

JOURNAL BEARINGS APPLIED IN THE CAMSHAFT SUPPORT ELEMENTS FOR HIGH PERFORMANCES ENGINES

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Abstract. *With the help of Hydraulic Lash Adjusters (HVA) that has the feature to reduce noise emission of passenger cars, we found out that HVA also combine decreasing maintenance costs with thermodynamical advantages. The objective of the last years that will continue being important in the near future is the emissions standardization, particularly in USA and Europe . Controlled by an "On Board Diagnostics ", the system requires a precise function in the complete system . The bibliography refers only to the oil film in the normal condition not considering the tilting . The journal bearing calculation is an important tool that improves the design time and optimize the design schedule . The important resource is the forces evaluation that shows the calculated values that influence the complete system .*

Keywords*.* *hydrodynamic bearing, pressure distribution, axial tilting.*

1. Introduction

For several years systems were developed for valvetrain system observing the efficiency and to help the power generation. The well-known tappet or lash adjusters (called HVA in that article or Hydraulic Valve Lash Adjusters) are one of the mechanisms that aided the noise reduction of passenger cars, as well as, assembly time improvement and system maintenance costs reduction agreeing with thermodynamic advantages (Maas, 2000). Nowadays most of the passenger cars are equipped with HVA, besides they provide a great performance they are also responsible for the substantial costs reduction. The decrease in emission levels is also important for the automobiles improvement, especially in Europe and US where the limits are getting so smaller that they are going hindering the engine project. HVA is one of the items that aids the emissions reduction, facilitating the designer work when wants to reach a lower level of emissions. With the new systems OBD (On board diagnosis) the vehicle electronics central demands a engine project more optimized. Through the years HVA is being more and more redesigned to achieve the customers requirements.

Some works has been published with these considerations, specially in the valvetrain system (Schmidt, 1995 e Buteschön, 1976). The objective of this work is the improvement of the project phase and the dimension work of a hydraulic system for forces and lubrication support. With some initial project data it is possible to evaluate the first requested work conditions. Inside of this context of boundary alteration conditions is extremely simple a re-evaluation and it facilitates the system optimization decreasing the development time.

2. Mathematical Model

The evaluation that will be treated here refers to the support elements for Roller Camshaft (Figure (1)). This support element that will be called "journal" is considered inside of the formulation as a conventional axis. The guidance hole where the axis or journal slides will be called "bearing" . Figure (2) shows an illustration to demonstrate the application.

The objective of the study is to evaluate the involved forces in the hydrodynamic system, reaching thus, the maximum force that the system can support in dynamic situations.. With the same importance we have the evaluation for the shaft tilting (fundamental requirement for the wear distribution in this component type). We came to the conclusion that the force in oil film direction should be higher to avoid vibrations in the system and lower in the tilting direction to facilitate the rotation of the shaft and to distribute the wear .

The mathematical model to describe the pressure distribution inside the bearing is based on Figure 3. The Reynolds Equation is applied to solve the hydrodynamic pressure distribution considering a laminar flux of a incompressible oil fluid film (Norton, 1996 ; Cavalca, 2000 ; Cataruzzi, 1998).

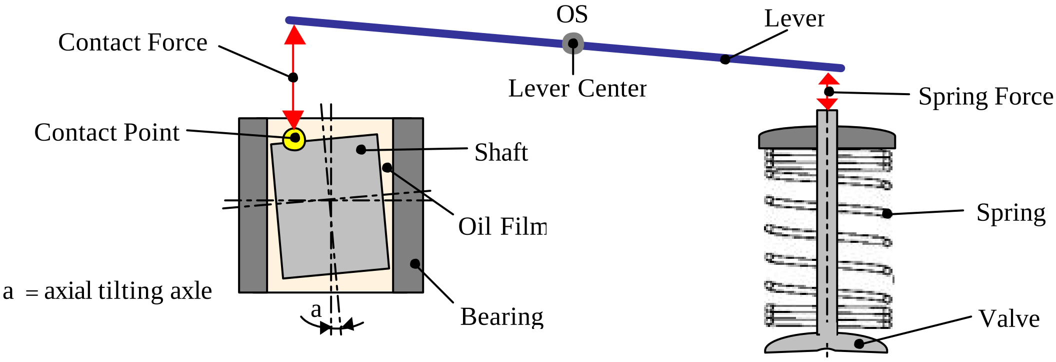


Figure 1. System Sketch

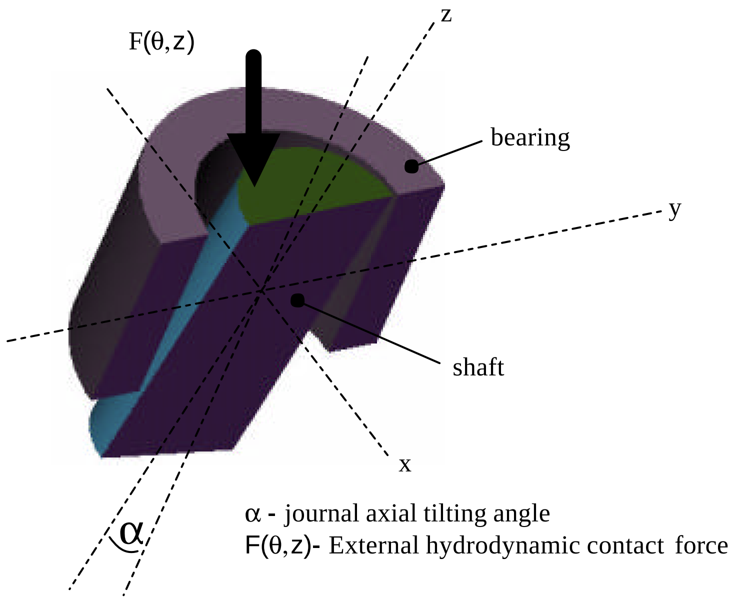
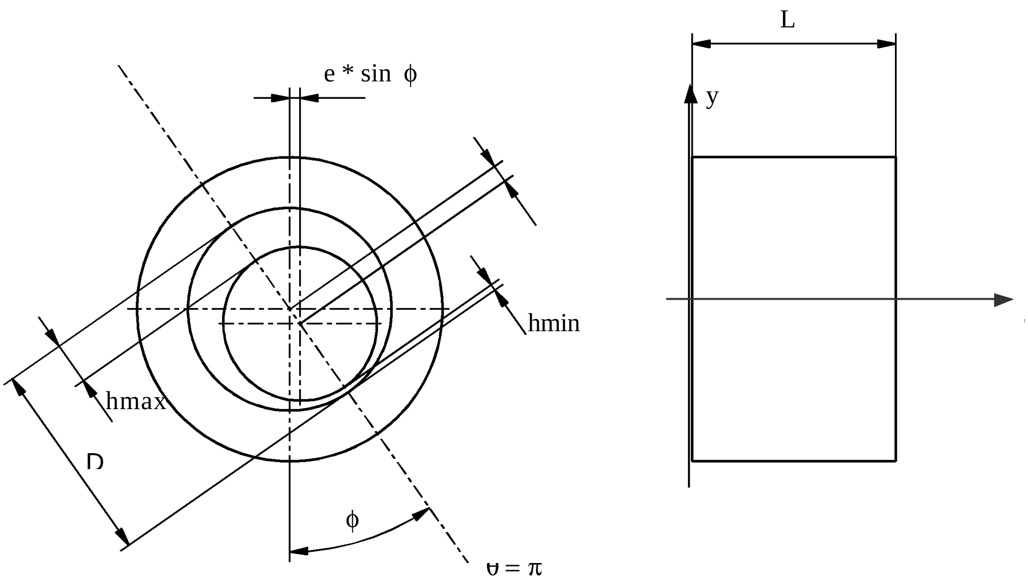


Figure 2 . External force applied in the journal and shaft axial tilting



Symbols list:

e	- eccentricity between journal and the bearing
ϕ	- eccentricity angle between journal and the bearing
L	- bearing width
hmin	- minimal oil film thickness
hmax	- maximal oil film thickness
D	- bearing internal diameter
x, y, z	- cartesian coordinates system
θ	- angular coordinate system
0_b	- geometric bearing center
0_j	- geometric journal center

Figure 3. Journal - bearing system

3. The hydrodynamic bearing

Hydrodynamic Lubrication refers to the supply of sufficient lubricant (typically an oil) to the sliding interface to allow the relative velocity of the mating surfaces to pump the lubricant within the gap and separate the surfaces on a dynamic film of liquid. This technique is most effective in journal bearings, where the shaft and bearing create a thin annulus within their clearance that can trap the lubricant and allow the shaft to pump it around the annulus. A leakage path exists at the ends, so a continuous supply of oil must be provided to replace the losses. This supply may be either gravity fed, or pressure fed. This is the system used to lubricate the crankshaft bearings in an internal combustion engine. Filtered oil is pumped to the bearings under relatively low pressure to replenish the oil lost through the bearing ends, but the condition within the bearing is hydrodynamic, creating much higher pressures to support the bearing load.

In a hydrodynamic sleeve bearing at rest, the shaft or journal sits in contact with the bottom of the bearing. As it begins to rotate, the shaft centerline shifts eccentrically within the bearing and the shaft acts as a pump to pull the film of oil clinging to its surface around with it. (The “outer side” of the oil film is stuck to the stationary bearing.) A flow is set up within the small thickness of the oil film. With sufficient relative velocity, the shaft “climbs up” on a wedge of pumped oil and ceases to have metal-to-metal contact with the bearing (Norton, 1996).

The whole system is based on the equation of the hydrodynamic lubrication proposed in 1886 by Dr. Osbourne Reynolds in the published article - " On the Theory of Lubrication and its Application to Mr. Beuchamp Tower's - Experiments, including in Experimental Determination of the Viscosity of Olive Oil ". In that case, series of experiments on relative movements among surfaces were made considering oil as a contact connection. We have the Reynolds Equation (Reynolds, 1886):

$$\frac{d}{dx}\times\left(h^3\times\frac{dp}{dx}\right)+\frac{d}{dz}\times\left(h^3\times\frac{dp}{dz}\right)=6\times m\times\left\{\left(U_0+U_1\right)\times\frac{dh}{dx}+2\times V_1\right\} \tag{1}$$

Where:

x,y,z	- bearing cartesian coordinates
P	- oil film pressure
H	- oil film thickness
μ	- oil kinematic viscosity
U_0, U_1	- circunferential velocities of journal and bearing, respectively
V_1	- radial velocity of journal

The oil film thickness is composed in relation to z and θ like indicated below :

$$H(q,z) = Cr - e(z) \times Cos\,(180 - q) \tag{2}$$

$$H(q,z) = Cr - e(z) \times \left(Cos\,180^{0} \times Cos\,q - Sin\,180^{0} \times Sin\,q\right) \tag{3}$$

$$H(q,z) = Cr + e(z) \times \cos(q) \tag{4}$$

The tilting is considered in relation to the eccentricity or tilting angel α :

$$e(z) = \left(L - \frac{L}{2} - z\right) \times sin\,a \tag{5}$$

$$e(z) = \left(\frac{L}{2} - z\right) \times sin\,a \tag{6}$$

Achieving finally the complete expression :

$$H(q,z) = Cr + \left[\left(\frac{L}{2} - z\right) \times sin\,a\right] \times \cos q \tag{7}$$

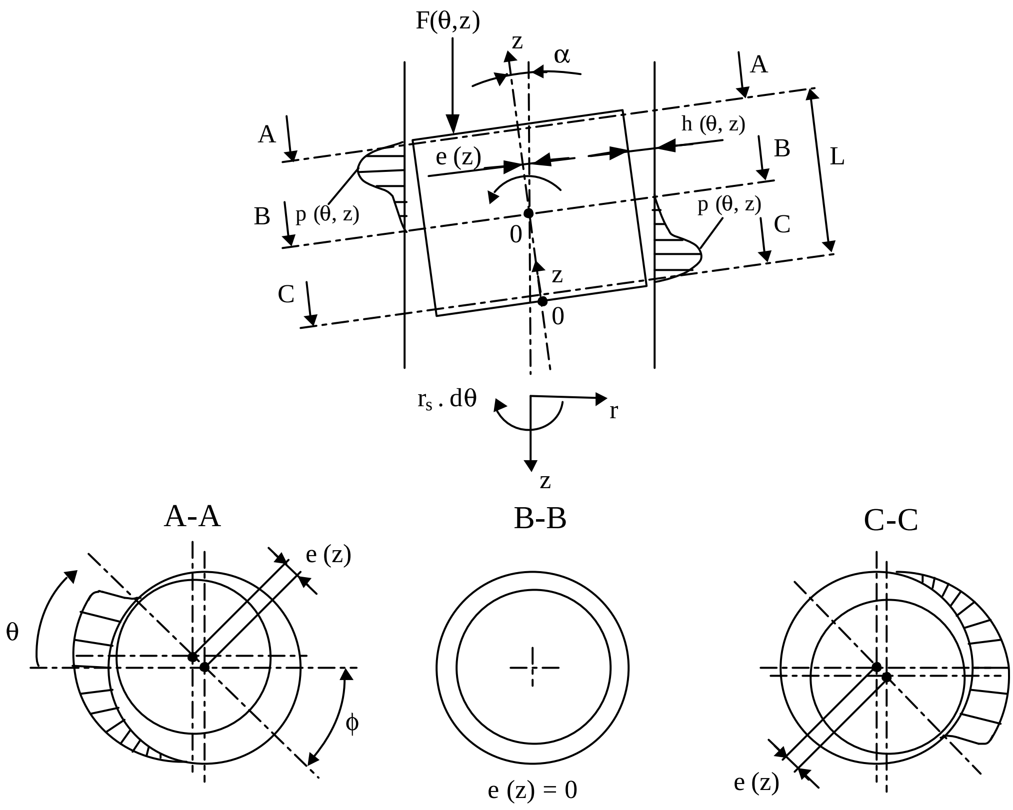


Figure 4. Pressure distribution circuferential in (θ) and (z) axial directions

4. The Finite Difference Method for Reynolds Equation

The pressure distribution in the HVA Support System for valve train system can be calculated through second order non partial homogeneous differentials which, in most of the cases, don't have full solution. The differential deduction of Reynolds is based on a balance of forces in a flux element for incompressible Newtonianians fluid which are subject to a laminar flux. The external forces are in balance with the pressure and friction.

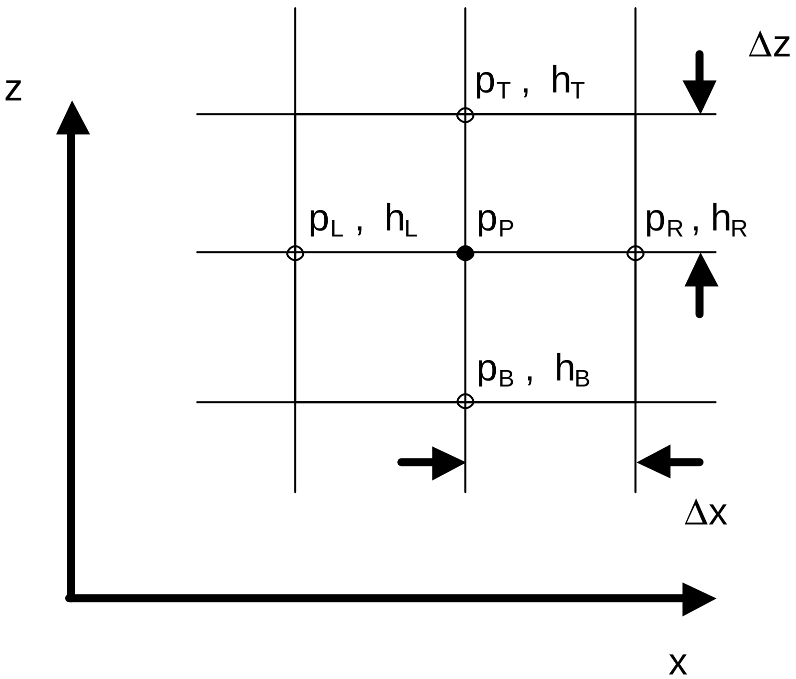
Considering the exposure, was achieved the dimensionless Reynolds Equation for a numeric solution, presented in an usual format as (Cavalca e Cattaruzzi, 2001; Pinkus, 1956,1959; Irretier, 2002; Nordmann, 1998):

$$p_n = \frac{18{,}84 \times \dfrac{(h_L - h_R)}{\Delta x} + \left(\dfrac{D}{L}\right)^2 \times \dfrac{h_T^3}{\Delta z^2} \times p_T + \dfrac{h_R^3}{\Delta x^2} \times p_R + \left(\dfrac{D}{L}\right)^2 \times \dfrac{h_B^3}{\Delta z^2} \times p_B + \dfrac{h_L^3}{\Delta x^2} \times p_L}{\left(\dfrac{D}{L}\right)^2 \times \dfrac{h_T^3}{\Delta z^2} + \dfrac{h_R^3}{\Delta x^2} + \left(\dfrac{D}{L}\right)^2 \times \dfrac{h_B^3}{\Delta z^2} + \dfrac{h_L^3}{\Delta x^2}} \tag{8}$$

We can consider Equation (2) in the format:

$$p_n = c_0 + c_1 \times p_T + c_2 \times p_R + c_3 \times p_B + c_4 \times p_L \tag{9}$$

In Equation (3) the values of c are constant and the values of p are unknown.



Δx , Δz - finite differences step or increment in x and z coordinates

p_P, p_R, p_T, p_B, p_L - pressure points (Principal Right, Top, Bottom, Left respectively)

h_R, h_T, h_B, h_L - oil film tickness (Right, Top, Bottom, Left respectively)

Figure 5. Oil film grid for finite differences evaluation

4.1. The oil film grid

Actually, to the evaluation of oil film pressure distribution will be of interest ½ of the axial direction due to the system axial symmetry tilting considered in this analysis (Koç, 1990).

Other consideration is that the bearing works as a partial arc bearing with positive pressure distribution in the bottom (0-180 degrees with respect to θ coordinate) and null pressure values in the top arc, during constant oil supply inside the bearing.

In the portion of interest, the oil film was discretized in a certain number of points in the circunferential (x) and axial (z) directions, where each point had its pressure calculated (Figure (4)). Obviously, the boundary condition of pressure zero on the bearing edge was used as well as in the half of its width, facilitating the equation system resolution.

5. Hydrodynamic Forces Solution in x, y and z directions

Once the pressure distribution was evaluated through the Finite Differences Process as well as the oil film grid discretization, a Gauss Seidel interative process was applied to achieve the pressure values in each point of the oil film. Gauss Seidel's interative process allows the increasing of pressure values accuracy.

The hydrodynamic forces in x, y and z directions can finally be evaluated from the projection of the resultant force F in each cartezian coordinate direction for each discretized point (Cavalca, 1998; Jacon, 2000; Cavalca e Idehara, 2001).

The sum of all projections in each direction gives the general hydrodynamic force components Fx, Fy and Fz:

$$F_X = \sum_{n=1}^{n} p_n \times Sina_n \times Cosq_n \times \Delta x \times \Delta z \qquad (10)$$

$$F_Y = \sum_{n=1}^{n} p_n \times Sina_n \times Senq_n \times \Delta x \times \Delta z \qquad (11)$$

$$F_Z = \sum_{n=1}^{n} p_n \times Cosa_n \times \Delta x \times \Delta z \qquad (12)$$

Once the global cartezian components of the hydrodynamic supporting force is defined, it is possible to proceed with the evaluation of the maximum permissible force in the top of the journal for the support condition, as well as the friction component with the roller camshaft system in accordance with the study situation presented .

$$F_R = \sqrt{F_X{}^2 + F_Y{}^2 + F_Z{}^2} \qquad (13)$$

6. Obtained Results

Some practical dimensional parameters are used in this application for passenger car with gasoline engine to evaluate some operational conditions:

Table 1. Design Parameters

Parameters	**Dimension**	**Units**
Bearing diameter (D)	8	mm
Eccentricity (e)	0,05	mm
Width (L)	6	mm
Diametral clearance (Cd)	200	μm
ϕ angle	25	degrees
α angle	0,95	degrees

Using the design parameters presented, the oil film pressure and forces distributions were obtained:

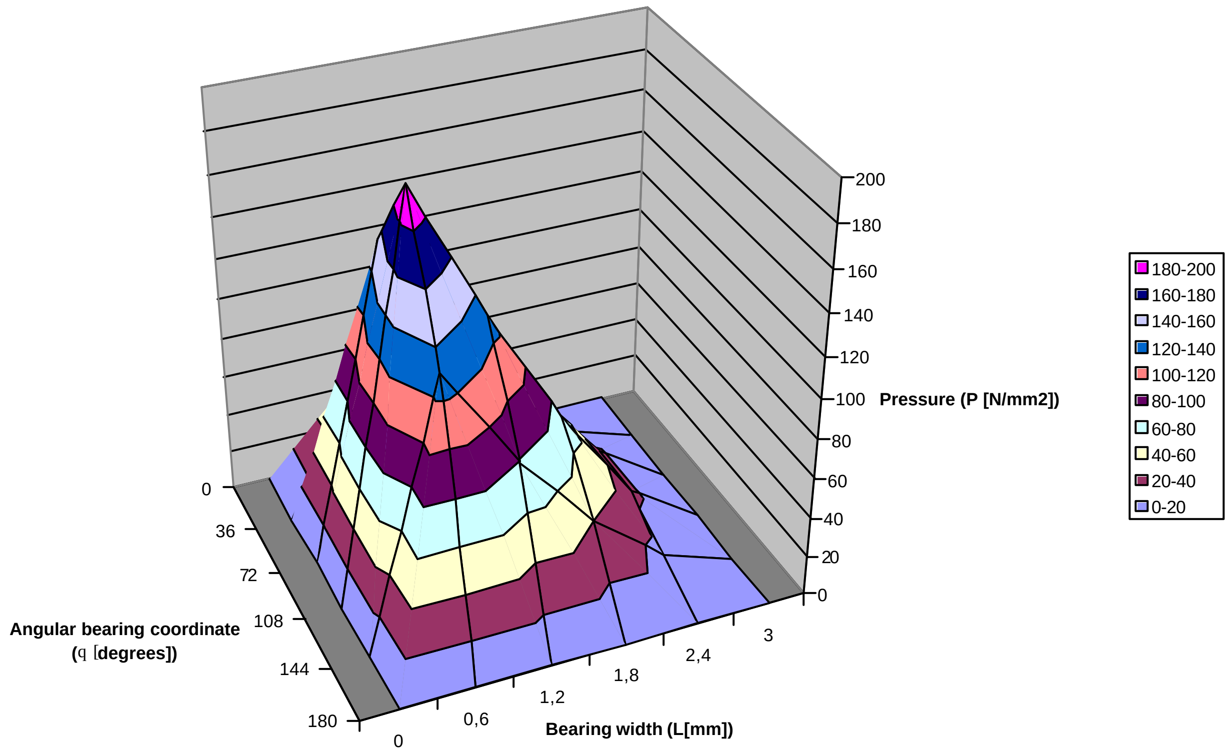
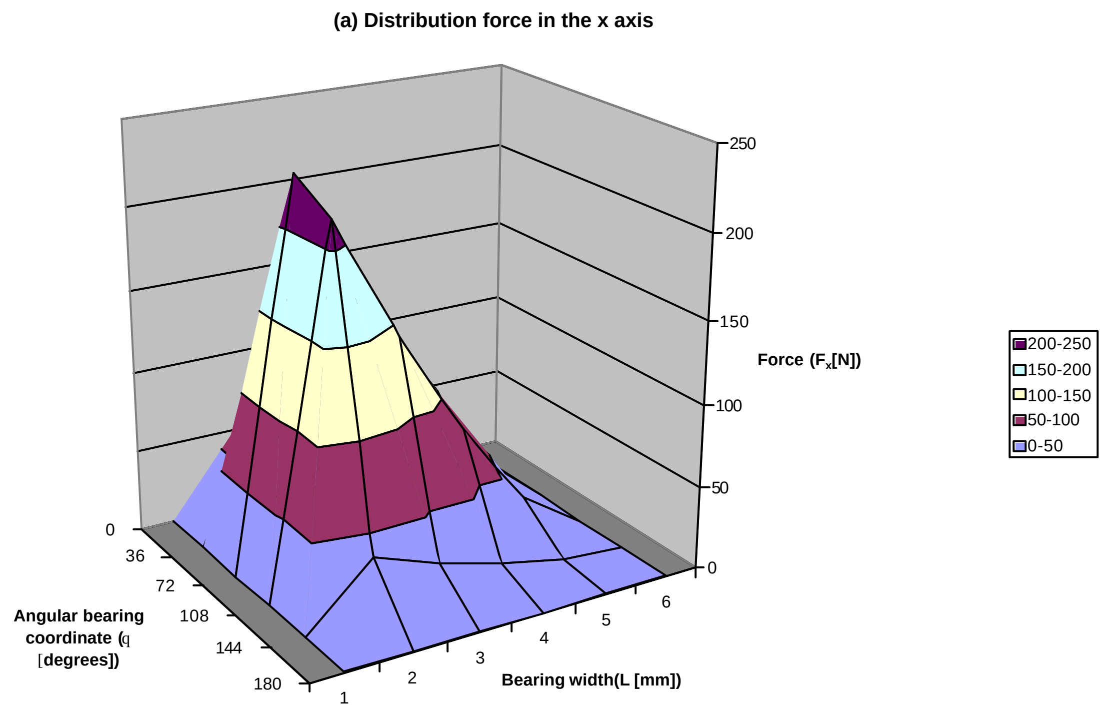


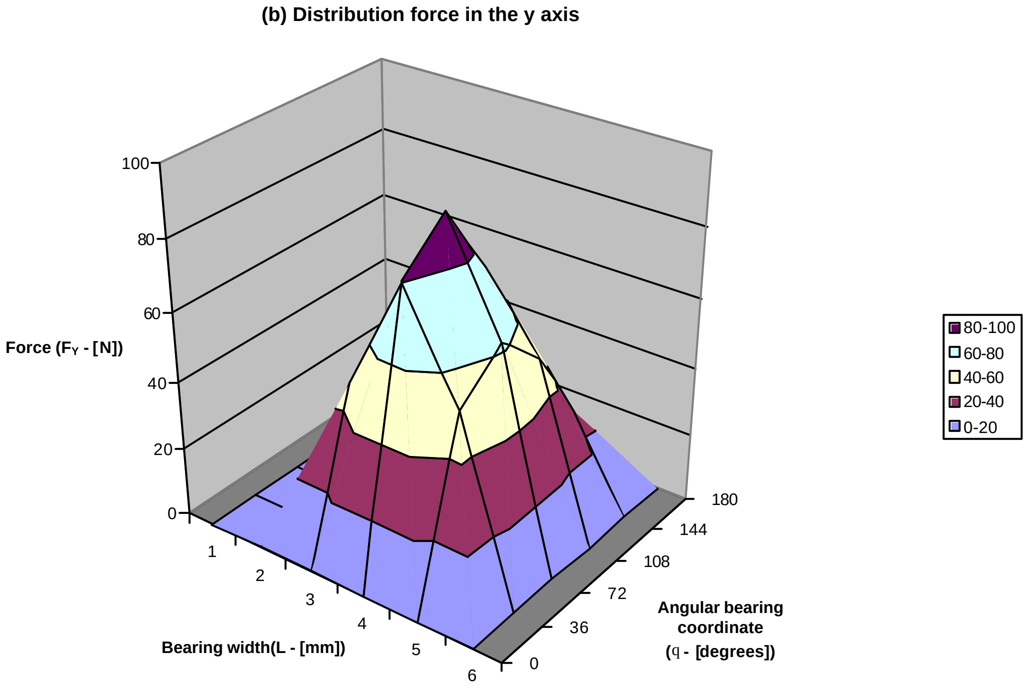
Figure 6. Pressure distribution

The pressure distribution of the oil film inside the bearing makes possible to find the effective forces in the system for the condition previously presented in Table (1):

***x** direction:*



y direction *:* Distribution force in the y axis



z *direction* :

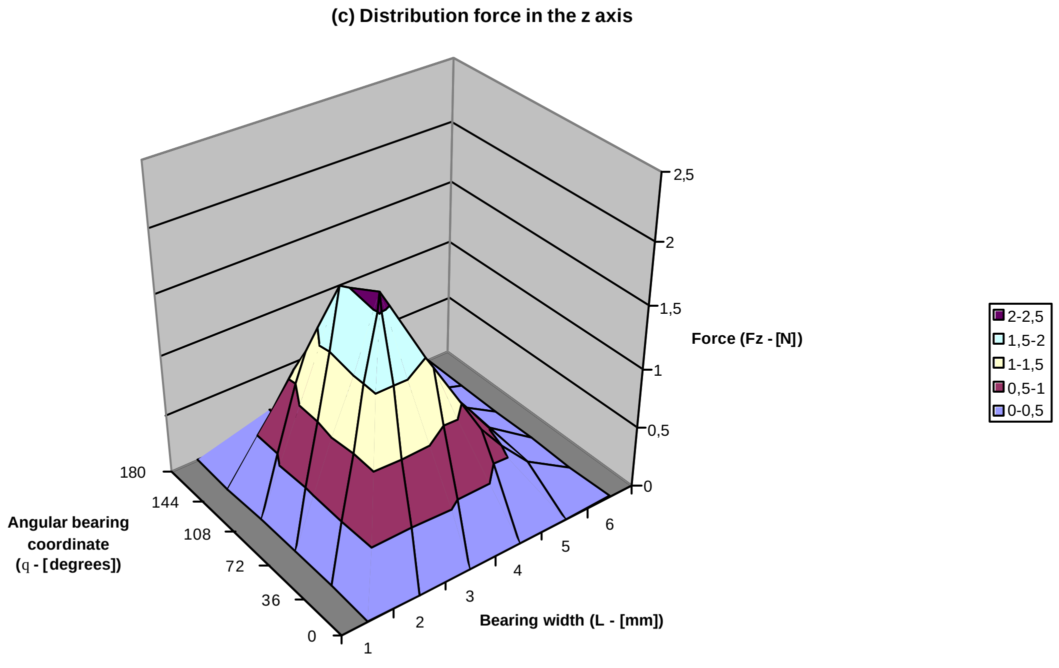


Figure 7. Hydrodynamic forces components (a)

Considering several eccentricity conditions we can evaluate the maximum forces as the graph below:

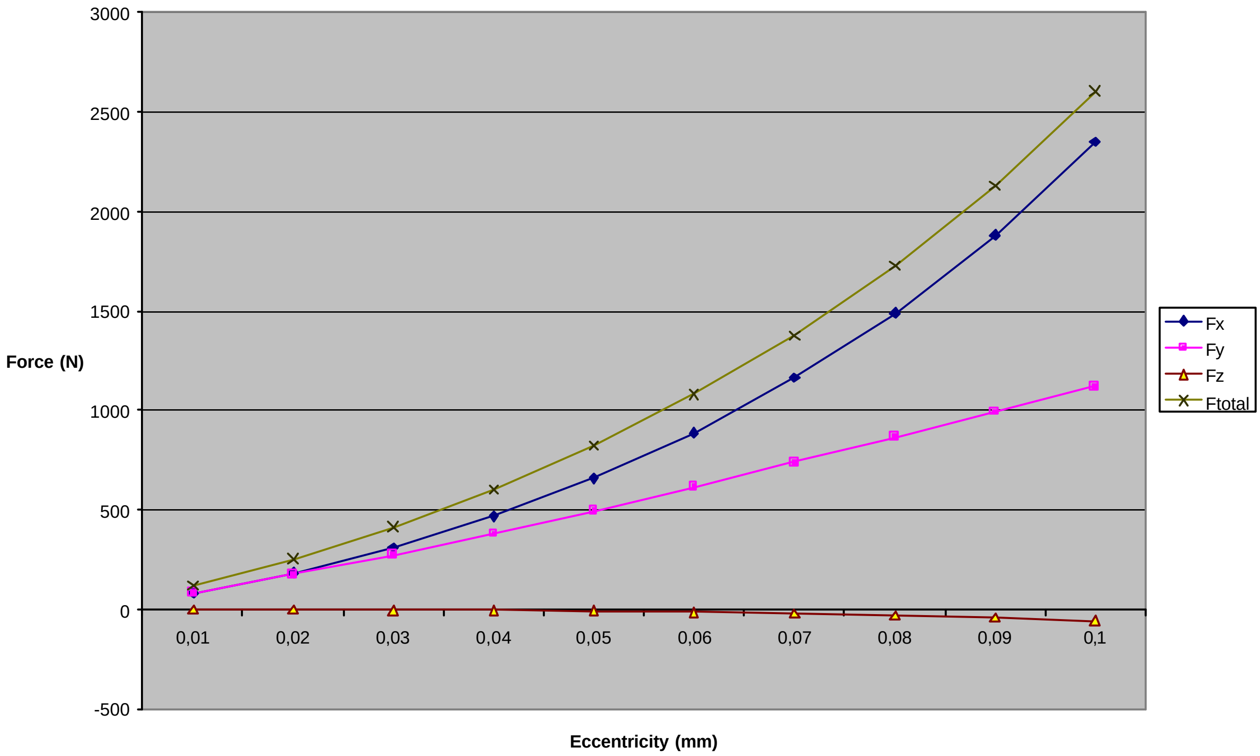


Figure 8. Hydrodynamic forces X shaft eccentricity

8. Conclusions

Starting from the obtained data we evaluated that the effective forces in z are relatively small so it facilitates the shaft support tilting, minimizing waste. The forces in y direction shown to be higher for high eccentricity values which indicates an instability area for eventual dynamic problems. The x values complement the dynamic forces required in the hole system . The experimental tests are necessary to prove some considerations .

9. Acknowledgement

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