Engines camshaft evaluation regarding tappets rotation

Alexandre Mastaler
Schaefler Brasil Ltda – Division INA/FAG – Av. Independência, 3500
Sorocaba, SP – Brazil
E-mail: alexandre.mastaler@br.ina.com

Katia Lucchesi Cavalca
UNICAMP – State University of Campinas, School of Mechanical Engineering, Department of Mechanical Design, Postal Box: 6051 – 13083-970 – Campinas, SP - Brazil
E-mail: katia@fem.unicamp.br

Abstract. *The last developments regarding the valve train system used in combustion engines leads to the simulation way. The aim is to recognize the potential improvement and to reduce the development time. The tappet is one of the central parts that influence the general dynamic system. The approached model reaches the hydrodynamic oil distribution and the forces/moments involved in the tappet system. The main idea is to analyze the tappet rotation and evaluate its influence in the system. The knowledge of this sub-system is important to evaluate ware and noise that came from this contact, as well as, friction evaluation for the command general optimization. To reduce wear, the tappet must rotate, but in some applications the tappet has only a linear displacement. Through the knowledge of pressure distribution, the forces in the sub-system are known. Considerations about friction, moments and tilting generated by the reactions in the tappet, enables the balancing conditions in the system. With these evaluations it is possible to notice the condition that allows the tappet rotation. The geometry can be optimized to achieve better results.*
Keywords: *- Valve train system, pressure distribution, axial tilting.*

1. Introduction

The emissions reduction and power improvement, for the modern passenger cars is a constant challenge. Emissions reduction is an issue mentioned many times to the engine designers, environmental friendly fuels need to be carried out in parallel with engine development, adding to the beliefs that fuels of biomass and renewable sources might account for about 20 % of Europe's fuel requirements. Natural gas is also significant, particularly in terms of lowering CO_2 levels, the company that introduces a power unit of this type meets EU4 emissions levels with CO_2 20% reduction (Birch, 2005). The decrease in emission level is also important for the automobiles improvement. In Europe and the US the limits are getting so small that they are hindering the engine projects. HVA (hydraulic valve adjusters) is one of the items that aids the emissions reduction, facilitating the designer work when they want to reach a lower level of emissions. With the new systems OBD (on board diagnosis) the vehicle electronics central demands a more optimized engine project. Through the years HVA is being more and more redesigned to achieve the customers requirements. Some works have been published with these considerations, specially with highlights in the valvetrain system (Alexander,2004 and Carney,2004). For some years we have been trying to achieve some parameters to design the entire camshaft / tappet contact . The experience showed that it is necessary to collect some information in the literature to improve the models. The experiments are expensive and demand an unavailable time. The trends in the market can not wait to the whole models, which means, that it is necessary to approach the models with experience .

2. Mathematical Model

The evaluation that will be treated here refers to the tappets (highlighted in Figure (1)). This tappet that will be called "journal" is considered inside of the formulation as a conventional axis (or shaft/journal). The guidance hole where the axis or journal slides along will be called "bearing". Figure (1) 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 condition of operation. With the same importance we have the evaluation for the shaft tilting (fundamental requirement for the wear distribution in this component type). The rotation on the tappet is necessary to reduce the wear in the contact with the camshaft, in this case the rotation distributes the wear and avoids the damage in some concentrated points. 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 an incompressible oil fluid film (Cavalca, 2001).

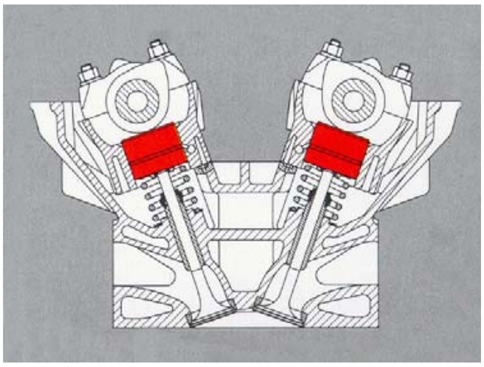

Figure 1. System Sketch

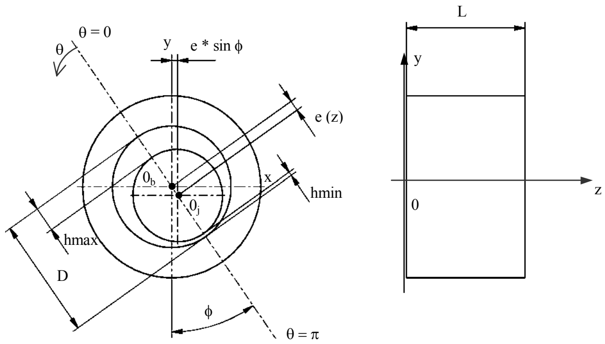

Figure 2. Journal - bearing system

Symbols list:

e	- eccentricity between the journal and the bearing
ϕ	- eccentricity angle between the 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

3. The hydrodynamic bearing

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 ".. We have the Reynolds Equation (Reynolds, 1886):

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

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 as indicated below :

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

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

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

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

$$e(z) = \left(L - \frac{L}{2} - z \right) \times sin \ \alpha \qquad (5)$$

$$e(z) = \left(\frac{L}{2} - z \right) \times sin \ \alpha \qquad (6)$$

Achieving finally the complete expression :

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

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, do not have complete solution. The differential solution of Reynolds equation is based on a balance of forces in a flux element for incompressible Newtonian fluid which is subject to a laminar flux. The external forces are in balance with hydrodynamic and friction forces. Considering the exposure, the dimensionless Reynolds equation presents a numeric solution (Cavalca e Cattaruzzi, 2001; Pinkus, 1956,1959; Irretier, 2002):

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

Equation (8) can be considered in the format:

$$p_n = c_0 + c_1 \times p_T + c_2 \times p_R + c_3 \times p_B + c_4 \times p_L \qquad (9)$$

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

Once the pressure distribution was evaluated through the Finite Difference Method 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.

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

$$F_X = \sum_{n=1}^{n} p_n \cdot Sen \ \alpha_n \cdot Cos \ \theta_n \cdot \Delta x \cdot \Delta z \qquad (10)$$

$$F_Y = \sum_{n=1}^{n} p_n \cdot Sen \ \alpha_n \cdot Sen \ \theta_n \cdot \Delta x \cdot \Delta z \qquad (11)$$

$$F_Z = \sum_{n=1}^{n} p_n \cdot Cos \ \alpha_n \cdot \Delta x \cdot \Delta z \qquad (12)$$

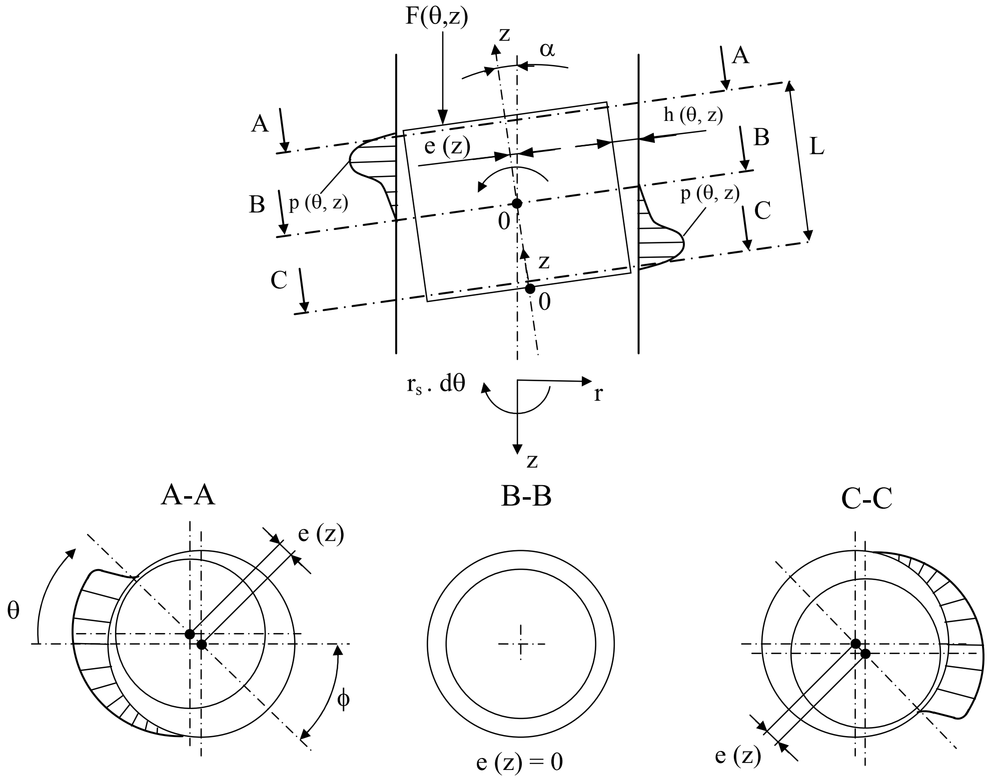

Figure 4. Pressure distribution circuferential in (θ) and (z) axial directions

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 on 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 .

$$F_R = \sqrt{{F_X}^2 + {F_Y}^2 + {F_Z}^2} \tag{13}$$

5.1 Moment Equilibrium to the Tappet

To the evaluation of the tappet equilibrium in the system of valve train command, the solution of the Reynolds differential equation through finite difference method, provides a pressure distribution for a certain operation condition of the oil film. The distribution of hydrodynamic forces acting in the tappet is reached. These values are considered in the equilibrium equations of the system.

In the simulations, considerations about the tappet tilting, radial displacement and tappet relation in respect of the guide hole will be approached. From then, it is possible to observe where to evaluate the action forces of the command and the reaction forces of the valve. There should be mentioned again that the considerations are to a determined operating condition, which means, the thermal effects of the aluminum and the steel due to a temperature variation is not taken into account. With these observations, it is possible to verify the performance of the system in respect of the spring forces and to the command shape design (Figure 4) (Schmidt,1995; Pischinger,1994 and 1999). Considering the contact point between the camshaft and the tappet, and the pressure distribution, the equilibrium condition is :

$$M_I = \sqrt{M_1^2 + M_2^2} \tag{14}$$

Where: M_I = Resultant moment from hydrodynamic forces
M_I = Moment in x direction
M_2 = Moment in y direction

$$M_1 = \int_0^{2\pi}\int_0^{h_S} p(\theta_n, z)\cdot \cos\theta_n \cdot h^{'}(z)\cdot r_S \cdot d\theta_n \cdot d_Z \tag{15}$$

$$M_2 = \int_0^{2\pi}\int_0^{h_S} p(\theta_n, z)\cdot \operatorname{sen}\theta_n \cdot h^{'}(z)\cdot r_S \cdot d\theta_n \cdot d_Z \tag{16}$$

Where: $p(\theta_n, z)$ = hydrodynamic pressure in the mesh referential (θ_n,z)
$h'(z)$ = axial coordinated z from reference O
r_S = tappet radius
θ_n = journal bearing circumferential coordinate

The final moment $\boldsymbol{M_I}$ resulting from the sum of all moments due to the hydrodynamic supporting forces, leads to the rotation of an angle $\boldsymbol{\beta_K}$ of a perpendicular plan to the top face of the tappet (Figure 4):

$$\beta_K = atag\left(\frac{M_1}{M_2}\right) \tag{17}$$

Where: β_K = angle of the resultant moments.

This internal tilting moment M_I must equilibrate the external tilting moment M_{KR}, that corresponds to the system loads due to the friction reactions force at the tappet top face and the contact normal force. The spring restoring force of the valve acts in the center of the tilting rotation of the tappet, which is the reference to the moment evaluation (Figure 4). As consequence of the geometrical eccentricity between the command and the tappet, with the radial value of I_H and the migration of the contact point I_B , which moves according to the position of the cam, the value of normal forces is quite necessary to the tilting moment M_{KL} calculation (Figure 5).

$$M_{KL}(\varphi_N) = F_N \cdot \sqrt{I_H^2 + I_B^2} \tag{18}$$

Where: F_N = normal force
I_H = radial distance of force F_N from the tappet/pivot reference center perpendicular to I_B
I_B = distance of the force F_N the A-A plane of the tappet/pivot.
φ_N = contact angle of F_N (Figure 5).

The moment due to the friction contact between camshaft and follower is also necessary, where axial distance h_B is the tilting component.

$$M_{Fat}(\varphi_N) = F_N \cdot \mu_E \cdot h_B \tag{19}$$

Where: μ_E = friction coefficient.
h_B = tilting axial component or axial distance from the reference center in z up to the top surface of the tappet.

Calculations of the tappet camshaft friction conditions (or finger follower with the pivot) give the equilibrium conditions. It is possible to calculate this values to obtain fixed and controlled boundary conditions, although this estimate does not insert in this work target.
The vector sum of both fractions in the geometrical space results in a mathematical expression to the extern tilting moment M_{KR} .

$$M_{KR}(\varphi_N) = \sqrt{M_{KL}^2 + (\mu_E \cdot F_N \cdot h_B)^2} \tag{20}$$

As a reference value to the system evaluation, a displacement angle (γ) associated to the external tilting moment M_{KL} is observed, as function of the camshaft angle of rotation and the M_{Fat} .

$$\gamma = atag\left(\frac{M_{KL}}{\mu_E \cdot F_N \cdot h_B}\right) \tag{21}$$

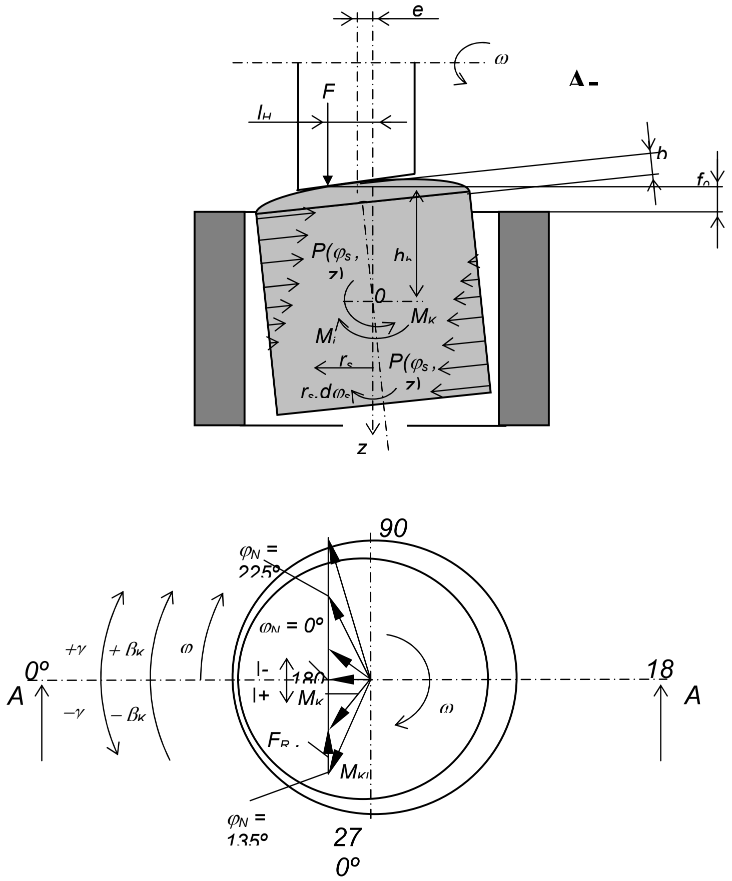

Figure 5. Scheme of the tappet.

With the tilting moment equilibrium to the tappet tilting angle interactive calculations, an equilibrium of the tilting moment in respect of the tappet axial reference is also accomplishable. The introduction of the tappet driver moment is obtained through the friction in the system camshaft/follower. This driver moment is the responsible for the tappet friction moment and for the determination of the normal force F_N, which can allow its rotation. The resultant moment is:

$$M_I = M_{Res}(\varphi_N) = \sqrt{M_{KL}^2 + M_{Fat}^2} \tag{22}$$

Expanding the function:

$$M_I^2 = F_N^2 \cdot \left(I_H^2 + I_B^2\right) + F_N^2 \cdot \left(\mu_E \cdot h_B\right)^2 \tag{23}$$

And isolating F_N:

$$F_N^2 = \frac{M_I^2}{\left(I_H^2 + I_B^2\right) + \left(\mu_E \cdot h_B\right)^2} \tag{24}$$

$$F_N = \frac{M_I}{\sqrt{\left(I_H^2 + I_B^2\right) + \left(\mu_E \cdot h_B\right)^2}} \tag{25}$$

F_N is the normal force needed in the contact with the camshaft according to Figure (4). With these values, the friction force F_{at} and the tappet moment M_A can be estimated:

$$F_{at} = \mu_E F_N \tag{26}$$

$$M_A = T = F_{at} \cdot I_H \tag{27}$$

Gscheidle (1998) verified experimentally that the values of the friction coefficient μ_E between the cam and the tappet surface vary according to the cam degree of penetration and, therefore, with its angular position, being more elevated when it is out of the base circle. (Mastaler,2004).

Calculations of the tappet camshaft friction conditions (or finger follower with the pivot) give the equilibrium conditions. It is possible to calculate this values to obtain fixed and controlled boundary conditions, although this estimate does not insert in this work target. The vector sum of both fractions in the geometrical space results in a mathematical expression to the extern tilting moment M_{KR}.

$$M_{KR}(\varphi_N) = \sqrt{M_{KL}^2 + (\mu_E \cdot F_N \cdot h_B)^2} \tag{28}$$

As a reference value to the system evaluation, a displacement angle (γ) associated to the external tilting moment M_{KL} is observed, as function of the camshaft angle of rotation and the M_{Fat}.

$$\gamma = atag\left(\frac{M_{KL}}{\mu_E \cdot F_N \cdot h_B}\right) \tag{29}$$

With the tilting moment equilibrium to the tappet tilting angle interactive calculations, an equilibrium of the tilting moment in respect of the tappet axial reference is also accomplishable. The introduction of the tappet driver moment is obtained through the friction in the system camshaft/follower. This driver moment is the responsible for the tappet friction moment, for the determination of the normal force F_N, which can allow its rotation. The resultant moment is:

$$M_I = M_{\mathrm{Res}}(\varphi_N) = \sqrt{M_{KL}^2 + M_{Fat}^2} \tag{30}$$

Expanding the function:

$$M_I^2 = F_N^2 \cdot \left(I_H^2 + I_B^2\right) + F_N^2 \cdot \left(\mu_E \cdot h_B\right)^2 \tag{31}$$

And isolating F_N:

$$F_N^2 = \frac{M_I^2}{\left(I_H^2 + I_B^2\right) + \left(\mu_E \cdot h_B\right)^2} \tag{32}$$

$$F_N = \frac{M_I}{\sqrt{\left(I_H^2 + I_B^2\right) + \left(\mu_E \cdot h_B\right)^2}} \tag{33}$$

F_N is the normal force needed in the contact with the camshaft according to Figure (4). With these values, the friction force F_{at} and the tappet moment M_A can be estimated:

$$F_{at} = \mu_E F_N \tag{34}$$

$$M_A = T = F_{at} \cdot I_H \tag{35}$$

Gscheidle (1998) verified experimentally that the values of the friction coefficient μ_E between the cam and the tappet surface vary according to the cam degree of penetration and, therefore, with its angular position, being more elevated when it is out of the base circle. (Mastaler,2004).

6. Obtained Results

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

Table 1. Design Parameters

Parameters	Dimension	Units
Bearing diameter (D)	16	mm
Eccentricity (e)	0,05	mm
Width (L)	13	mm
Diametral clearance (Cd)	200	µm
ϕ angle	25	degrees
α angle (depend on eccentricity)	0,95	degrees

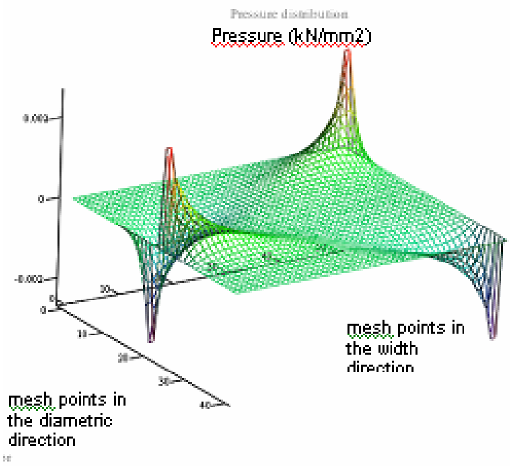

Figure 6. Pressure distribution.

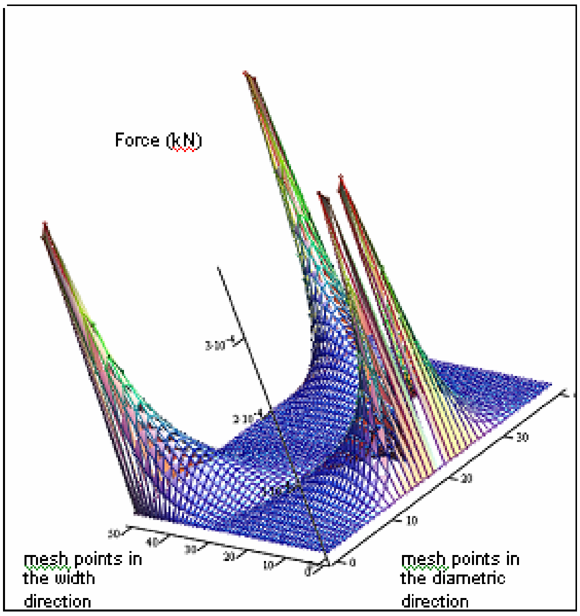

Figure 7. Hydrodynamic forces components.

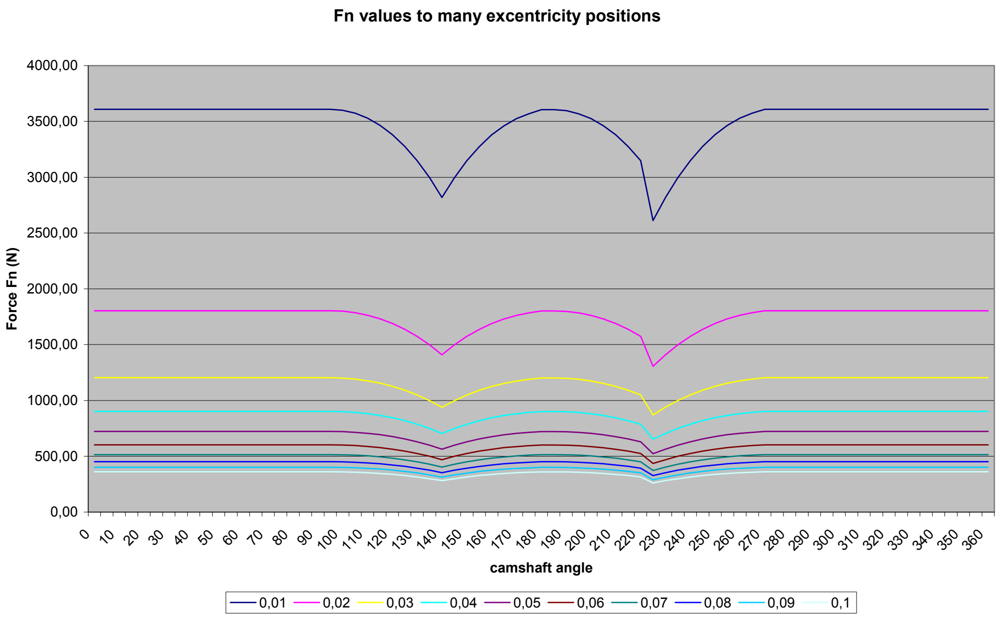

Figure 8. Normal force x camshaft angle

Using the design parameters presented and considering the fixed rotation of 1000 rpm to every condition , the oil film pressure distribution was obtained (Figure 6).

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) – Figure 7.

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

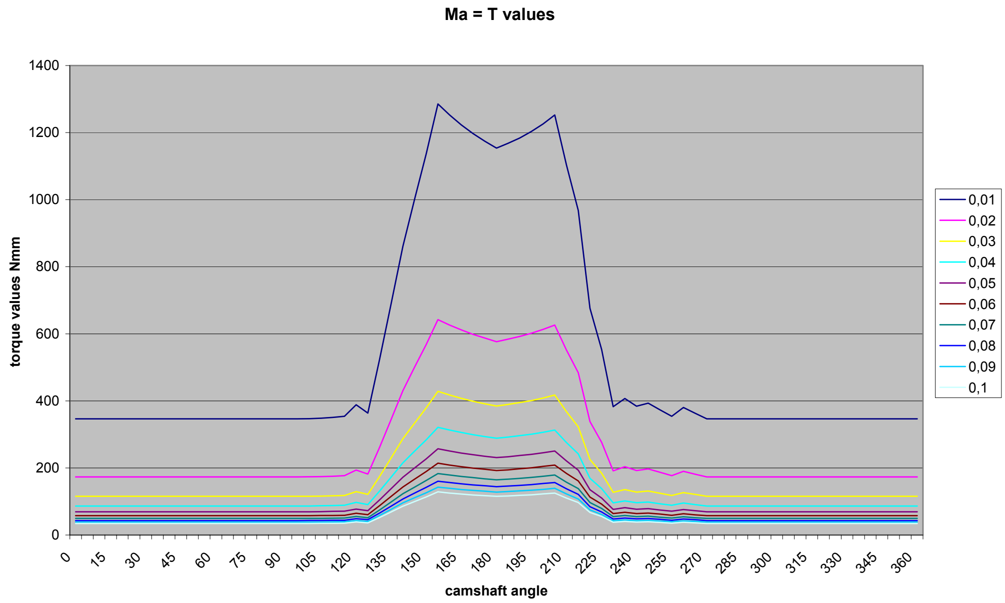

Figure 9. Torque x camshaft angle.

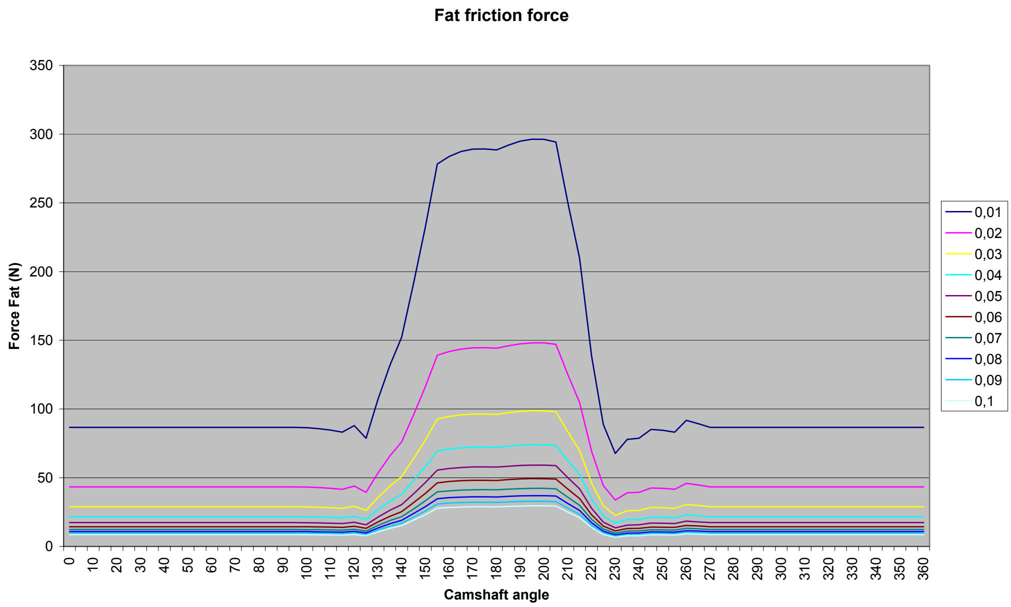

Figure 10. Friction force x camshaft angle.

8. Conclusions

Working with higher rotations ,the tappet achieves stability forces and lower eccentricity. To produce higher rotations with the contact tappet/camshaft, the design has to be calculated and tested . As it was shown the higher eccentricity results lower forces and instability. The eccentricity values among 0,01 and 0,02 mm are satisfactory and feasible for the manufacturing. Special contours can be investigated as reasonable solution to optimize the contact. To the next steps some multi-body theory analysis could be necessary to evaluate the whole system. The experimental tests are necessary to prove some considerations.

9. Acknowledgement

The authors would like to thank Schaeffler Brasil Ltda. division INA/FAG, Unicamp, Fapesp, CAPES and CNPq for the support of this work.

10. References

Alexander , David, 2004, *"Fueling the next generation"* – Automotive Engineering International -SAE International – USA , Vol. 112 number 9 – September , pg 73-77

Birch , Stuart , 2005 , *"Driving research "* , Automotive Engineering International – SAE International USA, Vol 113 number 04 – April - pg 166- 167

Carney, Dan ,2004, *"Internal-combustion engineering North America Global Viewpoints"* Automotive Engineering International , SAE International , Maio, vol.112 número 5 , pg 39-46.

Cavalca, K.L., Cattaruzi, E.F., 2001, *"A CAE Tool Dedicated to Design of Hydrodynamic Bearings"*- IJMEE – The International Mechanical Engineering Education, Manchester – UK, Vol.29, No.4, pg. 361-380.

Cavalca, K. L., Dedini F. G. , 1998, *"Experimental analysis of a tilting pad journal bearing influence in a vertical rotating system"* –5th International Conference on Rotor Dynamics, IFTo MM 1998 – Darmstadt – Germany – pp. 571-582

Castro H.F., Cavalca K.L., Idehara S.J. , 2004 , *"Updating method based on genetic algorithm applied to non-linear journal bearing model"* , 8th International Conference on Vibrations in Rotating Machinery , I MEch E 2004 , C623/091/2004 , Swansea – Wales , 2004 – pp. 433-444

Gscheidle , R. , 1998, *"Tabellenbuch Kraftfahrzeugtechnik"* – Europa-Lehrmittel GmbH & Co. – pg. 157-184.

Irretier, H., Lindemnan, S., 2002, *"Improvement of Damping Parameters of Flexive Rotors by Experimental Model Analysis and Model Updating"* – 6Th IFToMM, Sydney, pg. 388-395.

Kingsbury, A., 1931, *"On Problems in the Theory of Fluid-Film Lubrication , With an Experimental Method of Solution"* – Transactions ASME , pg. 59-75

Mastaler, A., Cavalca K.L., 2003, *"Journal Bearings Applied in the Camshaft Support Elements for High Performance Engines"* – Cobem 2003, São Paulo – Brasil

Mastaler, A., Cavalca K.L., 2004, *"Journal Bearing Theory Applied in the Simulation Camshaft Behaviour"* – XXV Cilamce 2004, CIL21-034, Recife- PE – Brasil

Pinkus, O., 1956, *"Analysis of Elliptical Bearings",* TRNAS. ASME, vol. 78, pg 965 - 973.

Pinkus, O., 1959, *"Analysis and Characteristics of Three-Lobe Mancal, Bearing of basic Engineering"* pg 49-55.

Pischinger , Dr. Techn. Dr. e. h. F , 1994, *"Verbrennungsmotoren – Vorlesungsdruck Band I und II – 15. Auflage"* – Lehrstuhl Für Nahgewandte Thermodynamik Reinisch – Westfälische Technische Hochschule Aachen – Trans-aix-press

Pischinger , S. , 1999 , *"Fahrzeugantriebe der Zukunft : Die Verbrennungskraftmaschine und ihre Alternativen "* – MTZ – Zukunft – Perspektiven des Verbrennungsmotors – Sonderheft 60 Jahre Motortechnische Zeitschrift

Reynolds, O., 1886, *"On the Theory of Lubrication and its Application to Mr. Beauchamp Tower's Experiments"-,* included in –*"Experimental Determination of the Viscosity of Olive Oil"* pg 188.

Schmidt, Dipl., Ing. Andreas, 1995 ,Dissertation Doktors-Ingenieurs aus Clausthal-Zellerfeld , *„Einfluss Tassenstößeldrehung auf die Reibung - und Schmierverhältnisse im System Nocken-Gegenläufer"* pg. 12-17.