JOURNAL BEARING PERFORMANCE IN GEAR PUMPS

Màrius GUTÉS, P.J. GÁMEZ MONTERO, Roberto CASTILLA,
Dr. Esteve CODINA
Technical University of Catalonia,
Fluid Mechanics Department, Terrassa Campus.
C/ Colon nº7-11 (08228) Terrassa. Barcelona. Spain

This paper is the current status summary of a Ph.D. work, which subject is the hydraulic external gear pumps, with pressure compensated lateral plates. Concretely, the aim is to determinate the behaviour of the plain journal bearing, i.e. found theoretically and experimentally, the orbits of the shaft v.s. the lateral floating plate, where the bearing is located. The Thesis is a part of an European project, concretely a Brite-Euram project centred on investigating the performance of an existing gear pump, focused to the industry site and, in this case, to the Spanish manufacturer of the pump being studied, the company Roquet, S.L. For this reason the aim of the finite element simulation is to give support to the experimental results, being these the most important part of this Thesis. Subsequently a summary of the theoretical work done, that is finding the orbits of the shaft by FEM integration of the 2D Reynold's equation will be explained. As well, measurements of the shaft's orbits, and movements of the lateral plates on a real unit under pressure are shown. These measurements are done without contact by laser probes.

Key words: gear pump, plain journal bearing, Reynold's Equation, orbit, floating lateral plate.

1 INTRODUCTION

This paper tries to summarise the current status of a Ph.D. work. The subject is the behaviour of a plain journal bearing on oleo-hydraulic external gear pumps. The bearings are important enough to be studied because if the shaft's orbit is not stable, or the bearing is not well designed, contact between the shaft and the bearing will appear. That will produce that the teeth of the gear wheel touches the inner part of the pump's body, and the immediate consequence will be the wear of the pump body and therefore a pump's lack of efficiency.

The plain journal bearings are fully used in hydraulics due to their small size, low price, and its capability of carrying load. Nowadays bearings with a teflon lining to minimise the consequences of the contact between the shaft and the housing are available on the market. Moreover firsts ceramic coatings for the inner surface of the bearings, or lateral plates completely in monolithic ceramics are being developed. In this last option, no bearing is needed, but the

lateral plate makes its function itself. Here the plain journal bearings with a metal basis, and an inner lining of teflon are studied.
To understand the philosophy of this work, the context has to be explained. This Ph.D. constitutes a part of a European project called *"Ecopump"*, which is a Brite-Euram project. Companies from all round Europe like Roquet S.L.-Pump manufacturer (Spain), Fuchs Petroleum A.G.-Oil manufacturer (Germany), Argoom-Sealing manufacturer (Italy), Alfredo Cardoso-Hydraulic platforms manufacturer (Portugal), ZFS-Investigation Center (Germany), CIMNE-Investigation Center (Spain), and Politechnical University of Catalonia (Spain) are involved. The purpose of the project is to determinate how the L-22, the most important pump of Roquet S.L., works, including the behaviour with dirty oils, ecological oils, i.e. vegetable oils, and so on, and this Thesis is related with one component of the pump, i.e. the plain journal bearing. Nevertheless, as the bearing is not alone, but assembled in the lateral floating plate, all together shall be investigated, and therefore, the work is to study the system: shaft-bearing-floating lateral plate.

As the objective is to help the industry by solving a concrete problem on a concrete unit, the philosophy is to focus all the effort on the real unit, and on the experimental site, for obtaining realistic and therefore useful results. It has to be said that a finite element code has been completely developed in the Fluid Mechanics Department, in order to solve the 2D Reynold's Equation, for any L/D ratio, with stiffness and damping effects behold, but excluding things like cavitation, misalignment, 3D behaviour, ... The reason is that the aim of the simulations done is to give support for understanding the measurements, and the ground because important aspects have not been taken into account, is because they would not, in this case, contribute to better conclusions.

2 THEORETICAL PART

As it has been mentioned, the theoretical work is to integrate the 2D Reynold's Equation in polar co-ordinates, for any L/D ratio, incompressible fluid, and unsteady state, that is:

$$\frac{1}{r_a}\frac{\partial}{\partial q}\left(\frac{h^3}{12h}\frac{\partial p}{\partial q}\right)+h^3\frac{\partial}{\partial y}\left(\frac{h^3}{12h}\frac{\partial p}{\partial y}\right)=\frac{w_b}{2}\frac{\partial h}{\partial q}+\frac{\partial h}{\partial t} \tag{1}$$

and therefore find a succession of positions (orbit) of the shaft relative to the bearing centre. For obtaining the orbits, the procedure is the following:

(i) Know the pump's geometry and the working pressure for the case we are studying.

(ii) Calculate the load (w_b) in time that the shaft has to carry (regarding the working pressure and the pump's geometry).

(iii) Give an initial position of the shaft related to the bearing's centre (to calculate ¶*h*/¶*q*)

(iv) Integrate, via Finite Element Method (FEM), the Reynold's Equation for a steady state (first eq. of the system (2)) to find the steady pressure (p_0). See Gutes, M. (1997)

(v) Use Lund's Method to linearise the load and the pressure [1]. $(p_x, p_{\dot{x}}, p_z, p_{\dot{z}})$

(vi) Substitute this p_0 in the other four eq. of system (2)

(vii) Integrate via FEM four pseudo-Reynold's Equations (system 2) , in order to determinate the linear pressures $(p_x, p_{\dot{x}}, p_z, p_{\dot{z}})$ and therefore the eight fundamental coefficients for the motion in 2D, i.e. the four stiffness and the four damping coefficients.

(viii) Use the above mentioned coefficients, to determinate the displacements in the two directions for a given time step, using a 5th Runge-Kutta scheme.

(ix) Calculate the new position, and take it as the initial position for the next time step. Go back to (iii)

$$h = c + e\,cosq = h_0 + \Delta x\ cosq + \Delta z\ senq \quad ; \quad \frac{dh}{dt} = \Delta \dot{x}\ cosq + \Delta \dot{z}\ senq$$

$$\left[\frac{1}{r_a}\frac{\partial}{\partial q}\left(\frac{h^3}{12h}\frac{\partial p}{\partial q}\right) + h^3\frac{\partial}{\partial y}\left(\frac{h^3}{12h}\frac{\partial p}{\partial y}\right)\right]\begin{bmatrix} p_0 \\ p_x \\ p_z \\ p_{\dot{x}} \\ p_{\dot{z}} \end{bmatrix} = \begin{bmatrix} \dfrac{w_b}{2}\dfrac{\partial h_0}{\partial q} \\ -\dfrac{w}{2}\left(sen\,q + \dfrac{3cos\,q}{h_0}\dfrac{\partial h_0}{\partial q}\right) - \left(\dfrac{h_0^3}{4h_{ar}^2}\dfrac{\partial p_0}{\partial q}\dfrac{\partial}{\partial q}\left(\dfrac{cos\,q}{h_0}\right)\right) \\ -\dfrac{w}{2}\left(cos\,q - \dfrac{3\,sen\,q}{h_0}\dfrac{\partial h_0}{\partial q}\right) - \left(\dfrac{h_0^3}{4h_{ar}^2}\dfrac{\partial p_0}{\partial q}\dfrac{\partial}{\partial q}\left(\dfrac{sen\,q}{h_0}\right)\right) \\ cos\,q \\ sen\,q \end{bmatrix} \tag{2}$$

[1] Generally there are three different methods for doing it so. (i) The Mobility method, which is the most simple, and does not take into account the damping effects. (ii) The Lund's method, which is a linear approximation, but considers the damping coefficients. (iii) Non-linear methods that are more complicated, and which need more computational effort. In this Ph.D. work the Lund's approximation has been adopted.

For more details see Gutes M. (1998).

The FEM code has been tested in two different ways. First, the integration of the Reynold's Equation has been checked, by running an infinitely long and infinitely short length bearing. In these cases, the numerical results can be compared with the analytical results [2]. On the other hand side, the numerical solution of the orbit has been also tested. The shaft has been left in an initial position, no load has been applied, and the shaft has reached the center of the bearing through a spiral way (see Figure 1)[3].

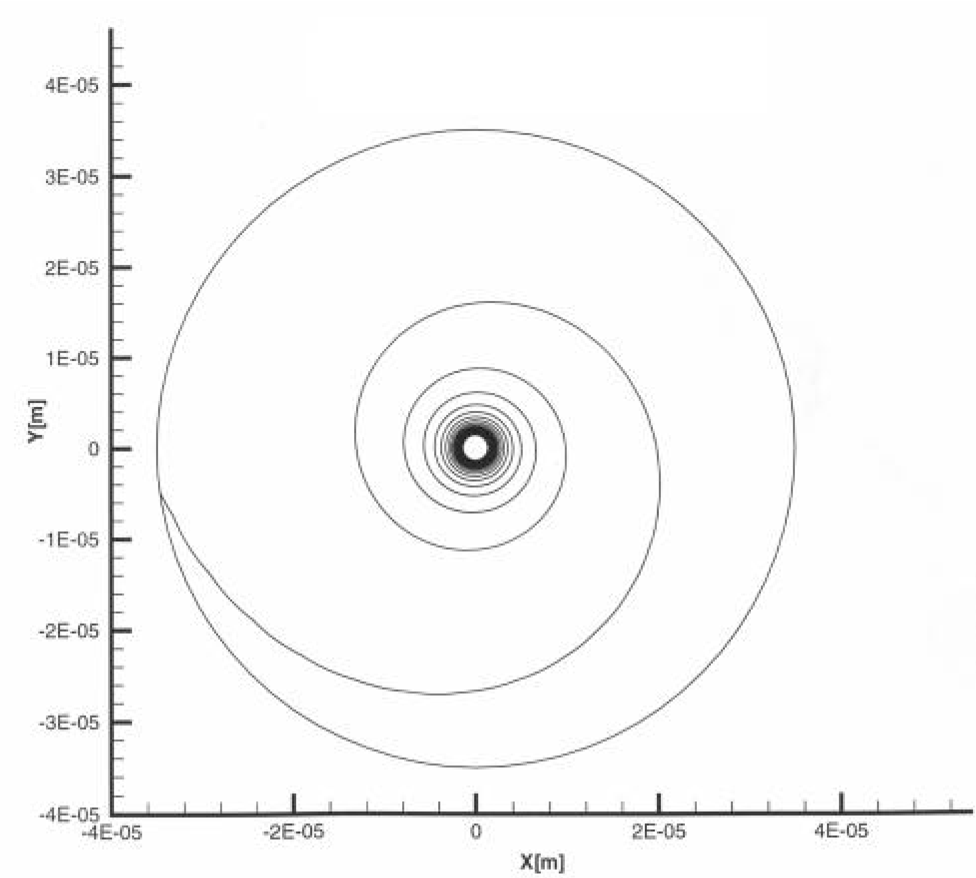


Figure 1: Orbit from an initial position without load

[2] If the domain of integration is cut and developed i.e., if we change from polar coordinates into cartesian coordinates *(x,y)*, and we assume that the length of the bearing is the *'y'* direction and that the *'x'* direction corresponds with '$2p\,r$', then in the infinitely long bearing, it can be assumed that there is no variation of the pressure with the *'x'* direction, and the analytical solution can be found. For infinitely short bearings the variation of pressure v.s. *'y'* can be neglected.

[3] This was a suggestion of Dr. Taylor (Leeds University).

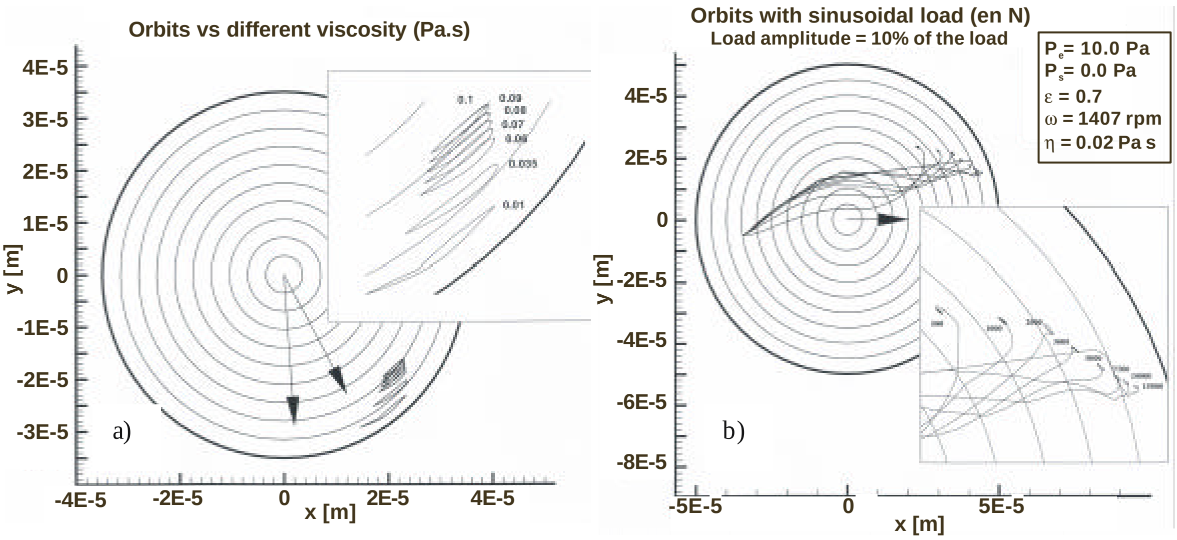


Figure 2 Theoretical orbits v.s. viscocity & v.s. load

Finally, some numerical results are shown. First, orbits are presented (Figure 2). All of them have been obtained for different sinusoidal loads[4] applied (i.e. from 500N up to 15000N). Here one can see how the eccentricity (ε) increases as the load increases, and at the same time how the attitude angle decreases. All of them are stable. It is also shown the behaviour due to the viscosity (Figure 2). Here it can be seen how the eccentricity (ε) increases as the viscosity decreases, and at the same time how the attitude angle decreases (less carrying capacity), so increasing the load has the same effect that decreasing the viscosity. Finally, the damping coefficients for a 0.02 Pa s viscosity, 1500 rpm. angular velocity, $5*10^{-5}$ eccentricity, and no load, are shown (Figure 3).

[4] The sinusoidal loads have an amplitude of the 10% of the load's mean value.

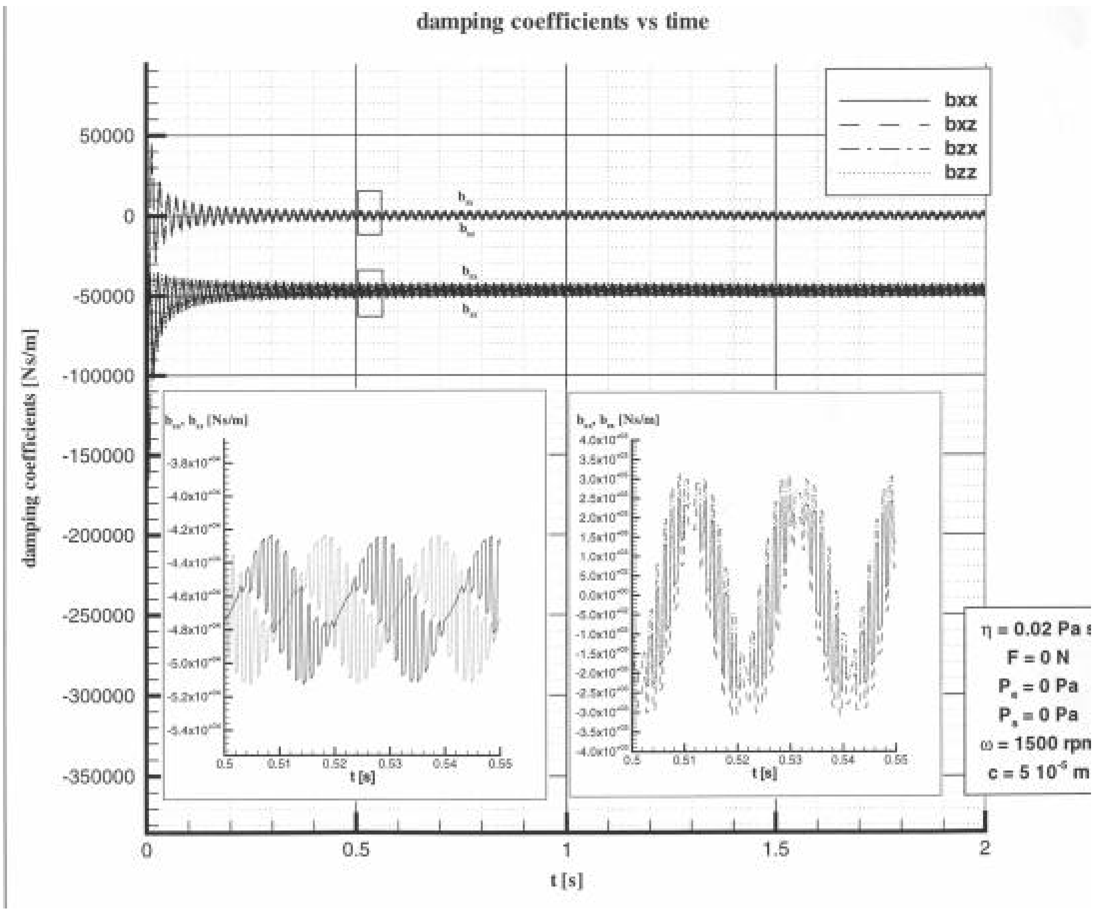


Figure 3 Damping coefficients

3 EXPERIMENTAL PART

As it has been mentioned, this is the most important part, because the aim was to contribute useful and real results to the pump manufacturer, The best way to do it is to study, and do the measurements on a real unit. Here we have found three problems:

(i) The real unit is not a big pump (180mm x 150mm external).

(ii) The measurements have do be done inside of the pump (remember that the movement of the shaft related to the journal bearing is what we need to measure).

(iii) The measurements have to be done under pressure.

This means that two different probes have to be installed in the lateral plate, where the bearing is located (one for *'x'* and one for *'y'* direction). These probes have to be small enough for not destroying the fluid film or the behaviour of the pump, but big enough to be able to assemble them. They need to be able to work under 200 bar, and finally they need also to be able to measure in a range from 0 up to 100μm without contact.

Different tests have been made with different probes. First an inductive probe of the company H.B.M. was tested, but it did not work. The range of measurement was O.K., and it was able to work under 250 bar, but it was to big to be installed in the pump, and problems with the time response happened. Then an Eddy Current sensor of the company $\mu\varepsilon$ was checked. It was very little, *1 mm* diameter and *4 mm* length. It was not able to work under pressure but the option of a special lining to work under pressure was able in the market. The problem was that, due to its sensitivity, an offset between it and the shaft, and a good alignment had to be ensured, but due to its size, it wasn't impossible to do it, and bad results where obtained.

As the measurements from inside of the pump were impossible[5], it was decided to measure the inside behaviour of the internal parts from the outside of the unit. To do it so:

(i) A little rod was assembled on the external part of the pump's body. This was the absolute reference (the absolute shaft position.), Figure 4.

(ii) Two little rods where installed in the lateral plate, and two holes made on the inlet pump's body[6] to allow them to go out, These were the relative references (the shaft position relative to the lateral plate, i.e. to the bearing), Figure 4.

(iii) As everybody knows, in an external gear pump, there is a long shaft (the driver shaft that is connected to the electrical motor), and a driven shaft, which does not go out of the pump as it's smaller. As the measurement needed is the shaft related to the bearing, two long shafts were needed. A long shaft, the driver one, and another shaft long enough to go out of the pump, and therefore to be able to measure it from the outside. So what was done was to set up a pump with two long shafts (manufacturer's driver shafts). One was used as driver shaft, and was connected to the electrical motor, and the other worked as a driven shaft, and was used to do the measurements, Figure 4.

(iv) Two pairs of laser probes (emisor-reciever for *'x'* and emisor-reciever for *'y'*) were installed in the test bench earlier constructed (Figure 5). These laser probes, of the company Keyence, are able to measure segments between two objects, without contact. Thus:

   (a) If the segment measured is between the shaft and the rod fixed on the front pump's cover, then the absolute motion of the shaft is obtained (Figure 6).

   (b) If the segment measured, is between one of the rods fixed on the lateral plate and the shaft, then the motion of the shaft relative to the lateral plate (i.e., to the bearing) can be obtained.

   (c) If the segment between the two rods of the lateral plate is measured, then the rotation of it can be obtained.

[5] If the pump was double or triple bigger, the measurements from inside of it should be possible.

[6] During the measurements it was checked that no cavitation appear.

(d) If the segment between one of the rods of the lateral plate and the fix rod of the cover of the pump is measured, and this measurement is combined with the measurement of point *(c)*, then the motion of the lateral plate in the *'xy'* plane can be found.

With this method, different measurements have been obtained: Absolute orbits of the shaft, relative orbits of the shaft to the bearing, movements of the lateral plate

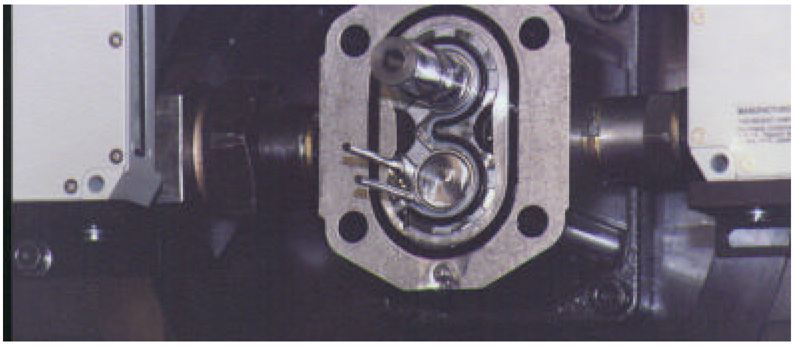


Figure 4: Test bench. Two little rods fixed on the lateral plate

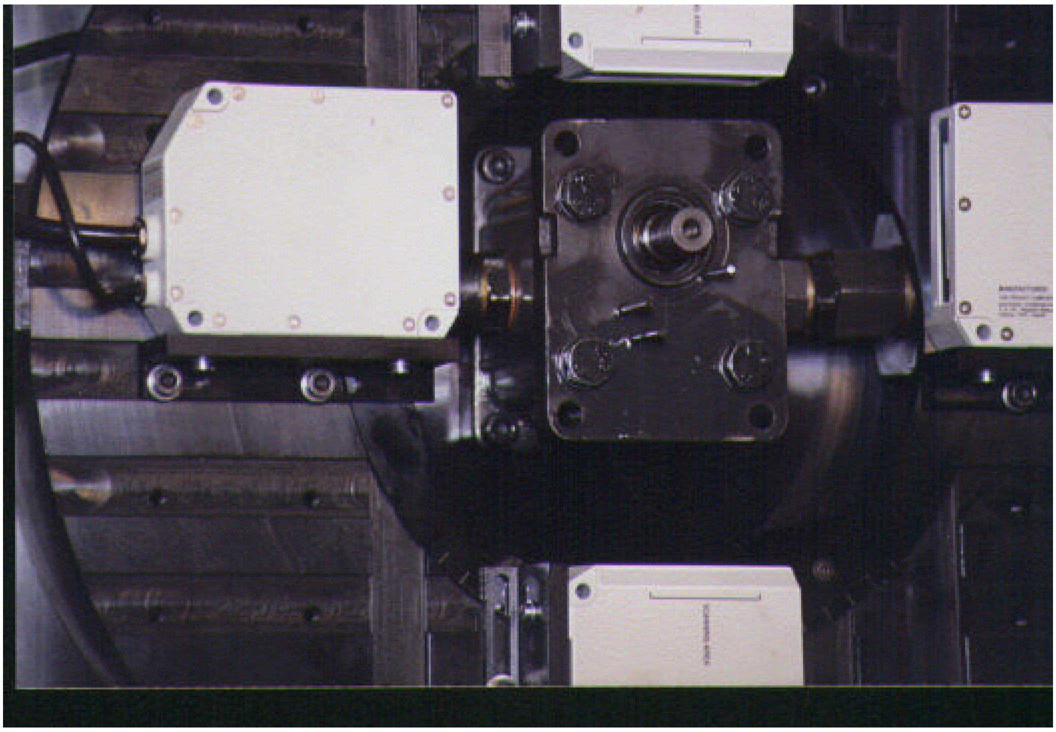
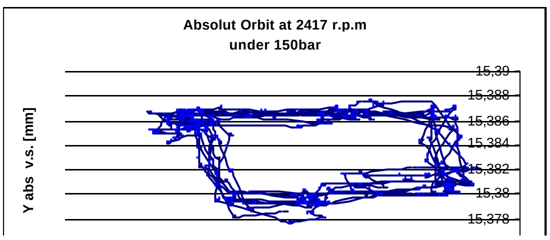


Figure 5: Test bench. Laser probes



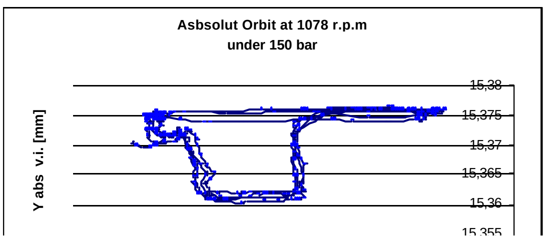


Figure 6: Absolute orbits of the shaft, at different rpm.

The behaviour of the lateral plate is very important for a good understanding of what's happening. Subsequently (Figure 7), a measurement of the lateral plate movement is shown. Here *"Delta* $X_{v.i.}$*"* means de displacement of the lower rod of the lateral plate in 'x' direction" produced for a translation, *"Delta* $Y_{v.i.}$*"* means de displacement of the lower rod of the lateral plate in 'y' direction" produced for a translation, *"Delta* $X_{v.i.}gir$*"* means de displacement of the lower rod of the lateral plate in 'x' direction" produced for a rotation, *"Delta* $Y_{v.i.}gir$*"* means de displacement of the lower rod of the lateral plate in 'y' direction" produced for a rotation, and *"gir[º]"* means the rotation of the lateral plate in the "*xy*" plane. It can be seen that the displacement, and the rotation behaviours of the lateral plate are opposite, that is, if the translation tries to produce a movement to the right, then a rotation appears, and rotates the lateral plate in order to produce a displacement to the left. It also can be appreciated a fixed range of the rotation (vibration of the lateral plate).

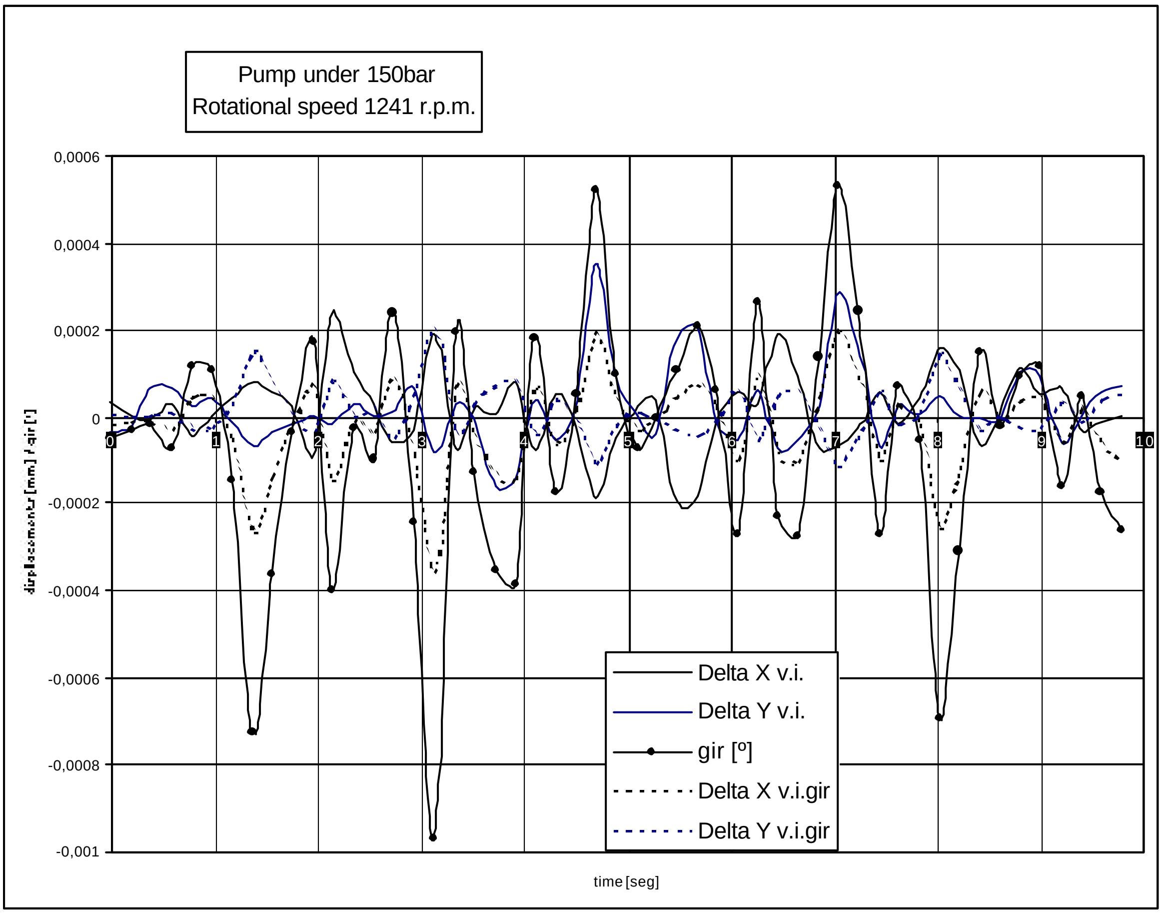


Figure 7: Lateral plate behaviour

4 FURTHER STUDIES

Until this point, the current status has been explained. With all of the numerical results, and as well, with all the experimental data, a final conclusion has to be reached. This will be the further study, to establish the behaviour.

5 ACKNOWLEDGEMENTS

I would like to thank the U.P.C., to give me the change for doing this Ph. D work, specially Dr. Codina, and also my colleges R. Castilla and P.J. Gámez Montero, to give support in theoretical and experimental research respectively.

6 NOMENCLATURE

a:	acceleration
b_{ij}:	fluid damping coefficient
c:	clearance
e:	eccentricity
h_0:	fluid film height
k_{ij}:	fluid stiffness coefficient
m_a:	shaft mass
p:	pressure
r_a:	shaft radius
θ:	angle – attitude angle
t:	time
w_b:	shaft angular speed
w_x:	carrying load in x direction
w_z:	carrying load in z direction
w_r:	load to carry
Δx:	x displacement
Δz:	z displacement
ϕ_l:	w_r 's angle
η:	kinematic viscocity

REFERENCES

Gutes, M.; Castilla, R.; Codina, E. (1997) Resolución numérica de la Ecuación de Reynolds para un cojinete cilíndrico, *XIII Congreso Iberoamericano de Ingeniería Mecánica, Facultad de Ingeniería Mecánica , Instituto Superior Politécnico José Antonio Echeverría, La Habana, Cuba.*

Gutes, M.; Castilla, R.; Codina, E. (1998) Movimiento teórico de un eje dentro de un cojinete hidrodinámico, *XIII Congreso Nacional de Ingeniería Mecánica, Escuela Técnica Superior de Ingenieros Industriales de Terrassa, Barcelona, España.*

Cameron, A. (1976) Principles of Lubrication, *Ellis Horwood Ltd., 1976*

Hamrock, B.J. (1994) Fundamentals of Fluid Film Lubrication, *McGraw-Hill., 1994.*