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Research Paper

Investigations of mechanical behavior in gear pump using design of software

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Abstract

This research study focused on the various aspects of tribological phenomenon using design of experiments approach, results are analyzed and compared to evaluate the gear performance. Though lot of research work had been carried out on the hydrodynamic gear under static loads, till date no one consider the parameter say stiffness, damping co-efficient, squeeze film effects and the pressure distribution at various crank angle under dynamic load conditions and its effects on inertia forces of gear pump. It covers sinusoidal load, journal speed, face width and mass flow rate with respect to variations in oil film thickness.
Frictional test rig was used to measure the frictional force and oil film thickness at various crank angles for every three minutes at an interval of every 15 degree crank angle. The response surface methodology (RSM) analysis shows appreciable increases in respect of performances in mass flow rate, face width, speed and loading against the Taguchi design of experiment results whereas the film thickness is almost same in both RSM and Taguchi design of experiment results. Particular attention is given in this research exertion to learn how the variation in gear dimensions influences the characteristics of the hydrodynamic gear pump.

Keywords: *Hydrodynamic Gear Pump, Oil Film Thickness, Mathematical Modeling, Taghuchi and RSM, Eddycurrent Cap Sensor*.

1. Introduction

A few researches has been carried out in dynamic analysis of gear pumping this work concerns external gear pumps for automotive applications, which operate at high speed and low pressure and presents a sensitivity analysis[3] about the influence of design and operational parameters on the pump dynamic behavior. The mathematical modeling was developed for eccentricity of shaft and pump, with the aim of including all the important effects, as well as to get a rather simple mode [2] l. There are three main sources of noise, vibration and friction considered: Angular pressure variation and gear meshing. The model has been taken in to account for comparison with experimental vibration data, in a wide range of operational parameters condition [1] s for several profile errors. This concept is concentrated on the dynamic analysis on the influence of the main design and operational parameters on the pump dynamic behavior.
In particular, the effect of operational pressure, speed and friction, the influence of the clearance in the hydrodynamic gear between tooth tip and pump case, and the effect of the operational parameters like stiffness, damping, coefficient of friction [6], film thickness, speed, friction, load, and torque of the gear will be thoroughly discussed in the mechanical behavior. Finally, the mathematical model and design will be a very useful and powerful tool in order to evaluate the dynamic improvements for noise [4], vibration and friction reduction.
The gear pumps are among the oldest and most commonly used pumps in the industry. It has become the main choice for fuel system designers due to long life, minimum maintenance, and high reliability, capability to operate with low lubricating fuel [3], low heat input to fuel, and low weight. 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, vibration and friction levels and unavoidable output pressure ripple [9].

For these reasons, the researchers have applied eddy current cap sensor with lumped-parameter on hydro dynamic model in order to study the dynamic behavior of an external gear pumps for automotive applications [6]. 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. Gear meshing phenomena have received particular attention in particular the time-varying meshing stiffness, damping effect and the tooth profile errors like [5] module, addendum, dedundum with respective pressure angle and arc of path of contact 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. One of the particular features of gear pump design is the use of hydrodynamic gear shaft support [12].
The non-linear dynamics of this kind of gear has been modeled using the theory of Childs called "finite impedance formulation and mathematical modeling". So, the model is highly non-linear [7]. The dynamic equations of motion for each degree of freedom are obtained in and are numerically integrated in Simulink environment. With the aim of reducing the integration time, the average positions of the shaft axes inside the journal gear are estimated before the integration of the dynamic model [8]. 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 [11] (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.

2. Materials and methods

2.1. Pump description

The pump body has been built in cast iron and material of the gear is steel. The tooth number of gear 1 is the same as that of gear 2. The Control Volumes (CVs) within the tooth gaps are variables dependent on the gear angle [3]. The subscripts are the index of these Control Volumes. The Outlet and Inlet volumes are two fixed volumes at the outlet and inlet ports respectively.
To produce a flow within a gear pump, fluid is carried by the CVs from the intake side of the pump to the discharge side of the pump through the transi-tion zone. As the gears rotate, these CVs increase their pressure to when reach the high-pressure chamber [11]. As the gear teeth mesh in the meshing zone, flu-id is squeezed out of each tooth gap by the mating tooth. When two tooth pairs contact, a trapped volume is generated. This may cause positive pressure peaks and the onset of cavitations. On the intake side, the gear teeth are coming out of the mesh [4]. The volumes of CVs increase so that fluid is inhaled into the tooth gaps. This process repeats itself for each revolution of the pump and therefore displaces fluid at a rate proportional to the pump speed.

2.2. Major contribution

In this section, we first give an overview of the pump model. Some components come from a free library HyLibLight based on Modelica [12].
The aim of this paper is to construct a hybrid model of the external gear pumps. It considers the leakages, compressibility of the oil, flow ripple, pressure distribution, etc. Specifically, the pump is modeled under the following assumptions:

1) The gears and the housing case are rigid; only the oil is compressible
2) The position of the shaft is known before the simulation and is fixed during the simulation.
3) The pressure at every single isolated region and fixed volume is well-proportioned.
4) The temperatures in all CVs are the same and constant.
5) The tooth numbers of the two gears are the same.

A pump is divided into 2n+2 control volumes, where n is the tooth number of a gear. Figure 1 shows he flows between those control volumes. This method is similar to but not the same as that in. There is no variation in the number of the control volumes which causes variation in the number of differential equations.

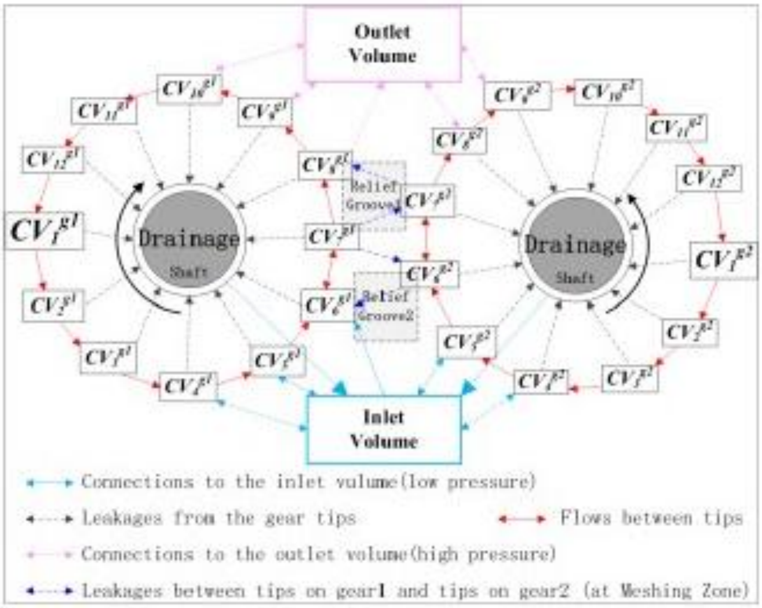


Fig. 1: Shows the Flows between those Control Volume

2.2.1. Experimental Data Analysis

Table 1: L9 Orthogonal Array

Exp. No	A	B	C	D
1	1	1	1	1
2	1	2	2	2
3	1	3	3	3
4	2	1	2	3
5	2	2	3	1
6	2	3	1	2
7	3	1	3	2
8	3	2	1	3
9	3	3	2	1

Table 2: OA of Experiment

Exp. No	A	B	C	D
1	1.3	25.50	1000	01.0
2	1.3	26.5	1100	10.0
3	1.3	28.00	1200	16.0
4	1.5	25.50	1100	16.0
5	1.5	26.5	1200	01.0
6	1.5	28.00	1000	10.0
7	1.7	25.50	1200	10.0
8	1.7	26.5	1000	16.0
9	1.7	28.00	1100	01.0

Table 3: Oil Film Thickness Values and S/N Ratios against Trial Numbers

Exp. No	Repetition 1	Repetition 2	Repetition 3	S/N ratio
1	0.025	0.022	0.028	-29.70
2	0.024	0.024	0.027	-29.40
3	0.022	0.022	0.023	-29.90
4	0.039	0.036	0.038	-28.10
5	0.020	0.046	0.048	-25.00
6	0.045	0.048	0.048	-26.20
7	0.019	0.013	0.015	-31.40
8	0.024	0.017	0.016	-31.20
9	0.018	0.014	0.016	-31.20

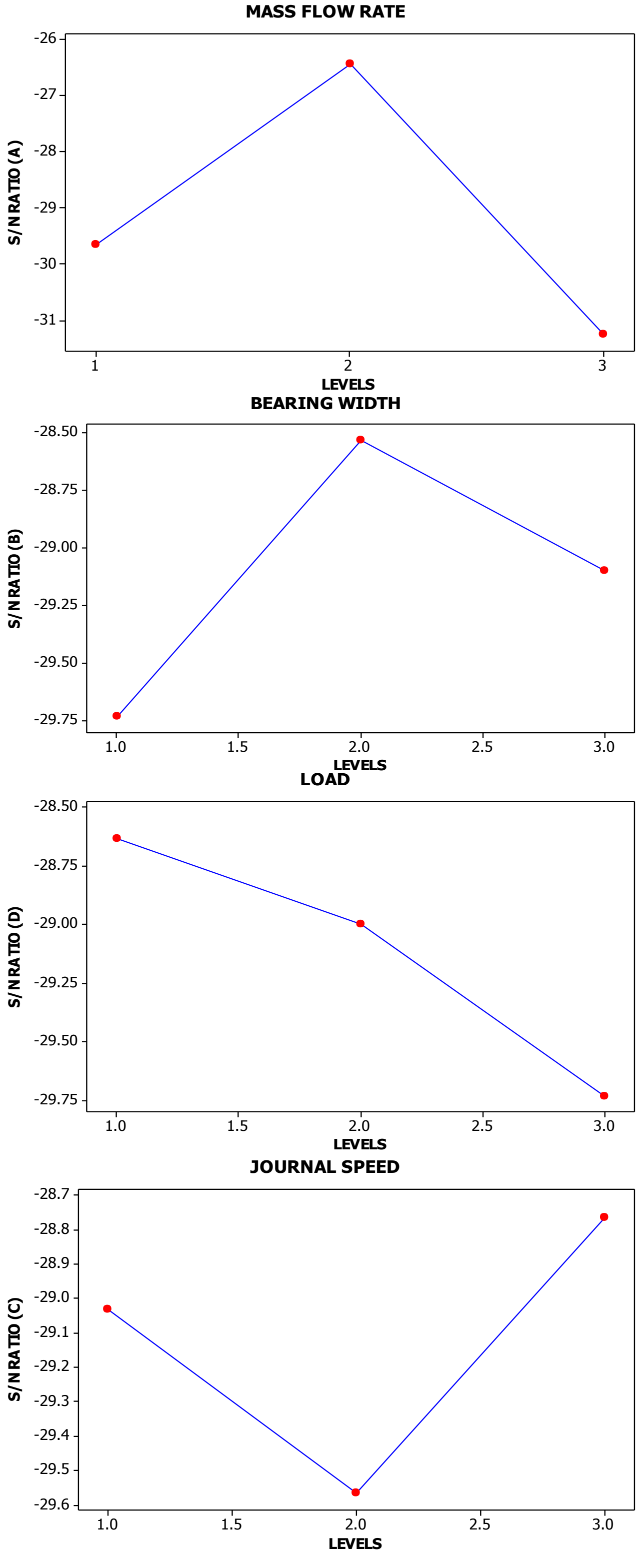


Fig. 2: Average Values of S/N Ratios for Each Parameter

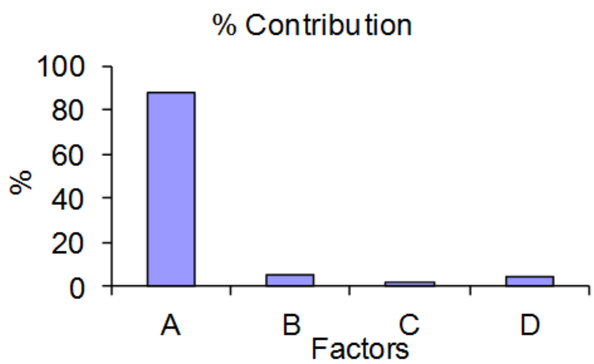


Fig. 3: Pareto Chart for % Contribution of Factor

Table 4: Anova Table

Source of Variation	Sum of Squares	DOF	Mean	clearance factor	Tangential at 5% level
A	36.38	2	18.19	16.84259	19
B	2.16	2	1.08	1	--
C	1	2	0.5	0.462963	19
D	1.88	2	0.94	0.87037	19

Table 4: Anova Calculations - Refer

Column	1	2	3	4	
FACTOR	A1 A2 A3	B1 B2 B3	C1 C2 C3	D1 D2 D3	
LEVEL SUM	-89.00 -79.00 -93.80	-89.20 -85.60 -87.30	-87.10 -88.70 -86.30	-85.90 -87.00 -89.20	
Total Sum	-262.10	-262.10	-262.10	-262.10	
Square	7921 6288.49 8798.44	7956 7327.36 7621.29	7586 17867 7447	7378 17569 956.64	
Sum of Square	23007.93	22905.29	22901.79	22904.45	
Mean Sum of Square	7669.31	7635.10	7633.93	7634.82	
Correction	7632.93	7632.93	7632.93	7632.93	Total
Total Sum of Squares	36.38	2.16	1.00	1.88	41.42
Contribution Ratio	0.88	0.05	0.02	0.05	
%of Contribution	87.83	5.22	2.40	4.54	100.00

Design of Surface Software

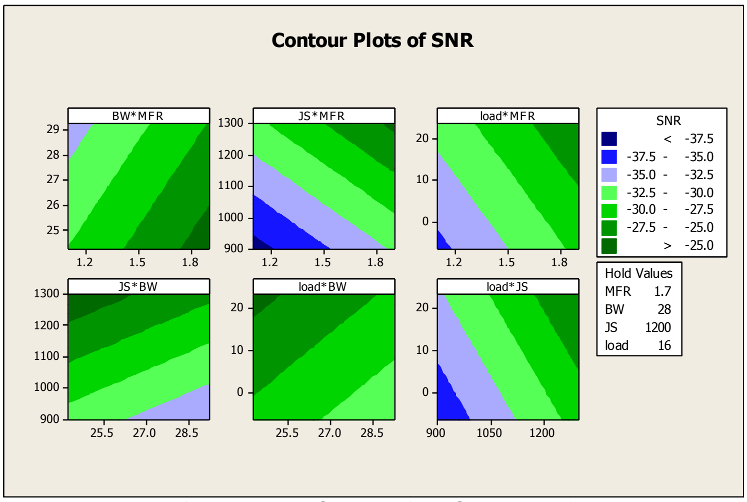


Fig. 4: Contour Plots SNR (Speed Noise Ratio)

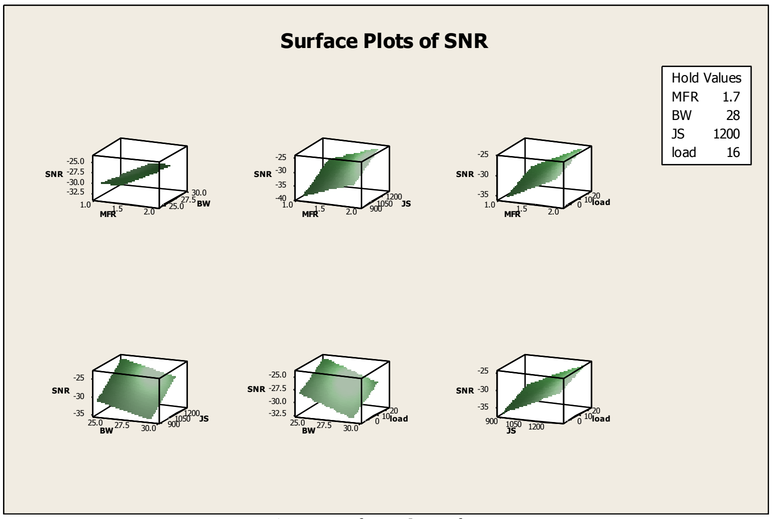


Fig. 5: Surface Plots of SNR

2.3. Influence of operational parameters: pressure and rotational speed, load, friction & torque

Figure 6 and Figure 7a 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 bars the circle in Figure 6 represents the maximum displacement of gear axes, allowed by the nominal clearance (hrn) of 0.0245 mm between gears and pump case.

The results depicted in Figure 6 show that as the output pressure increases, the eccentricity modulus of both gears increases, in fact the markers in Figure 6 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[8]. In fact, as the pressure forces increase, the new axis centre locations move reducing the actual clearance in the journal bearings 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 7a), exhibits a sudden variation in the first tooth spaces as depicted, due to the eccentricity direction, the minimum meatus height is located in the first tooth spaces [9]. 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 depend on the pressure drop Δ piBetween two consecutive vanes and on the meatus height hi Qi ∝ hi·Δ pi, subscript i refers to the generic meatus while the meatus height strictlydepends 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 meta as inversely proportional to h 3[11] . 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 3 in such a way that the sudden variation of the Normalized pressure distribution is more anticipated in the first isolated spaces.

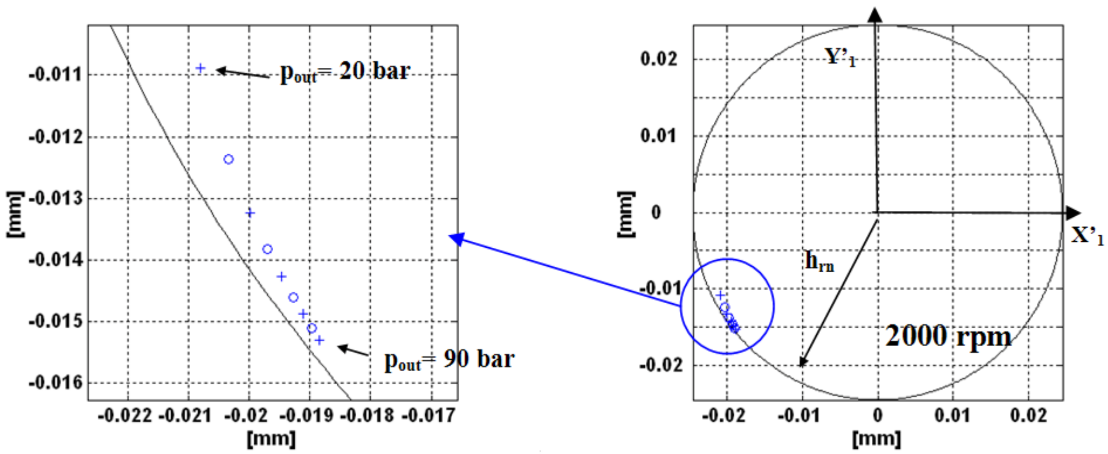
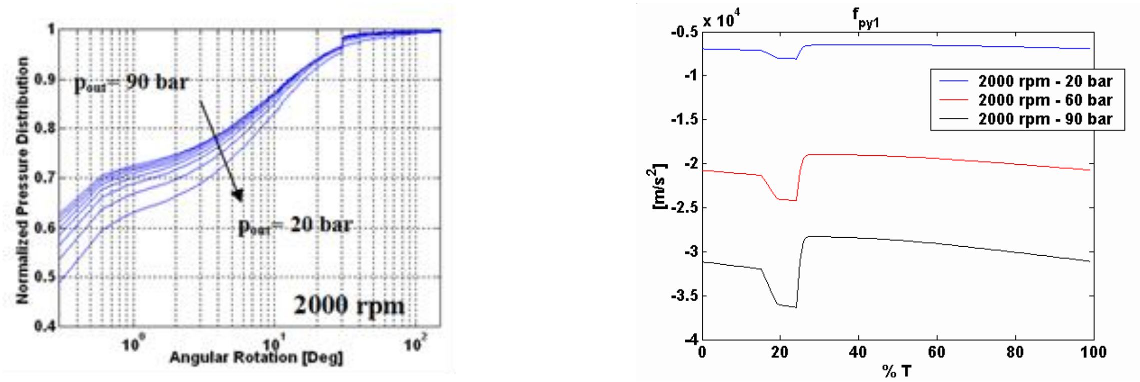


Fig. 6: Characteristics Curve for Pressure Variation and Speed Load, Friction

Figure 2 Stationary centre position for the driving gear with different output pressure values; the representation is in the reference frame of the circle represents the maximum displacement of gear axes, allowed by the nominal clearance hrn = 0.0245 mm between gears and pump case [12].

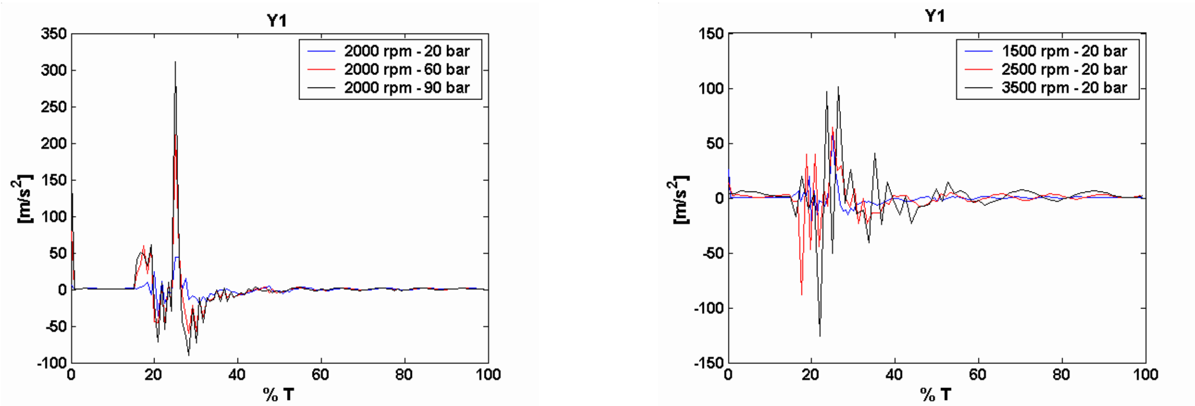
Furthermore, Figure 3b shows the influence of the pressure variation on the pressure force concerning the driving gear in Y-direction, for one meshing period T, starting at the instant when the second tooth pair comes in contact. It can be noted that the pressure force is noticeably affected by an increase of output pressure due to their direct dependence on sure distribution; the same behavior occurs for the other directions and for the driven gear, not shown hereafter.

On the other hand, for the gear accelerations, depicted in Figure 8a, 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 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.



(A) (B)

Fig. 7: Normalized Pressure Distribution (A) and Pressure Force in Y1-Direction (B) in Gear 1 for Different Output Pressure Values



(A) (B)

Fig. 8: Acceleration for the Driving Gear In Y1-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

In fact, as the rotational speed decreases, the bearing reaction tends to decrease too, due to the direct relation with the operational speed and the new axis centre location will change, decreasing the actual clearance in the gear this way, a new equilibrium between the dynamic forces is established [5]. 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, for analogous reasons. In addition, Figure 4b shows the influence of the operational speed variation on the acceleration in Y1-direction for gear 1: it can be noted that an increase of the operational speed determines an increase of the gear acceleration.

2.4. Effect of design parameters

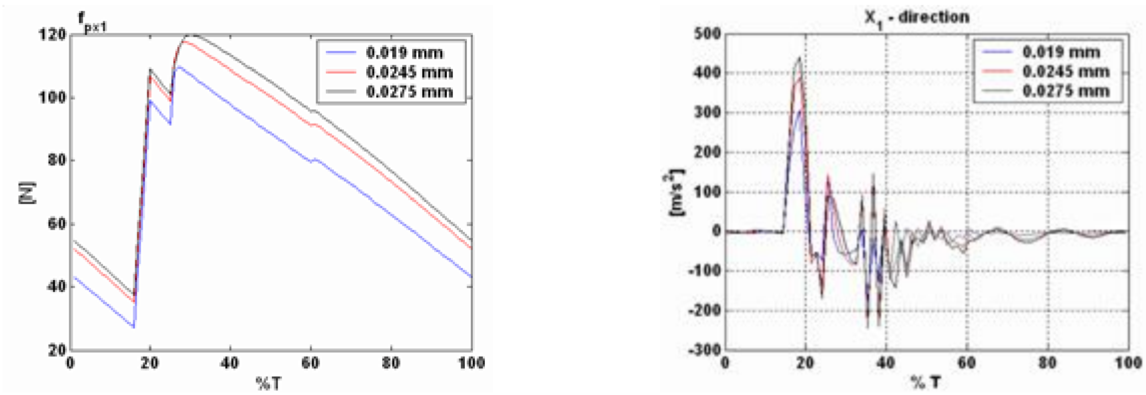
2.4.1. Effect of clearance modifications

In the pump under testing, the nominal clearance hrn between pump case and tooth tip has the same value as the nominal clearance Cr in the journal bearing, equal to 0.0245 mm [10]. 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 Cr and hrn will be shown. Reference 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 [3]. 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 1Collects the values of the actual minimum clearance Cr , χ in the journal bearings that obviously depends not only on the nominal clearances Cr but also on the eccentricity (Cr , $\chi = (1 - \chi)$ Cr). It can be noted that as the nominal clearance Cr 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)[5]. This pressure distribution behavior can be explained as a consequence of the increase of the eccentricity modulus (Table 1) which alters the ratios between two consecutives meatus heights (h3) 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 [6]. 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 [8], 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 5shows the clearance influence on stationary centre position (in modulus and direction angle) at 3350 rpm and 20 bars for gear 1 and influence on the actual minimum clearance.

Table 5: Clearance Influence on Stationary Centre Position

Nominal clearance [mm]	0.019	0.0245	0.0275
Eccentricity modulus χ (normalized)	0.900	0.935	0.946
Actual minimum clearance Cr , χ [mm]	0,0019	0,0016	0,0015
Eccentricity direction angle [deg] in the reference frame O1X'1Y'1	194.6	201.8	204.7

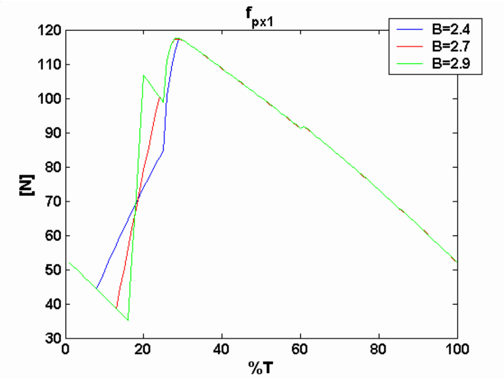


(A) (B)
Fig. 9: Clearance Influence on (A) Pressure Force and (B) Acceleration

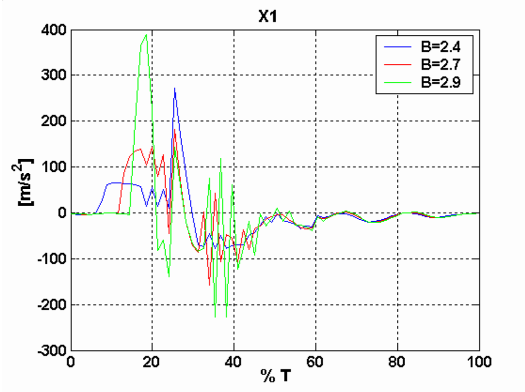
Figure 5 Clearance influence on (a) pressure force and (b) acceleration in X1-direction for gear 1 over one meshing period T; operational conditions of 3350 rpm and 20 bars.

2.4.2. Effect of the relief groove dimension

Figure 6a shows the pressure force in X1-direction for different kinds of relief grooves such a dimension has great influence on the performance of the gear pump, in fact it can be noted that with reduced length B[4], the discontinuities on the pressure forces are smoother due to the increase of the contemporaneous communication of the inlet and outlet chambers.
In fact, remembering equation 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%.



(A)



(B)

Fig. 10: (A) Pressure Force and (B) Acceleration of Gear

Figure 9 (a) Pressure force and (b) acceleration of gear 1 in X1-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 9b) 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[4], 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 [1].

3. Conclusion

In this work an extensive sensitivity analysis by using a design of experiments and operational parameters of a gear pump has been performed. In particular, the model has been developed in the design phase in order to evaluate performance of analysis of hydraulic gear pump modifications (stiffness, damping effect, and friction, load &pressure distribution of the gear pump) and in order to evaluate the influence of operational parameters such as load, speed, noise and vibration.

The graphical representation results have concentrated that the various variation in the operational range of the rotational speed gives a lower effect on the gear accelerations than the output pressure variation.

Therefore, the designer has to take into account that in order to reduce gear noise, vibrations and friction the path of arc of contact has to be increased, but in order to increase the pump performance; the clearance has to be maintained.

i) The Measurement of force and effects of oil fortifier in hydrodynamic gear pump under dynamic loading conditions and investigation of hydrodynamic gear pump under vertical sinusoidal fluctuating loads.

ii) The measurement of oil film pressure distribution, identification of dynamic stiffness, and damping Co-efficient.

iii) The film pressure increases with surface roughness effects under the constant mean Reynolds's number.

iv) Surface contours also arrived for gear strength analysis using design of software.

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

This research work will contribute the requirement of the industries like pump industry, automobile, aircraft, assembling and inspection & testing.

The novel technique which covers the Taghuchi and RSM (response surface method) for enabling the comparable results [10] which obtained by using design of experiments. The name of experiments is for measuring the frictional force, eddy current Cap sensor and oil film thickness.

The above techniques are combination of both.

References

[1] Andersson, BS 1991, '17th Leeds-Lyon Symposium on Elsevier Tribology Series', vol. 18, pp.503-506.

[2] Childs, D, Moes, H & Van Leeuwen, H 2001, 'Journal bearing impedance descriptions for rotordynamic application", Journal of lubrication technology The Mathworks, Matlab, Version 6.1, Release 12.1,vol.99, pp.198-214.

[3] Dalpiaz, G, Fernandez del Rincon, A & Poppi, ME 2003, 'Simulazione del comportamento dinamico di pompe and ingranaggi per servosterzo', Proceedings of XVI AIMETA, Ferrara, Italy, pp.9-12.

[4] Dalpiaz, G, Fernández del Rincón, A, Mucchi, E & Rivola, A 2005, 'Experimental validation of a model for the dynamic analysis of gear pumps', Proceedings of Novem 2005, Saint Raphael, France.

[5] Dalpiaz, G, Elia, GD, Mucchi, E & Fernández del Rincón, A 2006, 'Modeling run in process in external gear pumps', Proceedings of ESDA 2006, Torino, Italy.

[6] Dalpiaz, G, Elia,GD, Mucchi, E & Fernandez der Rincon, A 2006, 'Modeling run in process in external gear pumps'. Proceedings of esda2006-95466.

[7] Dalpiaz, G, Fernández Del Rincón, A & Mucchi, E, 'Modeling meshing phenomena in gear pumps'.

[8] Fernández del Rincón, A & Dalpiaz, G 2002, 'A Model for the Elastodynamic Analysis of External Gear Pumps', Proceedings of ISMA 2002, Leuven, Belgium, September 2002, pp.1387-1396.

[9] Hydraulic Systems in Design and Engineering Education using Modelica and HyLib. Modelica Workshop 2000 Proceedings: 33-40.

[10] C. Manoharan & Arunachalam VP 2007, 'Dynamic analysis of hydrodynamic bearing performance in IC engines using Taguchi and response surface methodology (RSM)', Springer, London Issn: 0268 – 3768 Doi: 10.1007 / S00170 – 007 – 0927 – X.

[11] Mucchi, E 2007, 'Dynamic analysis of external gear pumps by means of nonlinear models and experimental techniques', Ph.D. Thesis, EnDIF-Engineering Department in Ferrara, Università degli Studi di Ferrara, Ferrara, Italy.

[12] Proceedings of ISMA 2004, Leuven, Belgium, pp. 949-963.