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## PERFORMANCES OF TURBOCHARGER AT LOW SPEED

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

In order to know the power demand of the compressor with accuracy, a special torquemeter has been fitted in our test bench.

Many experiments have been done with different lubricating oil temperature. Results show, the real power requirement and that the compressor can't always be considered adiabatic. That the reason why the usual method is not applicable at low turbocharger speed.

## INTRODUCTION

Usually, the power of a compressor is deduced from the measurements of the inlet and outlet temperatures and the gas flow rate.

$$P = q_m C_p (T_{2t} - T_{1t}) \quad (1)$$

At low turbocharger speed the relative error done on temperature measurements becomes important due to the small difference of these two temperatures. Furthermore, heat transfers can't be neglected and equation (1) which assumed adiabaticity of the compressor can't be used. Hence, calculation of power by this method leads to erroneous results especially at very low engine speed.

Running at low speed is generally not an optimum range for the turbocharger. Unfortunately in automotive area this behavior is often encountered. For example when a diesel automotive engine is running at 1500 rpm at low torque, turbocharger speed is about 30000 rpm. So, improvement of engine necessitate a better knowledge of turbocharger characteristics at low speed.

In connection with a French automotive manufacturer it has been decided to fit our turbocharger test bench with a torquemeter in order to explore the functioning of a turbocharger between 30000 et 120000 rpm.

## 1. TURBOCHARGER TEST RIG

### 1.1 Test bench

Experiments have been conducted on our usual turbocharger test rig, which provides the experimental compressor and turbine performances. Some modifications have been done in order to receive the compressor and his torquemeter and a new lubricating unit. Scheme of the test rig is given figure 1.

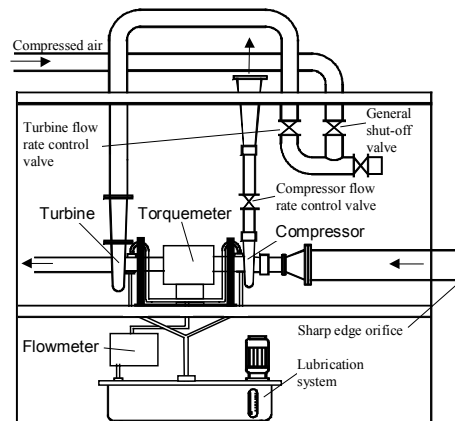


Fig.1 - Scheme of test rig

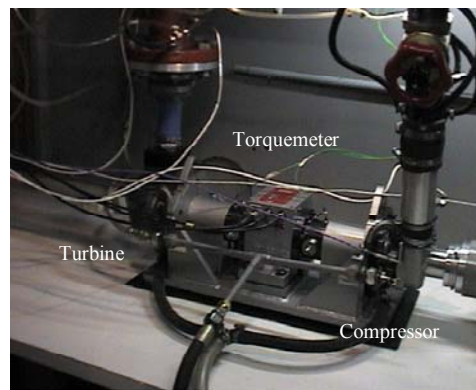


Fig. 2 – Test bench

The torquemeter, figure 2, is located between the turbine and the compressor. On the left side the turbine is fixed on the center housing and on the right side the compressor with an another center housing. Connecting of compressor and turbine are made with thin quill shafts. In these experiments turbine is only use to run the compressor and to adjust his rotational speed.

The turbine is fed by dry compressed air under steady flow. The air source is a tank of 700 m<sup>3</sup> under 25.10<sup>5</sup> Pa. The turbine flow rate is controlled by a valve. A second valve, downstream, allows to modify the compressor's operating conditions. Using these valves, the compressor performance graph may be described.

Both centers housing are fed by the lubricating unit with SAE 5-30W oil. Oil temperature and pressure are adjustable respectively from 20 to 120 °C and from 0.5 to 4 bar.

### 1.2 Measurements

All sensor signals are converted to 0-5 voltage and send to a data acquisition card fitted in a Personal Computer. Home made software has been developed in order to convert electrical data into physical value. Acquired point can be observed during acquisition procedure. For each signal 100 measurements are done at a scanning speed of 10 Hz. Only the mean values are registered.

Pressure are evaluated by different strain gauge transducers, adapted to measurements scale, and temperature by platinum resistance thermometers.

The mass air flow rate is determined by a sharp edge orifice.

### Performances of turbocharger at low speed

A mass flowmeter based on the Coriolis principle gives simultaneous measurements of oil flow rate and density. Oil feeding two centers housing, it is assumed that each center housing receive half of the total oil flow.

A special torquemeter has been set up for these experiments. Torque is measured from shaft twist which is deduced from phase difference between two teeth wheels located at each end of the shaft. The same device is able to give the rotational speed and hence the power. Main characteristics of the torquemeter are :

- Maximum speed: 120000 rpm
- Shaft rating torque: 0.4 N.m
- Shaft diameter: 2.46 mm
- Maximum power: 5kW
- Accuracy  $\pm 0.0016$  N.m
- Grease lubricating ceramic ball bearings

## 2. RESULTS

Experiments have been done with constant oil temperature of 60°C and the with variable oil temperature. Oil pressure has been kept constant and equal to 2 bar. Main results are given hereafter.

### 2.1 Constant oil temperature

In figure 3, compression ratio  $\tau_c = \frac{p_{2t}}{p_{1t}}$  is represented versus mass air flow rate  $q_m$

for different engine speed.

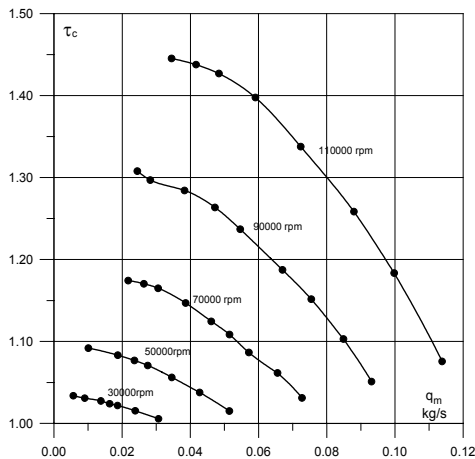


Fig. 3 – Compression ratio vs air flow rate.

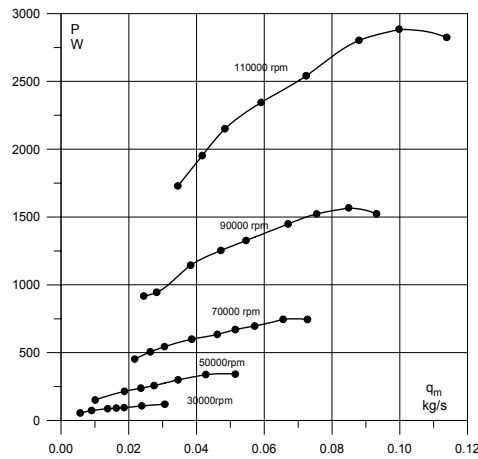


Fig. 4 – Power vs air flow rate.

In figure 4, the power P is the power, receive by the air flow rate, calculated by equation (1).

Isentropic efficiency is represented in figure 5. Although the evolution of this efficiency seems realistic, values can be discussed due to the way of calculation procedure and measurements. This subject will be dealt hereafter.

The mechanical power given to the compressor  $P_m$ , deduced from torquemeter's measurements, is represented in figure 6.

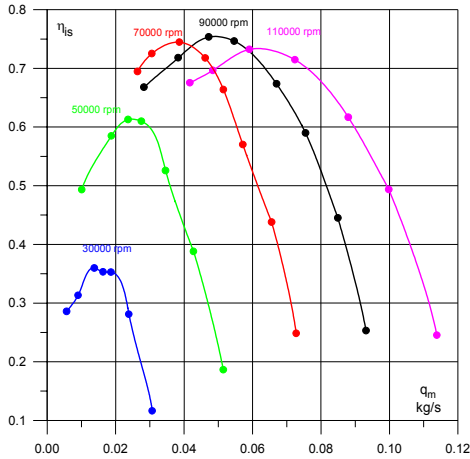


Fig. 5 – Isentropic efficiency vs air flow rate.

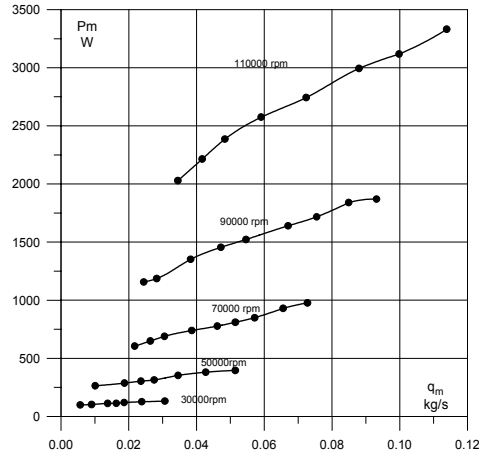


Fig. 6 – Mechanical power vs air flow rate.

The mechanical efficiency of the turbocharger, figure 7, can be expressed as the ratio of the power given to the air flow rate  $P$  to the mechanical power  $P_m$ .

At low speed 30000 and 50000 rpm, it is reached high levels of mechanical efficiency, up to 0.9, which doesn't seem realistic. Explanation of this phenomena will be clearly demonstrate by test at 30000 rpm with different oil temperature.

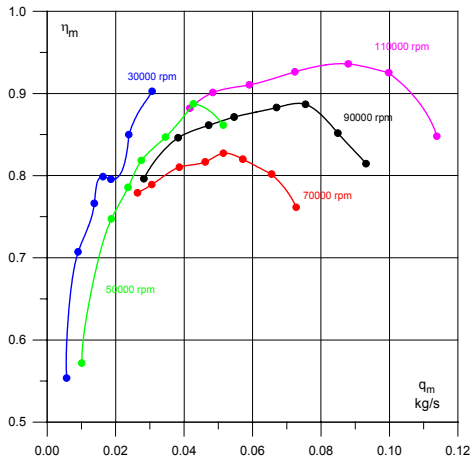


Fig. 7 – Mechanical efficiency vs air flow rate.

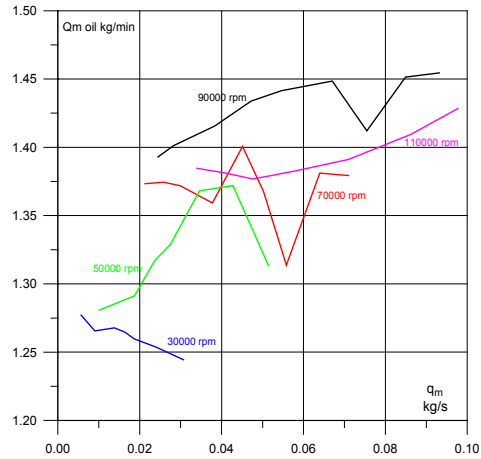


Fig. 8 – Oil flow rate vs air flow rate.

### Performances of turbocharger at low speed

The last n°8 figure, concerning these tests at constant oil temperature, relates to oil flow rate. Oil flow rate is the total oil flow rate in the two center housing.

Variation of mass oil flow rate is weak, varying from about 1.25 to 1.45, that is to say about 14 % for a turbocharger speed variation of 73%.

#### 2.2 Variable oil temperature

More interesting results are at 30000 rpm, because performances of turbocharger are more sensitive to the effect of oil temperature. So, it will be first focus on these tests and present results at 110000 rpm.

##### 2.2.1 Rotational speed 30000 rpm

In figure 9, it appears that the pressure ratio, is quasi-independent of the oil temperature.

In contrary there is a great variation of calculated power P. About 200 % for the extreme temperatures, figure 10.

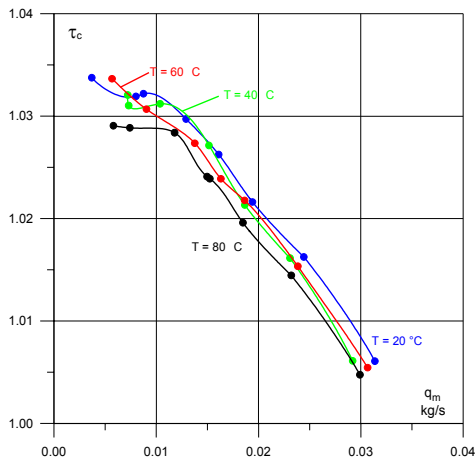


Fig. 9 – Compression ratio vs air flow rate.

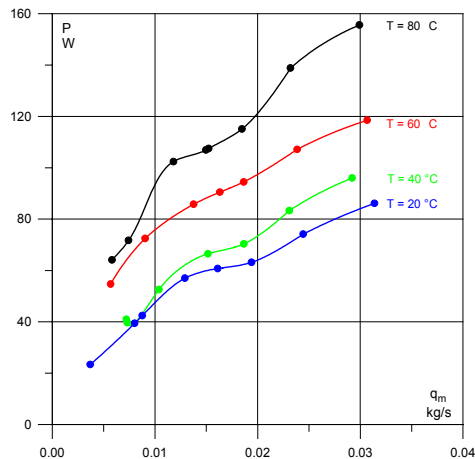


Fig. 10 – Power vs air flow rate.

Observations of the temperatures data directly measured at compressor inlet and outlet, show that inlet temperatures are varying from 20.5 to 22 °C. These variations are probably due to the change of ambient conditions.

Figure 11, the difference between outlet and inlet temperature is about 4 °C, for oil at 20°C, which is rather low and leads to inaccurate calculations. It can also be noticed that this difference is increasing with oil temperature. Theoretically, there is only one functioning point for a given rotational speed, gas flow rate and compression ratio. That means a unique power and then a unique temperature increase. The only explanation is that air flow rate receive heat from the lubricating circuit by conduction.

As an example at a mass gas flow rate of 0.03 kg/s the temperature difference for oil at 20°C and oil at 80°C is 2.5°C which corresponds to a power of 75 W. So, it appears that for an 80 °C oil temperature, as much energy is given to the flow by heat transfer than by compression work.

Due to this heat exchange, isentropic efficiency, figure 12, is decreasing when the oil temperature increase. This variation can be easily explain in a T,S diagram.

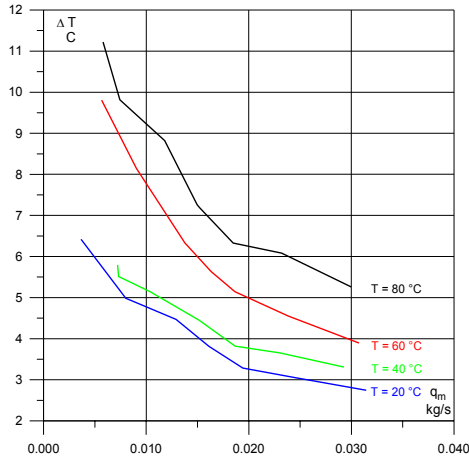


Fig. 11 – Difference of inlet and outlet temperature vs air flow rate.

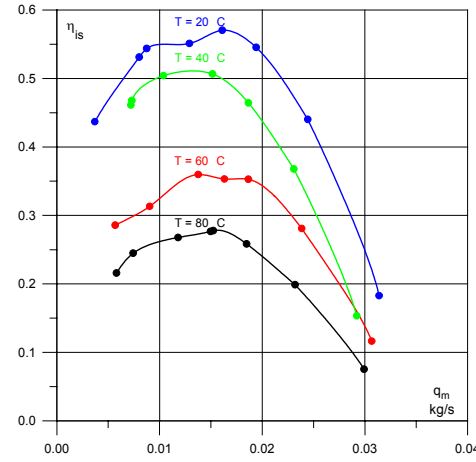


Fig. 12 – Isentropic efficiency vs air flow rate.

In spite of the fact that rotational speed is constant, variation of mechanical power, figure 13, is important. This effect is due to viscosity change which vary from  $15 \cdot 10^{-6} \text{ m}^2/\text{s}$  between 20 et 80 °C.

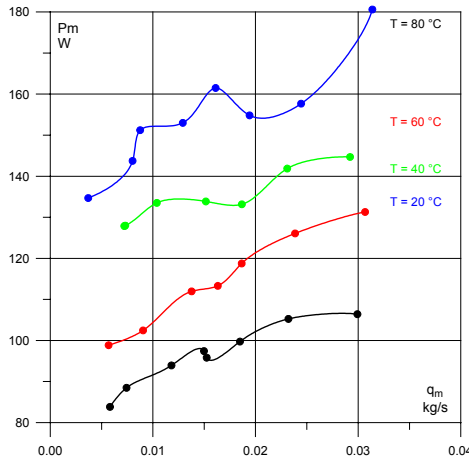


Fig. 13 – Mechanical power vs air flow rate.

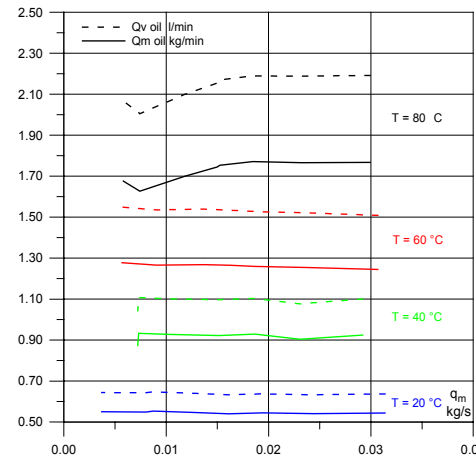


Fig. 14 – Oil flow rate vs air flow rate.

Variation of viscosity has also a more important effect on oil flow rate, figure 14, which changes of about 225 % for extreme oil temperature.

Calculation of mechanical efficiency is not consistent here, due to the impossibility of calculation of power given to the air flow. It is found mechanical efficiency greater than 1 for 80 °C oil temperature.

## Performances of turbocharger at low speed

### 2.2.2 Rotational speed 110000 rpm

Like for the previous test, results show that compression ratio is not affected by the oil temperature.

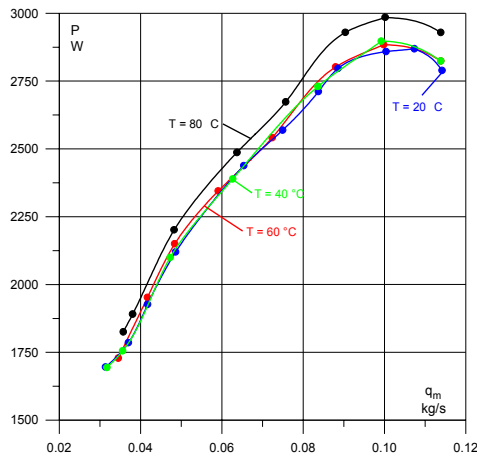


Fig. 15 – Power vs air flow rate.

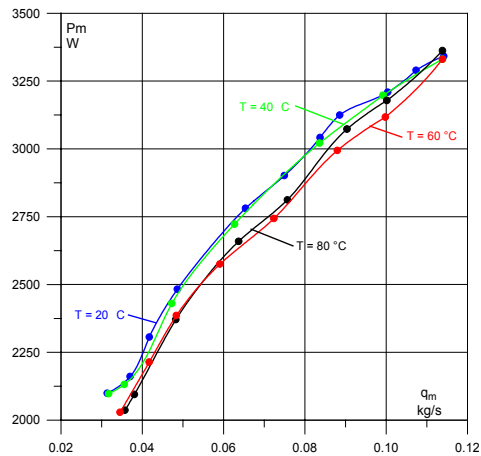


Fig. 16 – Mechanical power vs air flow rate.

Outlet compressor temperature is greater, typically varying from 45 to 75 °C, so heat exchanges are less important. Compression work is also more important, so heat exchanges have not a significant influence on power calculated  $P$  represented in figure 15. So, it results that isentropic efficiency is practically independent of the oil temperature.

A thin difference, figure 16, exist for the mechanical power which tends to decrease when oil temperature increase.

This result can be more clearly observed in figure 17 where mechanical efficiency is represented.

Increase of oil temperature, figure 18, leads to an increase of oil flow rate but not so important as for tests at 30000 rpm. Further experiments have demonstrated that at 80 °C, the oil flow rate is quasi similar at 30000 and 110000 rpm which underline the effect of oil viscosity.

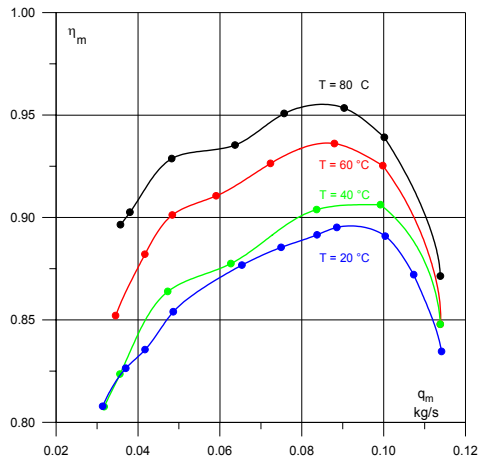


Fig. 17 – Mechanical efficiency vs air flow rate.

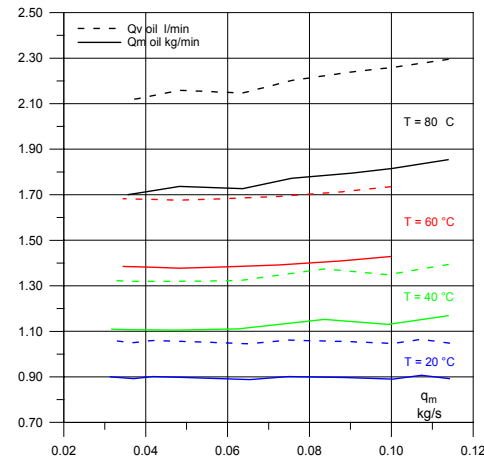


Fig. 18 – Oil flow rate vs air flow rate.

## CONCLUSION

These first experiments have emphasized the difficulties of estimating the efficiency of a turbocharger at low speed. It has been demonstrated that the compressor can't be considered adiabatic and that heat exchange between lubricating oil and air flow can't be neglected. That's why abnormal isentropic efficiency are generally calculated in this area.

Though the power given to the compressor shaft is directly measured, the mechanical efficiency can be appreciate for the above reason.

For a better knowledge of compressor efficiency, some other experiments are foreseen based on heat exchange reduction :

- tests with oil temperature close to compressor temperature outlet
- insulation of compressor scroll from center housing.

Or on mechanical losses reduction :

- replacing of journal bearings and thrust bearing by ball bearings.

Nevertheless, if efficiency of the compressor is well known, which can improve engine calculation and have a better comprehension of engine functioning, it mustn't be forget that it is also necessary to know the mechanical power demand of the compressor.

It has been seen that this power demand vary deeply with oil temperature, that is to say with oil viscosity, but some extra tests have shown that this power is also linked to the oil pressure.

For example running on a mean point at 30000 rpm with oil temperature of 20 °C and oil pressure of 3.5 bar the speed increase up to 50000 rpm by only setting oil at a pressure of 1.0 bar.

This important variation will be one of our future investigation, which can lead to a better knowledge of mechanical losses.