MECHANICAL POWER LOSSES OF TURBOCHARGER AT LOW SPEEDS

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Abstract: *One of the most efficient ways to reduce the pollution and fuel consumption of an automotive engine is to downsize the engine, whilst maintaining a high level of power and torque. This is achieved by using turbochargers.*
In urban, and often in suburban traffic conditions, the engine power demand is weak in relation to the maximum power available, so the turbocharger runs at low speed. To appreciate and improve engine performance, it is necessary to know the turbocharger's characteristics in this functioning area, characteristics which are not given by the turbocharger manufacturer.
The reason for this lack of information at low turbocharger speed has been explained and results of experiments have been presented [1, 2].
For accurate computations on engine performances, real values of turbocharger mechanical efficiency must be known. Unfortunately, by lack of information, mechanical efficiency is considered as a constant (0.8 to 0.9). At low speeds, efficiencies are much lower, as demonstrated in this paper.
Our turbocharger test bench has been equipped with a torquemeter, which measure the power on turbocharger shaft. If experiments are done in adiabatic conditions, power given to the air flow can be calculated and friction losses deducted. Direct measurements of friction losses will be also done by removing the compressor wheel.
Considering our test bench layout, the axial effort on turbocharger shaft is only due to the pressure on compressor wheel. There is no counter pressure effect on turbine side as on usual turbocharger. A special device has been added to our test bench in order to generate an axial load. This device will be depicted and some experimental results presented.
At least, some information will be given on a special turbocharger whose thrust bearing is equipped with strain gauges, which permit direct measurement of axial loads in real functioning conditions.

Keywords: turbocharger, friction losses, journal bearing, thrust bearing.

INTRODUCTION

Turbochargers generally run at speeds less than 70,000 rpm in the case of Urban Driving Cycle (UDC) and less than 100,000 rpm for Extra Urban Driving Cycle (EUDC) - figure 1, [3].

Unfortunately turbocharger performances are not well known in the range of low rotational speed (less than 90,000 rpm). The lack of information available for the low speed zone is a result of the testing procedure. Usually turbochargers are tested with hot gas feeding the turbine. Compressor power is calculated on the basis of adiabatic assumptions. For low rotational speeds, the temperature rise due to compression work is weak and the heat transfer, from turbine to compressor, is relatively significant, so this assumption is no longer valid. To improve knowledge on turbocharger

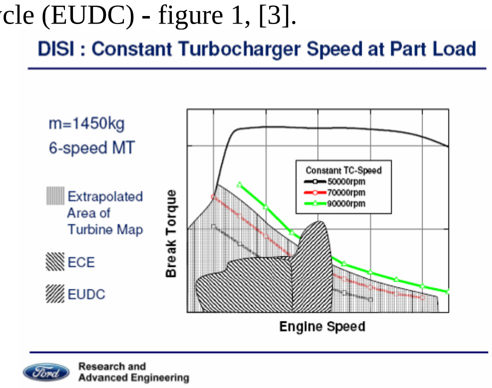

Fig. 1 : Turbocharger's functioning area on NEDC

performances, it is necessary to assess mechanical power losses [4]. Different kinds of experiments are conducted in research center. Honeywell Turbocharger Technologies [5] use a turbine carefully insulated (adiabatic conditions) and experiments are performed at 100°C. The compressor blades are removed, so compressor power can be neglected. The laboratory at Stuttgart University [6] has designed a specific test bench for the direct study of bearing losses. Turbocharger wheels are removed and axial loads are generated by an electromagnetic device. Torque is measured with a high degree of precision for a rotary strain gauge torque sensor. Results are expressed in terms of percentage of power and torque; while this does not provide access to the real value, the results of power or torque evolution are in agreement with our experiments.
An interesting method based on turbocharger inertia and measurements of speed deceleration has been proposed by the University of Hanover [7]. Unfortunately, it seems that the friction power determined by this method is overestimated.
Thanks to the torquemeter fitted on our test bench, friction power was measured according to two methods:
- Adiabatic measurements
- Direct measurements

TEST BENCH DESCRIPTION

<u>Torquemeter</u>
The torquemeter is set between the turbine and the compressor. The turbine is thus separated from the compressor and could be considered as a drive. The torquemeter allows a rotational speed of 120,000 rpm. Maximum torque is of 0.4 N.m and accuracy is ± 0.0016 m.
<u>Lubrication system</u>
Both central housing (compressor and turbine sides) are fed with SAE standard 15W40 oil. The oil inlet temperature is regulated thanks to an electrical heater and an oil/water cooler. The oil inlet pressure can be adjusted separately on the compressor and turbine side. Oil flow is measured by two Coriolis mass flowmeters.

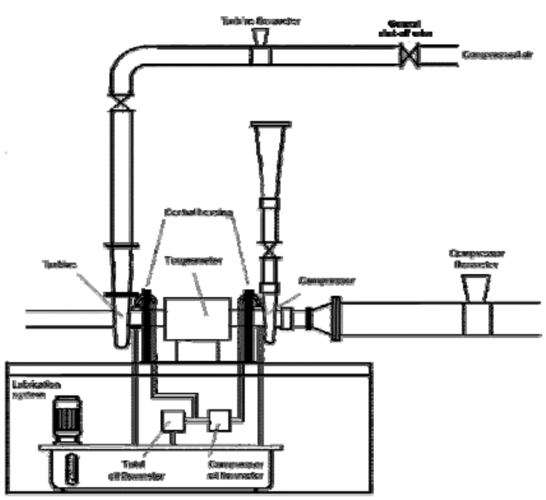
Fig. 2 : Sketch of the test bench

<u>Turbocharger measurements</u>

Air flow is measured with mass flowmeter at turbine and compressor inlet
A valve with an actuator before the turbine allows the flow rate entering the turbine to be adjusted, and thus allows the rotational speed to be set. Another controlled valve allows the air flow of the compressor to be adjusted and thus allows the pressure ratio to be set.
Temperatures of air flowing in and out the compressor are measured with PT100 sensors.
A small PT100 sensor has also been inserted inside the bronze part of the journal bearing. This sensor will provide information on the link between friction losses and thermal effects.
A sketch of the test bench is provided figure 2.

ORIGIN OF THE MECHANICAL LOSSES

The bearing system, presented in figure 3, is constituted of two semi floating journal bearings and a double effect thrust bearing.

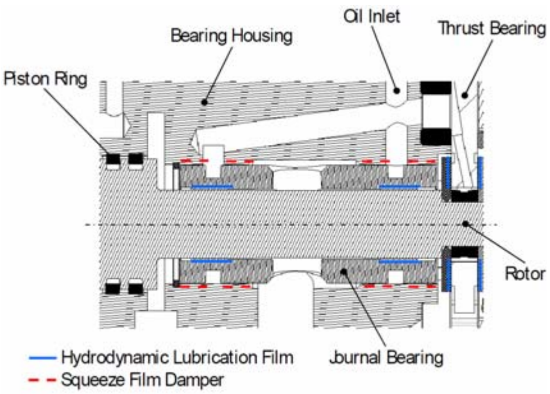

Fig. 3 : Sketch of bearing [6]

The journal bearings support the mass and dynamic efforts of the rotating assembly. The thrust bearing equilibrates the axial load on compressor side and turbine side.
Piston rings are used to prevent oil leakage.
The free floating journal bearing is constituted of an inner hydrodynamic oil film, where pressure is created to support the load, and an outer squeeze film, which dampens vibrations and ensure coolness of the centre housing.
The friction power of the bearings is mainly due to oil shearing effect. The principle variables of this effect are rotational speed and oil viscosity, which is highly linked with oil temperature

ADIABATIC EXPERIMENTS

Experiments have been carried out on the compressor which is fully insulated from the outside as showed in figure 4.
Moreover, in order to reduce heat exchange as much as possible, the compressor is fed with air and oil at the same temperature. In these conditions heat exchange is drastically reduced and compression can be considered adiabatic.
According to the first law of thermodynamic, for an ideal gas, the power received by the air flow is:

$$P_{air} = q_m c_p \left(T_{2_t} - T_{1_t} \right) \quad (1)$$

The torque and rotational speed of the shaft are given by the torquemeter, which provides the power given to the shaft; in its turn, it is the sum of the mechanical power bearing losses and the power given to the compressor wheel:

$$P_{shaft} = P_{losses} + P_{air} \quad (2)$$

So: $P_{losses} = C_{shaft}\omega_{shaft} - q_m c_p \left(T_{2_t} - T_{1_t} \right)$ (3)

In addition, power transmitted to lubricating oil may also be calculated:

$$P_{oil} = q_{oil} c_{oil} \left(T_{oil_o} - T_{oil_i} \right) \quad (4)$$

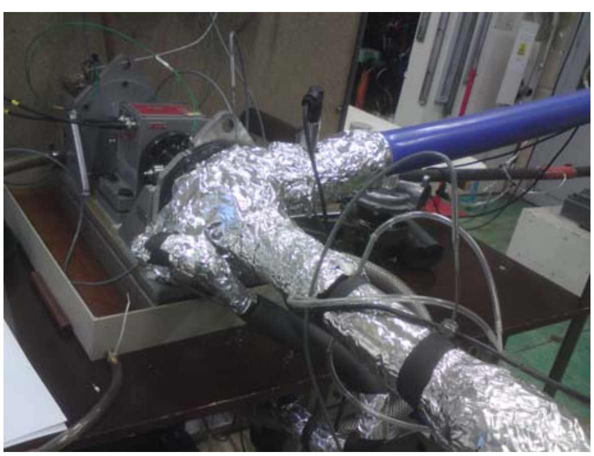

Fig. 4 : Compressor insulation

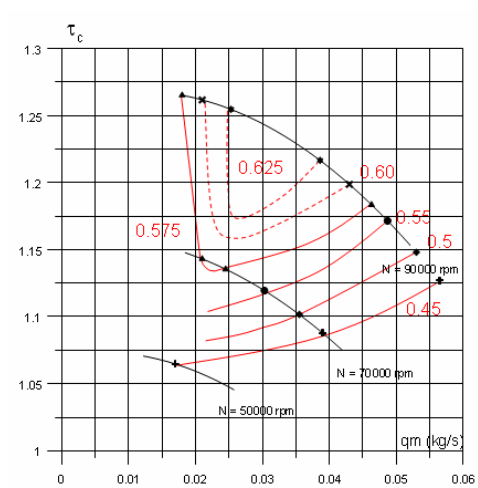

Fig. 5 : Compressor map

Experiments, in adiabatic condition permit to draw a compressor map (figure 5) at low speed with its isentropic efficiency. Values which cannot be determined with usual test bench procedure.

RESULTS ON FRICTION LOSSES

Figure 6 presents power measurements for constant lubricating oil temperature and calculation versus rotational speed

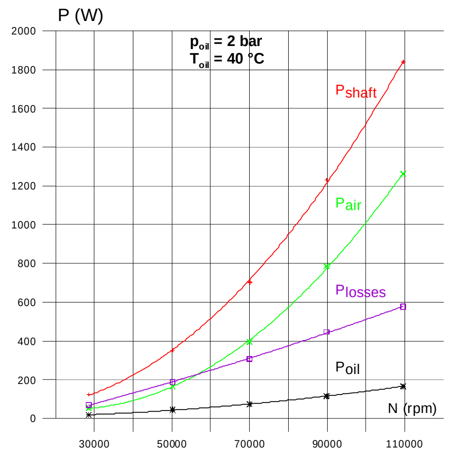

Fig. 6 : Power versus rotational speed Sketch of bearing

The power losses are the subtraction of shaft power minus the power of air (equation 2). One can notice that curve P_{air} cross curve P_{losses}. At this point mechanical efficiency is equal to 0.5. At lower speed more power is given to overcome friction than to compress air.
The oil power is the power measured between inlet and outlet of oil flow (equation 4).
Theoretically, the friction power must be equal to the power transmitted to lubricating oil. Some discrepancies appear. This difference seems to be due to the difficulty of measuring the real oil outlet temperature.

Influence of oil inlet temperature

In figure 7, friction power losses are plotted versus rotation speed for oil inlet temperature 40°C and 60°C and oil pressure of 2 bars. Obviously, losses with oil at 40°C are more important than losses at 60°C due to the change of viscosity:

$\mu_{40°C} = 0.1$ Pa.s $\quad \mu_{60°C} = 0.04$ Pa.s

According to Newton's Law ($P_{friction} = k\mu N^2$) this difference must be higher ($\mu_{40°C}/\mu_{60°C} = 2.5$) with respect to oil inlet viscosity (instead of ≅ 1.5).

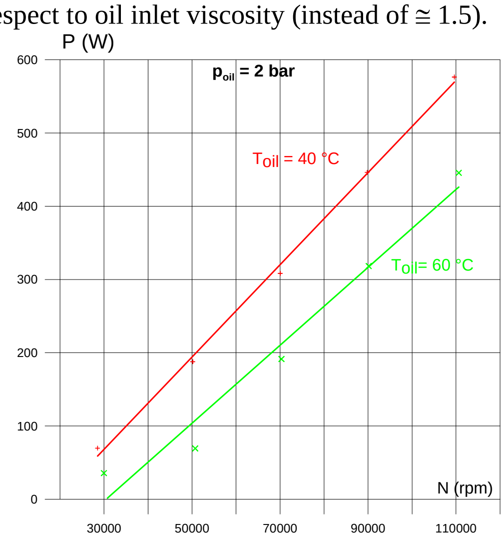

Fig. 7 : Power losses versus rotational speed Toil = 40°C and 60°C

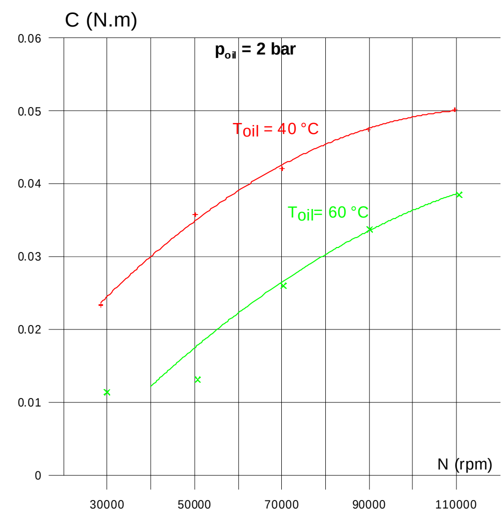

Fig. 8 : Torque friction versus rotational speed Toil = 40°C and 60°C

In figure 8, results are presented in term of friction torque. According to Newton's Law ($C_{friction} = k\mu N$), friction torque should be proportional to speed rotation; this not the case. Theses results are in agreement with the experiments of the Stuttgart laboratory [6].
The reason is that when the speed rotation increases, friction increases and oil temperature inside the bearing increases, too. So, viscosity decreases and friction losses decreases, as well.

Influence of oil inlet pressure

In figure 9, friction power versus rotational speed for inlet oil pressures of 2, 3 and 4 bars is drawn, for constant oil inlet temperature T_{oil} = 40°C. Results in terms of torque friction are presented in figure 10. Friction power or torque increases with lubricating oil inlet pressure. This behavior was observed in 2005, [1], but not explained.

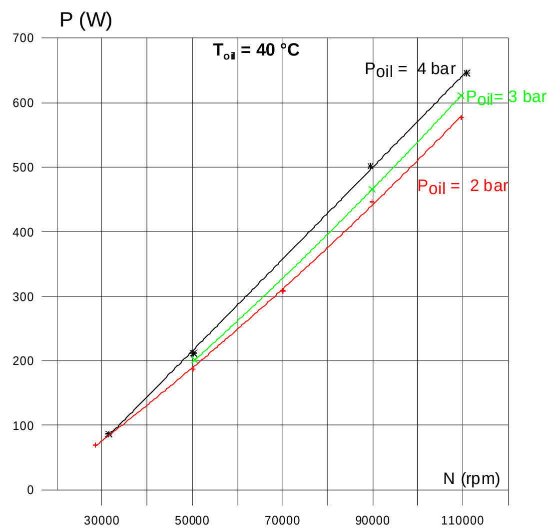

Fig. 9 : Power friction versus rotational speed Poil = 2, 3, 4 bars

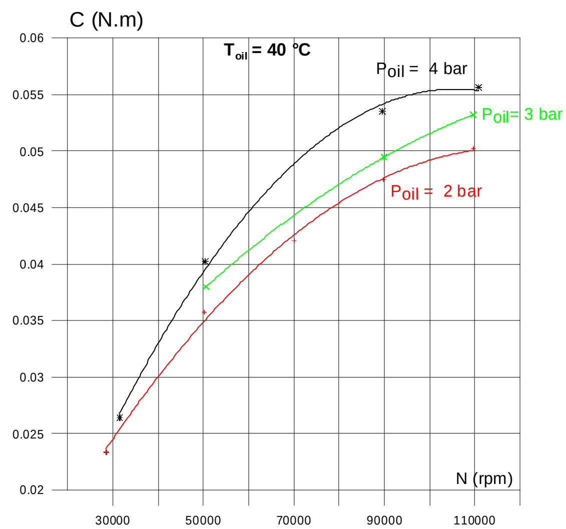

Fig. 10 : Torque friction versus rotational speed Poil = 2, 3, 4 bars

This effect, rather surprising, has been explained by computation results. Oil flow outer the bearing is about 100 times greater than the inner flow (inner gap 15 microns - outer gap 75 microns). The outer flow has a cooling effect. As pressure increases, oil flow increases and cooling effect is more important. So, temperature inside the bearing decreases, viscosity increases then friction losses increase.

Mechanical efficiency

In figure 11 is represented power given to the air flow and power given to the shaft versus air flow, for p_{oil} = 2 bar and T_{oil} = 40 °C.

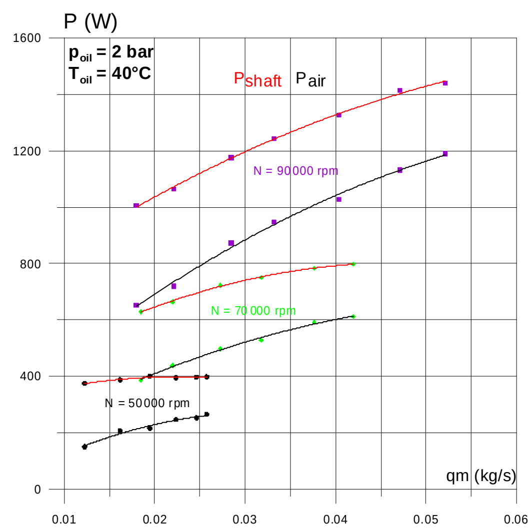

Fig. 11 : Power versus air flow

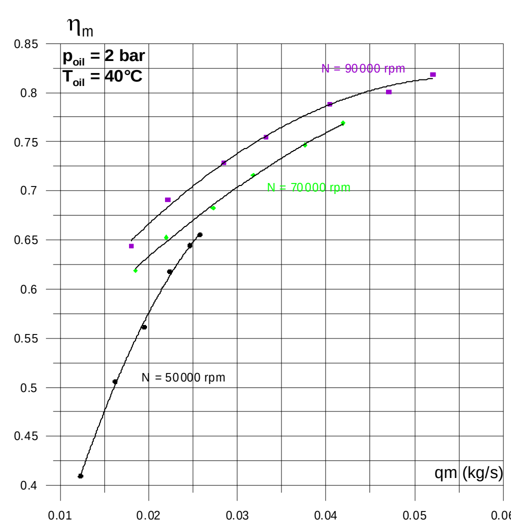

Fig. 12 : Mechanical efficiency versus air flow

It is observed that these curves are not parallel, and that the difference increases as air flow decreases. The reason is when air flow decreases pressure ratio increases and axial load increases, too, leading to more friction.
From theses measurements, mechanical efficiency is calculated and plotted versus air flow for the different speeds, in figure 12.

$$\eta_m = \frac{P_{air}}{P_{shaft}} = 1 - \frac{P_{losses}}{P_{shaft}} \tag{5}$$

Mechanical efficiency decreases as the air flow decreases. This is logical in regard to equation (5), assuming constant power losses (when P_{shaft} decreases, η_m decreases).
Power losses also increase when flow decreases, so this effect is here magnified.

DIRECT MEASUREMENTS OF FRICTION LOSSES

First goal of previous research was to set the compressor characteristics map at low speed. Experiments have shown that the procedure used, permit also to evaluate mechanical power losses. Opportunity of direct measurements of theses losses may be done by removing the compressor wheel. In figure 13, wheel has been replaced by a spacer.
Former tests done without compressor wheel revealed it was impossible to set a steady shaft speed, which avoid accurate measurements. On the new test bench, this problem is no more encountered, probably due to a good regulation of lubricating oil pressure and temperature. Tests results are presented, in figure 14.

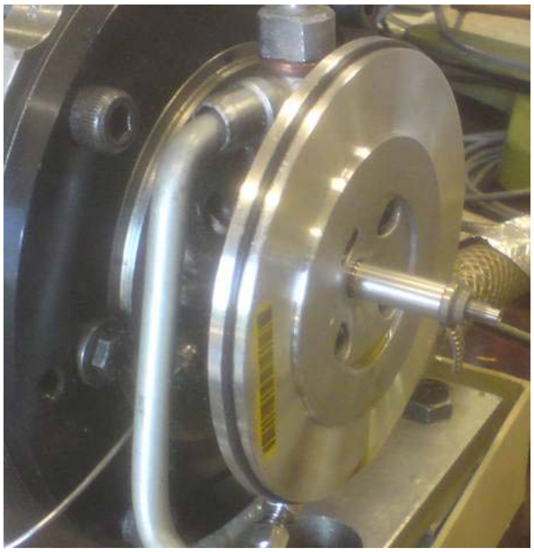

Fig. 13 : Direct measurements of friction losses

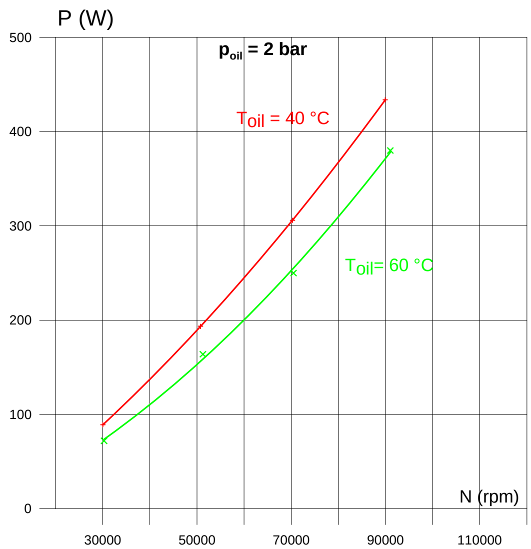

Fig. 14 : Power losses

EFFECT OF AXIAL LOADS ON FRICTION LOSSES

In real functioning conditions an axial effort is applied to the thrust bearing in compressor direction or turbine direction depending on functioning point. A special device, figure 15 has been drawn in order to generate an axial load available in both directions (reversible system). It is constituted of a magnet and a washer link to the shaft. The distance between theses two parts is adjustable and then axial loads, up to 50 N, can be generated.

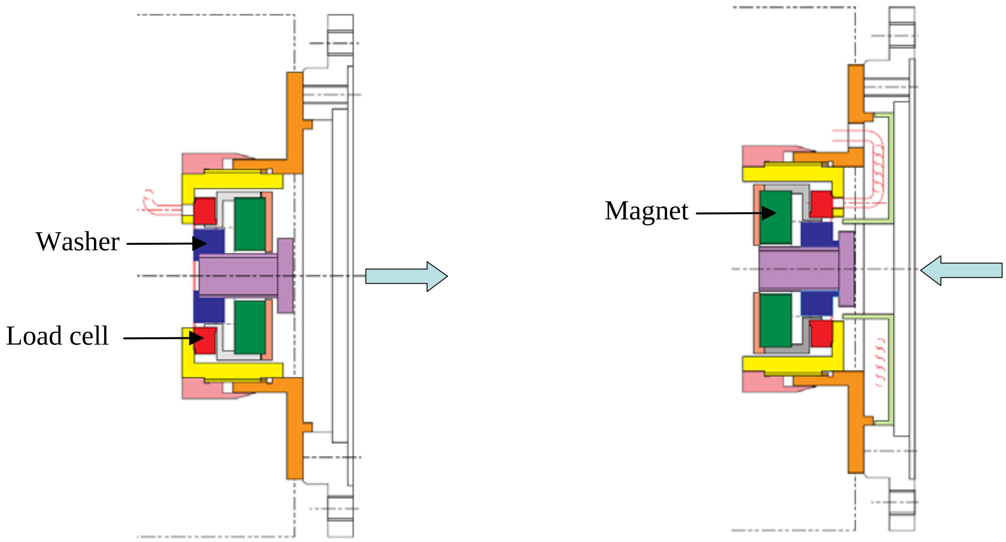

Fig. 15 : Sketch of special device for generating an axial loads

A picture of the system is given figure 16. Variation of power losses versus axial loads is given in figure 17. These variations seem in accordance with computation results and experiments done by other laboratories. However, a discrepancy appears between theses results and those done in adiabatic conditions. An explanation is rewarded.

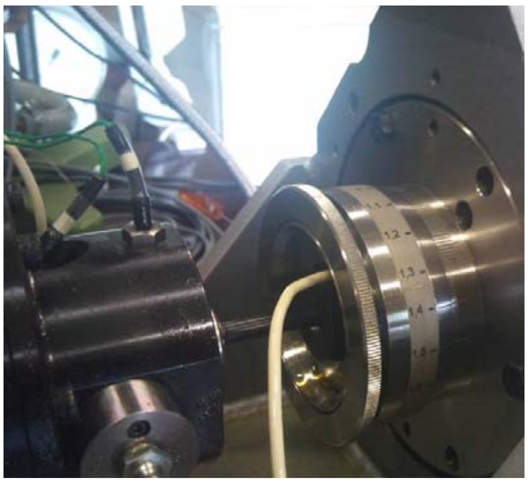

Fig. 16 : Special device for generating an axial loads mounted on test bench

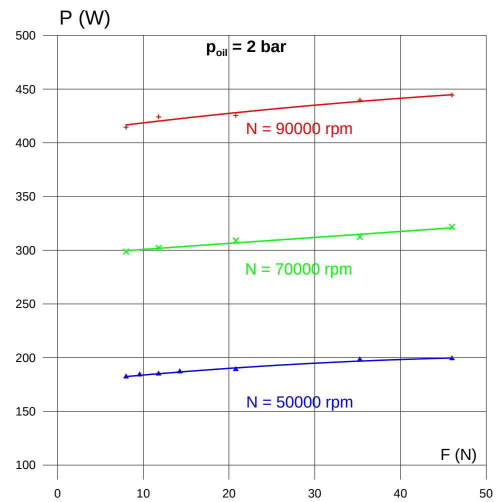

Fig. 17 : Power losses versus axial load

MEASUREMENT OF AXIAL LOADS ON FRICTION LOSSES

Previous system allows applying an axial load on thrust bearing but also it is possible to know the value of the load generated.

Last step of this study is to know the value of these loads in real functioning condition.

So, the thrust bearing part has been machined. The six pads thrust bearing (figure 18) is maintained by three brackets and equipped with strain gages in order to measure axial load and equipped with strain gage.

Figure 19 is a snapshot of the thrust bearing after some tests done on an engine.

Fig. 18 : Machined thrust bearing.

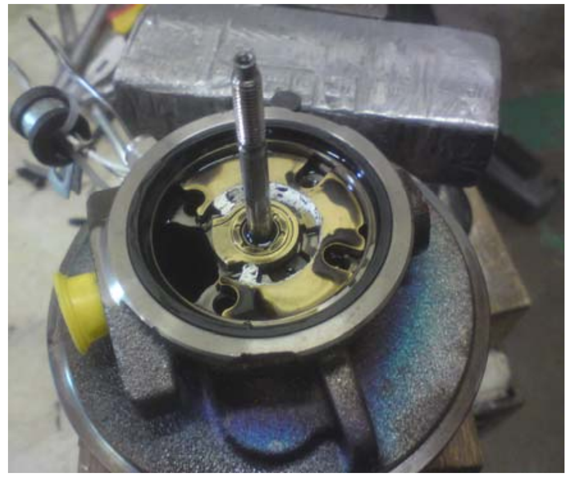

Fig. 19 : Thrust bearing assembly

CONCLUSION

The test bench developed at the CNAM laboratory equipped with a torquemeter allows measurements of turbocharger's performances in the low speeds area and calculation of mechanical efficiency.
Two ways of measurements of power losses have been used:
- an indirect measurement with a functioning of the compressor in adiabatic conditions,
- a direct measurements compressor, the wheel being removed.

The effect of lubricating oil pressure and temperature on friction losses has been stated.

A special device has been fitted on the test bench to apply an adjustable axial load on turbocharger shaft. This opportunity will be helpful to separate power losses due to journal bearing from the ones due to thrust bearing. In parallel, some CFD computations are done to assess theses friction losses [8].
A turbocharger has been equipped with strain gauges in order to measure the axial load in real functioning conditions. Tests are in progress.

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