Determination of heat flows inside turbochargers by means of a one dimensional lumped model

P.Olmeda *, V.Dolz, F. J. Arnau, M. A. Reyes-Belmonte

Universitat Politécnica de València, CMT-Motores Térmicos, Camino de Vera s/n, 46022 Valencia, Spain.

Abstract

In the present paper a methodology to calculate the heat fluxes inside a turbocharger from diesel passenger car is presented. The heat transfer phenomenon is solved by using a one dimensional lumped model that takes into account both the heat fluxes between the different turbocharger elements, as well as the heat fluxes between the working fluids and the turbocharger elements.

This heat transfer study is supported by the high temperature differences between the working fluids passing through a typical diesel turbocharger. These flows are the hot exhaust gases coming from the diesel engine exhaust passing through the turbine, the fresh air taken by the compressor, and the lubrication oil passing through the housing. The model has been updated to be used with a new generation of passenger car turbochargers using an extra element in the heat transfer phenomenon that is the water cooling circuit.

This procedure allows separating the aerodynamic from the heat transfer effects, permitting study the behavior of compressor and turbine in a separated way.

Keywords: Turbochargers Heat Transfer Lumped Models

*P. Olmeda. CMT-Motores Térmicos, Universitat Politécnica de València, Camino de Vera s/n, 46022 Valencia, Spain.
Phone: +34 963877650 Fax: +34 963877659 e-mail: pabolgon@mot.upv.es

1. INTRODUCTION

One of the biggest difficulties in turbomachinery industry is the accurate efficiency determination of its basic components, the compressor and the turbine. Understanding the phenomena affecting this efficiency determination is a key point in the field of automotive engines due to the widespread use of turbocharged diesel engine. In normal operation conditions, the traditional way to calculate the turbocharger efficiency [1] provides inaccurate results, mainly at lower speeds and powers [2]. This inaccuracy comes from the no consideration of the heat fluxes effects and the mechanical losses in the turbocharger axis in those methodologies. In order to improve the turbocharger efficiency determination a mathematical one-dimensional model of the turbocharger is presented in this paper.

2. LUMPED MODEL

Several studies about the heat transfer phenomena inside the typical turbocharger used in diesel engine applications have demonstrate the radial temperature distribution in a cross sectional area is negligible compared to the axial temperature distribution [3]. That evidence allows the simplification of the heat transfer problem inside the turbocharger considering it as a one-dimensional problem [4] instead of the three-dimensional case.

The one-dimensional lumped model proposed for the turbocharger under study is shown in Figure 1. The thermal model is based on the electrical analogy [5]. As it is shown, there are four working fluids in the lumped model. The two used in the power generation are the hot exhaust gases passing through the turbine and the fresh air taken by the compressor and used later to feed the engine. Besides, there is a lubrication oil source in the axis allowing the high rotational speeds of the turbocharger, but also removing part of the heat transferred inside the turbocharger. Finally, there is a cooling circuit surrounding the central part of the turbocharger to insulate thermally the turbine from the compressor. Each of these working fluids has been represented as a node in Figure 1.

The turbocharger has been separated into five different regions. Each region has been introduced as a metal node in the model and represents the surface temperature. Both metal cases, the turbine (T) and compressor (C), are represented as single nodes but the turbocharger housing has been divided into three different regions (H_1,H_2,H_3) due to both its complicate geometry and that the oil circuit and coolant circuit are placed inside that small element. So that, the node named H_1 is placed near the turbine case, the H_3 is near the compressor case and the one named H_2 is just in the center of the housing, where the lube oil and cooling circuits are placed. Metal nodes in the lumped model are connected between them by means of metal conductance and connected with the working fluids by means of a convective conductance. There are extra convective heat transfer ways between the exhaust gas and the H_1, the air and the H_3 and the oil with the H_3 due to the internal construction of the turbocharger. Finally, the equivalent electric model of the turbocharger includes capacitors in each of the metal nodes in order to represent the phenomena of accumulation or release of thermal energy of those nodes during a transient evolution.

3. SOLVING PROCEDURE

The objective of the solving procedure is to fill down all parameters of the lumped model. Once this information has been obtained, the lumped model can be used in order to determine the real power used by the turbine and the compressor. For example at the turbine side, the traditional way to calculate real power is based in the enthalpy difference between inlet and outlet (equation 1).

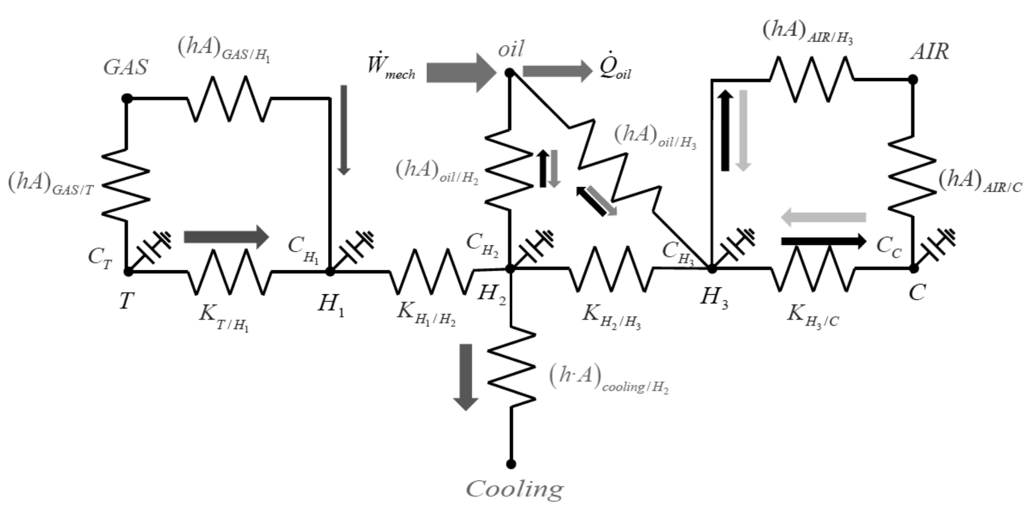

Figure 1: 1-D Lumped Model for the turbocharger. Heat fluxes are represented with arrows.

$$\dot{Q}_{fluid} = \dot{m}_{fluid} \cdot \int_{Tin}^{Tout} c_p \cdot dT \qquad (1)$$

Nevertheless in real the whole energy from the exhaust gases is not used in power generation. An important part of that energy is transmitted by convection to the turbine case and later to the rest of the turbocharger by means of metal conduction. Another part of the energy will be transmitted to the surroundings as a convective and radiative heat flux. This effect is not consider in the present paper as the model has been designed for a fully insulated turbocharger. And finally, other part of this energy is used to heat up the turbine case during transient operation (energy accumulation). In order to simplify the heat transfer phenomena in the turbine, it has been assumed that heat transfer procedure occurs before the expansion process. This phenomena is shown in Figure 2-(b) and leads to the turbine real power is lower than the difference in enthalpy between the turbine inlet and outlet.

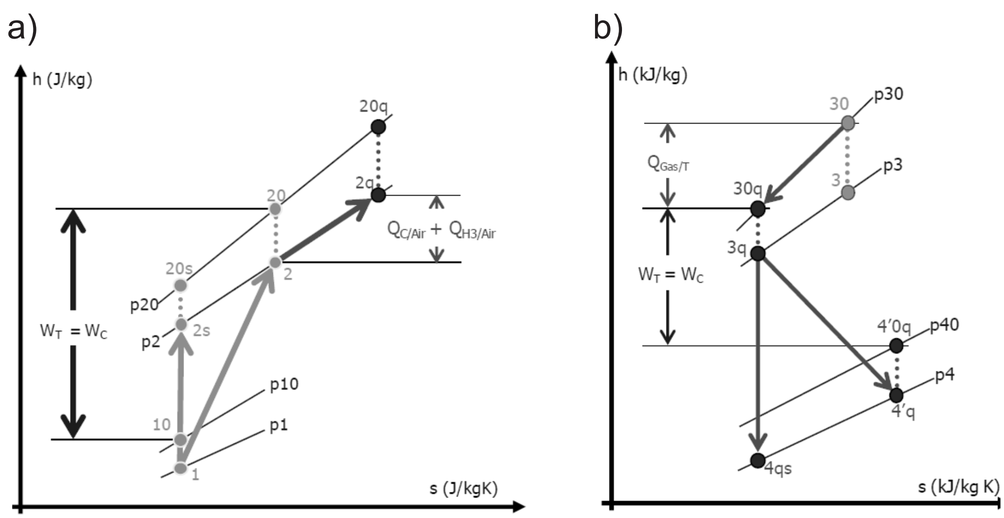

Figure 2: Adiabatic and diabatic operation in the h-s diagram of a) Compressor. and b) Turbine.

At the compressor side, the outlet air can released part of its thermal energy to the compressor case (by convective phenomena) and then to the rest of the turbocharger (by metal conduction). But during some operative conditions, this air can be heated up with the energy from the rest of the turbocharger depending on the temperature drop. Part of that energy can be accumulated in the compressor case during transient operative conditions. In order to simplify it has been assumed that the heat transfer phenomena occurs after the compression work as shows the Figure 2-(a). In this figure compressor is receiving a heat flux from the neighboring node H_3 due to the operative conditions.

The mathematical model scheme is based on the energy balance equation applied at each of the metal nodes. The expression in discrete-time way form is represented in equation (2).

$$m_i c_v \frac{dT^i}{dt} = \sum_j K_{ij} \left(T^j - T^i\right) + \sum_k \dot{q}_{k \to i} + \sum_l h_{li} A_{li} \left(T_l - T^i\right) \tag{2}$$

Summarizing, the lumped model has 16 unknowns to be determined. These are, 4 metal conductance (K_{T/H_1}, K_{H_1/H_2}, K_{H_2/H_3}, $K_{H_3/C}$), 5 metal capacitance (C_T, C_{H_1}, C_{H_2}, C_{H_3}, C_C) and 7 convective heat transfer coefficients ($(hA)_{GAS/H_1}$, $(hA)_{GAS/T}$, $(hA)_{oil/H_2}$, $(hA)_{oil/H_3}$, $(hA)_{cooling/H_2}$, $(hA)_{AIR/H_3}$, $(hA)_{AIR/C}$). Finally, to close the turbocharger power balance a model for mechanical losses characterization is needed. This model will be introduced as an energy source into the lumped model ($\dot{W}_{mech}$) and it will take into account the increase in lube oil temperature due to friction losses. In order to feed the model with that information a set of experimental tests campaign is needed. That methodology is based on the simplification of that complex lumped model by designing some special tests. The whole methodology is described more in detail in [4].

The first step in lumped model solution is the calculation of metal conductances. In order to determine them, a special experiment in a thermo-hydraulic test bench has been used [6]. The experiment consisted of generating a high temperature drop between turbine and compressor, as in real operative conditions, but passing hot oil through turbine case and cold water through compressor case. In addition, the turbocharger axis was blocked avoiding compression and expansion phenomena. Lubrication oil nor coolant liquid were used and the turbocharger was fully insulated to avoid the external convection and radiation. This experiment allows the direct determination of metal conductance as the ratio between the thermal power released (and absorbed) by the working fluids passing through the turbine and compressor, divided by the temperature difference between the calculation nodes, using equation (2). This procedure is only valid when all surface temperatures are steady (no temperature changes along the time) and so there is no energy accumulation in the nodes.

Metal capacitance determination was obtained using this same test rig, but modifying the testing procedure in order to generate transient flow conditions. Detailed information about its determination is showed in [4].

Once metal nodes properties (conductance and capacitance) have been determined, other tests were carried out to characterize convective properties of the turbocharger. The general expression for that problem is based on Sieder-Tate correlation [7]. This expression is showed in equation (3). Where l represents the working fluid (exhaust gases, air, oil, coolant liquid) while i represents a metal node in the lumped model. So that convective conductances are related with Reynolds' number, Prandtl's number and the ratio of fluid viscosities (estimated at the wall temperature and at the stream temperature).

$$(h \cdot A)_{l/i} = k \cdot a \cdot \mathrm{Re}^m \cdot \overset{n}{\mathrm{Pr}} \cdot \left(\frac{\mu}{\mu_0}\right)^o \tag{3}$$

In that equation, Reynolds' number takes into account the effect of mass flow variations in the heat transfer problem while Prandtl number is related with temperature variations. A test campaign varying both parameters is needed in order to determine constants from that equation (a,m,n,o). Convective conductance from experiments can be calculated using the general expression for convection problem, showed at equation (4). Where DTML represents the log mean temperature difference between inlet and outlet temperatures to the working fluid and its wall surface temperature as it is shown in equation (5),

$$\dot{Q}_{conv} = (h \cdot A)_{l/i} \cdot DTML_i^l \tag{4}$$

$$DTML = \frac{(T_{in} - T_{wall}) - (T_{out} - T_{wall})}{\ln \frac{(T_{in} - T_{wall})}{(T_{out} - T_{wall})}} \tag{5}$$

Convective power ($\dot{Q}_{conv}$) from equation (4) can be calculated also with the temperature drop along the working fluid circuit by means of equation

(1). Equating (1) and (4) allows the experimental calculation of convective conductance (hA). The Levenberg-Marquardt algorithm [8] has been used to determine constants (a,m,n,o) from equation (3).

Finally, an expression to correlate mechanical losses in the turbocharger axis with its operative conditions is essential in the turbocharger lumped model determination. This information will be used to decouple the increase in oil temperature by heat transfer phenomena (transmitted from turbocharger components) and its increase produced by the friction losses located in the shaft. The mechanical losses characterization is based on the methodology proposed by [9]. This methodology consists of a quasi-steady test campaign in a gas stand facility [10] where the turbine inlet temperature, compressor outlet temperature and lube oil inlet temperature are remained as similar as possible in order to reduce the internal heat fluxes. Therefore mechanical power in the shaft can be calculated directly by means of the increase in lube oil temperature. Due to the difficulty of keeping those temperatures as similar as possible, the heat fluxes effects have been taken into account in mechanical power definition (6) and mechanical efficiency definition (7).

$$N_{Mech} = \left(\dot{m} \cdot c_p \cdot \Delta T\right)_{Oil} + Q_{H1/H2} + Q_{H2/H3} \tag{6}$$

$$\eta_{Mech} = 1 - \frac{\dot{W}_{Mech\ Loss}}{\left[\dot{m}_T \cdot c_p \cdot (T_3 - T_4) + \dot{Q}_{H2/H1}\right]} \tag{7}$$

4. MODEL APPLICATION

Once lumped-model information has been fully determined, the model can be used to determine heat fluxes between the different measured nodes. The system of equations representing the lumped model is shown in matrix form in (8). Where $\mathbf{K}$ is a $9x9$ matrix storing the whole information from lumped model. This matrix can be subdivided into four submatrix as indicates equation (9).

$$\mathbf{K} \cdot \mathbf{T} = \mathbf{T_k} \tag{8}$$

$$\mathbf{K} = \begin{bmatrix} \mathbf{I} & \mathbf{0} \\ (\mathbf{h} \cdot \mathbf{A})_{\mathbf{i/fluid}} & \mathbf{K_{i/j}} \end{bmatrix} \tag{9}$$

Submatrix $\mathbf{I}$ represents the $4x4$ identity matrix, $\mathbf{0}$ is a $4x5$ zero matrix. Columns from submatrix $(\mathbf{hA})_{\mathbf{i/fluid}}$ are each one of the working fluids (exhaust gases, lube oil, coolant liquid and air) meanwhile there is a row per each metal nodes (T,H_1,H_2,H_3,C). Therefore, elements from submatrix $(\mathbf{hA})_{\mathbf{i/fluid}}$ are zero if there is not a physical connection between node i and fluid j. Columns and rows from submatrix $\mathbf{K_{i/j}}$ represent the metal nodes. This submatrix is tridiagonal and symetric. Elements for the main diagonal are the sum of conductances (conductive and convective) connected with metal node i but with a minus sign, and elements from upper and lower diagonal are metal conductance between node i and j.

Finally, vectors of temperatures are,

$$\mathbf{T} = \left[\begin{array}{ccccccccc} T_{T,in} & T_{OI} & T_W & T_{DO} & T_T & T_{H1} & T_{H2} & T_{H3} & T_C \end{array} \right]^t \tag{10}$$

$$\mathbf{T} = \left[\begin{array}{ccccccccc} T_{T,in} & T_{OI} & T_W & T_{DO} & 0 & 0 & 0 & 0 & 0 \end{array} \right]^t \tag{11}$$

As it is shown in the system of equations, there are five independent equations (each non-identity line in matrix (9)) meanwhile there are six unknonws (one per each measurement node and one more for the diffuser outlet temperature). In order to solve this system of equations extra information is obtained from turbocharger maps where turbine, compressor, oil and cooling inlet conditions are well known. Besides, it has been assumed that the point measured in the map corresponds to the one defined by the inlet conditions and the efficiency from map are estimated and the real mass flow are calculated as it is shown in equation (12). Since the map was obtained without having in mind the heat fluxes, an iteration is performed until the diffuser outlet temperature is obtained. Then, the aerodynamic efficiency of the compressor will be known. With this efficiency, and iterative process is followed in order to obtain the rest of temperatures. A comparisson of the results obtained using directly the map data and those obtained by applying the model to the measures are shown in Figure 3.

It has been observed that in the case of turbine, a deviation of less than 2% between measurement and model results is observed, while using map data the higher is the temperature the higher is the deviation. In case of compressor, results do not show any clear trend but model predicts accurately the behaviour deviation less than 1%. Once the model has been proved to predict the turbocharger behaviour, a study on the importance of heat fluxes has been performed for all the operative conditions of the turbocharger.

$$m_T^{map} = \frac{N_{is,c}^{map}}{\eta_{TG}\eta_C \Delta h_{is,T}^{map}} \quad (12)$$

This information is kept in Figure 4-(a) where the energy transferred to the turbine case and later transmitted to the rest of the turbocharger is shown. That energy is expressed in dimensionless form dividing it by the exhaust gases energy. As it is shown, an important ratio of the available energy at turbine entrance is not used during the expansion procedure and passes directly to the rest of the turbocharger as a heat flux. These ratios are near 45% for the lowest speeds and loads as it has been said traditionally [11]. Figure 4-(b) shows a similar study but performed the compressor. As it is represented, the lower is the turbocharger speed the higher is the ratio between the energy passed from the compressor case to its air, compared with compressor power with a ratio around the 10% of the whole compressor energy. This kind of information can be used to obtain turbocharger adiabatic maps using the maps provided by manufacturers or from an engine gas stand (both obtained at high temperature).

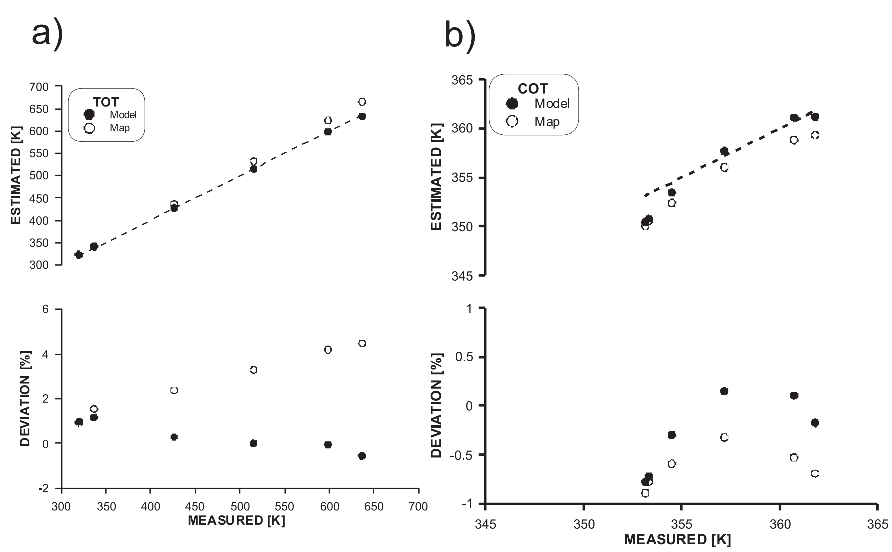

Figure 3: Comparisson between model and map outlet temperatures for the a) Turbine. and b) Compressor.

5. CONCLUSIONS

A one-dimensional lumped model for heat fluxes calculation inside an automotive engine turbocharger has been presented. The whole turbocharger has been split into different metal nodes. Parameters needed in the lumped model are, metal conductance representing the heat transfer path between two consecutive metal nodes and metal capacitance that is directly related with the amount of energy that a metal node can store. It is also needed the heat transfer coefficient of the turbocharger cooling circuit and lube oil circuit. And also heat transfer coefficients for turbine and compressor cases. Finally, mechanical losses to the axis will be considered as an energy source in the lumped model. The application of this model into a real turbocharger has shown the importance of internal heat fluxes when the turbocharger is running at lower rotational speeds. Besides, this application allows obtaining adiabatic maps for turbine and compressor using the maps given by turbocharger manufacturers.

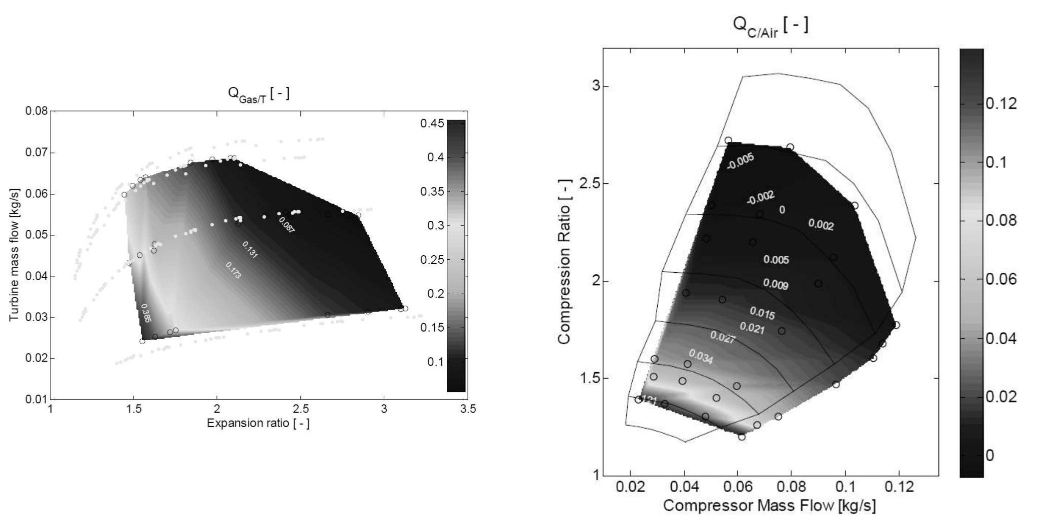

Figure 4: Heat fluxes importance in the turbocharger, a) Heat flux between exhaust gases and turbine case. b) Heat flux between air and compressor case.

NOTATION

c_p	Specific heat at constant pressure ($Jkg^{-1}K^{-1}$)
c_v	Specific heat at constant volume ($Jkg^{-1}K^{-1}$)
C_i	Metal capacitance for the metal node i (JK^{-1})
$DTML$	Log mean temperature difference (K)
(hA)	Convective conductance (WK^{-1})
k	Thermal conductivity of the fluid ($Wm^{-1}K^{-1}$)
K	Thermal conductance (WK^{-1})
m	Mass (kg)
N_{mech}	Mechanical power (W)
Pr	Prandtl number (-)
$\dot{q}$	Source energy term (W)
$\dot{Q}$	heat power (W)
Re	Reynolds number (-)
T	Temperature (K)

Greek letters

η_{mech}	Mechanical efficiency (-)
μ	Dynamic viscosity ($Pa \cdot s$)

Subscripts

i, j	Consecutive metal nodes
l	Working fluid
DO	Diffuser outlet
OI	Oil inlet
$T/H/C$	Turbine/Housing/Compressor metal nodes
W	Cooling liquid

Numbers

$3-4$	Turbine inlet-outlet

References

[1] N. Watson, S. Janota, Turbocharging the internal combustion engine, MacMillan Publishers Ltd, London, 1982.

[2] S. Marelli, M. Capobianco, Steady and pulsating flow efficiency of a

waste-gated turbocharger radial flow turbine for automotive application, Energy 36 (1) (2011) 459 – 465.

[3] S. Shaaban, J. Seume, Analysis of turbocharger non-adiabatic performance, in: Proc. 8th Int. Conf. on Turbochargers and Turbocharging, 2006.

[4] J.R. Serrano, P. Olmeda, A. Páez, F. Vidal, An experimental procedure to determine heat transfer properties of turbochargers, Measurement Science and Technology, 2010 21.

[5] A. J. Torregrosa, P. Olmeda, J. Martín, C. Romero, A tool for predicting the thermal performance of a diesel engine, Heat Transfer Engineering 32(10) (2011) 891–904.

[6] A. J. Torregrosa, A. Broatch, P. Olmeda, J. Martín, A contribution to film coefficient estimation in piston cooling galleries, Exp. Therm. Fluid. Sci. 34 (2010) 142–51.

[7] D. P. Incropera, Frank P & DeWitt, Fundamentals of Heat and Mass Transfer, 4th Edition, Wiley, New York, 2000.

[8] K. Levenberg, A method for the solution of certain non-linear problems in least squares, The Quarterly of Applied Mathematics 2 (1944) 164–168.

[9] F. Payri, J.R. Serrano, P. Olmeda, A. Páez, F. Vidal, Experimental methodology to characterize mechanical losses in small turbochargers, in: Proceedings of ASME Turbo Expo 2010: Power for Land, Sea and Air, Glasgow, June 2010.

[10] J.R. Serrano, V. Dolz, A. Tiseira, A. Páez, Influence of environmental conditions and thermodynamic considerations in the calculation of turbochargers efficiency, SAE World Congress & Exhibition 2009-01-1468.

[11] M. Jung, R. Ford, K. Glover, N. Collings, U. Christen, M. Watts, Parameterization and transient validation of a variable geometry turbocharger for mean- value modelling at low and medium speed-load points, SAE Technical Paper 2002-01-2729.