

A NUMERICAL MODEL FOR THE SIMULATION OF EXTERNAL GEAR PUMPS

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ABSTRACT

External gear pumps are the most widely employed source of power in hydraulic applications, allowing good performance and low manufacturing costs. The present paper reports the description of a numerical model for the simulation of these kind of machines, developed by the authors. The model has been implemented using the software AMESim[®], introducing new in-house C++ models, with the main aim of an accurate prediction of the pressure ripple at the delivery port and the calculation the flow rate time course through the pump. In fact the reduction of flow pulsations is one of the crucial targets in the development of these machines.

The fluid dynamic model of the pump is based on a finite volumes framework: each chamber presents uniform fluid properties and is connected with the adjoining ones by variable orifices. The pump is described by the interaction between a fluid dynamic model, that predicts the instantaneous pressures and the flow rates between the chambers, with a geometrical sub-model for the evaluation of the actual values of the variable volumes (defined by teeth, housing, and side wear plates) and the throat areas, as functions of the shaft angular position.

This paper reports a deep description of the fluid dynamic model and a comparison between its numerical results and data available from experiments, showing how a correct evaluation of the flow when the chambers expand (and oil is pushed from the supply tank) or decrease (and oil is ejected into the outlet port) allows a good prediction of the pump characteristics.

The results reported in the paper show how the model can be a powerful tool for design and development of external gear pumps, e.g. it easily allows the prediction of the maximum and minimum pressure reached inside the teeth space volumes, giving important information to prevent cavitation and damages as well as to reduce the pump noise. Moreover the simulation environment easily allows the evaluation of the pump behaviour when it works in a given hydraulic system.

KEY WORDS

External gear pump, simulation, flow ripple, pressure ripple

NOMENCLATURE

n : shaft speed
 p : pressure
 p_m : pump maximum constant pressure
 t : time
 V : volume
 \dot{V} : volume flow rate
 ρ : fluid density
 \mathcal{G} : angular position
 ω : angular velocity

Subscripts

in : entering
 out : leaving

1,2 : driver gear (1) and driven gear (2)
 i,j : indexes of tooth space volume

Abbreviations

CV: control volume

INTRODUCTION

Fixed displacement pumps remain the most widely used sources of power in hydraulic applications, despite many present researches oriented to the evolution and cost reduction of variable displacement machines. In this perspective external gear pumps still have the role of prime actor. These machines combine the manufacturing simplicity, leading to reduced costs, with good efficiencies;

moreover their application area is extremely wide [1,2] spacing from the low pressure (such as the case of charge or lubricating pumps) to the medium-high pressure applications, including open loop and closed loop circuits. Notwithstanding gear pumps are relatively simple to manufacture, their operation involves a lot of complex phenomena of both mechanical and fluid nature; for this reason these pumps have been captivating the interest of many researchers for more than five decades. The first important studies pertain to Wilson [3] and Castellani [4]; in the late eighties Nervegna and Mancó proposed a simulation model for external gear pumps [5] and some studies about the experimental evaluation of the pressure transients [6] in the machine. After this time, several papers [7-10] concerned the analysis of the inter-teeth pressure, of the leakages, of the pressure and the force distribution on the gear side faces. Other researches analyzed the performance of the pump bearings and the fluid borne noise generated at the suction port [11,12], while in [13] the effects of changing teeth geometry on pump flow ripple is described.

Nowadays at the Industrial Engineering Dept. of the University of Parma the authors are developing a simulation model for external gear pumps in cooperation with Casappa S.p.A., an important Italian fluid power industry. The present paper refers to the first step of the research, presenting the basic model structure and the first significant results that have been reached.

For the model implementation the authors have chosen the simulation software AMESim[®]. Some new in-house modules have been created using AMESet[®], writing sections of code in C++ language. These have been added to a new user-defined library that is used to build up the complete model of the pump. The model is lumped parameter based; the chosen framework has the purpose of evaluating the flow and pressure course at the suction and delivery port and also in each tooth space volume and the leakages within the pump. The geometrical calculations were performed with CAD/CAE based tools provided by the manufacturer. These tools, not described in the paper, interact with the developed AMESim[®] model for the evaluation of all the pump geometrical characteristics (namely the flow areas between the volumes and the instantaneous volume entity) as a function of the position of the gears, accounting for the real shape of teeth and the geometry of the balancing bearing blocks.

In order to verify and calibrate the model, experimental test were performed on a stock pump using a rig specific for pump characterization.

NUMERICAL MODEL DESCRIPTION

The lumped parameter framework

According to the chosen approach, the pump is subdivided in a number of control volumes in which fluid properties are assumed uniform and only time dependent. As reported in fig. 1a, the model considers a control volume (CV) for each tooth space volume of both gears. Under the hypothesis of same number of teeth on the driver and driven gears, fig. 1a highlights how, as the shaft rotates, the generic tooth space volume $V_{1,i}$ of driver gear always meshes with the corresponding $V_{2,i}$ of the driven gear. The

solution adopted to describe the fluid dynamics of two conjugated teeth space volumes $V_{1,i}$ and $V_{2,i}$ is similar to the one presented in [7,10], although the method developed in the mentioned work is suitable only for the prediction of the inter teeth volumes pressure transient, while in this work the model is conceived to characterize the whole pump operating. Following each tooth space volume as a separated CV during the whole rotation of the gear, the model differs also from the scheme proposed in [5], which analyzes the pump using an eulerian approach for the CVs. Fig. 1b shows the definition of CVs for the inlet and the delivery volumes of the pump.

The model takes into account the connections between every tooth space volume with its surroundings, and the changing of net volume in the meshing zone. Eq. (1) gives the pressure course inside a generic CV as a function of fluid properties, the geometric volume variation and the net mass transfer with the adjacent CVs.

$$\frac{dp_{CV}}{dt} = \frac{1}{V_{CV}} \frac{dp}{d\rho} \left[\sum \dot{m}_{in,CV} - \sum \dot{m}_{out,CV} - \rho_{CV} \frac{dV_{CV}}{d\theta} \omega \right] \quad (1)$$

The flow areas connecting each tooth space volume with its surroundings and the actual values of volumes are considered depending on the shaft angular position.

Fig. 2 summarizes the framework on which the model is based. During a shaft rotation teeth go into mesh and the actual value of each CV changes; besides, with the gear rotation, it can be connected to several other chambers by means of variable orifices, whose areas follow a precise trend.

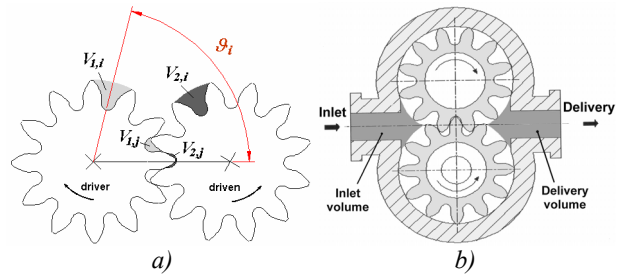


Figure 1 – Control volumes assumed in the model: teeth space volumes (a); inlet and delivery volumes (b)

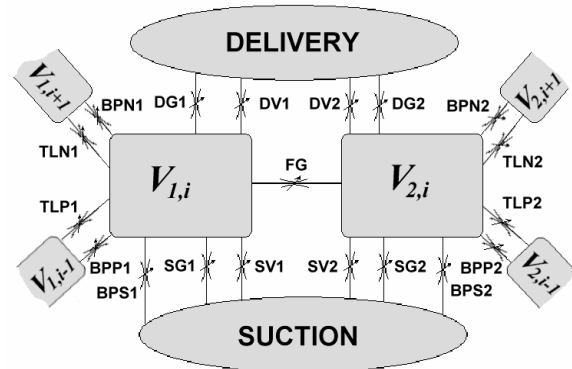


Figure 2 – Basic scheme of connections between the CVs of corresponding teeth space volumes

Referring to fig. 2, abbreviations have been adopted to identify the different flow areas:

- FG represents the connection between the two corresponding teeth space volumes when they are meshing (fig. 3a);
- DV1 (or DV2) is the connection between $V_{1,i}$ (or $V_{2,i}$) and the delivery volume through the gear whole depth (fig. 3a);
- SV1 (or SV2) is the connection at the opposite side, to the suction volume (fig. 3a);
- DG1 (or DG2) represents the connection between V_1 (or V_2) and delivery port through the bearing blocks recesses (fig. 3b). Fig. 4 shows also how the possible presence of side grooves (commonly used to compensate axial forces) can be considered by the DG1 (or DG2) orifices (DG1' and DG2' indicate the backflow grooves, if they are present);
- SG1 (or SG2) is the analogous connection with the suction port (fig. 3b);
- TLP1 and TLN1 (or TLP2 and TLN2) pertain to the leakages between adjacent teeth space volumes, due to the clearances between teeth tip and pump casing, as shown in fig. 5a (TLN refers to the next tooth space volume, while TLP to the previous one);
- BPP1 and BPS1 (or BPP2 and BPS2) refers to leakages through the lateral clearances between gear side faces and bearing blocks internal surfaces, as shown in fig. 5b (BPP: flow between the volume and the previous one, BPN: flow with the next volume);
- BPS1 (or BPS2) indicates the leakage flow between the tooth space volume $V_{1,i}$ and the drain line (fig. 5b), which, in the analyzed case, returns to the suction line.

A correct evaluation of the effect of orifices displayed in figs. 3 and 4 is necessary to analyse a solution widely used for the reduction of the pressure peak due to the compression of the inter teeth trapped volume between two adjacent contact points. This solution consists in a proper design of recesses [4,5], that makes the trapped volume never isolate from inlet and delivery (e.g. fig. 6). As shown in fig. 6, SG2 comes up before the closure of DG1. In this way pressure peaks – or excessive pressure drops – are avoided, but there is a flow that comes back from delivery to the inlet port. For this reason it is important to obtain a proper design of recesses inducing a tiny loss of flow while the inter teeth pressure does not rise too much.

As explained above, every connection between each control volume is accounted as equivalent orifice, whose area is described as a function of the angular position \mathcal{G}_i of the tooth space volume $V_{1,i}$ (see fig. 1a). In particular, that function is null outside a defined interval (for example for each $V_{1,i}$ and $V_{2,i}$ FG is greater than zero only in the meshing zone). Consequently, a correct implementation of the model strictly depends on the accurate evaluation of these geometrical functions.

Flow rates between adjacent volumes are evaluated considering the incompressible steady-state turbulent flow equation for orifices; according to the solution implemented in AMESim® standard models (described in [14-16]); the influence of the efflux dynamics on the correlation between \dot{V} and Δp is described by a discharge coefficient that depends on Reynold's number.

Leakage flows (represented in fig. 2 by BPN, BPP, BPS,

TLN, TLP), are approximated by the modified Poiseuille equation, for fully developed laminar conditions, accounting of relative motion of boundary surfaces [14].

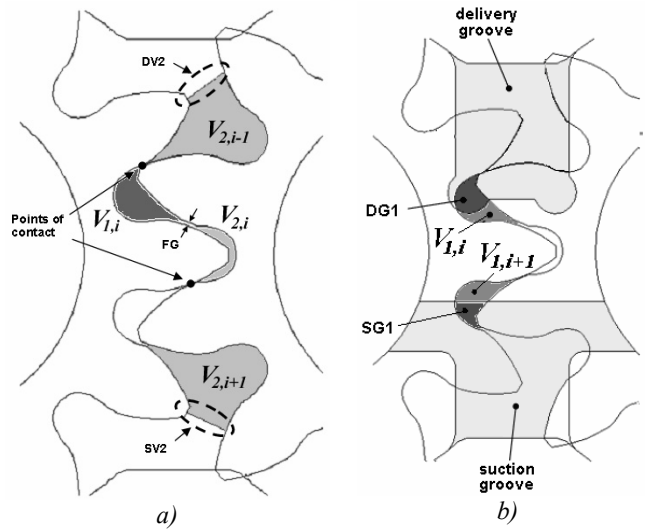


Figure 3 – a) connection between V_2 and delivery (DV2), suction (SV2) and V_1 (FG). b) connection between V_1 and suction (SG1) and delivery (DG1) through the recesses in the bearing blocks

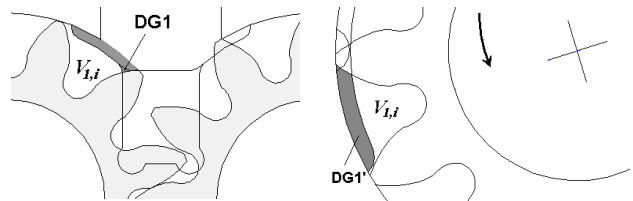


Figure 4 – Particular connections represented by DG1 due to the delivery groove and to the backflow grooves (DG1')

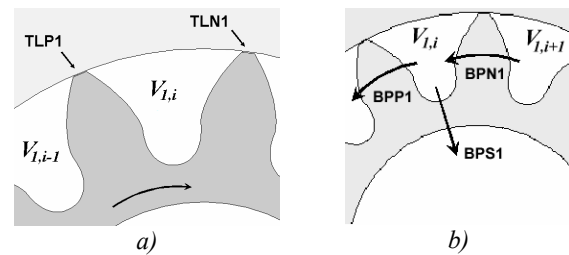


Figure 5 – Evaluation of leakages: between adjacent teeth space volumes (a) and in the clearances bounded by gears lateral sides and bearing blocks internal surfaces (b)

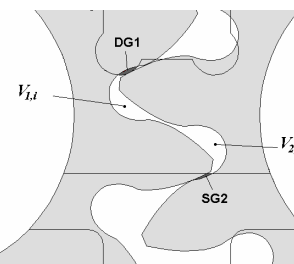


Figure 6 – Inter-teeth trapped volume between two adjacent contact points in the meshing zone

Model implementation

In fig. 7 the AMESim[®] diagram of the entire pump is presented. A major advantage of the generated pump simulation model lies in its suitability as a core element (supercomponent) inside the AMESim[®] simulation environment: it is therefore possible to study the behaviour of the machine in several circuit dispositions appraising the pump-circuit interaction. The majority of sub-models have been developed in-house, in C language, through the AMESet facilities [16]. The simulation model follows the framework of fig. 2, it can be easily adapted to different external gear pump geometries. The model has been conceived in order to allow the user to describe the pump by a number of parameters easy to define. Main pump features (e.g. number of teeth per gears, throat areas of inlet and delivery ports, etc.) and pump operating conditions are declared at the beginning of each simulation. The most important sub-models in the sketch of fig. 7 are indicated with the label “Multi Ch”. In the sketch of the model, the upper “Multi Ch” model refers to driver gear (fig. 1), while the lower pertains to the driven gear. The C-routine of such model implements the scheme displayed in the figure: the “Multi Ch” icon represents an array of hydraulic chambers (CVs) whose behaviour is governed by eq. (1). The course of the volume of each chamber and its first derivative come from an external ASCII file, while mass flow rates exchanged with the connected volumes are evaluated with external orifices models (only the leakage flows TLN_i and TLP_i are implemented into the sub-model itself). All these orifices are modelled with a in-house “Multi Orifice” component (fig. 7), in which a number of

variable orifices equal to the number of teeth of each gear is implemented in a C language subroutine. The variable chambers located at the bottom left and bottom right of the model (fig. 7) compensate the discrepancies of volume that would come from the chosen volume framework. In fact, when two teeth start meshing, there is a portion of volume that instantaneously passes from the tooth to the delivery (or suction) volume, as represented in fig. 8.

A prerequisite of the described model, in order to achieve a good prediction of flow as a function of pump operating conditions, is the knowledge of all necessary geometrical data, namely the variable volumes of teeth space and the throat areas of variable orifices, as a function of gear position. In the literature (e.g. [5,8]) it is possible to find simplified analytic expressions of teeth space volumes as a function of gear angular position, however the real trend and the throat areas of all orifices are strongly dependent on design of gear and recesses in the bearings plates. In this work, with the aim of comparing simulation results with available experimental data, a deep analysis of pump geometry for a particular stock pump was carried out. In particular, the evaluation of all needed geometrical data was performed through CAD/CAE tools, starting from the drawings.

RESULTS

The model results have been verified through the comparison with some available experimental data. In fact test campaigns were carried out on a stock pump whose main features are summarized in tab. 1. Details concerning the test rig utilized are reported in [17].

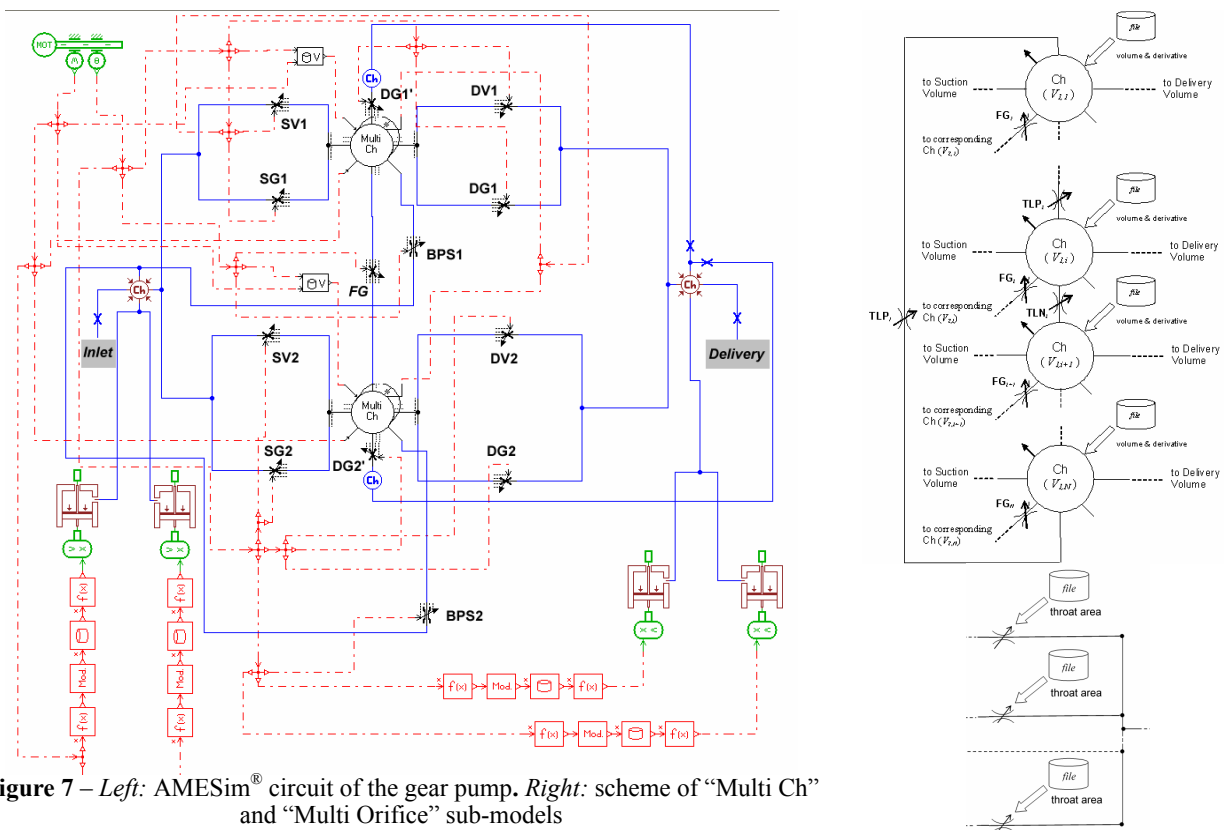


Figure 7 – Left: AMESim[®] circuit of the gear pump. Right: scheme of “Multi Ch” and “Multi Orifice” sub-models

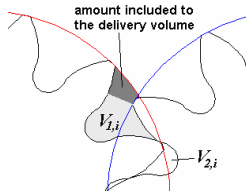


Figure 8 – Portion of volume assigned to delivery



pump name	PLP20.11.2 standard	
displacement	[cm ³ /r]	11.23
maximum speed	[r/min]	3500
minimum speed	[r/min]	600
number of teeth per gear		12

Table 1 – Main features of the pump

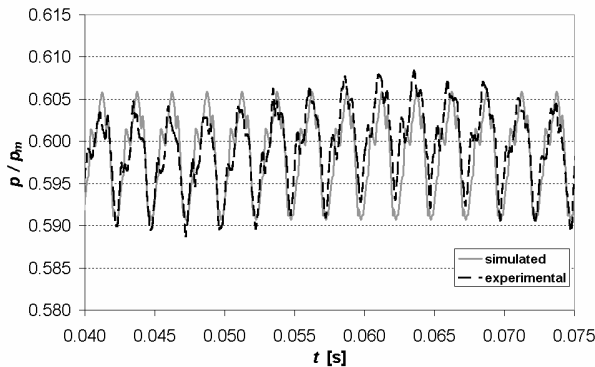


Figure 9 – Delivery pressure ($n = 2000$ r/min)

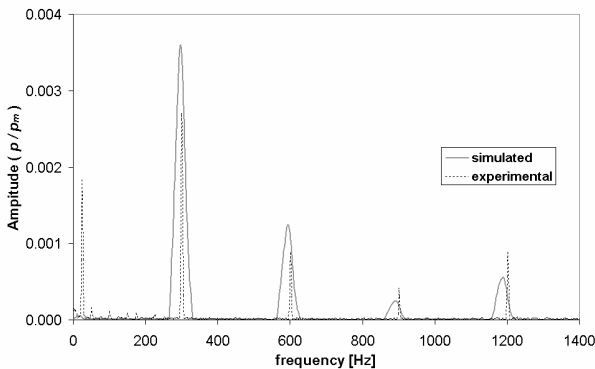


Figure 10 – FFT courses ($n = 1500$ r/min, $p/p_m \approx 0.8$)

The pressure measured at pump delivery, for a constant pump speed, and the measured pump flow rate were later compared with the numerical data, obtained simulating the whole test circuit used for pump characterization. In particular the delivery pipeline downstream of the pump was modelled through a distributed line models that account for wave propagation and frequency dependent friction phenomena, according to the standard models provided by the software [16].

Calibration parameters required by the code, namely all the discharge coefficients, were first assumed from the data reported in [7,18] for similar conditions, later they were slightly adjusted to better match with test data. The model has been validated through the comparison between predicted and measured pressure at the delivery port for

several working conditions (e. g. fig. 9). These simulations were performed assuming the pump operating with the nominal geometrical parameters, neglecting the deformation of pump structure with pressure and the initial period of adjustment. From fig. 9, the 12 teeth of each gear are noticeable observing the 12 pressure peaks during a shaft revolution (about 0.03 s at 2000 r/min), corresponding to 12 couples of teeth going into mesh. For all the considered cases the pressure ripple characteristic of this pump is well reproduced in detail by the simulation code. As displayed also in the representation of fig. 10, the experimental course highlights a low frequency trend – not observable in the calculated data – with a period equal to the time required for an entire gear revolution. Other peaks in the spectrum are well predicted, taking into account the approximation imposed by the windows method used for the FFT calculation to avoid leakage. The reason of this discrepancy could be the eccentricity of gears caused by the dimensional tolerances.

One of the most important results provided by the simulations is the trend of pressure inside a tooth space volume (fig. 11). The figure shows both pressure and tooth space volume versus gear angular position. From the figure it is possible to observe how the volume decreases when the tooth goes into mesh, with the typically parabolic trend (similar to those reported in [5,10]). The figure is very useful for understanding the working concept of the pump: starting from the condition of tooth volume outside the meshing zone (for ϑ approximately greater than 30°), it is possible to notice how the pressure begins to increase as effect of the aforementioned leakage flows (fig. 5) when the pump casing starts covering the tooth tip. The pressure suddenly reaches the delivery pressure when the backflow groove (DG1' see fig. 3) starts working (for $\vartheta \approx 150^\circ$, point B of fig. 11). The described increase follows an initial smooth increment (from point A), caused by the leakage from the adjacent tooth space volume in which pressure has already reached the delivery value. After point B, the pressure inside V_i follows the same trend as the delivery one, until the tooth reaches again the meshing zone. In fact in this period V_i is connected with the delivery through DG1' and DG1, or DV1 (figs. 3 and 4). Fig. 11 points out the effect of the shape of the backflow groove adopted in the analyzed pump (whose geometry is confidential): it realizes a further connection between the tooth space volumes in the zone marked with the letter C. The final peak marked with D corresponds to the minimum value of tooth space volume, reached in the meshing zone. As previously described, the maximum pressure value directly affects noise and fluctuations at pump delivery, so that its prediction is fundamental for the evaluation of pump performance. It is also important, in order to verify the design of the recesses and avoid any conditions of trapped volume isolated from inlet or delivery, the prediction of the minimum pressure value related to the quick increase of volume V_i (zone E). In this case the model is useful to predict any possible onset of cavitation.

Figure 12 shows the calculated flow rate at the delivery port. This result permits the estimation of important parameters as the flow ripple and the volumetric efficiency. In particular, the course of \dot{V} in fig. 12 gives a time average value of flow rate of about 15.4 l/min; this results

is consistent with the measured volumetric efficiency of this pump, even if the model deals with the leakages in a simple manner.

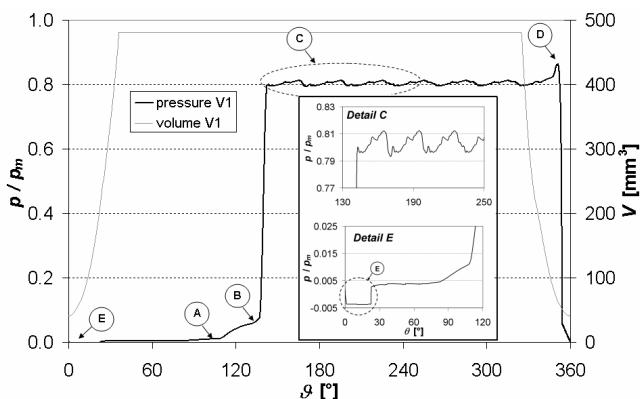


Figure 11 - Simulated pressure inside a tooth space volume V_1 ($n = 1500$ r/min, delivery pressure = 200 bar)

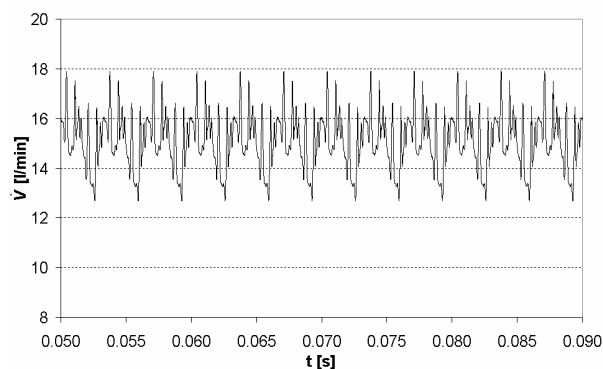


Figure 12 – Predicted mass flow rate at pump delivery ($n = 1500$ r/min, $p = 200$ bar)

CONCLUSION

This paper describes a numerical model for the simulation of external gear pumps. The model, implemented in AMESim[®] environment, is based on a lumped parameter framework and it stands out for its ease of use and simulation swiftness. The model has been developed and validated using as reference a commercial pump produced by Casappa S.p.A. Results provided by the model permit to analyze the pump operation deeply, showing the trend of some parameters which are difficult to measure (e.g. the inter-teeth pressure transient) and giving plenty of useful information to understand the influence of the main design parameters (e.g. the design of the recesses on the bearing blocks). Despite the model requires an accurate definition of all the pump geometrical features (as functions of the gear position) it easily allows the characterization of the fluid dynamics of the whole pump, with a wide versatility of simulation. Future works will take advantage of the approach adopted for the definition of the control volumes in the model for prediction of dissymmetric geometrical conditions (for example due to possible damage of some teeth, or to consider the actual eccentricity of gears while pump is operating).

ACKNOWLEDGEMENTS

The authors are grateful to Prof. G.L. Berta for his precious suggestions and help and to M. Guidetti (Casappa S.p.A.) for test and experimental data.

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