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Elasto-dynamic analysis of a gear pump–Part III: Experimental validation procedure and model extension to helical gears

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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 nonlinear 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's 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 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 globally, 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 evolution 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	centre distance of gear pair
b	face width of both gears
B	relief groove dimension
C_j	viscous damping coefficient of tooth pair j
C_T	torsional viscous damping coefficient of the driving shaft
E_j	profile error of tooth pair j
f_{bxk}	bearing reaction applied to gear k in direction X
f_{byk}	bearing reaction applied to gear k in direction Y
f_{mg}	meshing force
f_{mgj}	meshing force of tooth pair j
f_{mgm}	mean meshing force over one pitch
f_{pxk}	pressure force applied to gear k in direction X
f_{pyk}	pressure force applied to gear k in direction Y
F_P	high-pressure pipe force
g	backlash along the line of action
J_G	moment of inertia of pump and plate with respect to the centre of mass
J_k	moment of inertia of gear k
K_j	stiffness of tooth pair j in case of spur gears
K_T	torsional stiffness of the driving shaft
m_k	mass of gear k
M_m	motor driving torque
M_{pk}	pressure torque applied to gear k
m_{pp}	mass of pump and plate
r_{bk}	base radius of gear k
R_{pk}	radius of the pitch circle of gear k
$R_{X'12}$	sum of the force sensor reactions in X'_1 direction concerning sensors 1 and 2
$R_{X'34}$	sum of the force sensor reactions in X'_1 direction concerning sensors 3 and 4
$R_{Y'23}$	sum of the force sensor reactions in Y'_1 direction concerning sensors 2 and 3
$R_{Y'14}$	sum of the force sensor reactions in Y'_1 direction concerning sensors 1 and 4
t	periodic time ($0 \le t < T$).
T	meshing period.
$\ddot{x}'_C, \ddot{x}'_D$	C and D accelerometer accelerations in direction X'_1.
x_k, y_k	coordinates of the centre of gear k in reference frame $O_kX_kY_k$.
x^*_k, y^*_k	coordinates of the centre of gear k in the SEP in the reference frame $O_kX_kY_k$.
x'_G, y'_G	centre of mass of pump and plate in reference frame $O'_1X'_1Y'_1$
$\ddot{x}'_{O_1}, \ddot{y}'_{O_1}$	plate acceleration in point O_1 in direction X'_1 and Y'_1
$\ddot{y}'_A, \ddot{y}'_B$	A and B accelerometer accelerations in direction Y'_1

Greek symbols

α_w	pressure angle in working condition
β	helix angle on the pitch circle
$\tilde{\varepsilon}$	contact ratio
γ_m	proportionality constant between viscous damping coefficient and stiffness of a tooth pair
γ_T	proportionality constant between viscous damping coefficient and stiffness of the driving shaft
θ	angular coordinate
θ'	angular displacement of force plate and pump casing in reference frame $O'_1X'_1Y'_1$
θ_k	angular displacement of gear k in reference frame $O_kX_kY_k$
θ_0	angular displacement of the electrical drive in reference frame $O_kX_kY_k$
ω_k	mean angular speed of gear k in steady-state operational conditions

Subscripts

$j=a, b, c, d$	denotes pairs of teeth
$k=1, 2$	denotes gears.
$_L$	stands for linearized

1. Introduction

Noise Vibration and Harshness (NVH) is one of the most important topics in the automotive industry [3]. Acoustic and vibration comfort is another of the key features in a new design and consequently suppliers of automotive components need to satisfy more restrictive requirements defined by final manufacturers. Moreover, in order to achieve the goals defined by the final customer, the component suppliers need to increment the resources on design and testing tasks. This scenario increases the interest on modelling the dynamic behaviour of mechanical systems as a way to improve the initial design reducing testing efforts. A good dynamic model could be a useful and powerful tool for the identification of noise and vibration sources and for design improvements. Models should be simple but at the same time should provide a prediction of the dynamic behaviour in a sufficiently accurate and reliable fashion. The development of this kind of tool requires a good analysis of the system in order to define the most important phenomena that must be included in the model as well as the formulation or modification of theories that allow the description of each system element.

In this work a numerical model of the vibratory behaviour of an external gear pump for automotive application is presented. Part I and II ([1,2]) describe in detail the non-linear lumped-parameter kineto-elastodynamic model. Part III is devoted to the experimental validation and to the model extension to helical gears; Part IV presents the pressure formulation improvements.