

Computational Fluid Dynamics Calculations of Turbocharger's Bearing Losses

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ABSTRACT

Fuel consumption in internal combustion engines and their associated CO₂ emissions have become one of the major issues facing car manufacturers everyday for various reasons: the Kyoto protocol, the upcoming European regulation concerning CO₂ emissions requiring emissions of less than 130g CO₂/km before 2012, and customer demand. One of the most efficient solutions to reduce fuel consumption is to downsize the engine and increase its specific power and torque by using turbochargers. The engine and the turbocharger have to be chosen carefully and be finely tuned. It is essential to understand and characterise the turbocharger's behaviour precisely and on its whole operating range, especially at low engine speeds. The characteristics at low speed are not provided by manufacturers of turbochargers because compressor maps cannot be achieved on usual test bench.

Experiments conducted in our laboratory on a special test rig equipped with a high-precision torque meter, demonstrate that compressor performances in this area cannot be deduced from adiabatic assumption. Nevertheless, our study suggests that as long as torque at the shaft end is measured and mechanical power losses are known, the effective power provided to the air flow can be calculated. Tests and calculations reveal that these mechanical power losses cannot be evaluated by general physical laws. A better knowledge of these losses is

required. In this paper, a CFD model of a turbocharger journal bearing system is proposed. The real behaviour of what occurs in the bearing system (such as leakage flow, heat transfer from the inner film to the outer through the brass bearing material) has been computed with the energy equation. The bearing system performance is presented against the rotational speed at various oil inlet temperatures and pressures. The impact of those parameters has been studied in detail and presented in this paper. It is demonstrated that the oil temperature rise decreases the friction torque along the rotational speed by making the viscosity drop. Moreover, an increase of the oil inlet pressure results into a higher friction torque. This paper provides an analysis of this trend showing the link between oil inlet pressure, oil mass flow and thermal exchange inside the bearing. Results also present the variation of oil mass flow along the entire speed range and its distribution between the inner and outer clearances.

INTRODUCTION

This paper presents a 3D CFD model for turbocharger journal bearing friction losses. When analysing a journal bearing, the generalised Reynolds equation [1] is used. This equation issued from mass continuity and momentum conservation has been solved in a non dimensional form, with various values of L/D ratio and eccentricity ratio [2]. Computation results are sum up in tables which permit to calculate the functioning

parameters of many types of journal bearing, but mainly for large machines. Typically this type of bearing has a 100 mm diameter, 100 mm long, operates at 3000 rpm, and could carry a 50000 Newton load. In the case of the turbocharger bearings, these are less than 10 mm diameter and operate between 30000 and 230000 rpm. The applied static load is typically about 1 Newton. Yet, these tables were used to compute functioning parameters of the turbocharger's journal bearing [3]. Although, the calculated Sommerfeld number was too high to be included in this table, a linear approximation was considered between friction number and the Sommerfeld number but this approximation did not give the expected results. The computed power is too high and the approximation is unreliable.

Then, it was decided to solve the Reynolds equation by a finite difference method using the Gauss-Seidel iterative method in the case of turbocharger bearings [4]. This study gave good results, but the friction power was still overestimated compared to experiments [5]. This was due to the isothermal assumption which neglects viscous heating and modification of the viscosity. This is the reason for developing the 3D CFD model which is presented in this paper.

PRESENTATION

TURBOCHARGER PERFORMANCES

Compressor power is generally obtained from temperature and gas flow rate measurements, which leads sometimes to erroneous values, typically at low turbocharger speeds.

On the compressor map presented in [Figure 1](#), the circled zone, is the low speed zone which is unknown. The lines of 100,000 to 200,000 are iso-speeds in rpm and the lines of 0.60 to 0.77 are iso-efficiency lines.

Usually, the compressor power is determined using the first law of thermodynamics:

$$dh_t = \delta q - \delta w$$

With the adiabatic assumption $\delta q = 0$, the power provided to the air flow is:

$$P_{air} = q_m c_p (T_{2_t} - T_{1_t})$$

When a turbocharger operates at low speed, the error made on temperature measurement becomes significant due to the small temperature rise. Furthermore, heat transfer cannot be neglected, and the previous equation cannot be used to calculate the power.

The knowledge of friction power losses and the use of a special test bench equipped with a torquemeter which give the mechanical power provided to the turbocharger shaft will permit to calculate the power given to the air flow:

$$P_{air} = P_{meca} - P_{friction}$$

The principle of this test bench is presented in [Figure 2](#).

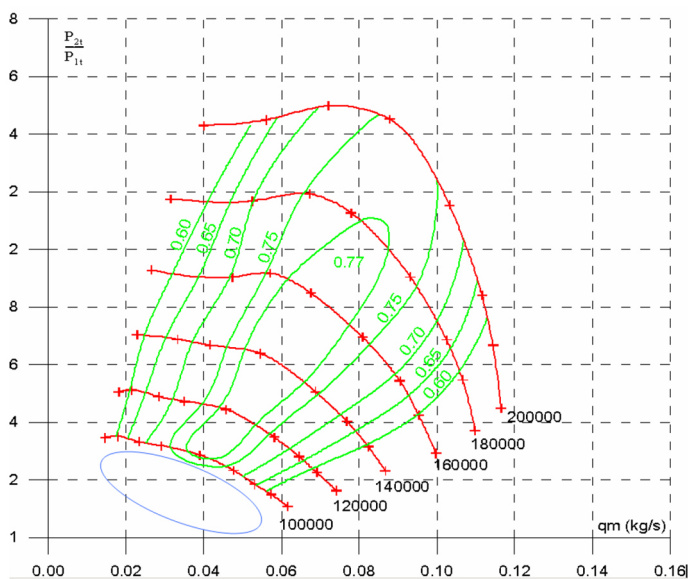


Figure 1. Typical compressor map given by manufacturer

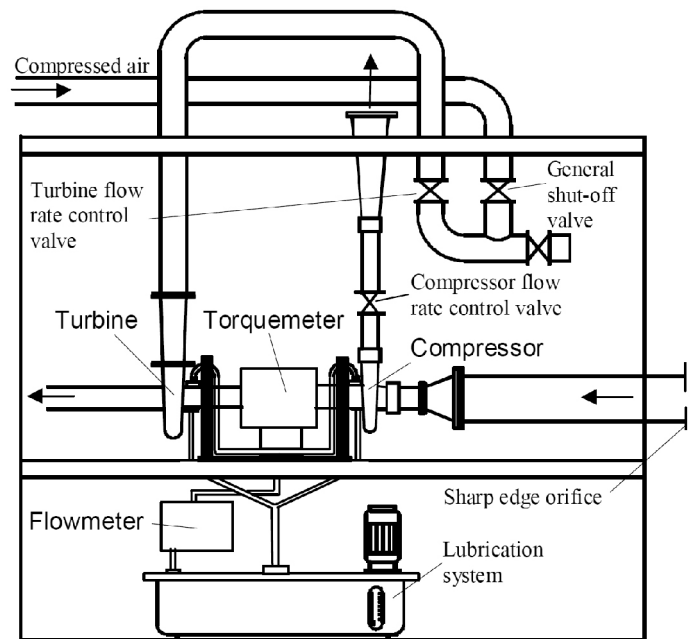


Figure 2. Turbocharger test bench with torquemeter [5].

TURBOCHARGER BEARING UNIT

The journal bearing consists in:

- The journal; the shaft or rotor,
- The bearing; the bush part.

General arrangement of journal bearing is represented in [Figure 2](#). The bearing unit is composed of two journal bearings of length 3.8 mm.

The turbocharger rotational speed is between 30,000 rpm and 240,000 rpm for automotive engines.

The bearing unit is lubricated with standard engine oil 5W30. The feed oil pressure is in the range 0.2 MPa and 0.4 MPa, and the temperature is between 20°C and 120°C.

The total load is estimated to be equal with the weight of shaft, turbine wheel and impeller: $W=1,128\text{N}$.

This study focuses on the turbocharger behaviour at low rotational speeds, less than 110,000 rpm, and on the influence of oil temperature and rotational speed on this. Only one of the two bearings is considered as a simple journal bearing.

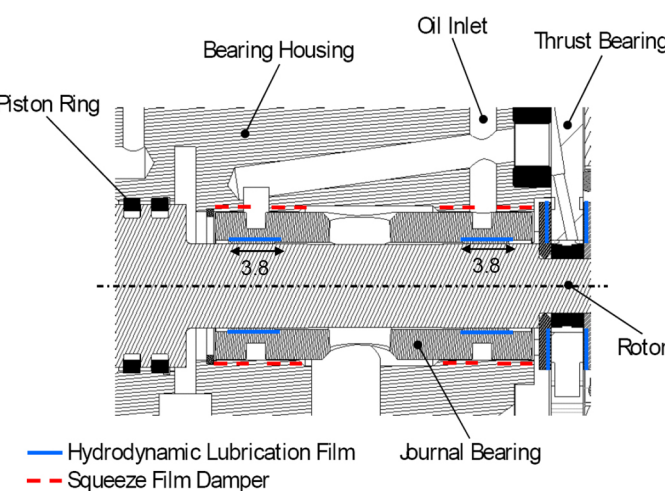


Figure 3. Sketch of the hydrodynamic bearing unit [6]

The coordinate system and the corresponding geometry of the journal bearing are shown in [Figure 4](#).

The bearing's radius is R_b and the journal's radius is R_j . The journal axis O_j is at a distance e from the bearing axis O_b . With geometrical consideration the film thickness h can be defined by the following expression using bearing coordinates: $h = C(1 + \varepsilon \cos \theta)$

With C is the radial clearance, and $\varepsilon = e / C$ is the eccentricity ratio.

The film thickness varies from its maximum value $h_0 = C(1 + \varepsilon)$ at the bearing's angle $\theta = 0$ to its minimum value $h_1 = C(1 - \varepsilon)$ at $\theta = \pi$.

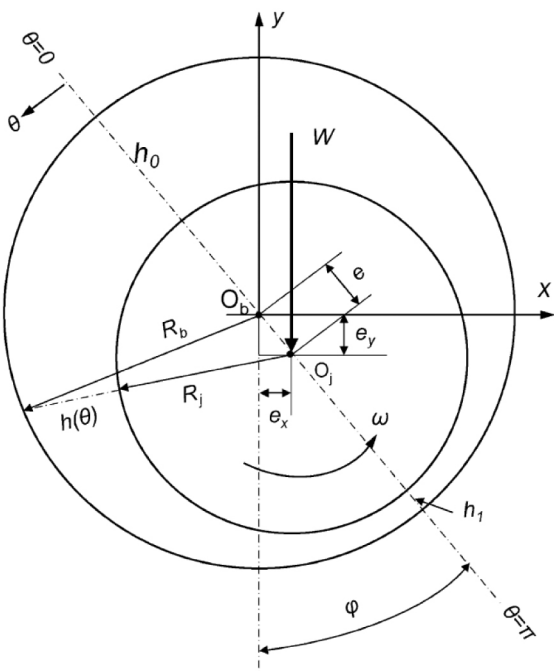


Figure 4. Schematic geometry and parameters

The characteristics of the bearing are:

Inner	diameter 7 mm clearance 15 μm
Outer	diameter 13 mm clearance 75 μm
Circumferential groove	width 1.6 mm height 1 mm
4 feeding holes	diameter 1.4 mm
Inlet pipe	diameter 1.4 mm
Load force	0.564 N

The direction of the load force W is $-y$ axis.

MODEL

In [reference \[3\]](#), authors show the inability of classical approaches to predict performances of turbocharger bearings which operate at a very high speed with high viscosity and weak load force. For this reason in [reference \[4\]](#), a 3D CFD model was developed with simplified thermal conditions but with geometry close to the real part. This model gave good results compared to experiments, but it was not able to

predict the influence of inlet pressure. Thus this work was enhanced with a more complex geometry which takes into account both inner and outer oil film and with coupled thermal conditions.

ASSUMPTIONS

Aligned bearing

Assuming aligned bearing allows the use of the symmetry condition, only half the bearing is modeled, it divides computation time by two. It also permits to consider that both bearing are the same.

Laminar flow

Reynolds number is the criterion commonly used to characterize the operating regime. Generally the regime is considered to be laminar if the Reynolds number is less than 2000.

$$\Re_e = \frac{\rho V L}{\mu}$$

Reynolds number is expressed by this formula:

The maximal value of the Reynolds number is calculated in this paragraph.

Maximum angular velocity:
maximum $N=110000$ rpm and R the average

$$V = R\omega = R \frac{\pi N}{30}$$

radius of the bearing.

Thickness of fluid: $L = C$

Viscosity at 80°C $\mu = 0.01273 \text{ Pa.s}$

Density at 80°C $\rho = 805.4 \text{ kg / m}^3$

$$\Re_e = \frac{\rho V L}{\mu} = \frac{\rho R \frac{\pi N}{30} * C}{\mu}$$

$$\Re_e = 38.26 < 2000$$

The Taylor criterion for the transition between laminar flow and Taylor vortices regime was applied and it shows that in the case of turbocharger journal bearing the flow could be assumed to be laminar.

Reynolds condition on pressure

Reynolds condition on pressure has been considered as being the most adequate in comparison with experimental results.

(See [Figure 5](#) after last section of paper.)

In this model, cavitation was not considered. From the simulation point of view, the bearing is divided into two regions. One is the active region where the oil hydrodynamic pressure is created and the other one is inactive and corresponds to the region where normally cavitation could occur. To simplify this calculation, the inactive zone is that where computed pressure is negative, and a user defined function was created to set negative pressure values to operating pressure at each iteration. This means that pressure in the inactive zone is the atmospheric pressure, which is not far from the saturation pressure.

OIL PROPERTIES

Oil dynamic viscosity SAE 5W30 is drawn in [Figure 6](#).

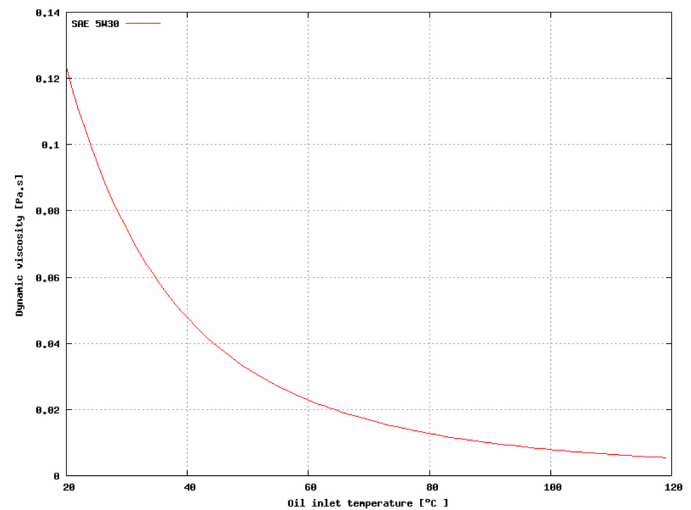


Figure 6. Dynamic oil viscosity vs temperature

It can be noticed that the oil viscosity is divided by 6 when the oil temperature changes from 20 to 80°C . The variation of oil viscosity has to be taken into account in the calculation model. The oil inlet temperature and the oil temperature variation will be important parameters.

The oil viscosity can be expressed using Vogel equation

$$\mu = a e^{\frac{b}{T-c}} \text{ with:}$$

$a=7.48\text{E}-5$, $b=1005.2$ and $c=157.45$

T is the absolute temperature in Kelvin and μ the dynamic viscosity in Pa.s.

The cooling oil mass flow depends on the oil density. So the variation of oil density has to be taken into account in the model calculation. A linear relation can be used to compute