ABSTRACT
The measured performance maps of turbochargers which are commonly used for the matching process with a combustion engine are influenced by heat transfer and friction phenomena. Internal heat transfer from the hot turbine side to the colder compressor side leads to an apparently lower compressor efficiency at low to mid speeds and is not comparable to the compressor efficiency measured under adiabatic conditions. The product of the isentropic turbine efficiency and the mechanical efficiency is typically applied to characterize the turbine efficiency and results from the power balance of the turbocharger. This so-called ‘thermo-mechanical’ turbine efficiency is strongly correlated with the compressor efficiency obtained from measured data. Based on a previously developed one-dimensional heat transfer model, non-dimensional analysis was carried out and a generally valid heat transfer model for the compressor side of different turbochargers was developed. From measurements and ramp-up simulations of turbocharger friction power, an analytical friction power model was developed to correct the thermo-mechanical turbine efficiency from friction impact. The developed heat transfer and friction model demonstrates the capability to properly predict the adiabatic (aerodynamic) compressor and turbine performance from measurement data obtained at a steady-flow hot gas test bench.

1 INTRODUCTION
The reduction of fuel consumption and emissions is the most dominant challenge in powertrain development. In the past few years gasoline direct injection (GDI) downsizing was the key engine technology used to reduce CO2 emissions while offering excellent transient performance. The concept of downsizing is to combine a GDI engine with a turbocharger, where the compressor delivers air at a higher density to the engine and more fuel can be burnt compared to a naturally aspirated engine. The exhaust gas of the engine drives the turbocharger turbine which is directly coupled by a shaft with the compressor and delivers the demanded compressor work for the required boost pressure of the engine. The engine operates at higher brake mean effective pressure.
(BMEP) levels and the same power and torque can be reached with a smaller displacement compared to a naturally aspirated engine. Smaller displacement results in a reduction of weight of the engine, reduced friction losses, lower throttling losses as well as lower fuel consumption and CO₂ emissions.

The matching of the turbocharger with the GDI engine relies on simulation and measurement results and is an essential step to achieve optimum powertrain efficiencies. In the commonly used zero- or one-dimensional gas dynamics simulation models, the turbine and compressor performance maps are looked-up from measurements of speed, pressure ratio, mass flow rate and isentropic efficiency obtained at a steady flow hot gas test bench.

Due to the fact that pressure and temperature are measured at positions which do not coincide with the respective system boundaries of the turbomachinery, the measured turbocharger performance maps do not represent the real aerodynamic performance of the turbomachinery (Fig. 1).

In addition, heat transfer within the turbocharger is significant especially at low to mid shaft speeds. In this operating range, the compressor and the turbine are working at small pressure ratios and mass flow rates and the change in enthalpy due to work transfer is small in comparison to the heat transfer. For this reason, heat transfer phenomena are strongly affecting the measured gas temperatures.

Ignoring heat transfer will directly lead to a poor prediction of gas temperatures at the compressor outlet, the turbine inlet as well as the turbine outlet. Subsequently, the matching process, the layout of the inter-cooling and the after-treatment devices is affected.

Due to the impact of heat transfer and the difficulty to measure the turbine outlet temperature directly, the product of the isentropic efficiency and the mechanical efficiency is typically applied to characterize the turbine efficiency. This product results from the power balance of the turbocharger and will further on be called ‘thermo-mechanical’ turbine efficiency (Eqs. (1) and (2)). It is strongly correlated with the diabatic compressor efficiency obtained from measured data at the test bench boundaries (Eq. (3)).

$$0 = \eta_m P_i + P_c = \eta_m \eta_i m_t \Delta h_{ia,t} + m_c \Delta h_{c,\text{dia}} \quad (1)$$

$$\Rightarrow \eta_m = \frac{m_c C_p,c (T_{02} - T_{01})}{\bar{m}_t C_{p,t} T_{03} \left[1 - \left(\frac{P_{01}}{P_{03}}\right)^{\frac{\gamma - 1}{\gamma}}\right]} \quad (2)$$

$$\eta_{ia,c,\text{dia}} = \frac{\Delta h_{ia,c}}{\Delta h_{c,\text{dia}}} = \frac{T_{01} \left[\left(P_{02} / P_{01}\right)^{\frac{\gamma - 1}{\gamma}} - 1\right]}{T_{02} - T_{01}} \quad (3)$$

The mass flow rate and shaft speed are corrected/reduced to the inlet conditions to retain Mach similarity between different performance maps:

$$m_{\text{corr,c}} = m_c \frac{P_{\text{ref}}}{P_{01}} \sqrt{\frac{T_{01}}{T_{\text{ref,c}}}} \quad (4)$$

$$m_{\text{red,t}} = m_t \sqrt{T_{03}} \quad (5)$$

$$n_{\text{red,t}} = \frac{n}{\sqrt{T_{03}}} \quad (6)$$

Consequently, three main errors must be taken into account when using measured turbocharger performance maps for the matching process:

1. Discrepancy in terms of system boundaries between test bench and turbomachinery
2. Impact of internal and external heat transfer on the gas temperatures measured at test bench boundaries
3. Impact of friction power on turbine efficiency

This work focuses on the impact of internal and external heat transfer phenomena as well as friction power on measured turbocharger performance maps. By applying one-dimensional heat transfer and friction models, the obtained corrected performance maps improve the accuracy in terms of predicting gas temperatures as well as the turbocharger matching simulations. Within the layout process and the assessment of design changes, the application of the proposed models ensure, that no wrong conclusions are drawn from measured performance maps which are not corrected from heat transfer and friction impact.
Following the previously discussed investigations, the processes is more significant than the impact of heat transfer. The contribution of heat transfer before and after the compression process from the compressor side is the dominating effect causing errors in the magnitude of heat transfer after the compression process to or after the compression process. From a combination of one-modeling as well as on the distribution of heat transfer before and after the compression process. Between the polytropic and isentropic analysis. But for automobile turbochargers are relevant, there is no significant difference in the measured turbocharger performance maps from the impact of heat transfer and friction. Fundamental work was done by Rautenberg et al. [2] and Malobabic [3] who investigated the influence of heat transfer at two different arrangements between compressor and turbine side. The results showed clearly that the compressor outlet temperature is influenced not only by the hot turbine temperature but also by the distance between the compressor and the turbine side. The investigations revealed that the obtained efficiencies under diabatic conditions can not be used to evaluate the aerodynamic performance of the turbomachinery.

Casey and Fesich [4] showed that an isentropic analysis of the compressor and turbine efficiency is thermodynamically incorrect and a polytropic analysis should be applied. The advantage of a polytropic analysis is that heat transfer can take place before, during or after the compression process whereas for a correct isentropic analysis only heat transfer after the compression process is allowed. But the work also revealed that for low speeds and pressure ratios, where heat transfer phenomena at automotive turbochargers are relevant, there is no significant difference between the polytropic and isentropic analysis. These findings were confirmed by a work done by Burke et al. [5]. The work focuses on investigations on one-dimensional modeling as well as on the distribution of heat transfer before and after the compression process. From a combination of one- and three-dimensional modeling they showed that at low speeds the magnitude of heat transfer after the compression process to or from the compressor side is the dominating effect causing errors in predicting gas temperatures. At higher shaft speeds the distribution of heat transfer before and after the compression process is more significant than the impact of heat transfer. Following the previously discussed investigations, the processes undergone by the exhaust gases on the turbine side and the air on the compressor side can be separated as follows (Fig. 2):

1. Isobaric heating or cooling \((1\rightarrow 1^*; 3 \rightarrow 3^*)\)
2. Adiabatic compression/expansion \((1^* \rightarrow 2^*; 3^* \rightarrow 4^*)\)
3. Isobaric heating or cooling \((2^* \rightarrow 2^*; 4^* \rightarrow 4)\)

Measuring the pressures and temperatures at the turbomachinery boundaries is usually not possible (Fig. [2] 1*–2* and 3*–4*). Therefore, many different modeling approaches have been developed over the past decades to obtain the aerodynamic performance of the turbomachinery from the measured values at the test bench boundaries.

Shaaban and Seume [6] derived from an analytical and experimental investigation that the compressor peripheral Mach number and the compressor heat number, which relates the amount of heat transfer to the enthalpy at the impeller inlet, are the two most important parameters affecting the compressor performance under diabatic conditions. An increase of the compressor heat number leads to a decrease in diabatic compressor efficiency.

Burke [7] presented a simple two-mass model of a complete turbocharger which provides significant improvements of the predicted gas temperatures at compressor and turbine side. The approach relies on empirical aerodynamic performance maps which are considered to be adiabatic. Their heat transfer model was applied to these measurements in order to derive the respective adiabatic performances.

Casey and Fesich [4] published an investigation on diabatic flows and their impact on the efficiency evaluation. This work presented a heat transfer model developed from non-dimensional analysis. By applying a constant amount of heat transfer, the results showed that their model is capable to estimate the apparent loss in efficiency due to heat transfer at low speeds.

Sirakov and Casey [8] showed in their experimental results that heat transfer does not affect the overall performance of the turbocharger because it has no influence on the pressure ratio characteristics. The presented heat transfer model is based on non-dimensional analysis and allows a conversion from diabatic work coefficients to adiabatic work coefficients to obtain the adiabatic compressor efficiencies.

3 EXPERIMENTAL SETUP AND INSTRUMENTATION

Extensive experimental investigations of three automotive turbochargers with varying boundary conditions were performed on the hot gas test bench of the University of Hannover. All three turbochargers comprise the same coolant circuit of the bearing housing and oil circuit for the journal and axial bearings. Turbocharger A delivers a maximum compressor power of about 32 kW and has compressor and turbine wheel diameter of 49 mm and 39 mm, respectively. Turbocharger B and C deliver a maximum compressor power of about 20 kW and have compressor and turbine wheel diameter of 42 mm and 36 mm, respectively.
3.1 Standard Instrumentation

Static pressure and temperature were measured at standard positions (1-2-3-4) on the compressor and turbine side as schematically shown in Fig. 1. Static pressure measurements were carried out using four static pressure taps of 0.5 mm diameter at each measuring position. These taps were uniformly distributed around the pipe circumference and provide an average value of the static pressure at each standard measuring position. Depending on the level of the measured pressure, transducers with measuring ranges of 0-0.35 bar, 0-3.1 bar and 0-3.5 bar were used for pressure measurement. Three NiCrNi-thermocouples (K-type, class 1, 3 mm) were applied at the turbine inlet and outlet. The temperatures at compressor inlet and outlet were measured by three resistance thermometers (Pt100, type 404, 1.5 mm).

Two ABB Sensyflow devices with a maximum error of less than 1% were used for measuring the turbine and compressor mass flow rates. Turbocharger rotational speed was measured with an eddy current sensor with a maximum linearity of ±0.2% of the full scale reading (±800 RPM). Oil and coolant volumetric flow rates were measured with turbine flowmeters with a linearity of ±0.15% - ±1% of the measuring range and the inlet and outlet temperature were measured with NiCrNi-thermocouples (K-type).

Experimental error analysis was carried out which revealed the following maximum measurement uncertainties:

<table>
<thead>
<tr>
<th>Measurement</th>
<th>Unit</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure</td>
<td>[Pa]</td>
<td>±1200</td>
</tr>
<tr>
<td>Temperature</td>
<td>[K]</td>
<td>±4.5</td>
</tr>
<tr>
<td>Mass Flow Rate</td>
<td>[kg/s]</td>
<td>±0.0015</td>
</tr>
<tr>
<td>Speed</td>
<td>[RPM]</td>
<td>±344</td>
</tr>
<tr>
<td>Compressor efficiency</td>
<td>[%]</td>
<td>±2.5</td>
</tr>
<tr>
<td>Turbine efficiency</td>
<td>[%]</td>
<td>±4.5</td>
</tr>
</tbody>
</table>

TABLE 1. MAXIMUM UNCERTAINTIES OF THE MEASUREMENT DEVICES AND TURBOMACHINERY EFFICIENCIES

3.2 Additional instrumentation

From previous measurements with turbocharger A, performed on the same test bench it can be concluded that the dependency of the temperature distribution on the peripheral direction can be neglected [9]. The measured temperatures of the compressor housing, made from aluminum alloy, were almost constant along the circumference (Pos. 2 and 3 in Fig. 3) because of the very high thermal conductivity of the compressor housing. The temperature of the compressor housing is strongly correlated to the compressor operating point as well as to the coolant temperature. It was also observed that the surface temperatures at the compressor inlet were in the range of the compressor outlet temperature but heat transfer upstream of the impeller has a negligible effect on the compressor performance [5].

For this reason, the temperature of the compressor housing of turbocharger B and C was only measured at one peripheral position (Pos. 1 in Fig. 4). The turbine housing of turbocharger B and C was instrumented accordingly to Fig. 3 to apply the proposed thermal model for the turbine side by Schinnerl et al [9]. The additional instrumentation for all three turbochargers can be summarized as follows (Fig. 3 and 4):

- Thermocouples on the internal surfaces along the compressor housing at the impeller inlet and the volute (1, 2)
- Thermocouples at the contact area between compressor and bearing housing (3)

On the turbine side additional thermocouples were applied at the following positions (Fig. 3):

- At the contact area between turbine and bearing housing in four different peripheral positions (6–9)
- On the outer and inner surface of the volute at two peripheral positions (10–11)
- On the outer surface of the turbine housing outlet (12)

Turbocharger B and C have the same thermodynamic layout of the compressor stage but differ in the design of the diffuser backplate as shown in Fig. 4. For turbocharger C, the diffuser backplate is not a separate component, but is an integral part of the bearing housing.
4 EXPERIMENTAL RESULTS AND ANALYSIS

All three turbochargers were tested at a turbine inlet temperature of 600°C and the efficiencies as well as the pressure ratios of the turbine and compressor are defined by using total-to-static and total-to-total conditions, respectively. This standard test conditions are commonly used in automotive turbocharger industry to obtain turbocharger performance maps from steady-flow hot gas test benches and are applied accordingly to the SAE turbocharger nomenclature and terminology [10].

Lüddecke et al. [11] showed that the status of the coolant temperature is the dominating boundary condition for heat transfer phenomena within automotive turbochargers for gasoline engines. Therefore, all three turbochargers were tested under two different coolant temperatures to show the impact of internal heat transfer on the compressor and turbine efficiency. This variation in coolant temperatures is used to validate the thermal models against the experimental results.

4.1 Non-insulated vs. insulated measurement pipes

The first investigation addresses turbocharger performance measurements with insulated measurement pipes to minimize external heat transfer from the turbomachinery boundaries to the test bench boundaries and vice versa (Fig. 1). This investigation was conducted by testing turbocharger B at a constant coolant and oil temperature of 90 °C. Figure 5 shows the difference in compressor efficiency between measurements with non-insulated and insulated measurement pipes. The largest differences (up to 5%) were obtained at the lowest shaft speeds near the surge line of the compressor map where the compressor power is relatively small compared to the amount of heat transfer [1]. Consequently, with non-insulated measurement pipes the compressor efficiency is overestimated at low shaft speeds due to heat losses at the measurement pipes caused by the lower measured compressor outlet temperature.

Additionally, the thermo-mechanical turbine efficiency will be overestimated in these operating ranges where external heat transfer has an impact on the evaluation of compressor efficiency (Eq. (2)). A simple one-dimensional pipe flow heat transfer model was applied to account for heat losses at the compressor outlet measurement pipes. The respective heat transfer coefficients were estimated according to Baehr and Stephan [12]. The estimated difference in gas temperature was used to recalculate the compressor efficiency for the non-insulated condition and to compare it against the compressor efficiency obtained for the insulated conditions.

Figure 6 presents the remaining difference in the compressor efficiencies and reveals that a simple one-dimensional pipe flow model is capable to almost eliminate the differences between non-insulated and insulated conditions at the measurement pipes. Nevertheless, insulating the measurement pipes should be a standard for testing turbocharger performance to avoid a misinterpretation in terms of compressor and turbine efficiency.

4.2 Non-dimensional characterization of compressor maps

The compressor inlet flow rate can be represented by the inlet flow coefficient $\phi_0$, which normalizes the volume flow rate $\dot{V}_c$ to the outer compressor wheel diameter $d_2$ and the circumferential speed $u_2$.

$$\phi_0 = \frac{\dot{V}_c}{d_2^2 u_2}$$

\[ Eq. (7) \]
The compressor pressure ratio \( p_{02}/p_{01} \) can be non-dimensionally represented by the head coefficient \( \psi \).

\[
\psi = \frac{\Delta h_{i,s,c}}{u_2^2} = \frac{C_{p,c} T_1}{u_2^2} \left( \Pi^{(\gamma - 1)/\gamma} - 1 \right) \quad (8)
\]

The adiabatic work coefficient \( \lambda \) relates the compressor work input \( \Delta h \) to the outer circumferential speed of the impeller \( u_2 \).

\[
\lambda = \frac{\Delta h_c}{u_2^2} = \frac{C_{p,c} T_1}{\eta_{i,s,c} u_2^2} \left( \Pi^{(\gamma - 1)/\gamma} - 1 \right) \quad (9)
\]

These three non-dimensional quantities \( \phi_{01} \), \( \psi \) and \( \lambda \) transfer the dimensional characterization of the compressor performance, mapped as a function of mass flow rate and pressure ratio, into a non-dimensional characterization. This transfer enables to compare compressor performance maps for geometrically similar compressors independently from the actual compressor size.

### 4.3 Variation of coolant temperature

The turbochargers A, B and C were operated at coolant temperatures of 90°C and 40°C and at a constant coolant volumetric flow rate of 10 liters/min and with insulated measurement pipes. The measurement results of the head coefficients in Figure 7 for Turbocharger A are congruent with the findings of Sirakov and Casey [8].

The variation in coolant temperature and therefore varying internal heat transfer on the compressor side does not exhibit an influence on the aerodynamic performance of each turbocharger. As expected, turbocharger B and C showed the same values of the head coefficients on constant speed lines within the typical experimental error. However, the influence of varying coolant temperature on the work coefficient (which is proportional to the compressor efficiency) can be seen in Fig. 8. A significant rise in the work coefficient can be observed at low speeds due to the apparently higher measured compressor outlet temperature. This is not a thermodynamic effect but rather due to the heat transfer impact which is incorporated as additional work input in the efficiency calculation.

### 4.4 Comparison of turbocharger B and C

Turbocharger B and C have the same thermodynamic compressor layout but differ in the design of the diffuser backplate (Fig. 4). Although, the two turbochargers show almost the same
FIGURE 9. WORK COEFFICIENTS OF TURBOCHARGER B AND C FOR A COOLANT TEMPERATURE OF 90 °C

performance in pressure rise on each constant speed line, the work coefficients differ significantly due to the different diffuser backplate (Fig. 9). The difference in the design of the diffuser backplate has two major impacts on the heat transfer at the compressor side. First, there is no insulating air gap for heat transfer between the air flow and the bearing housing for turbocharger C. In addition, the bearing housing of turbocharger C consists of a material which has a three times higher thermal conductivity than the diffuser backplate of turbocharger B.

For operating points where the compressor outlet temperature is below the temperature of the compressor housing, heat transfer to the compressor side takes place and leads to apparently higher work coefficients (Fig. 9) at low to mid speeds. For operating points where the compressor outlet temperature is above the temperature of the compressor housing, heat transfer from the compressor side to the bearing housing takes place and leads to apparently lower work coefficients.

4.5 Results from friction test bench

All three turbochargers comprise the same coolant circuit of the bearing housing and the same oil circuit for the journal and axial bearings. By neglecting manufacturing tolerances and the influence of axial forces on the bearing friction, it can be assumed that all investigated turbochargers should exhibit the same friction power at same speeds.

The friction measurements without axial load variation were performed on a friction test bench of FEV Germany [13]. Based on the measurement results, Li [14] validated his ramp-up simulations with the same boundary conditions and turbochargers and could achieve a satisfactory agreement between measurement and simulation.

The results of these simulations were then used to develop a simple friction power model which includes the radial as well as the axial bearing friction losses. The radial bearing friction power is modeled in dependency to the square of the angular speed related to the maximum angular speed. The influence of the axial bearing friction power is modeled linearly dependent on the angular speed.

To eliminate any impact of heat transfer phenomena and to consider the influence of the actual size of the compressor, the maximum change in isentropic enthalpy at the compressor side was identified as appropriate quantity. This approach leads to the following simple model for the overall bearing friction power.

\[ P_f = a_1 + a_2 \Delta h_{is,c,max} \left( \frac{\omega}{\omega_{max}} \right)^2 + a_3 \omega \]  

The coefficients \( a_1, a_2 \) and \( a_3 \) are fitted to the ramp-up simulation whereas \( a_1 \) and \( a_3 \) remain the same for all three turbochargers but \( a_2 \) changes because of the different maximum isentropic enthalpy range of the tested compressors. Figure 10 shows the results of the ramp-up simulation done by Li [14] compared to the fitted analytical friction power model.

FIGURE 10. RAMP-UP SIMULATIONS COMPARED TO ANALYTICAL FRICTION POWER MODEL FOR AN OIL TEMPERATURE OF 90 °C

5 CORRECTION METHODOLOGIES FOR COMPRESSOR AND TURBINE PERFORMANCE MAPS

The approach of this work is to correct the measured turbocharger performance maps from heat transfer and friction impact by non-dimensional analysis of the compressor side and is closely related to the work presented by Casey and Fesich [4] and Schinnerl et al. [9].

Following the investigations carried out by Casey and Fesich [4], the adiabatic work coefficient

\[ \lambda_{ad} = \left( 1 + \frac{k}{\phi_2} \right) \left( 1 - (1 - \mu) + \phi_2 \tan \beta_{2,bl} \right) \]
can be calculated with the knowledge of the flow coefficient $\phi_2$ at the impeller outlet

$$\phi_2 = \frac{m}{b_2 D_2 \pi \rho_2 u_2}$$  \hspace{1cm} (12)

and the slip factor $\mu$. For the calculation of the flow coefficient $\phi_2$, the outlet conditions at the impeller must be known. For this reason, the validated model of Schinnerl et al. \[9\] is used to calculate the respective impeller outlet conditions in terms of pressure and temperature for all three turbochargers. Note that this model sets the calculated conditions at the diffuser inlet equal to the conditions at the impeller outlet.

The calculation of the slip factor $\mu$ is also implemented and results in a value of about 0.825 for each investigated turbocharger.

The blade angle $\beta_{2,bl}$ in Eq. (11) is in general negative for automotive applications to obtain a wide compressor map and the factor $k$ takes into account the disc friction power whereby a value of 0.004 is used.

5.1 Non-dimensional heat transfer correction methodology for the compressor side

The apparently additional work input by heat transfer after the compression process, as shown in Fig. 2, can be calculated from the difference between the diabatic and the adiabatic work coefficient

$$\frac{q_{12}}{u_2^2} = \lambda_{dia} - \lambda_{ad}$$  \hspace{1cm} (13)

where $q_{12}$ is the specific heat flow per unit mass flow and can be expressed as

$$q_{12} = \frac{\dot{Q}}{\rho_2 \phi_2 u_2 \pi b_2 D_2}$$  \hspace{1cm} (14)

Since the adiabatic work coefficient is known by applying the model of Schinnerl et al. \[9\], a heat transfer model is needed to calculate the heat transfer to or from the compressor side after the compression process. In case of forced convection, Newton’s law of cooling (Eq. (15)) can be applied and the Nusselt number for estimating the heat transfer coefficient $h$ can be expressed in a general form by Eq. (16), where the constant $a$ is about 0.7.

$$\dot{Q}_{conv} = h A (T_e - T_w)$$  \hspace{1cm} (15)

$$N_u \propto \text{Re}^a \text{Pr}^{1/3} = \frac{h L}{k_e}$$  \hspace{1cm} (16)

The Reynolds number $\text{Re}$ in Eq. (16) is defined by

$$\text{Re} = \frac{u_2 \rho L}{\mu}$$  \hspace{1cm} (17)

where $L$ represents a characteristic length for calculating the heat transfer coefficient $h$ after the compression process as follows

$$L = L_{diff} + L_{vol}$$  \hspace{1cm} (18)

The travelling length $L_{diff}$ through the diffuser is calculated according to Stanitz \[15\] and decreases with increasing flow coefficient due to the higher meridional component of the air flow. Hence, the impact of heat transfer within the diffuser decreases due to the decreasing travelling length of the air flow operating the compressor from surge line to choke line.

The travelling length through the volute is represented by a mean length according to Shaaban \[16\]. The conditions for the air flow through the volute in terms of the travelling length and the impact of heat transfer apply accordingly to the conditions explained for the air flow through the diffuser.

By combining Eqs. (13) and (15) and lumping together the remaining quantities into a single constant $c$, a simple heat transfer model is developed

$$\frac{q_{12}}{u_2^2} = \frac{u_2^2 L^{a-1} c (T_w - T_{02})}{u_2^2} = \lambda_{dia} - \lambda_{ad}$$  \hspace{1cm} (19)

where $T_w$ is the measured temperature of the compressor housing and $T_{02}$ is the calculated total temperature at the impeller outlet (diffuser inlet). The constants $a$ and $c$ have to be fitted to the experimental results of $\lambda_{dia}$ for each constant speed line. This simplified heat transfer model allows to calculate the heat transfer to or from the compressor side, depending on the temperature gradient $(T_w - T_{02})$, whether is positive or negative. Additionally, the decreasing influence of heat transfer phenomena with increasing circumferential speed $u_2$ is taken into account because in case of forced convection, the exponent $a$ in Eq. (16) is always lower than unity \[12\].

The adiabatic efficiency can then be evaluated from the correlation between work coefficient and efficiency as follows

$$\eta_{is,c,ad} = \frac{\lambda_{dia}}{\lambda_{ad}} \eta_{dia}$$  \hspace{1cm} (20)

5.2 Heat transfer and friction correction methodology for the turbine side

In theory, the thermo-mechanical efficiency corrected by influences of from heat transfer and friction impact, must be equal
to the isentropic turbine efficiency calculated from the temperatures at the turbine inlet and outlet (Eqs. (21) and (22)).

As described in chapter 1, the measured temperatures at the test bench boundaries on the turbine side cannot be directly used to calculate the isentropic turbine efficiency. Hence, modeling the heat transfer between the test bench and turbomachinery boundaries is necessary to simulate the heat transfer at the turbine volute and turbine housing outlet and eventually at the measurement pipes in case of non-insulated conditions.

\[
\eta_{is,t} = \frac{\dot{m}_c C_{p,e} (T_{02} - T_{01}) - \dot{m}_c q_{12} + P_f}{\dot{m}_t C_{p,t} T_{03} \left[ 1 - \left( \frac{p_4}{p_{03}} \right)^{\gamma-1} \right]} \tag{21}
\]

\[
\eta_{is,t} = \frac{C_{p,t} (T_{03} - T_4)}{C_{p,t} T_{03} \left[ 1 - \left( \frac{p_4}{p_{03}} \right)^{\gamma-1} \right]} \tag{22}
\]

To overcome the discrepancies in the system boundaries, Schinnerer et al. [9] have proposed a one-dimensional heat transfer model for the turbine side which improves the prediction of the turbine wheel inlet and outlet temperature and therefore enables the calculation of the isentropic turbine efficiency directly. The results of the simulated isentropic turbine efficiencies will be compared to the thermo-mechanical turbine efficiencies, corrected from heat transfer and friction impact, in the following section.

### 6 RESULTS AND DISCUSSION

This section describes the correction methodology for heat transfer and friction impact on measured turbocharger performance maps and is split up into compressor and turbine side, respectively.

All results are shown for the cases of insulated measurement pipes, constant coolant volumetric flow rate and constant oil inlet temperature and pressure.

#### 6.1 Correction of heat transfer impact at compressor side

Figure 11 shows the diabatic work coefficients (solid lines) of turbocharger A obtained from the measurement results in comparison to the adiabatic work coefficients (dashed lines). A significant lower diabatic work coefficient is obtained, especially at low speeds, if the apparently additional work input due to heat transfer is cancelled out from the measurement results. The adiabatic work coefficients form almost a single band of points which are only dependent on the flow coefficient \( \phi_2 \), the slip factor \( \mu \) and the disc friction factor \( k \) and represent the compressor performance under adiabatic conditions (Eq. (21)).

Figure 12 shows the different trends in the measured compressor efficiencies of turbocharger A for a coolant temperature of 90 °C and 40 °C (solid lines). By calculating the adiabatic compressor efficiencies (dashed lines) with the usage of Eq. (20), a satisfying agreement between both corrected measurement results could be achieved.

The results of the heat transfer correction method in terms of calculating the adiabatic work coefficients for turbocharger B and C are presented in Fig. 13. The measured diabatic work coefficients (solid lines) of turbocharger C show at low speeds a higher impact of heat transfer due to the absence of an insulating air gap and the higher thermal conductivity of the bearing housing in comparison to the separate diffuser backplate of turbocharger B (Fig. 4).

As for turbocharger A, the correction from heat transfer impact
6.2 Correction for heat transfer and friction impact at turbine side

The results of the correction methodologies for the turbine side are exemplarily shown for turbocharger A and are similarly to that obtained for turbocharger B and C. Following the argumentations in section 4.5, all three turbochargers should exhibit the same amount of friction power at a given speed and the thermal model of the turbine side is applied accordingly for each turbocharger.

The impact of friction power on the turbine efficiencies is present over the whole turbine performance map and therefore more speed lines are shown in the following figures than in the figures of the heat transfer investigations on the compressor side. First, the thermo-mechanical turbine efficiencies $\eta_{tm}$ obtained from measurements with a coolant temperature of 90 °C are recalculated by inserting the corresponding adiabatic compressor efficiencies $\eta_{is,c,ad}$ in Eq. (2) instead of the measured diabatic compressor efficiencies (Eq. (3)). The comparison between the measured thermo-mechanical turbine efficiency (blue lines) and the thermo-mechanical turbine efficiency calculated with the adiabatic compressor efficiency (red lines) is shown in Fig. [15]. At low to mid speeds, where heat transfer is significant (on the compressor side), the correction for heat transfer of the measured thermo-mechanical turbine efficiency by the usage of the adiabatic compressor efficiency results in significantly lower turbine efficiencies (up to 10%). Within the matching process, the turbine efficiencies at low to mid speeds are crucial for transient response of the turbocharger and therefore accurate turbine performance maps in this operating range are required. The friction power on each speed line is only dependent on the angular speed and therefore constant on a constant speed line.
By the usage of Eq. (10), the thermo-mechanical turbine efficiency corrected from heat transfer impact and friction impact is obtained by applying Eq. (21).

Figure 15 shows the thermo-mechanical turbine efficiency corrected from heat transfer and friction impact (black lines) compared to the adiabatic thermo-mechanical turbine efficiency (red lines) and the measured one (blue lines). The trends in Figure 15 show that on a constant speed line the influence of the friction power on the turbine efficiency decreases because of the increasing turbine power. In addition, the turbine power increases overproportionally compared to the friction power and therefore the impact of friction power is less at higher speeds than at the lower speed lines.

By applying the heat transfer model for the turbine side of Schinnerl et al. [9], the isentropic turbine efficiencies according to Eq. (22) are calculated for turbocharger A. In theory, comparing the isentropic turbine efficiencies to the thermo-mechanical turbine efficiencies corrected for heat transfer and friction impact (Eq. (21)), must result in the same values.

Figure 16 presents the comparison of the corrected thermo-mechanical turbine efficiency to the isentropic turbine efficiency directly calculated from the temperatures calculated at the turbomachinery boundaries. Over the whole turbine performance map an agreement between both approaches can be achieved within 2.5% which means a quite satisfactory result for accurate predicting the turbine performance maps in terms of turbine inlet and outlet temperatures for the matching process.

7 CONCLUSIONS AND OUTLOOK

The measured performance maps of three different turbochargers are corrected with respect to heat transfer and friction phenomena by one-dimensional models to predict the adiabatic (aerodynamic) compressor and turbine performance.

The first part of the current work addresses the importance of insulating the measurement pipe at the compressor outlet to minimize external heat transfer. Neglecting the temperature losses to the ambient in case of non-insulated measurement pipes results in an apparently higher compressor efficiency of up to 5% at low speeds and therefore leads to apparently higher thermo-mechanical turbine efficiencies.

Non-dimensional analysis of the compressor stage of different turbochargers allowed to develop a simple heat transfer model to predict the adiabatic work coefficients. The heat transfer after the compression process is calculated in dependency of the travelling length of the air flow and the temperature gradient between the impeller outlet total temperature and the temperature of the compressor housing.

Two turbochargers (B and C) with the same thermodynamic layout of the compressor stage but with different design of the diffuser backplate are investigated. At the same boundary conditions of coolant and oil temperature, the work coefficients of both turbochargers show a significant difference at low to mid speeds where heat transfer phenomena are relevant. The calculated adiabatic work coefficients form almost a unique line of points due to the removal of the apparently additional work input caused by heat transfer. The adiabatic compressor efficiencies show the same trends and differ in a range of ±1.5%.

Based on measurements and ramp-up simulations, a simple analytical friction model is developed to correct the turbine efficiency from friction impact. The thermo-mechanical turbine efficiency corrected by heat transfer and friction impact are compared to the isentropic turbine efficiency calculated from the simulated temperatures at the turbine inlet and outlet. In theory, both corrected turbine efficiencies should exhibit the same values and this is achieved by the application of the heat transfer and friction models within a band of ±2.5%.

Further work should focus on the application of the proposed models on measurements for enhanced turbine performance maps to cover a wider range of turbine efficiencies on constant speed lines. The analytical friction model can be improved in terms of accounting for the influence of axial thrust on the axial bearing friction.

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Nomenclature

\[ A = \text{area (m}^2) \]
\[ b = \text{diffuser width (m)} \]
\[ C_p = \text{specific heat at constant pressure (J kg}^{-1} \text{K}^{-1}) \]
\[ d = \text{diameter (m)} \]
\[ h = \text{specific enthalpy (J kg}^{-1}) \]
\[ k = \text{thermal conductivity (W m}^{-1} \text{K}^{-1}) \]
\[ L = \text{characteristic length (m)} \]
\[ \dot{m} = \text{mass flow rate (kg s}^{-1}) \]
\[ n = \text{rotational speed (s}^{-1}) \]
\[ p = \text{pressure (Pa)} \]
\[ P = \text{Power (W)} \]
\[ \text{Nu} = \text{Nusselt number} \]
\[ \text{Pr} = \text{Prandtl number} \]
\[ \dot{Q} = \text{heat flux (W)} \]
\[ \text{Re} = \text{Reynolds number} \]
\[ T = \text{temperature (K)} \]
\[ u = \text{angular velocity (m}^1 \text{s}^{-1}) \]
\[ \dot{V} = \text{volumetric flow rate (m}^3 \text{s}^{-1}) \]
\[ \eta = \text{efficiency} \]
\[ \phi = \text{flow coefficient} \]
\[ \gamma = \text{ratio of specific heats} \]
\[ \lambda = \text{work coefficient} \]
\[ \mu = \text{dynamic viscosity (kg m}^{-1} \text{s}^{-1}) \]
\[ \omega = \text{angular speed of shaft at bearing position (s}^{-1}) \]
\[ \psi = \text{head coefficient} \]
\[ \rho = \text{density (kg m}^{-3}) \]

Subscripts

0 = stagnation (temperature, pressure)
1 = compressor stage inlet (test bench boundary)
1* = compressor stage outlet (turbomachinery boundary)
2 = compressor stage inlet (test bench boundary)
2* = compressor stage outlet (turbomachinery boundary)
3 = turbine stage inlet (test bench boundary)
3* = turbine stage outlet (turbomachinery boundary)
4 = turbine stage outlet (test bench boundary)
4* = turbine stage outlet (turbomachinery boundary)
a = after compression/expansion
adi = adiabatic
b = before compression/expansion
bl = Blade
c = compressor
corr = corrected to inlet conditions
dia = diabatic
diff = diffuser
f = friction
g = gas
is = isentropic
m = mechanical
red = reduced to inlet conditions
ref = Reference temperature/pressure (298.15K)/(10^5 Pa)
t = turbine
tm = thermo-mechanical
vol = volute
w = wall

Acronyms

BH = bearing housing
BMEP = brake mean effective pressure
CO\textsubscript{2} = carbon dioxide
GDI = gasoline direct injection
RPM = revolutions per minute
TC = Turbocharger

REFERENCES


