}$	axial volumetric flow rate through clearance $h_{b,i}$
$Q_{d,i}$	volumetric flow rate between isolated tooth space i and the drainage circle through clearance h_d
$Q_{d,t}$	volumetric flow rate between the trapped volume and the drainage circle
$Q_{f,i}, Q_{f,i+1}$	volumetric flow rate between isolated tooth spaces through clearance h_f
$Q_{f,t,in}, Q_{f,t,out}$	volumetric flow rate between the trapped volume and the inlet and outlet volumes through clearance h_f
$Q_{h,i}, Q_{h,i+1}$	volumetric flow rate between isolated tooth spaces through clearances h_i, h_{i+1}
$Q_{T,in,atm}$	turbulent volumetric flow rate between the inlet volume and the reservoir
$Q_{T,out,atm}$	turbulent volumetric flow rate from the outlet volume to the hydraulic circuit
$Q_{T,t,in}, Q_{T,t,out}$	turbulent volumetric flow rate between the trapped volume and the inlet and outlet volumes, respectively
t	time
V	volume of the generic control volume
V_i	volume of the tooth space i
V_{in}, V_{out}	volumes of the inlet and outlet volumes
V_0	nominal volume of the tooth space.
V_t	volume of the trapped volume
$V_{t0}, V_{in0}, V_{out0}$	values of the trapped volume, inlet volume and outlet volume when $\theta = 0$, i.e. in the initial condition when a new meshing begins

Greek symbols

ΔQ	difference between the volumetric flow rates, coming into a control volume and coming out
θ	angular coordinate
$\theta_a, \theta_b, \theta_{t,end}$	angles defining the beginning of the connection with the inlet volume, the end of the connection with the outlet volume and the end of the trapped volume, respectively (see Fig. 4b)
μ	Oil dynamic viscosity
ρ	oil density
ω	angular speed
ω_k	Mean angular speed of gear k in steady-state operational conditions

Subscripts

i	denotes tooth space
$k = 1,2$	denotes gears
n,m	number of isolated tooth spaces in gears 1 and 2, respectively
$\|_k$	applied to gear k

1. Introduction

External gear pumps exhibit a complex interrelationship between gear meshing, clearances between components and pressure pulsation that makes difficult the definition of general design procedures. As a consequence, large amounts of time and money are spent during the development of new designs. This requires huge testing efforts in order to refine the noise and vibration behaviour and simultaneously achieve good hydraulic efficiency. A good dynamic model could be a useful and powerful tool for the identification of noise and vibration sources and design improvement allowing "Design Right First Time" which leads to shorter time-to-market and reduced costs as compared to conventional "Test, Analyze & Fix". Following this objective, in Part I and II ([1,2]), the authors have presented a numerical model for the dynamic analysis of an external gear pump for automotive applications (called hereafter *PModel*). Fluid pressure distribution around the gears, which is time-varying, is computed and included as a resultant external force and torque acting on each gear. Gear meshing phenomena have received particular attention, the time-varying meshing stiffness [3] and the tooth profile errors, the effects of the backlash between meshing teeth, the oil squeeze and the possibility of tooth contact on both contact lines have been included in the model. One of the particular features of gear pump design is the use of hydrodynamic journal bearings