

General Procedure for the Determination of Heat Transfer Properties in Small Automotive Turbochargers

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ABSTRACT

These days many research efforts on internal combustion engines are centred on optimising turbocharger matching and performance on the engine. In the last years a number of studies have pointed out the strong effect on turbocharger behaviour of heat transfer phenomena. The main difficulty for taking into account these phenomena comes from the little information provided by turbocharger manufacturers. In this background, Original Engine Manufacturers (OEM) need general engineering tools able to provide reasonably precise results in predicting the mentioned heat transfer phenomena.

Therefore, the purpose of this work is to provide a procedure, applicable to small automotive turbochargers, able to predict the heat transfer characteristics that can be used in a lumped 1D turbocharger heat transfer model. This model must be suitable to work coupled to whole-engine simulation codes (such as GT-Power or Ricardo WAVE) for being used in global engine models by the OEM. Moreover, the procedure must be capable to predict heat transfer effects using available data as external geometrical parameters of the turbocharger.

To reach these several purposes, a description of the procedure to obtain correlations for heat transfer behaviour to be used in a lumped model is given. The procedure is based on several generalised correlations obtained from the evaluation of heat transfer properties of different turbochargers. The validity of the procedure is confirmed by simulations performed in GT-Power environment compared to experimental results both in a gas stand and in an engine test bench. The results of the validation show an acceptable level of accuracy using the proposed procedure. Furthermore, the advantages of the procedure are shown by comparing its results with the results of standard GT-Power simulations. In these last simulations only turbocharger maps are used without a turbocharger heat transfer model.

Further analysis of these results evidences that the simple procedure of using general correlations for heat transfer properties in a lumped model is accurate enough to predict turbine outlet temperature much better than standard GT-Power model. This parameter is crucial for after treatment modelling and design as well as for two stage turbocharging.

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INTRODUCTION

Traditionally, turbochargers behaviour has been modelled neglecting heat transfer phenomena [1, 2], especially when a whole-engine simulation is carried out [3, 4]. However, in the last decade several authors claimed about the importance of considering heat transfer phenomena to generate improved models for turbocharger and engine modelling. The effect of heat transfer on efficiency measurements in turbocharger hot gas stands has already been considered by several authors [5]. These authors pointed out that calculated efficiency in both compressor and turbine differ from adiabatic efficiency if hot gas is flowing through the turbine. The reason behind this is the difficulty in measuring the fluid temperature difference due to pure aerodynamics phenomena, decoupled from possible

heat fluxes that cannot be neglected for many operative points [6]. This deviation in map measurement affects turbocharger and engine 1D modelling [7] as compressor and turbine efficiencies are needed to compute outlet temperature and power of both machines, properly. Compressor outlet temperature prediction is crucial in setting correct boundary conditions for the intercooler and combustion chamber. Turbine outlet temperature prediction is important in increasing the precision of after treatment modelling and in improving the prediction of the available energy for a second turbine. Furthermore, it is also important for modelling exhaust energy recovery systems. Finally, correct power output is necessary to calculate turbocharger speed which affects to overall turbocharger performance.

In order to take into account the influence of heat transfer phenomena several studies have been performed by different authors. Bohn et al. [8, 9] carried a numerical and experimental study to evaluate the different heat flux paths inside the turbocharger. From this study a correlation was proposed based on turbocharger geometry and dimensionless Reynolds and Nusselt numbers but no predictions of outlet temperatures have been provided. Baines et al. [10], based on an experimental survey, stated that standard convective correlations can be used for internal and external heat transfer and they provided general coefficients for several turbochargers. The conductive heat transfer of these authors is based on thermal conductivity of casing materials using published data. However, neither outlet temperature predictions are shown in the paper nor independent conductive heat fluxes study to assure the validity of thermal conductivity values is given. Romagnoli et al. [11] developed a complex model for evaluating heat fluxes in a turbocharger working on engine but only one turbocharger is checked against the model. Cormerais et al. [12] proposed an equivalent resistance model and gave a correlation to account convective heat fluxes for a tested turbocharger but the generality of such correlation was not proved.

The interest in the subject has not decreased and several papers have appeared in the last year focusing on different aspects of the problem [13]. Sirakov and Casey [14] have thoroughly studied the effect of heat transfer on compressor and turbine efficiencies and they have proposed a method to obtain adiabatic maps of the turbocharger from hot measurements. However, the improvement in turbine and compressor outlet temperature prediction using this method has not been shown. Mohd et al. [15] have performed 3D-CFD simulations on a turbine volute taking into account heat transfer phenomena and analysing temperature distribution. However, such approach cannot be adapted to whole engine simulations due to computational costs, so it must be coupled to 1D simulation codes [16]. Other studies [17, 18] have been centred in considering turbocharger heat transfer in a whole 1D engine model but a lot of tests on engine bench are needed for this procedure.

In the present paper a generic procedure to account heat transfer phenomena in small automotive turbochargers is provided. It is based on an experimental study of four different turbochargers and a lumped capacitance model to account for heat fluxes [19, 20]. The main objective of the paper is to show the generality of the procedure, i.e. it can be applied to any small turbocharger giving good results. Secondary objective is to prove the usefulness of the procedure in predicting different turbocharger variables like compressor and turbine outlet temperatures. The predictions were made using a 1D gas dynamic code coupled with the heat transfer lumped model.

The paper is structured in four main sections. A brief description of the experimental methodology and of the lumped capacitance model will be the first part. In the second and third parts the correlations for conductive conductances and for

convective heat fluxes will be addressed. Later, a comparison between simulations performed by a 1D gas dynamic code using the correlations and experimental results will be provided. Finally, the discussion of the results and the conclusions will be exposed.

LUMPED CAPACITANCE HEAT TRANSFER MODEL

Heat transfer in a turbocharger is a complex problem with several heat paths and high temperature gradients due to the small size of the turbomachine and the presence of fluids at high and low temperatures. In order to account for all the possible heat fluxes of an operating turbocharger in the simplest way the lumped capacitance model of Figure 1 [21] is proposed.

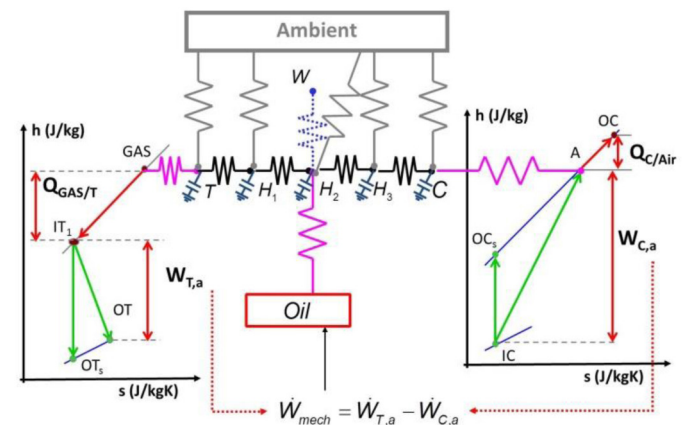


Figure 1. Lumped capacitance heat transfer model for turbochargers [21]

A lumped approach reduces the turbocharger thermal system to a discrete number of nodes assuming constant temperature in each of them. This geometry simplification of the turbocharger in discrete nodes implies the assumption that radial temperature distribution in a cross sectional area is negligible compared to the axial one. Several turbocharger heat transfer studies [8, 22] confirm this assumption. Therefore, considering the previous assumptions, three-dimensional turbochargers heat transfer problem can be simplified to a discrete approach.

Five discrete nodes have been considered for the metal, as can be observed in Figure 1: one for turbine casing (T), one for compressor casing (C) and three for the housing (H1 close to turbine, H2 in central housing and H3 close to compressor). In this way the most critical region (housing), where temperature gradients can be high is composed of more nodes, allowing more accurate estimation of heat fluxes [23]. The operating fluids have been divided in five nodes: hot turbine inlet gases (GAS), compressor outlet air (A), coolant (W), lubricating oil (OIL) and ambient (AMBIENT). The metal nodes can exchange heat by internal convection with the working fluids (gas, air, oil and water), by conduction with other metal nodes, by radiation and external convection (natural or forced) with the

ambient. To close the model, friction losses in lube oil are added as a power source [24]. For accounting transient heat transfer each node has an associated capacitance which models the thermal inertia [20].

In order to use the described model, the different conductances between the nodes must be known. In a previous work [25] these conductances were measured for a given turbocharger unit. Special tests using incompressible flow (thermohydraulic test bench) were performed to obtain the conductive conductances (from metal node to metal node) values. In such tests the temperature of the nodes was measured by means of thermocouples welded to the metal casings. Three thermocouples were used for each node in a radial distribution checking the assumption of negligible radial temperature differences. In the thermohydraulic test bench the turbocharger units were externally insulated to avoid heat transfer to the ambient. In addition, turbocharger shaft was blocked to avoid any mechanical power transfer, and no oil or coolant has been supplied to the corresponding ports. Finally, thermal oil was passed through the interior of compressor and turbine casings with a given temperature difference. The whole procedure assured that pure conductive heat transfer problem was separated from the convective heat transfer and turbocharger mechanical power transfer problems. With a proper combination of transient and steady state tests and using temperature at the nodes, the conductive conductances and capacitances were obtained. The values of these parameters will be used in the following sections in order to develop correlations for them based on external geometry.

The internal convection has been already solved in previous works by using experimental data from gas stand and avoiding external heat transfer by insulating the turbocharger [20, 21]. The procedure is based on knowing the conductive conductances from the specific tests described in previous paragraph and measuring temperature at the nodes; then the convective heat fluxes can be calculated [13]. A wide range of Nusselt and Reynolds numbers can be covered by testing the turbocharger in different conditions. Therefore, the experimental information is suitable for convective correlations development for each convective conductance of the model. These correlations provide the model the capability to determine proper convective heat fluxes for any operative condition of the turbocharger. In previous works [21] specific correlations for each tested turbocharger were developed. However, the experimental data of several turbochargers can be used to fit a general correlation for each convective branch of the model.

External convection and radiation can be determined from experimental data with exposed to ambient turbocharger tested in gas stand [19]. Radiation determination is based on a simplified model of the turbocharger geometry [iError! No se encuentra el origen de la referencia.]. External convection is based on correlations for forced and natural convection [26].

Finally, friction losses in lube oil are obtained by a physical procedure described in [24]. This procedure is based on a simplification of Navier-Stokes equations for the journal and axial bearings of a given turbocharger. This methodology requires experimental data from gas stand at adiabatic operation of the turbocharger as well as detailed geometry of turbocharger bearings. From this information four constants must be fitted to match the measured values of mechanical losses at turbocharger shaft.

Using the described procedures and experiments for a given turbocharger it is possible to characterize all conductances and capacitances of the lumped model shown in [Figure 1](#). Conductive conductances and capacitances are assumed to be constant values and the convective ones are described by correlations since their dependence on turbocharger operative conditions. Following heat fluxes at each fluid (gas, air, oil and coolant) can be predicted for any turbocharger operative conditions. This methodology requires high quantity of empirical data to fit the model for a specific turbocharger, as the following tests must be performed at different operative conditions:

- Tests using incompressible fluids in turbine and compressor ports for conductive conductances and capacitances determination.
- Hot tests at gas stand insulating the turbocharger for internal convective heat fluxes determination.
- Hot tests at gas stand with exposed to ambient turbocharger for external convection and radiation determination.
- Adiabatic tests at gas stand for the fitting of the four constants of mechanical losses model.

Without any doubt, the experimental effort required to properly determine the parameters of the model for a given turbocharger is high. In order to save these experimental costs a general methodology will be proposed in the following sections. This methodology is based on previous experience and only requires external turbocharger geometry for the determination of all the heat properties of the lumped model.

GENERAL CORRELATIONS FOR HEAT TRANSFER MODELLING FITTING

The general correlations of the heat transfer properties in order to use the model described in the previous section must provide all the capacitances and conductances of [Figure 1](#). The work can be split in conductive and convective conductances, in addition to a general procedure for accounting external heat transfer (convection and radiation) and mechanical losses model constants determination.

Turbochargers Description

In [Table 1](#) the main data of the turbochargers that will be used for correlation development is presented.

Table 1. Turbochargers characteristics (A sketch of geometrical acronyms is shown in Figure 2)

Turbocharger #	T#1	T#2	T#3	T#4	T#5
Engine displacement (l)	1.6	1.6	2.0	1.2	2.0
Engine type	petrol	diesel	diesel	petrol	diesel
Comp. wheel diam. (mm)	40	46	48.5	40	49
Turb. wheel diam. (mm)	38	37.5	40	37.5	39
VGT	No	Yes	Yes	No	Yes
Water cooled?	No	No	Yes	Yes	Yes
Turbine case external diameter, D_T (mm)	106	122	110	80	110
Turbine case external length, L_T (mm)	40	70	70	30	75
Compressor case external diameter, D_C (mm)	62	123	125	104	115
Compressor case external length, L_C (mm)	34	32	30	27	35
Bearing housing external diameter, D_H (mm)	51	70	60	57	60
Bearing housing external length, L_H (mm)	37	31	30	35	30

Turbochargers #1 to #4 will be used for the development of the correlations while T#5 will be used to check the ability of the correlations to predict heat transfer parameters. The selected turbochargers have different external geometry, different rotor geometry and are used in engines of different displacement. Moreover, three of the turbochargers are water cooled while the remaining two are not. This variety of turbochargers will allow getting more accurate correlations, i.e. better prepared to fit turbochargers of different characteristics.

Capacitances Determination

In order to calculate steady and transient heat transfer of the turbocharger conductive conductances and capacitances must be determined. Both parameters are assumed to be constant for any operative condition of the turbocharger. In order to provide general correlation of these parameters several steps must be performed.

The division of the turbocharger must correspond to the node definition of the lumped model. In Figure 2 a scheme of the turbocharger is given where the lengths and diameters of each component and plane are shown.

The capacitance of each node, in first approximation, can be calculated as the product of the mass of the node and the specific heat capacity of the material of the node given in Table 2, as shown in Eq. (1)

$$C_i = m_i \cdot c_i = \rho_i \cdot V_i \cdot c_i \quad (1)$$

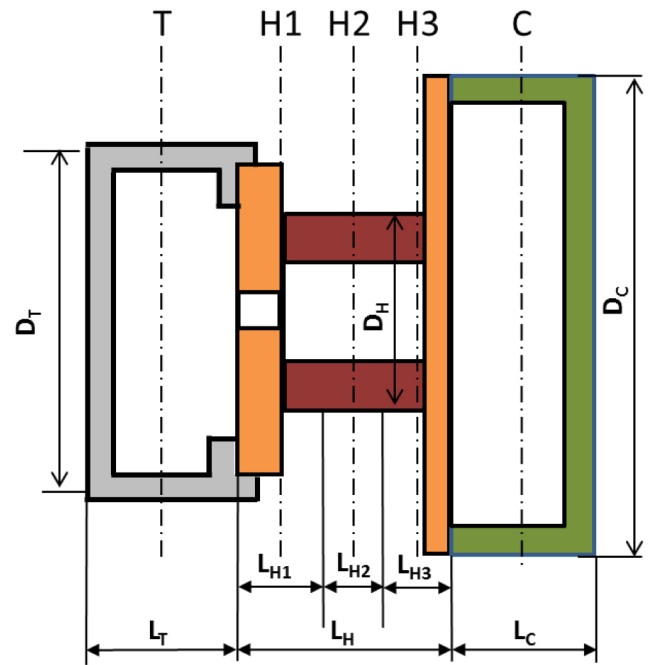


Figure 2. Turbocharger geometry simplification (Details of dimensions are shown in Table 1)

Table 2. Turbocharger material data

Part	Material	c ($\text{Jkg}^{-1}\text{K}^{-1}$)	ρ (kg m^{-3})	k ($\text{Wm}^{-1}\text{K}^{-1}$)
Turbine	SIMO iron casting	460	7100	36
Housing	FT200 iron casting	525	7059	48.4
Compressor	Aluminium	860	2670	205

However, the division of the housing in three nodes does not allow the usage of the above expression directly. The capacitance of housing casing must be distributed between the three nodes. Furthermore, the division of the turbocharger in nodes is rather artificial so small contribution of the turbine node capacitance to the capacitance of the adjacent housing node (H1) can exist, as well as in the case of the compressor to the H3 node. Assuming linear contribution in each case the capacitance determination can be done using Eq. (2), where four fitting constants (α , β , γ and ε) must be calculated. The constraints for the constants which avoid contributions higher than unity or lower than zero are given in Eq. (3). In the case of central housing to avoid zero contribution $\beta + \gamma$ must be inferior to unity. The same sum must be superior to zero to avoid node H2 to take the contribution of the whole physical housing of the turbocharger, which must be necessarily distributed between the three housing nodes.