

Importance of Mechanical Losses Modeling in the Performance Prediction of Radial Turbochargers under Pulsating Flow Conditions

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ABSTRACT

This work presents a study to characterize and quantify the mechanical losses in small automotive turbocharging systems. An experimental methodology to obtain the losses in the power transmission between the turbine and the compressor is presented. The experimental methodology is used during a measurement campaign of three different automotive turbochargers for petrol and diesel engines with displacements ranging from 1.2 l to 2.0 l and the results are presented.

With this experimental data, a fast computational model is fitted and used to predict the behaviour of mechanical losses during stationary and pulsating flow conditions, showing good agreement with the experimental results. During pulsating flow conditions, the delay between compressor and turbine makes the mechanical efficiency fluctuate. These fluctuations are shown to be critical in order to predict the turbocharger behaviour.

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INTRODUCTION

Nowadays internal combustion engines, ICE, are facing two main problems the reduction on both pollutant emissions and fuel consumption without losing their performance in order to fulfill different economical-areas' regulations like the European norm Euro VI [1].

In order to fulfill these norms, engine sizes have been reduced in recent years (downsizing) with a very high compression of the inlet air (supercharging), what is usually performed by means of a turbocharger. Engines efficiency largely depends on turbochargers efficiency.

The efficiency of a turbocharger depends on the ability to transform the energy of engine exhaust gases to mechanical energy in the turbine and, later on, transform this mechanical energy by compressing the fresh air in the compressor. Unfortunately, some of the mechanical energy is lost in the transmission shaft and the systems attached to it (the lubrication system, bearings, etc.) These energy losses between the turbine and the compressor are commonly called mechanical losses, which are not easy to quantify in this type

of system because of both the turbochargers' small sizes and their high rotating speed [2]

Traditionally, the study of the turbochargers has been performed in medium and high loads, i.e. compressor pressure ratios above 1.5, where mechanical losses are much smaller than the rest of the energy flow in a turbocharger; but the importance of the urban cycles in the homologation of engines (where loads are very low) has led to the necessity of improving turbochargers performance in these particular operating points. In this low load cases, heat and friction losses can be very important (in some cases are even higher than the compressor mechanical power) [3], so it is very important to quantify them as properly as possible.

Theoretical studies using CFD determined the need of considering the change of the lubricating oil temperature when passing through the bearings, in view of the fact that better results were obtained than in the case of considering isothermal condition [4]. Moreover, Lihua et al. [5], studying the design of hydrostatic spindle, arrived to the conclusion of considering not only the temperature drop but also the pressure distribution of the lubricating oil across the journal



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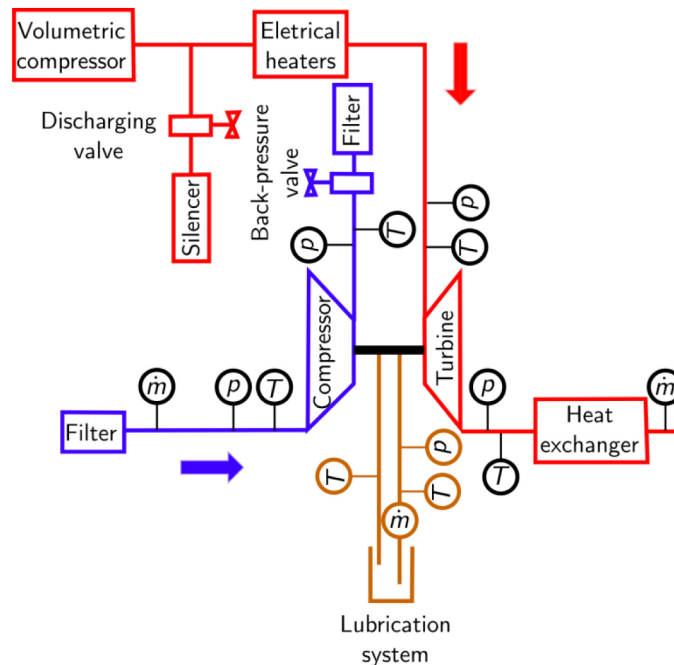


Figure 1. Turbocharger gas stand.

in order to obtain more accurate results. Chen [6] studied the rotordynamic characteristics of large turbochargers, but no indications for small turbochargers are mentioned.

Experimental studies about mechanical losses on turbochargers are usually performed in cold tests in which the temperatures of the different fluids (oil, refrigerant and compressor and turbine air) are kept almost constant [7]. In this way, heat transfer effects are usually neglected and mechanical power can be determined by means of oil temperature drop. Another possibility is to use a torquemeter [8], but this approach implies the modification of the turbocharger. Also Schorn et al. [9, 10] published friction measurement results, even their approach requires a special test bench and preparation of the turbocharger, it allows separating all individual effects like oil temperature/pressure and thrust load on friction power.

Payri et al. [11] presents an empirical model in which the dependency of mechanical losses are fitted to dimensionless numbers: Reynolds, Prandtl and a dimensionless pressure difference, which expression was based on the work of Hu et al. [12], who determined that the axial thrust of the turbocharger is mostly determined by the static pressure difference of compressor impeller and the turbine's back plate, which is related to the compressor impeller outlet and turbine inlet pressure. Payri et al [11] made dimensionless the pressure difference with oil pressure later on. Podevin describes cold tests done using a torquemeter, determining that oil pressure is important due to the change on oil mass flow [8]. Deligant et al. used a torquemeter and a magnetic axial load in order to study such a load influence on friction power, being for the tested conditions up to a 12 % of the friction power [13]. Finally, in Deligant's work [13] the

dependency on oil pressure seems to be smaller than on oil temperature that directly affects the oil viscosity.

A physical model developed by the authors has been adjusted with experimental data obtained from a specific turbocharger test bench, in which a cold test campaign similar to that described by Payri et al. [11] was performed, so instantaneous mechanical losses can be computed.

The first part of the paper concerns about the experimental methodology and the main parameters measured to characterize mechanical losses. Then, a physical model is fitted with the experimental data and used to compute instantaneous behaviour of the turbocharger mechanical efficiency. Finally the main conclusions of the work are outlined.

EXPERIMENTAL METHODOLOGY AND RESULTS

An experimental campaign to measure several turbocharger models has been done. The experiments were done in quasi-adiabatic conditions as described in [7, 11], both with stationary and pulsating flow in the turbine. Pulsating flow was generated as described in the work from Serrano et al. [14].

In quasi-adiabatic tests, turbine inlet, compressor outlet and oil inlet temperatures are maintained as close as possible and the turbocharger is thermally insulated from outside. As at low and medium turbocharger speeds the temperature at the inlet of the oil needs to be low to minimize internal heat transfer between the different parts of the turbocharger, its viscosity should be similar of that found during urban driving

Table 1. First turbocharger temperatures.

n	$T_{oil,in}$	$ (T_{oil,in} - T_{tu,in})_{mean} $	$ (T_{oil,in} - T_{co,out})_{mean} $
[krpm]	[°C]	[°C]	[°C]
30	26.7	2.4	1.6
50	32.6	3.3	0.3
70	43.9	2.5	1.1
90	60.0	3.1	1.7
110	76.1	1.3	0.6
130	96.5	1.1	1.5
150	124.3	0.3	3.7
170	144.1	1.5	12.6

cycles when the engine is still cold and the oil temperature is relatively low.

The turbocharger gas stand is represented schematically in [figure 1](#). The whole test rig has thermal insulation to minimize heat exchange with the ambient. The temperature at the inlet and the outlet of the lubrication system is measured with RTDs and the mass flow rate is measured using a Coriolis flow meter. The turbine mass flow rate is measured with a thermal mass flow meter, the temperature of the inlet and outlet flow is measured with an array of 4 type K thermocouples and its pressure with piezoresistive pressure transducers, as described in [15, 16]. The compressor side is measured as the turbine. Also, the wall temperatures of the housing, the turbine and compressor casings are measured with an array of 15 type K thermocouples.

The first tested turbocharger has variable geometry vanes. This device is commonly used in a 2.0 liter diesel engine. It contains a water-cooling system and its bearing is composed by a plain fixed journal bearing and a separated thrust bearing. The second turbocharger is also VGT, it is used in a 1.6 liter diesel engine, it has no water-cooling system and it has a plain floating ring journal bearing and a separated thrust bearing. The third turbocharger has a waste-gate and it is used frequently in 1.2 liter petrol engines. Its housing is water-cooled and its bearing system is formed by a plain floating ring journal bearing and a separated thrust bearing.

The mechanical losses are corrected due to spurious heat flow still remaining after quasi-adiabatic tests. The correction is performed by measuring the temperature at different parts of the turbocharger and using its heat transfer characteristics (conductances) obtained in specific tests as described in [17, 18], so only power losses are taken into account:

$$\dot{W}_{oil} \approx \dot{m}_{oil} \cdot c_{oil} \cdot (T_{oil,out} - T_{oil,in}) - \dot{Q}_{oil} \quad (1)$$

The mechanical efficiency is computed as:

$$\eta_{mech} \approx 1 - \frac{\dot{W}_{oil}}{\dot{W}_{tu}} \quad (2)$$

Power losses, mechanical efficiency and heat transfer to the oil are plotted against turbine power in [figures 2, 3 and 4](#) for all three tested turbochargers.

The mean measured temperatures during the quasi-adiabatic tests are shown in the [tables 1, 2 and 3](#), which show the mean inlet temperature of the oil used during the quasi-adiabatic tests for each tested turbocharger speed, as well as the mean temperature difference between the oil and turbine inlet temperatures and the oil inlet and the compressor outlet temperatures. The differences tend to be small in order to fulfill the quasi-adiabatic requirements. The oil temperature was kept constant for each speed line.

As can be seen in [figures 2, 3, 4](#), the mechanical efficiency becomes appreciably small in the quasi-adiabatic case for low turbine speeds. The turbine power is small, so small friction losses represent an important fraction of it, and also the oil viscosity is higher than in normal operating conditions due to the low oil temperature, increasing the friction. Nevertheless, during urban driving cycles the oil inlet temperature can be even smaller [11]. As seen in [figures 2, 3 and 4](#), the heat transfer to the oil, even in quasi-adiabatic conditions, is an important fraction of the power losses in the lubrication system, and it is expected that its importance grows as the turbine temperature rises [19]. These results show that the heat exchange between the lubrication system and the rest of the turbocharger should not be neglected to compute the real power dissipated in the bearings. [Figure 3](#) shows the biggest effects of heat transfer due to the fact that the second turbocharger is just oil-cooled.