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A NEW METHODOLOGY TO DETERMINE THE ROTATIONAL SPEED OF A FLOATING RING IN A SHORT RADIAL JOURNAL BEARING

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Abstract: *Turbocharger rotors for the spark and compression ignition engines are resistant steels manufactured in order to support the high exhaust gas temperatures. The power of these systems is up to 10 kW and the operating speeds are important, ranging between 30000 ÷250000 rpm. The turbocharger lubrication complicates the model, because the classical hydrodynamic theory is difficult to be applied in order to characterize the floating bush bearings operating conditions. Also, the rotational speed of the floating short ring inside the turbocharger is difficult to be determined. The paper proposes a study using Mathcad to find a simplified method for mathematical computation of a floating ring from journal bearing using the equilibrium friction torque acting on the floating ring as result of the two journal bearings: between the shaft and the floating ring and between the floating ring and the housing.*
Key words: *floating ring, journal bearing, HD lubrication, clearance, friction torque.*

1. INTRODUCTION

Turbocharger is a key component in modern engines in order to increase their power. There are also great benefits by supercharging the engine because it is possible to obtain the same power output considering that the engine size and weight are more and more reduced. Turbocharging combined with this "downsizing" is also an effective way to reduce the engine consumption [1, 2]. There are various methods to increase the engine power, but one of the most used is to increase the air flow into the cylinder [1,2].
A turbocharger consists of a turbine and a compressor located on a shaft.
The turbocharger running is initiated by the engine exhaust gases acting upon the turbine. Therefore, this component is subjected to high temperatures exceeding 1200 K and the working turbine rotation speeds can reach 100000 ÷ 250000 rpm [3]. On the other shaft end, the compressor who aspires fresh air and compresses it is positioning. The resulting air pressure (commonly 1.5÷2.7 bar) is called boost pressure and the turbocharger systems are measured by its amount. Therefore, the fuel/air combination substantially improves the engine volumetric efficiency with consequences on the engine power, which could be considerable increased and also, the fuel consumption is reduced [3].
The engine technological development also requires the improvement of turbochargers dynamic characteristics, therefore theoretical research becomes more significant and necessary in order to realize an exhaustive covering model. First, they were one-dimensional codes expressing quasi-static approach based on the turbocharger rotors characteristic maps usually provided by the manufacturers [4]. However, the performance data from maps are recorded on the test bench in a steady operation of the turbocharger system, while in a real situation the turbine and the compressor operate under unsteady conditions caused by the engine pulsations. As consequence, another series of codes with varying degree of complexity were resulting in different approaches taking into account of heat transfer, mechanical losses and unsteady behavior during the engine operation [9].
In the last years, the models are verified through experimental measurement procedures with accelerometers and laser [5, 6, 7].
Contribution to improve the knowledge in this field are related to the lubrication problems studied both experimentally, by measuring the thermodynamic variables in turbocharger test benches, and theoretical, by means of mathematical problem in Mathcad.

2. TURBOCHARGER GENERAL CONSIDERATIONS

An automotive turbocharger consists in a turbine acted by the exhaust engine gas, a compressor for air supply and a steel shaft.
The turbine and the compressor are usually supported on floating ring or semi-floating ring bearings, free to rotate, and the oil film also contributes to the complex turbocharger system behavior.
The floating bush ring is commonly used because it is an efficient and cheap component. The design concept for the floating bush bearing was to reduce friction losses and to realize the vibrations damping. Therefore, the inner clearance is smaller than the outer clearance, so the bushing rotation is a fraction of shaft speed. The floating ring speed is determined by the friction torque balance between the inner and outer oil films. The rotation of the outer surface of the floating bush bearing acts as a film damper [9].
The pressure distribution is mainly generated by the rotational speed of the floating ring, according to HD lubrication behavior.
Figure 1 presents the general view of a turbocharger with its most important components [8, 9].

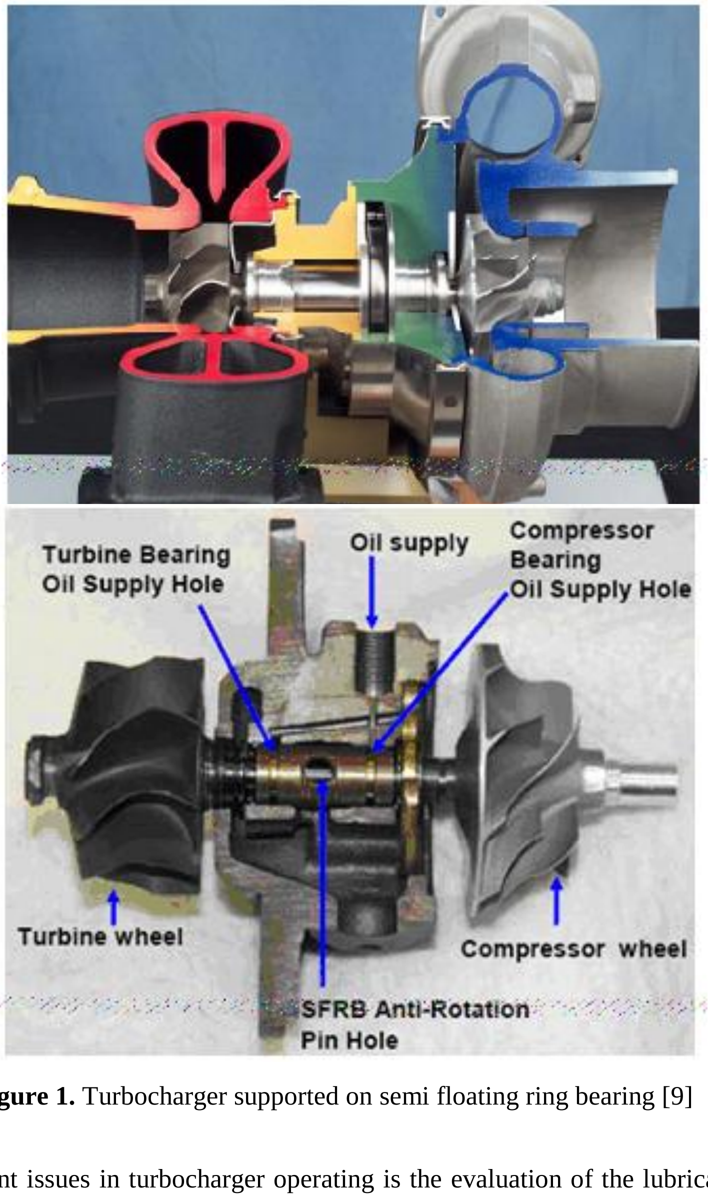


Figure 1. Turbocharger supported on semi floating ring bearing [9]

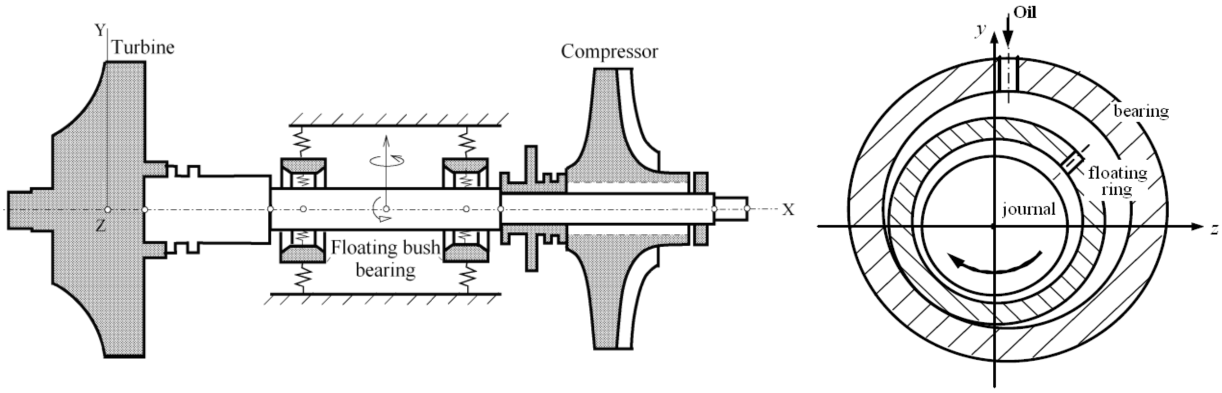
One of the most important issues in turbocharger operating is the evaluation of the lubrication regimes in both journal bearings (between shaft and floating ring and between floating ring and the housing). The phenomena complexity leads to a difficult analysis.
In the present paper the authors propose a simplified methodology to solve the HD lubrication problems in the short floating ring journal bearing.

3. A NEW METHODOLOGY TO SOLVE HD TURBOCHARGER LUBRICATION

A simplified concept, illustrating a possible algorithm which can be applied to the turbocharger model (Figure 2), include the following steps:

- solving the Reynolds equations for a short journal bearing, realized between shaft and floating ring, considering an initial value of the floating ring angular speed;
- evaluation of the oil friction torque developed between shaft and floating ring;
- solving the Reynolds equations for a short journal bearing realized between floating ring and housing;
- evaluation of the oil friction torque developed between floating ring and housing;
- evaluation of the friction torques equilibrium acting on the floating ring.

The steps are reiterated for various floating ring speeds until the balance of the friction torques is realized.
Then, the HD parameters for both journal bearings are determined (minimum film thicknesses, pressure distribution, normal and tangential forces).



(a) (b)

Figure 2. Turbocharger model with two floating rings (a), and a floating journal bearing (b) [9]

For the two short journal bearings presented in the Figure 2-a, the following geometrical and operating parameters must be considered in the start of the simulating program: *d* – shaft diameter, *J1* – diametric clearance between shaft and floating ring, *J2* - diametric clearance between floating ring and housing, *B*- floating ring width, *ns* – rotational speed of the shaft, *nr*- rotational speed of the floating ring, η – oil dynamic viscosity, *Fr* – radial force acting on a short journal bearing.
The first step (S1) supposes an initial value settlement for rotational speed of the floating ring, *nr*.
The second step (S2) supposes a value imposed for angular position of the angle $\beta 1$ between direction of the radial force *Fr* and center line for the bearing realized between shaft and floating ring.
In the third step (S3) the relative eccentricity $\varepsilon 1$ from the radial forces equilibrium is determined.
So, the pressure distribution in the first short bearing realized between shaft and floating ring is given by following equation [10]:

$$p(\theta, y) = -\frac{24 \cdot \eta \cdot n1 \cdot \varepsilon 1 \cdot \sin(\theta)}{J1^2 \cdot (1 - \varepsilon 1 \cdot \cos(\theta))^3} \cdot \left(y^2 - \frac{B^2}{4} \right) \tag{1}$$

where rotational speed *n1* is given by relation:

$$n1 = ns + nr \tag{2}$$

The equilibrium equation between the pressure acting on the journal bearing and radial force is given by following equation [10]:

$$\int_0^{\pi} \left[-\frac{24 \cdot \eta \cdot n1 \cdot \varepsilon 1 \cdot \sin(\theta)}{J1^2 \cdot (1 - \varepsilon 1 \cdot \cos(\theta))^3} \right] \cdot \cos(\pi - \alpha + \beta) d\alpha \cdot \int_{-B/2}^{B/2} \left(y^2 - \frac{B^2}{4} \right) dy + Fr = 0 \tag{3}$$

The nonlinear equation (3) is solved and a resulted value for the eccentricity $\varepsilon 1^*$ has been obtained. The new value of the eccentricity $\varepsilon 1^*$ is compared with the imposed value of the eccentricity $\varepsilon 1$. The step S3 is repeated until the initial value for $\varepsilon 1$ corresponds to the calculated value by equation (3), $\varepsilon 1^*$.
The calculus continues in the step S2 to obtain the real value of the angle β.
The two components of the normal and tangential forces N and T has been determined by the following equations [10]:

$$N = \int_0^{\pi} \left[-\frac{24 \cdot \eta \cdot n1 \cdot \varepsilon 1 \cdot \sin(\theta)}{J1^2 \cdot (1 - \varepsilon 1 \cdot \cos(\theta))^3} \right] \cdot \frac{d}{2} \cdot \cos(\pi - \alpha) d\alpha \cdot \int_{-B/2}^{B/2} \left(y^2 - \frac{B^2}{4} \right) dy \tag{4}$$

$$T = \int_0^{\pi} \left[-\frac{24 \cdot \eta \cdot n1 \cdot \varepsilon 1 \cdot \sin(\theta)}{J1^2 \cdot (1 - \varepsilon 1 \cdot \cos(\theta))^3} \right] \cdot \frac{d}{2} \cdot \sin(\pi - \alpha) d\alpha \cdot \int_{-B/2}^{B/2} \left(y^2 - \frac{B^2}{4} \right) dy \tag{5}$$

The resulted value of the angle β^* is determined by the following equation:

$$\beta^* = \arctan\left[\frac{N}{T} \right] \tag{6}$$

The new value of the angle β^* is compared with the initial value of the angle β. If the two compared values are different, the steps S2 and S3 are repeated until the initial value imposed for the angle β is obtained.
If the step S2 and S3 are finalized, then the friction torque in the first journal bearing is calculated by the following equation [10]:

$$M_{f1} = \frac{\eta \cdot n1 \cdot d^3 \cdot B \cdot \pi^2 \cdot (2 + \varepsilon 1)}{2 \cdot J1 \cdot (1 + \varepsilon 1) \cdot \sqrt{(1 - \varepsilon 1^2)}} \tag{7}$$

In equation (7) the eccentricity $\varepsilon 1$ have the real value which was obtained as a successive calculus according to the steps S2 and S3.

In the fourth step (S4), all the previous steps S2 and S3 are repeated for the second journal bearing realized between floating ring and housing, maintaining the initial value for the rotational speed of the floating ring, *nr*.
The equations (1) – (7) are used with new following parameters: initial eccentricity $\varepsilon 2$, diametric clearance $J2$ and the rotational speed $n2 = nr$.
Finally, the friction torque in the second journal bearing, M_{f2} has been obtained according to following equation [10]:

$$M_{f2} = \frac{\eta \cdot nr \cdot d^3 \cdot B \cdot \pi^2 \cdot (2 + \varepsilon 2)}{2 \cdot J2 \cdot (1 + \varepsilon 2) \cdot \sqrt{(1 - \varepsilon 2^2)}} \tag{8}$$

A real value for the initial rotation speed of the floating ring *nr* must satisfy the following equilibrium equation for the floating ring:

$$M_{f2} - M_{f1} = 0 \tag{9}$$

The calculus is finished when the difference between the two friction torques is null (with an accepted tolerance), otherwise the step S4 must continue. So, a new value for the rotational speed of the floating ring must be supposed and the S2, S3 steps for both journal bearings must be repeated.
Finally, when the equation (9) is satisfied, the minimum film thickness for both journal bearings is determined as function of relative eccentricity by equations:

$$h_{\min,1,2} = \frac{J1{,}2}{2} \cdot (1 - \varepsilon 1{,}2) \tag{10}$$

4. SIMULATION RESULTS

The new methodology was applied for a floating ring radial journal bearings with the following geometrical parameters: B = 5 mm, d = 7 mm, J1 = 0.02 mm, J2 = 0.005 mm, the rotational speed of the shaft, ns = 10000 rot/min and the radial force Fr = 1000 N. The oil viscosity was η =0.1 Pas.

Finally, after we have realized a lot of steps S1, S2, S3 and S4, it was obtained the real rotational speed of the floating ring *nr = 3561 rot/min*, that means a rotational speed of 35.61% from the rotational speed of the shaft. This result is in good correlation with the literature values [2, 3, 4].

5. CONCLUSIONS

A new simplified methodology to solve the HD lubrication in a floating ring short radial journal bearing has been developed by the authors.

The methodology is based on a successive computation steps used in a short radial journal bearing operating in HD lubrication regime.

Two short HD journal bearings have been simultaneous solved with the classical relations having as restrictive condition the equilibrium of the viscous friction torques acting of the floating ring from the two journals bearings: one realized between the shaft and the floating ring and the second realized between floating ring and the housing.

The existence of an accurate solution for the rotational speed of the floating ring determined by the above proposed methodology suggests that the methodology can be used as a first approximation in solving the complex problems generated by the thermal effects and damping phenomena specific to the turbocharger.

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