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On the Simulation of Crankshaft Dynamics

DOMENICO GUIDA
Department of Mechanical Engineering
University of Salerno
Ponte Don Melillo, 84084 Fisciano (SA)
ITALY
guida@unisa.it

Abstract: - A theoretical model to investigate on the influence of the fluid film force on the crankshaft dynamics is proposed. This analytical approach is more effective than a numerical investigation and makes it possible to cover not only the individual case but also the whole class to which the system in question belongs. This model allowed us to set up a first code in *Mathematica 4.0* for quick simulations of the journal orbit of the main bearings in full load conditions. The code core is the analytical formulation of the fluid film force proposed in [1].

Key-Words: - Bearings, Finite Journal Bearing, Hydrodynamic Lubrication, Fluid Film Force.

1 Introduction

The engine block and crankshaft are coupled by hydrodynamic bearing forces of the main bearings. Hydrodynamic bearings often are calculated by solving Reynolds' equation using a finite element procedure. This method is very expensive in computer time because of non-linear function of the fluid film force and is not suitable when we want analyze the dynamic behavior of a structure in a fluid force field.

In general, the analytical approach to analyze the behavior of a system is more effective than a numerical investigation and makes it possible to cover not only the individual case but also the whole class to which the system in question belongs. The analysis of many aspects of the dynamic behavior of a rotor on lubricated bearings, such as the time transient analysis, requires an analytical model in order to provide a rapid description of the hydrodynamic force field governing the interaction between journal and bearing. In fact, the use of pure numerical transient analysis incurs the computational expense of solving the two-dimensional Reynolds equation with the numeric methods in order to determine the fluid film force at every time step of the numerical integration procedure of the motion equations.

In the present paper a new analytical approximate formulation of the fluid film force in finite lubricated bearings is used [1] in order to set up a very quick code to analyze the dynamics of journal.

The main input data for the computation is the geometric data: diameters, widths and clearances of the bearings, masses of the cranks, counterweights, connecting-rod and pistons; the cylinder pres-sure diagrams of the engine for all operating engine speeds; values and directions of additional constant load vectors; oil viscosity. The main output data is the journal orbit..

2 Mathematical Model

With reference to the journal bearing in Figure 1 the pair of coordinates identifies the generic point T on the internal surface of the bearing while the intersection point P of the journal axis on the bearing middle plain is identified by the pair of coordinates [4]. The thickness of the oil film, which is assumed to be independent from the axial coordinate, is:

$$\bar{h} = C(1+\varepsilon\cos\theta)$$

assuming the film pressure distribution proposed by Warner [1] :

$$\bar{p}_w(\theta,z) = \bar{p}_L(\theta)\,\gamma(\bar{z},\lambda)$$

where

$$\bar{p}_L(\theta) = \bar{p}_\infty(\theta) - \bar{p}_a = 6\mu\omega\left(\frac{R}{C}\right)^2 \times$$

$$\times\left[\frac{\varepsilon(1-2\dot{\varphi})\sin\theta}{2+\varepsilon^2} - \dot{\varepsilon}\cos\theta\right]\times\left[\frac{1}{1+\varepsilon\cos\theta} + \frac{1}{(1+\varepsilon\cos\theta)^2}\right]$$

and:

$$\gamma(\bar{z},\lambda) = \left\{1 - \frac{Cosh(\bar{z}q\lambda)}{Cosh(q\lambda)}\right\}$$

are respectively the Sommerfeld solution for the cavitated infinite long bearing [1] and Warner's flow correction factor with q and λ indicated in the following relations:

$$q^2 = \frac{\int_\alpha^{\alpha+\pi} h^3\left(\frac{dp_L}{d\theta}\right)^2 d\theta}{\int_\alpha^{\alpha+\pi} h^3 p_L{}^2\, d\theta} \qquad \lambda = \frac{\mathrm{L}}{\mathrm{D}}$$

Where α is calculated from equations [2]:

$$\begin{cases}(2+\varepsilon^2)\dot{\varepsilon}\cos\alpha - \varepsilon(1-2\dot{\varphi})\sin\alpha = 0\\(2+\varepsilon^2)\dot{\varepsilon}\sin\alpha + \varepsilon(1-2\dot{\varphi})\cos\alpha \ge 0\end{cases}$$

and the pressure for $\theta = 0$.

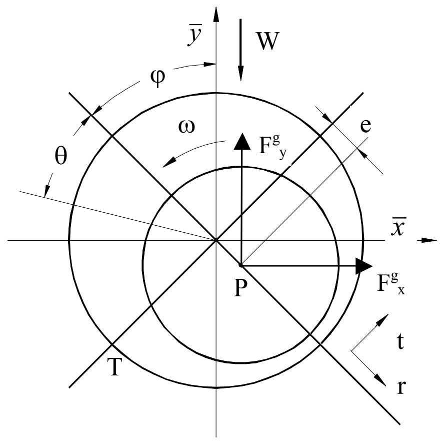


Fig.1 – Definition of hydrodynamic parameters

$$\begin{Bmatrix} f_r \\ f_t \end{Bmatrix} = \int_{-1}^{1} \int_{\alpha}^{\alpha+\pi} p_w(\theta) \begin{Bmatrix} -\cos(\theta) \\ \sin(\theta) \end{Bmatrix} d\theta \, dz$$

On the other hand the differential equation of motion in reference rotating frame can be write in the following form:

$$M \begin{Bmatrix} \ddot{\varepsilon} - \varepsilon\dot{\phi}^2 \\ \varepsilon\ddot{\phi} + 2\dot{\varepsilon}\dot{\phi} \end{Bmatrix} = 6\mathbf{f} + \mathbf{Q}(\mathbf{w} + \mathbf{f_g}) \qquad \phi = \varphi + \frac{3\pi}{2}$$

where with:

$$\mathbf{f} = \begin{Bmatrix} -f_r \\ +f_t \end{Bmatrix}; \quad \mathbf{f}_g = \begin{Bmatrix} F_{gx} \\ F_{gy} \end{Bmatrix}; \quad \mathbf{w} = \begin{Bmatrix} w_x \\ w_y \end{Bmatrix};$$

$\mathbf{Q}$;

$$S = \frac{\mu\omega RL}{W}\left(\frac{R}{C}\right)^2, \quad M = \frac{\overline{M}}{M_0}, \quad p = \frac{\overline{p}}{p_0},$$

$$M_0 = \frac{SW}{\omega^2 C}, \quad p_0 = 6\mu\omega\left(\frac{R}{C}\right)^2$$

are indicated the: fluid film force vector in the rotating frame [1], constant load vector in fixed frame, variable load vector in fixed frame, fixed-rotating frame rotation matrix, Sommerfeld number, dimensionless mass, dimensionless pressure, reference mass and reference pressure.

3 Results and Conclusions

In Figures 2, 3, 4 and 5 the dimensionless orbit of the journal centre for the following operating conditions are shown:

M_p = 2.0 Kg	Crankshaft mass/numbers of main bearings
M_a = 0.5 Kg	Mass connecting rod, piston, spin
M_r = 0.4 Kg	Mass rotating
r = 0.05m	Crank length
l = 0.15 m	Connecting rod length
μ = 0.008 Pa s^{-1}	Oil viscosity
D = 2R= 0.055 m	Bearing diameter
L = 0.025 m	Bearing length
λ = L/(2 R)	Aspect Ratio
C = 25, 80 μm	Radial clearance

In fig. 2 the results for a radial clearance of 25 μm and 80 μm with 2000 rpm in full load conditions are shown. In figures 4, 5, 6 and 7 the results of the simulations for 3000, 4000 and 6000 rpm are indicated, respectively.

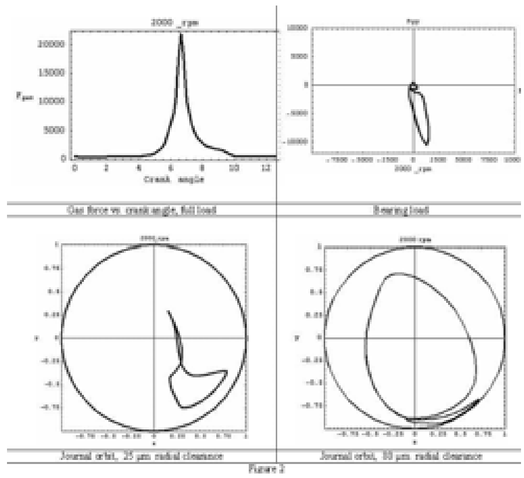


Figure 2

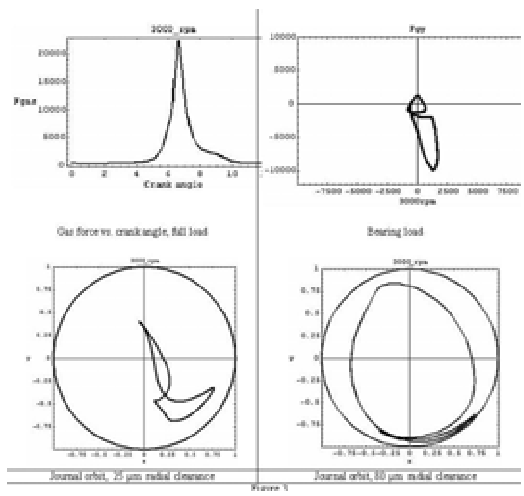
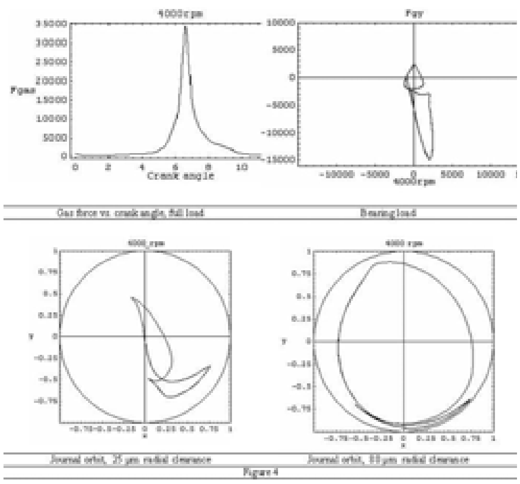
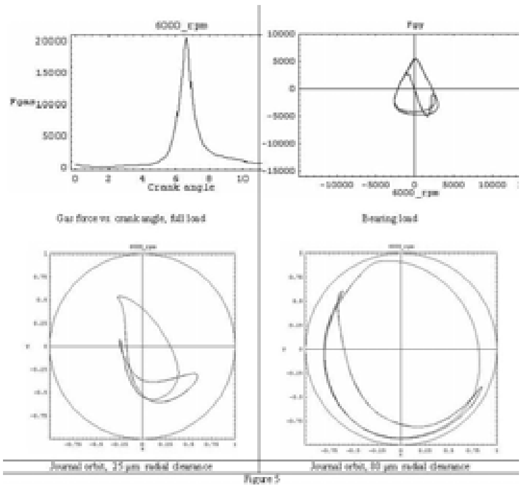


Figure 3





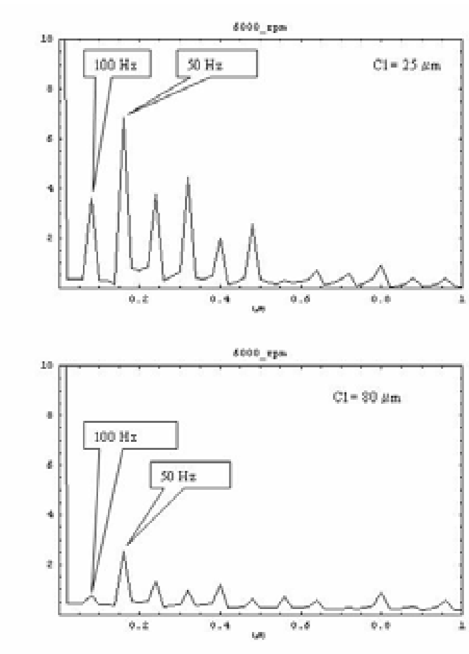


Figure 5 - Dimesionless eccentricity spectra
Radial clearance, 0.25 µm, 0.80 µm

The code is based of following assumptions:

1. finite lubricated journal in laminar, incompressible and isoviscous flow;
2. negligibility of fluid inertia so as to justify the use of the standard form of the Reynolds equa-tion;
3. Warner's flow factor correction for the pressure field;

Analysing the simulations proposed in figure 2,3,4,5 the following considerations can be made:

a) increasing radial clearance the dimensionless minimum thickness of the oil film decrease
b) the dynamics of journal induce subarmonic frequencies of order ½;

The code proposed allows us not only quick simulations but can be used as tool in the dynamic analysis of multi-body systems in fluid film force due to the lubrication mecha-nism. The present paper has shown the first results of a project on the dynamic analysis of crankshaft. This project will be developed for each main bearing type used in real industrial applications.

Nomenclature

$C =$	Radial clearance
$D=2R =$	Bearing diameter
$e =$	Eccentricity
$F_r, F_t =$	Components of the fluid film force in rotating system frame
$f_r, f_t =$	Dimensionless components of the fluid film force in rotating system frame
$\bar{h} =$	Oil film thickness
$h = \bar{h} / C =$	Dimensionless oil film thickness
$L =$	Bearing length
$\bar{p} =$	Pressure
$p_0 = 6\mu\omega(R/C)^2 =$	Reference pressure
$p = \bar{p} / p_0 =$	Dimensionless pressure
$R =$	Bearing radius
$S = \mu\omega RL/W \left(R/C\right)^2 =$	Sommerfeld number
$W =$	Load
$\mu =$	Dynamic viscosity
$e =$	Eccentricity
$\varepsilon = e / C =$	Eccentricity ratio
$\varphi =$	Attitude angle
$\lambda = L/D =$	Aspect ratio
$\mathrm{M} =$	Journal mass
$m = M\omega^2 C /(SW) =$	Dimensionless journal mass
$\bar{x}, \bar{y} =$	Coordinates of journal centre in fixed system frame
$x = \bar{x}/C; y = \bar{y}/C$	Dimensionless coordinates of journal centre
$Fgx =$	$\frac{1}{2}\begin{pmatrix} mr\omega^2 Sin(\omega t) + \\ (Fgas(\omega t) - \\ m_a\omega^2 r(Cos(\omega t) + \\ \lambda Cos(2\omega t) \end{pmatrix}$
$Fgy =$	$-\frac{1}{2}\begin{pmatrix} -mr\omega^2 Sin(\omega t) + \\ (Fgas(\omega t) - \\ m_a\omega^2 r(Cos(\omega t) + \\ \lambda Cos(2\omega t) \end{pmatrix}$

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