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Heat transfer in turbocharger turbines under steady, pulsating and transient conditions

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ABSTRACT

Heat transfer is significant in turbochargers and a number of mathematical models have been proposed to account for the heat transfer, however these have predominantly been validated under steady flow conditions. A variable geometry turbocharger from a 2.2 L Diesel engine was studied, both on gas stand and on-engine, under steady and transient conditions. The results showed that heat transfer accounts for at least 20% of total enthalpy change in the turbine and significantly more at lower mechanical powers. A convective heat transfer correlation was derived from experimental measurements to account for heat transfer between the gases and the turbine housing and proved consistent with those published from other researchers. This relationship was subsequently shown to be consistent between engine and gas stand operation: using this correlation in a 1D gas dynamics simulation reduced the turbine outlet temperature error from 33 °C to 3 °C. Using the model under transient conditions highlighted the effect of housing thermal inertia. The peak transient heat flow was strongly linked to the dynamics of the turbine inlet temperature: for all increases, the peak heat flow was higher than under thermally stable conditions due to colder housing. For all decreases in gas temperature, the peak heat flow was lower and for temperature drops of more than 100 °C the heat flow was reversed during the transient.

1. Introduction

Turbocharging internal combustion engines is set to increase rapidly as this is a key technology to deliver fuel economy savings for both Diesel and spark ignition engines (Taylor, 2008). Using a compressor to provide higher air flows to an internal combustion engine increases the power density and allows smaller engines to be used in more high power applications, reducing overall weight and friction. The matching of a turbocharger with an internal combustion engine is a crucial step in the development process and relies on simulation of the engine air path system. In these models, turbochargers are represented by characteristic maps, which are defined from measurements of pressure ratio, shaft speed, mass flow and isentropic efficiency taken from a gas stand. Whilst the mass flow, pressure ratio, and speed can be measured directly, the efficiency has to be calculated from measured gas temperatures. For both turbine and compressor, enthalpy changes in the working fluids are equated to work changes during the characterisation process.[1] Any heat transfer affecting these gas temperature measurements will cause errors in the characterisation process. Conversely, when the characteristic maps are subsequently used in engine simulations to predict engine performance; if heat transfers are ignored then a poor prediction of gas temperatures for inter-cooling and after-treatment will arise. Consequently there is a twofold interest in understanding and modelling heat transfer in turbochargers:

1. To improve the accuracy of work transfer measurements during characterisation.
2. To improve the prediction of gas temperatures in engine simulations.

Current practice ignores heat transfers and limits investigations to operating conditions where heat transfer are small compared to work transfers; these conditions prevail for the compressor at higher turbocharger speeds but heat transfer is always significant

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[1] Some specialist facilities use a turbine dynamometer to measure turbine work directly, however these rarely used for automotive turbochargers in industrial applications.

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in the turbine. Parametric curve fitting techniques are then used to extrapolate to the lower speed region (Moraal and Kolmanovsky, 1999).

This work focuses on heat transfer in the turbine which represents the principal heat source for turbocharger heat transfer and strongly affects the gas temperature entering after-treatment systems. In particular, this paper aims to assess the applicability of gas-stand derived heat transfer models to on-engine conditions where flows are hotter, pulsating and highly transient.

2. Background

A number of studies into heat transfer in turbochargers have been presented over the past 15 years. The first studies focussed on quantifying the effects of heat transfer on steady flow gas stands by comparing the work transfers that would be measured based on temperature changes for different turbine inlet temperatures (Chesse et al., 2011; Cormerais et al., 2006; Shaaban, 2004; Serrano et al., 2007; Baines et al., 2010; Aghaali and Angstrom, 2013). Cormerais et al. (2006) presented the most extreme changes in operating conditions, varying turbine inlet temperature from 50 °C to 500 °C with a thermally insulated turbocharger and observed up to 15% points change in apparent compressor efficiency. Baines et al. (2010) measured losses of 700 W at 250 °C turbine inlet gas temperature (TIT) which is considerably lower than the 2.7 kW measured for a similar turbocharger by Aghaali and Angstrom with turbine inlet temperatures ranging 620–850 °C (Aghaali and Angstrom, 2013). Baines et al. (2010) also estimated heat transfer to ambient as 25% of total turbine heat transfer, however at 700 °C TIT, where temperature gradients to ambient were much higher, Shaaban (2004) estimated this at 70%.

A number of modelling approaches have been used ranging from 3D conjugate heat transfer, giving a detailed insight to the heat transfer processes (Bohn et al., 2005; Heuer and Engels, 2007), to simple 1D models for use with engine simulations. The most basic approach adopted to improve the correlation of engine models to experimental data consists of empirically adapting or *correcting* turbine maps using efficiency multipliers (Aghaali and Angstrom, 2013; Jung et al., 2002). This approach is typically parameterized to estimate heat energy directly using an exponential function that decays with increasing mass flow or turbine power and is tuned to match measured data from an engine or vehicle dynamometer. Whilst this approach can improve the accuracy of engine models, it is not predictive and alternative models have been proposed.

In practice heat transfer will occur through the turbocharger stage (Casey and Fesich, 2010), however a common assumption in 1D models assumes that heat transfer and work transfer occur independently (Cormerais et al., 2009; Olmeda et al., 2013; Serrano et al., 2010); this is represented schematically on enthalpy–entropy diagrams in Fig. 1. The actual processes undergone by the gases are shown between states 1–2 and 3–4 for compressor and turbine respectively. The split of work and heat transfer is shown by the intermediate states 1′, 2′, 3′ and 4′ such that flow through the turbine is composed of the following stages:

1. A heating or cooling at constant pressure (processes 1–1′ and 3′-3).
2. An adiabatic compression/expansion (processes 1′–2′ and 3′–4′).
3. A heating or cooling at constant pressure (processes 2′–2 and 4′–4).

Based on this analysis it is obvious that any measurement of temperature change across the turbine or compressor will include both the work and heat transfers, and that any estimate of work based on the total enthalpy change will include an error equal to the net heat transfer (Eq. (1)).

$$\Delta h_{act} = \Delta h_{work} + q_b + q_a \quad (1)$$

The isentropic efficiencies used in engine simulation codes are described for compressor and turbine in Eqs. (2) and (3) respectively. These equations describe the transitions between 1′–2′ and 3′–4′.

$$\eta_{s,c} = \frac{\Delta h_{s'}}{\Delta h_{work,c}} = \frac{c_{p,c}\left[T_{01'}\left(\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}} - 1\right)\right]}{\Delta h_{act,c} - q_{b,c} - q_{a,c}} \quad (2)$$

$$\eta_{s,t} = \frac{\Delta h_{work,t}}{\Delta h_{s',t}} = \frac{\Delta h_{act,t} - q_{b,t} - q_{a,t}}{c_{p,t}\left[T_{03'}\left(1 - \left(\frac{P_4}{P_{03}}\right)^{\frac{\gamma-1}{\gamma}}\right)\right]} \quad (3)$$

In Eqs. (2) and (3) it is common to define efficiencies using total conditions at points 1, 2 and 3 (and hence 1′, 2′ and 3′) and static conditions at point 4 (and 4′). For clarity, these distinctions have been omitted from Fig. 1.

The major issue that arises in applying Eqs. (2) and (3) is that it is not possible to directly measure $T_{1'}$ $T_{2'}$, $T_{3'}$ and $T_{4'}$ because they are not well defined spatially within the turbocharger. Consequently, for industrial mapping, operation is assumed to be adiabatic, i.e. $q_a = q_b = 0$, $T_1 = T_{1'}$; $T_2 = T_{2'}$, $T_3 = T_{3'}$ and $T_4 = T_{4'}$. This assumption holds for a compressor operating at higher shaft speeds where the heat transfer is small compared to the work transfer (Serrano, Olmeda, Arnau, et al., 2013). On the turbine side, the condition of adiabatic operation can only be achieved in special laboratory conditions and commonly turbine work is estimated either through compressor enthalpy rise or using a turbine dynamometer (Szymko et al., 2007).

The 3D conjugate heat transfer modelling undertaken by Bohn et al. (2005) showed that heat transfers between the working fluids and the housing could occur in either direction and could change

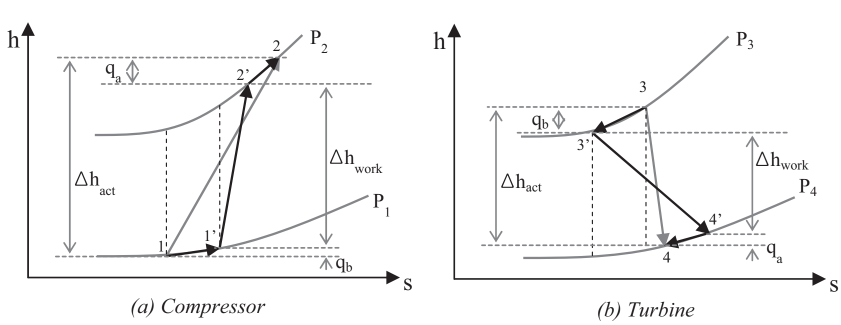

Fig. 1. Apparent and assumed compression and expansion processes in (a) compressor and (b) turbine.