

Importance of Heat Transfer Phenomena in Small Turbochargers for Passenger Car Applications

Jose Serrano, Pablo Olmeda, Francisco Arnau and Miguel Reyes-Belmonte
Universitat Politècnica de València

Alain Lefebvre
RENAULT SAS

ABSTRACT

Nowadays turbocharging the internal combustion engine has become a key point in both the reduction of pollutant emissions and the improvement of engine performance. The matching between turbocharger and engine is difficult; some of the reasons are the highly unsteady flow and the variety of diabatic and off-design conditions the turbocharger works with. In present paper the importance of the heat transfer phenomena inside small automotive turbochargers will be analyzed. These phenomena will be studied from the point of view of internal heat transfer between turbine and compressor and with a one-dimensional approach. A series of tests in a gas stand, with steady and pulsating hot flow in the turbine side, will be modeled to show the good agreement in turbocharger enthalpies prediction. The goodness of the model will be also shown predicting turbine and compressor outlet temperatures. An accurate prediction of these parameters is a key factor to make easier the design of intercooler and aftertreatment devices.

CITATION: Serrano, J., Olmeda, P., Arnau, F., Reyes-Belmonte, M. et al., "Importance of Heat Transfer Phenomena in Small Turbochargers for Passenger Car Applications," *SAE Int. J. Engines* 6(2):2013, doi:10.4271/2013-01-0576.

INTRODUCTION

Traditionally heat transfer effects have been neglected in turbomachinery studies due to the relative small exchange surface and the occurring high flows. Nevertheless this assumption is only valid for large turbomachines, but not true for the small turbochargers used in passenger car applications, mainly at low load operative points and transient conditions [1,2]. Such conditions correspond to urban traffic, according to NEDC (New European Driving Cycle) tests, where the turbocharger speed is less than 90 krpm. In such conditions the heat transfer to the compressor (from the turbine and oil) can be equal to the energy transferred by the compressor shaft [3]. Therefore, improving turbocharger simulation codes and turbo-engine matching in these operative conditions becomes quite relevant in the automotive industry; since turbocharging the downsized engines is the most feasible way to reduce fuel consumption and control pollutant emissions [4,5].

Nevertheless, the information provided by turbocharger manufacturers is not usually measured under adiabatic conditions due to the difficulties and limitations of such testing procedure. So the efficiency given in a turbine map is not the one defined as isentropic as it also contains the

influence of heat and friction losses. Nonetheless, if the overall simulation of the turbocharger wants to be improved it would be desirable obtaining information about isentropic efficiency. It is due to it will remain the same for any given configuration (in a gas stand or in an engine) because it is subject only to the internal gas dynamic processes occurring in the turbomachine. If a direct parameterization of the turbocharger is performed just using extrapolation from turbocharger maps, unsatisfactory modeling results will be obtained as shown by [6].

For the aforementioned reasons having a turbocharger Heat Transfer Model (HTM) would be important when manufacturer maps are used. In such conditions the HTM will provide the turbocharger isentropic efficiency by discounting the heat fluxes occurring when the map was measured according to, for example, SAE J826 Standard [7].

Several studies have shown that the turbocharger is part of a complex thermal system containing different heat flows between compressor, turbine, engine housing, lubrication oil and environment [8,9]. In normal operative conditions heat flows from the turbine to the turbocharger housing and later arrives to the compressor, worsening its efficiency [10,11,12,13]. For instance when turbine inlet temperature

increases from 50°C to 500°C a nearly 15% lower compressor efficiency is measured for turbocharger speeds lower than the 50% of the maximum tip speed [14]. In the housing the heat is partially removed by the lubricating oil and the coolant circuit, in case it exists. In addition, the turbocharger exchanges energy to the surroundings by means of radiation and a mixed convection (free and forced depending on the set-up). A schematic diagram of these energy fluxes can be observed in Figure 1.

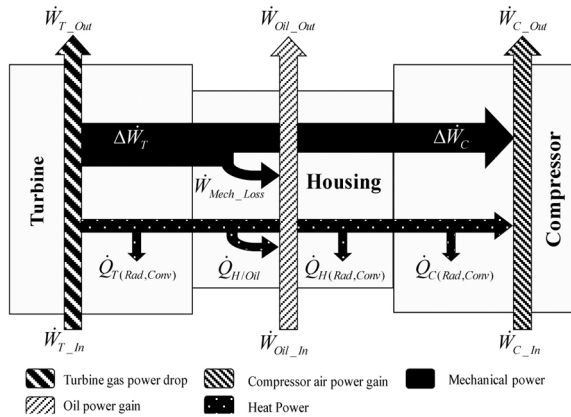


Figure 1. Main energy fluxes inside a turbocharger without water cooling.

Several methodologies to characterize heat transfer phenomena in turbochargers have been proposed. They range from fast one-dimensional models for the whole turbocharger to the accurate conjugated FEM-CFD models. One-dimensional models to characterize both the internal and external heat transfer of turbochargers are based in the thermodynamic analysis of diabatic processes.

Figure 2 shows diabatic processes compared with the ideal ones (isentropic). The isentropic evolution, from 1 to 20_s (Figure 2a), reflects the ideal situation where the compression process will take place under the assumptions of being adiabatic and reversible. In this situation, compressor outlet temperature will be calculated using the isentropic evolution and state equation between the inlet and outlet conditions [15]. If the compressor process is assumed to be adiabatic but not reversible; total compressor outlet temperature (T_{20}) will increase compared to the isentropic case. The ratio of enthalpy changes between both processes is known as compressor isentropic efficiency. Furthermore, assuming that compressor is receiving some heat flux from the turbocharger unit (whether the turbine case or the lubricating oil), its real total outlet temperature will be even higher (T_{20q}) than the adiabatic total outlet temperature (T_{20}). In the case showed in Figure 2a, air at the compressor inlet (T_{10}) will receive a heat flux that will increase its total temperature (T_{10q}). Later, the compression process will take place (T_{20q}) and finally the total temperature at the compressor outlet will be reduced as a consequence of losing a heat flux to the compressor case (T_{20q}).

At low rotational speeds, thermal energy will be transferred to the compressor. At moderate and high rotational speeds compressor also receives heat from the turbocharger housing but at certain point compressed air temperature will be higher than casing temperature and heat transfer will take place from the air to the compressor casing and later to the surroundings reducing its total temperature (T_{20q}).

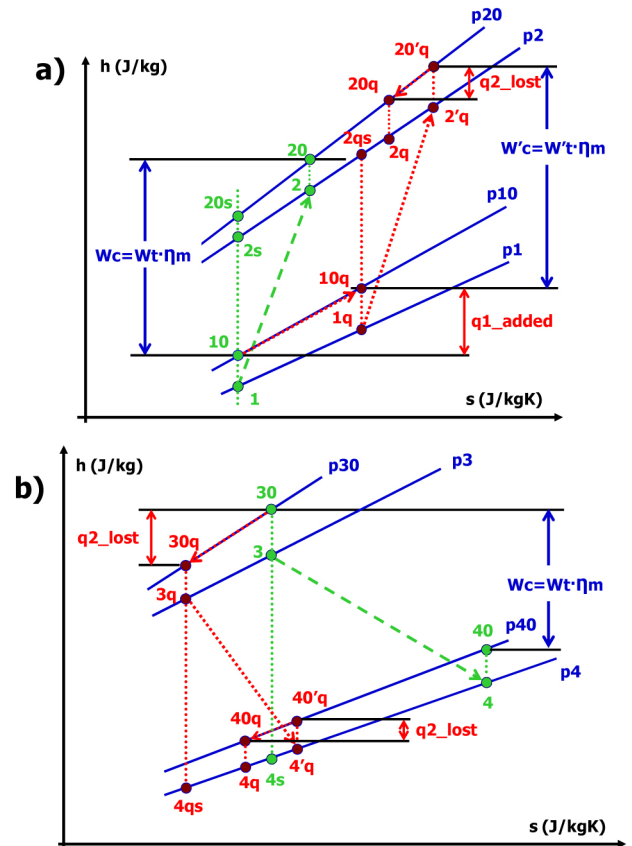


Figure 2. Entropic diagram for the compressor (a) and turbine (b)

Even in a rigorous thinking the compressor could not be considered as adiabatic under any circumstances [16], heat transfer effects will be minor at high rotational speeds, especially if compared with its mechanical power.

A new efficiency could be defined between the ratio of enthalpy changes between the ideal situation and the real one ($T_{20q}-T_{10}$). This efficiency is known simply as compressor efficiency and is typically provided by manufacturer maps. One of the consequences of this thermal energy transferred to the compressor will be the reduction in compressor efficiency given in manufacturer maps due to the increase in outlet temperature that can lead to an underestimation of turbocharger speed when computing it from solving the energy balance in the turbocharger. That temperature increase must be taken into account for the intercooler design; this effect will become important at engine cold start, idle and partial-load engine operation. In those situations heat transfer

to the compressor will increase due to the low rotational speeds and the proximity of engine operating points to the compressor surge line [17].

The same processes can be distinguished during turbine operative conditions. In this case, [Figure 2b](#) shows that the isentropic process takes place between points 30 and 4s. Turbine isentropic outlet temperature can be calculated, if the expansion process keeps adiabatic but not reversible, then another final state is defined with the name of adiabatic (point 4, which is based on keeping constant turbine effective power). The ratio of enthalpy change between the adiabatic and the isentropic states, shown in [Figure 2b](#), comes from the definition of turbine efficiency in manufacturer maps. If a diabatic evolution is considered (between points 30 and 40q), turbine total outlet temperature could be even lower than the value predicted by the isentropic evolution. That is justified as the real evolution is cooled and so turbine efficiency defined between initial and final states will be higher than the adiabatic one, and could be even higher than one. [Figure 2b](#) shows that an important error in predicting static turbine outlet temperature can take place.

One of the main discrepancies among the 1D models proposed in the literature is related to the way how heat is removed with respect to mechanical power: previously, after or a combination of both. The model proposed by Baines et al. [18] is mainly focused in heat transfer based on convection processes. Baines et al. [18] assume that the heat transferred with the compressor air takes place after the energy input. It means in the diffuser and in the volute, which is justified by the larger surface exposed to the air compared to the impeller area. Although heat transfer must exist in the impeller due to conduction along the shaft, it will be relatively small due to the lubrication oil and the small diameter of the shaft. This hypothesis is confirmed by the fact that compressor constant speed lines are not modified when the turbine inlet temperature increases, what ensures that aerodynamic losses do not depend on the heat fluxes [19]. However compressor efficiency lines show important changes what confirms the existence of a heat flux from the hot turbine to the cold compressor. The differences are higher at the low load area, where the compressor is far away from the adiabatic behavior. If heat transfer only takes place after the impeller, the mechanical power transferred in adiabatic conditions will be the same whatever will be the turbine inlet temperature. Using the same argument of exposed areas, in the turbine the bigger heat transfer will take place at volute up to turbine tongue and so before extracting power in the rotor [18]. In this case, heat transfer will affect the work transfer process. The main advantage of this technique is the simplicity for latter programming in an engine computational code, as the whole heat transfer takes place only in one place. In addition, experimental instrumentation will be easier and so the validation of the model.

Models proposed by Bohn [20] and Shaaban [16,21] make a distinction between the heat transfer that takes place before and after the impeller. Values from literature about thermal

conductivity of all casing materials and surface emissivity were used by Bohn et al. [22] instead of a global turbocharger characterization. Internal convection between each of the working fluids (exhausted gases, air and oil) to the corresponding cases (turbine, compressor and housing case) was characterized by means of a general equation based on Nusselt number but fitting constants using experimental data.

The study of Bohn et al. [22] revealed the importance of the lubricating oil removing part of the internal heat transfer from the turbine side to the compressor side. The importance of the lubricating oil is also confirmed as compressor heat flux is not so strong dependent with turbine inlet temperature. Similar results were obtained by Yamagata [23] but only focusing in high turbocharger speeds. Yamagata [23] study concludes that temperatures of compressor components (back plate, impeller back and impeller hub) vary proportionally to the compressed air temperature released by compressor, being the effects of turbine inlet temperature relatively small.

Models presented by Sidorow et al. [24] and Romagnoli et al. [25] propose the division of compression and expansion processes into three stages. For the compressor it is considered that the heat transfer takes place at two locations on the flow path according to [9]. A first heat flux increases compressor temperature before the compression process. After that the adiabatic compression occurs. Finally a heat flux after the compression will increase the adiabatic outlet temperature up to the measured temperature. A similar approach is considered for the turbine, assuming that the diabatic expansion process is divided into three stages. Firstly a heat flux is released at turbine inlet conditions; next adiabatic expansion and finally another release of heat flux.

In the present paper the importance of internal heat transfer inside automotive turbochargers will be analyzed. For that purpose two different turbochargers in size and configuration will be studied under steady and pulsating hot flow conditions, just at the turbine side. The importance of including a HTM in 1D engine simulation code will be observed in the prediction of compressor and turbine outlet temperatures and also in the prediction of turbocharger power balances.

TURBOCHARGER HEAT TRANSFER MODEL DESCRIPTION

In order to study heat transfer phenomena inside the turbocharger it has been divided into several measurement planes. Those planes are introduced according to the big temperatures differences between the hot turbine, the “cold” compressor and the lubricating oil passing through the central housing [20,22]. Those differences force a heat flux from the hot side (the turbine) to the cold side, typically to the oil but also to the compressor. Differences in radial surface temperature are small compared to the axial distribution what allows the simplification of the internal heat transfer problem into the 1D hypothesis [21].

The proposed turbocharger thermal model is based on the electrical analogy [26]. In this kind of models, the turbocharger is considered as a thermal network consisting in a finite number of nodes, whose thermal inertia is characterized by a thermal capacitance, and linked with other nodes by means of thermal conductance. Once the structure is divided into nodes, the energy conservation equation can be written for each node. It means the sum of heat fluxes between nodes, convective heat fluxes and other heat fluxes in a time interval equals the change in sensible energy of the node (1).

$$m_i c_v \frac{T_i^{t+\Delta t} - T_i^t}{dt} = \sum_j K_{ij} (T_i^{t+\Delta t} - T_j^{t+\Delta t}) + \sum_k q_{k \rightarrow i} + \sum_l h_{li} A_{li} (T_l^{t+\Delta t} - T_i^{t+\Delta t}) \quad (1)$$

Writing equation (1) for each of the n nodes gives a set of linear, implicit equations of the form,

$$\left(K + hA + \frac{I}{\Delta t} C \right) T_i^{t+\Delta t} = Q + \frac{I}{\Delta t} C \cdot T_i^t + hA \cdot T_l^{t+\Delta t} \quad (2)$$

In stationary conditions ($t = t + \Delta t$), equation (2) reduces to:

$$(K + hA) T_i^t = Q + hA \cdot T_l^t \quad (3)$$

Assuming the fluid temperatures, T_{bc} as boundary conditions and rearranging equation (3) it can be obtained:

$$\begin{bmatrix} I & 0 \\ hA & K \end{bmatrix} \begin{bmatrix} T_{bc} \\ T_{unknown} \end{bmatrix} = \begin{bmatrix} T_{bc} \\ 0 \end{bmatrix} \quad (4)$$

The proposed one-dimensional lumped model for the automotive turbocharger consists on a thermal resistor network presented in Figure 3. There are 5 metal nodes named T, H1, H2, H3 and C corresponding to the cases for the turbine, housing and compressor [27,28]. Housing case has been split into three measurement planes, one near the turbine (H1) other near the compressor back plate (H3) and the other in the center of the housing (H2). That division is justified by the complexity of housing internal geometry and the later simplicity of studying the internal heat fluxes with the lubricating oil and the coolant circuit in that kind of turbochargers. The boundary conditions are represented by five convective nodes; turbine gas (IT), lubrication oil (oil), and compressed air (IC-OD), ambient and cooling liquid (W) in case it exists. These nodes are characterized by their average temperatures.

Metal nodes are connected between them by means of conductive conductances K_{ij} being calculated the heat flux by means of Fourier's Law (5).

$$\dot{Q}_{ij}^{cond} = K_{ij} \cdot (T_i - T_j) \quad (5)$$

Conductive conductance between adjacent metal nodes in the turbocharger will be constant for any operative condition as this property only depends on internal geometry and material.

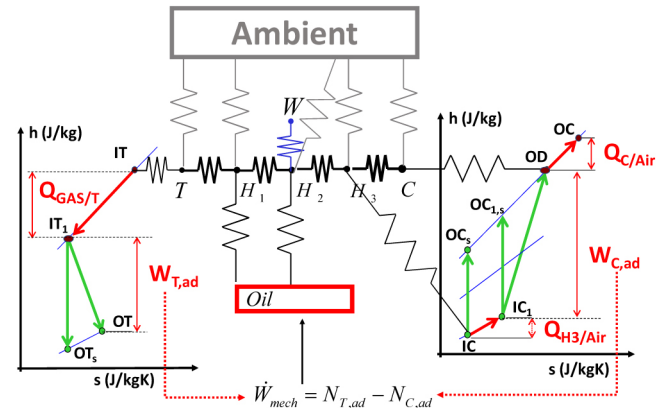


Figure 3. One-dimensional lumped model for a turbocharger with coolant circuit.

Metal nodes can store heat energy during transient performance introducing thermal capacitances in order to simulate that effect. Those properties will be obtained on a specific test bench [27,29] following the methodology described in [28]. Metal nodes are connected with fluid nodes (working fluids or ambient) by means of convective conductances hA_{li} being calculated the heat flux by means of Newton's cooling law (6).

$$\dot{Q}_{li}^{conv} = hA_{li} \cdot (T_l - T_i) \quad (6)$$

Convective conductance will depend on turbocharger geometry (wetted surface) but also on turbocharger operative conditions (fluid velocity and temperature). Thermal correlations for those heat transfer paths will be wisely determined from turbocharger measures under steady hot flow conditions. Those correlations will be based on Nusselt' number definition assuming that turbocharger internal cases (turbine and compressor volutes, lubricating ports and cooling ports) can be modeled as single pipes. With that assumption Nusselt number correlations will be in the form of Dittus-Boelter (7) and Sieder-Tate (8) correlations [30].

$$Nu_D = 0.023 Re_D^{4/5} Pr^n \quad (7)$$

$$Nu_D = 0.027 Re_D^{4/5} Pr^{1/3} \left(\frac{\mu}{\mu_s} \right)^{0.14}$$