APPARENT AND REAL EFFICIENCY OF TURBOCHARGERS UNDER INFLUENCE OF HEAT FLOW

Mehdi Nakhjiri¹, Peter F. Pelz¹*, Berthold Matyschok¹, Lorenz Däubler², Andreas Horn²

¹Technische Universität Darmstadt
Fluid Systems Technology
Magdalenenstraße 4, Darmstadt, 64289, Germany
mehdi.nakhjiri@fst.tu-darmstadt.de
* peter.pelz@fst.tu-darmstadt.de
berthold.matyschok@fst.tu-darmstadt.de

²IAV GmbH
Rockwellstraße 16, 38518 Gifhorn, Germany
lorenz.daeubler@iav.de
andreas.horn@iav.de

Abstract
The measured and usually known apparent efficiency of a turbocharger compressor is due to heat fluxes from the turbine to the compressor not comparable with the real efficiency of the machine determined under adiabatic condition. Hence, there is a need to make the distinction between apparent and real efficiency which is discussed here. Furthermore, and more important, we introduce a physical based and hence reliable scaling procedure to determine the real efficiency of compressor out of manufacturers’ non-adiabatic measured map. Finally, we present an analysis to show the significance of heat transfer on the performance of compressor and turbine. Based on this the overall turbocharger efficiency and the turbine efficiency are discussed and reevaluated. Most important, only the real efficiencies are needed for the system engineer to match a turbocharger with a combustion engine.

1. Introduction
Manufacturers’ performance maps are based on temperature, pressure and mass flux measurements at the control surfaces of the test stand (see Fig.1). Since the internal heat flux is neglected the efficiency given by the manufacturer has to be named “apparent” efficiency. The definition of the apparent efficiency is well known to be given by the ratio of measured total enthalpy difference \( h_t \) to isentropic enthalpy difference \( h_{ts} \):

\[
\eta_{app}^{\pm 1} = \frac{[h_{ts}]}{[h_t]}, \tag{1}
\]

where the positive exponent +1 holds for the compressor and the negative exponent -1 for the turbine.

Due to the temperature difference between turbine and environment as well as between turbine and compressor and resulting heat flows, the turbocharger operation cannot be characterized as adiabatic. The measured total enthalpy differences include heat flows which constitute uncertainties and distort the performance maps. Again as a result, manufacturers’ measured efficiency is an apparent
efficiency based on the assumption that heat remains within the control volume of test stand:

\[ \eta_{\text{app}} = \eta, \quad \text{for} \quad \dot{Q} = 0, \]
\[ \eta_{\text{app}} \neq \eta, \quad \text{for} \quad \dot{Q} \neq 0. \]  

(2)

Therefore, it is needed to analyze the compressor and turbine within distinct control volumes as sketched in Fig.1. Further, heat flow through bearing housing needs to be prevented or at least significantly diminished which is a technical challenge.

Hence, the objective of this paper is to reversely approach the performance maps of compressor and turbine according to the compressor and turbine control volume from manufacturers’ maps generated based on the control volume of the test stand.

Figure 2 exhibits compressor measurements at three different levels of heat flow \( \dot{Q} \) (corresponding to three different turbine inlet temperatures). We obtain that there is almost no influence of heat flow on pressure ratio \( \Pi_t := \frac{p_{t2}}{p_{t1}} \) in the compressor. This allows the conclusion that heat transfer increases the temperature of the air flow through the compressor mainly after the compression process. Otherwise deterioration in pressure ratio would be expected which refers to diverging isobars in an h-s-diagram. However, higher measured temperature and accordingly higher specific total enthalpy at the outlet results in overestimation of work input \([h_{t1}]_{\text{adiabatic}} > [h_{t1}]_{\text{adiabatic}} \). This can be seen as apparent work input and leads to underestimation of efficiency at low speed. Hence, the amount of heat transferred to the compressor needs to be corrected from the measured work input \([h_{t1}]_{\text{adiabatic}} \) in order to gain the real compressor efficiency.

Heat flow considerations as well apply to the turbine with regard to the control volume in Fig.1. Different engine load points generate different exhaust gas temperatures and pressures. The temperature of exhaust gas corresponds to its working capacity. The turbine gives off heat into the environment and bearing housing decreasing the working capacity of the exhaust gas which has consequences for the work output of the turbine and its efficiency.

2. Compressor Performance

The maps in this section are generated based on measurements on a hot gas test stand. Three different turbine inlet temperatures 800 °C, 600 °C and 300 °C are explored, while the latter is the lowest possible turbine inlet temperature on the hot gas test stand. Measuring technique and standard values are arranged according to SAE J922. Considerations in this section are exemplarily demonstrated on one turbocharger.

Based on the first law of thermodynamics, the dimensionless work input of compressor, i.e. the power coefficient reads:

\[ \lambda := \frac{2[h_t]}{u_2^2} = \frac{2}{u_2^2} (w_{\text{th}} + w_{\text{par}} + q_C), \]

(3)

\[ \lambda = \lambda_{\text{th}} + \lambda_{\text{par}} + q_{C+} \]

(\( \lambda_{\text{th}} \)=theoretical, \( \lambda_{\text{par}} \)=parasitic, \( C=\text{compressor} \)). Here the shaft power is written as sum of the theoretical and parasitic technical work \( P_s = (w_{\text{th}} + w_{\text{par}}) \dot{m} \), and the heat flow reads \( \dot{Q}_C = q_C \dot{m} \).
Figure 3 shows the measured power coefficient at turbine inlet temperature of $T_{t3} = 800$ °C which is plotted against outlet flow coefficient $\varphi := 4\dot{m}/\rho_2 u_2 \pi D_2^2$. In Eq.3, $\lambda_{th}$ refers to the theoretical work of a specific machine according to Euler’s equation dating back to 1775, which is expected to be unique and feature a straight line. However, the power coefficient curves in Fig.3 show different characteristics! The increased slope of curves towards low flow coefficient refers to parasitic losses (see [1] and [2]) which lead to higher work input at low flow coefficient. Further, Fig.3 exhibits increased power coefficient with decreasing speed. The reason for this is not a Reynolds or Mach number dependence of the parasitic losses but change of density due to pressure ratio at the rotor and different pressure recovery in the radial vaneless diffuser. Change of density is connected to rotational speed and results in different values for outlet flow coefficient at different speed. This implies horizontal shifting of power coefficient curves at different speeds and we observe increased power coefficient! The head transfer due to non-adiabatic turbocharger operation which we consider to be dominating at low
speed can clearly be seen in Fig.6. Further, we expect that heat leads to apparent increase of work input while parasitic losses cause power consumption and raise the real work input. In order to prove this, we compare both the work input and output of the compressor at different levels of heat transfer achieved through different turbine inlet temperatures. From Fig.3 and Fig.5 we record higher power coefficient at higher turbine inlet temperature. At the same time we observe unchanged work output of the compressor which is characterized by the pressure ratio \( \Pi_t \) in Fig.2. Hence, the increased power coefficient from heat addition does not perform useful work. This argues for apparent increase of work input due to heat transfer. This also implies that the air is mainly heated up after the compression process. Otherwise deterioration in pressure ratio would be expected which refers to diverging isobars in an h-s-diagram. Unchanged pressure ratio \( \Pi_t \) and the corresponding isentropic change of state means unchanged pressure coefficient \( \psi = 2[h_{ts}] / u^2 \) so that the consequence of the apparently increased power coefficient \( \lambda \) is apparent deterioration of efficiency (see Fig. 4):

\[
\eta_{C,\text{app}} = \left[ \frac{h_{ts}}{\Pi_t} \right] = \frac{\psi}{\lambda} = \frac{\psi}{\lambda_{th} + \lambda_{par} + \eta_{C,\text{ad}}}. \quad (4)
\]

From Fig. 4 we recognize that the influence of heat transfer on efficiency decreases with increasing speed. At high speed the efficiency curves do not vary with turbine inlet temperature. The apparent deterioration of efficiency due to heat is maximal at lowest speed. We characterize a vertical shift of the peak efficiency \( \eta_{opt} \) of each low speed curve with turbine inlet temperature. Hence, the efficiency measured at \( T_{13} = 300 \, ^\circ C \) best matches the real efficiency i.e. adiabatic efficiency, considering that a smaller amount of heat is transferred to the compressor due to reduced temperature gradient. However, it is to note that even at \( T_{13} = 300 \, ^\circ C \) there is still significant heat addition at low speed (see also Fig.5). We conclude that the adiabatic efficiency is of a higher level and assume further vertical shifting of efficiency curves at low speed. So it is necessary to characterize an upper limit for the vertical shifting in order to determine the peak efficiency \( \eta_{opt} \). In the following we introduce three different approaches to gain the real efficiency from manufacturers’ performance maps.

1. First method is most easy to apply. It is based on the experience that the real peak efficiencies are all aligned on one single straight line in the \( \eta-\varphi \)-plane.
2. Second method takes an adiabatic power coefficient vs. flow coefficient curve as reverence. This reference curve follows from an analysis of measurements at different temperatures and rotational speeds.
3. The heat flow follows directly from temperature profile measurements by applying Fourier’s heat flow law (see [9]).

In the following all three methods will be applied and the results discussed. Already at that point the different needed effort of the three methods should be made clear. The first method needs only the apparent efficiency vs. outlet flow coefficient given by the turbocharger manufacturer. The remarkable point about the method is its simplicity in application. The second method needs the power coefficient measured at different temperature levels. This is usually not given by the manufacturer. The third and last method applied here and introduced in [9] needs surface temperatures, which are again usually not available.

The fundamental message is that all three methods give similar appropriate results in terms of validation. This can clearly be seen for the test case shown in Fig.7.

2.1. First Method: Scaling the Apparent Efficiency to the Real Efficiency

Considering the peak efficiency \( \eta_{opt} \) at high speed, i.e. high Mach number, it is obvious that the peak efficiency points all are lined up on a single straight line as shown in Fig.4. Taking this observation as a rule, low speed curves up to \( 0.59 \, u_{\text{max}} \) should be ignored at the moment. Later we will show how the low speed curves fit into the overall picture by
considering the real efficiency. It can be seen that there is a linear shift in the location of peak efficiency \( \eta_{\text{opt}} \) towards lower flow coefficient \( \varphi \) with increasing speed \( u \) or Mach number. In one point there is agreement with the scaling theory given by Pelz et al. [3] (see also [4], [5]): higher peak efficiency \( \eta_{\text{opt}} \) is reached at higher flow coefficient \( \varphi \). Here the Reynolds number does not play the dominant role but the Mach number. In this context, we rely e.g. on Naumann [6]. Naumann points out that the drag coefficient of a sphere is solely a function of Reynolds number at low Mach numbers. With increasing Mach number the impact of Reynolds number gets more and more insignificant and finally disappears at high Mach numbers where the drag coefficient only depends on the Mach number. This general behavior is also true for the flow around the compressor blades.

Back to the connecting line regarding the linear shift of peak efficiency at high speed, it can be specified (see e.g. Fig.4):

\[
\frac{\Delta \eta_{\text{opt}}}{\Delta \varphi_{\text{opt}}} = \text{const.} \quad (5)
\]

Introducing dimensionless loss coefficient, Eq.4 can be rewritten by definition as \( (1 - \eta_C) := \frac{\psi_{\text{loss}}}{C} \). Based on the model of Pelz et al. [3] the total differential is given as:

\[
-\text{d} \eta_C = (1 - \eta_C) \frac{\text{d} \psi_{\text{loss}}}{\psi_{\text{loss}}} \quad (6)
\]

Terms of second order in inefficiency \( 1 - \eta_C \) are neglected. Assuming that the relative change in dimensionless loss coefficient \( \psi_{\text{loss}} \) results from the relative change in drag coefficient \( c_d \), we obtain the following relation for linear case [3]:

\[
-\Delta \eta_C \approx (1 - \eta_{\text{opt}}) \frac{\Delta c_d}{c_d} \quad (7)
\]

According to Pelz’s relation \( \frac{\Delta c_d}{\Delta \varphi} = \text{const.} \) (see Pelz et al. [3]), Eq.7 can be rewritten as:

\[
-\frac{\Delta \eta_C}{\Delta \varphi} \approx C \frac{1-\eta_{\text{opt}}}{c_d} \quad (8)
\]

which is the basis of the following procedure. \( C \) is a machine typical constant determined by the specific shape of the rotor. This confirms the observed linear shift of peak efficiency expressed in Eq.5. We assume that the change of density in the vaneless diffuser is small compared to the change of density in the rotor. Hence, the latter primarily determines the outlet flow coefficient.

Now from the trend of connecting line regarding the location of peak efficiency \( \eta_{\text{opt}} \) at high speed (marked with an arrow in Fig. 4), the real level of efficiency at low speed should be concluded. Here, the hypothesis is that the peak efficiency \( \eta_{\text{opt}} \) increases with decreasing speed while the shift occurs linearly. Hence, the real level of efficiency would be highest at lowest speed. (see Fig. 6 in comparison to Fig. 5 and the comparison of all three methods in Fig. 7.) As a result, a rule of thumb can be presented in order to estimate the real compressor efficiency at low speed based on manufacturers’ measured map and without any knowledge about the power coefficient:

First, the measured efficiency should be plotted versus outlet flow coefficient. Assuming that the effect of heat flow at high speed is negligible, the peak efficiency connecting line can be evolved based on the efficiency curves at high speed (at least two curves are needed providing two points to define a straight line). This connecting line would automatically meet all peak efficiencies which are not affected by heat addition. We find the peak efficiencies at low speed, which are affected by heat addition, below this connecting line. These efficiency curves need to be shifted vertically until the peak efficiency of each curve meets the connecting line. The value of outlet flow coefficient at peak efficiency does not change.

This guideline can be applied without knowing the critical speed which was identified at 0.71\( u_{\text{max}} \) for the turbocharger considered above. A reasonable
speed range should be provided in manufacturers’ maps.

The procedure of vertical shifting may include uncertainty regarding the assumption that the shape of the efficiency curves does not change due to heat flow. The shape of the corrected efficiency curve by vertical shifting exactly corresponds to the shape of the apparent measured efficiency curve. However, the consistency in Figure 7 indicates reasonable level of uncertainty.

2.2. Second Method: Adiabatic Reference Power Coefficient versus Flow Coefficient

The second method is based on measurements at different temperatures and rotational speeds. In order to verify the hypothesis, the efficiency curves at low speed should be reevaluated comparing the increased power coefficient curves shown in Fig. 3 with adiabatic power coefficient. The latter includes the theoretical work input and parasitic losses:

$$\lambda_{ad} := \lambda_{th} + \lambda_{par} = \lambda - q_{c+}. \quad (9)$$

We use the adiabatic power coefficient to define the real efficiency of compressor adjusted for heat:

$$\eta_{c} = \frac{\psi}{\lambda_{ad}} = \frac{\psi}{\lambda - q_{c+}}. \quad (10)$$

Correction of efficiency based on power coefficient considerations has already been applied in other works. Sirakov and Casey [7] propose an iterative approach deriving the theoretical power coefficient (corresponding to the adiabatic one) from an approximate analysis introduced by Casey and Schlegel [8]. This approach makes use of Euler’s equation and only allows for disk friction power as parasitic loss. There are two uncertainties about this approach. First, parasitic losses are additionally made up of recirculation loss and leakage loss. The power consumption and accordingly the increase of power coefficient due to recirculation is reported to be dominating disk friction and leakage at low flow rate by Qiu et al. [1] and under entire working conditions by Oh et al. [2]. This should be considered when deriving the theoretical work input. Second, the approach makes use of a constant dimensionless heat transfer coefficient which is regarded as critical in the same paper. The constant value is supposed to be valid for all working points of an individual turbocharger.

The approach of this work bases on test data generated on three different turbochargers; each exposed to three different levels of heat transfer. The correction of heat is carried out comparing the power coefficient at the different heat levels. In doing so, Fig. 4 shows that 0.71$u_{max}$ is the lowest speed at which the efficiency does not change due to heat transfer. Though, we observe discrepancy in power coefficient curves comparing the curves at 0.71$u_{max}$ and the one at $u_{max}$ (see Fig.3). So, this discrepancy cannot be due to heat, but due to change of density and resulting change of outlet flow coefficient, as already discussed above. Similar behavior could be observed for all three compressors tested. As a result, only lower speeds need to be corrected from heat addition (The critical speed 0.71$u_{max}$ is valid for the turbocharger used here and can vary for other turbocharger design and size).

Further, the turbocharger operation at $T_{t3} = 300^\circ C$ is exposed to a significant lower level of heat transfer compared to operation at 600$^\circ C$ or 800$^\circ C$, but still not adiabatic at low speed. This issue can be observed in Fig.5 as the measured power coefficient curves at 0.44$u_{max}$ and especially at 0.28$u_{max}$ exhibit significant discrepancy to the curves at higher speed. For estimation of the adiabatic power coefficient at low speed, however, the measured power coefficient curves at $T_{t3} = 300^\circ C$ can provide assistance. As already concluded, we know that the measured power coefficient curve at 0.71$u_{max}$ is almost free of heat transfer phenomena. Further, it includes the theoretical work input as well as parasitic losses. Hence, we assume the power coefficient curve at 0.71$u_{max}$ to be the adiabatic power coefficient for lower speeds based on the consideration that change of density gets more insignificant with decreasing speed. Thus, the real
efficiency results from Eq.10. Figure 6 shows the corrected efficiency map. Again, the connecting line regarding the location of peak efficiency $\eta_{opt}$ is inserted which exactly coincides with the connecting line in Fig.4. The connecting line meets the corrected low speed peak efficiencies, as well. This is a fundamental result of this work.

### 2.3. Third Method: Heat Flow from Temperature Profile Measurement

In a final step, we determine the heat flow directly from temperature profile measurements by applying Fourier’s heat flow law (see [9]). This method is not described in detail here and serves as validation. The fundamental message is that we record similar appropriate results regarding the level of efficiency for all three methods. The difference between the efficiency curve corrected by vertical shifting and the corrected efficiency curve using the heat model remains insignificant. We notice higher discrepancy in the shape of the corrected efficiency curve resulting from adiabatic power coefficient (cross symbol in Fig.7) particularly at higher flow. Nevertheless, the three methods, in particular, the rule of thumb can be applied to revaluate the apparent efficiency at low speed with a gain of about 20% points within an uncertainty range of ±2%.

It provides a simple approach for a fast and reliable estimation of efficiency at low speed based on manufacturers’ maps and without any a priori knowledge. It is to note that peak efficiency $\eta_{opt}$ at low speed will not increase without bound. The value of $\Delta\eta_{opt}$ shrinks with decreasing speed since $\Delta\varphi_{opt}$ attains smaller values due to lower pressure ratio. In appendix A the results of an alternative turbocharger are displayed.

### 3. Turbine Performance

Turbocharger overall efficiency (index o) is generally referred to adiabatic flow based on the control volume of test stand in Fig.1 and reads the ratio of isentropic change of state of compressor to turbine:

$$\eta_{o,app} := \frac{\dot{m}_C(h_{t_{3/5}}-h_{t_5})}{\dot{m}_T(h_{t_{3/5}}-h_{4/5})}. \quad (11)$$

The index with (') refers to the apparent values, i.e. non-adiabatic measured quantities downstream of the compressor and upstream as well as downstream of the turbine (see also Fig. 9 and Fig. 11). Further, turbine efficiency is introduced as the combined aerodynamic and mechanical efficiency:

$$\eta_{TM,app} := \frac{\eta_{o,app}}{\eta_{C,app}} = \frac{\dot{m}_C(h_{t_{3/5}}-h_{t_5})}{\dot{m}_T(h_{t_{3/5}}-h_{4/5})}. \quad (12)$$

This definition of efficiency considers only the work transfer regardless of heat flow phenomena. In other words, it relies on the implicit assumption that the total enthalpy difference between the two measuring planes upstream and downstream of the turbine is entirely implemented in the rotor. Therefore, the temperature based calculation of power and efficiency involves uncertainties. On the one hand, we need to account for the real compressor efficiency from last section, i.e. the real work input (the effect is displayed in Fig. 12). On the other hand, heat loss from turbine body to the environment and the bearing housing decreases the working capacity of the exhaust gas prior to expansion at the rotor. This is investigated in the following sections.

![Fig. 7: Comparison of efficiency correction procedures; correction via vertical shifting, correction using adiabatic power coefficient and using the heat model in [9].](image-url)
3.1. Heat Loss and Reduced Working Capacity

We assume that heat transfer mainly occurs in the volute of the turbine. As a result, we specify heat loss prior to expansion. Figure 8 exhibits the magnitude of heat flow from turbine body based on temperature profile measurements (see Nakhjiri [9] for more details regarding the estimation of heat loss). We record substantial heat loss over the entire speed range. This reduces the total temperature of the gas which corresponds to its working capacity. In Fig. 8 heat loss is related to the square of speed in order to establish a relationship to the level of turbine work output depending on speed. The results suggest higher relative heat loss at low speed.

Keeping the reduced working capacity in mind, we need to investigate if the effect of heat loss on turbine efficiency corresponds to the results shown in Fig. 8. For this, we first analyze the effect of heat loss on flow quantities at turbine inlet prior to expansion. In Fig. 9 the turbine operation is resolved into two stages in series: heat transfer and work transfer. For the first stage we assume a lossless pipe flow with constant cross section which is exposed to heat flow \( q_T = \dot{Q}_T/\dot{m}_T \). This is regarded as a simplified volute flow.

![Fig. 9: Pipe flow exposed to heat transfer and Expansion process in series.](image)

The energy balance with lost heat \( q_T \) reads in differential form:

\[
\dot{h} + \frac{\frac{c + dc}{2} - \frac{c^2}{2}}{2} = q_T + dt.
\]

Neglecting second order terms and considering the equation of state \( h = e + p/\rho \) we obtain:

\[
de + \frac{1}{\rho} \frac{dp}{\rho^2} + cdc = dq_T.
\]

Hence, heat flow leads to change of internal energy, static pressure, density as well as velocity; also known from Thermodynamic literature. In order to determine the order of variations, we consider the control volume in Fig. 9 in a 0D analysis. The following relation results from the balance of mass and the balance of momentum:

\[
p_{3r} + \rho_3 c_3^2 = p_3 + \rho_3 c_3^2
\]

Additionally, taking the definition of sound speed \( a := \sqrt{\gamma p/\rho} \) and Mach number \( Ma := c/a \) into account, we obtain the following relations for the change of state only dependent on the Mach number:

\[
\frac{p_3}{p_3} = \frac{1 + \gamma Ma_3^2}{1 + \gamma Ma_3^2},
\]

\[
\frac{\rho_3}{\rho_3} = \frac{Ma_3^2}{1 + \gamma Ma_3^2},
\]

\[
\frac{T_3}{T_3} = \frac{Ma_3^2}{1 + \gamma Ma_3^2}.
\]
Now the reduction of total temperature results from the heat flow and can be directly determined from the balance of energy:

\[ \frac{T_{t3'}}{T_{t3}} = \left( \frac{\text{Ma}_3^2}{\text{Ma}_3^2} \right) \left( \frac{1+\gamma \text{Ma}_3^2}{1+\gamma \text{Ma}_3^2} \right) \frac{1}{2+(\gamma-1)\text{Ma}_3^2} \]  \tag{19}

Hence, with measured inlet temperature \( T_{t3'} \) and known \( q_T \) based on from [9], Mach number \( \text{Ma}_3 \) results from Eq.19 and Eq.20. Thus, the thermodynamic state at positions 3 and 3' are explicitly identified using Eq.16-18 as shown in Fig.10 for a test case. We record marginal change of pressure (below 0.1% points) which is not included in Fig.10. This allows treating heat loss as isobaric change of state in the turbine.

Hence, the expansion ratio does not change. However, the measured expansion ratio is to be referred to reduced total enthalpy difference. This is founded upon the diverging isobars as shown in the h-s-diagram in Fig.11. Reduced total enthalpy difference means reduced work output and has consequences for the turbine efficiency.

### 3.2. Apparent and Real Turbine Efficiency

Non-adiabatic measurement results in apparent turbine efficiency as already discussed at the beginning of section 3. Therefore, it is needed to analyze the turbine within distinct control volumes of turbine and bearing housing as sketched in Fig.1. According to definition in Eq.12, two points need to be taken into account in order to gain the real turbine efficiency:

- The real compressor efficiency, i.e. the real compressor work input which results in deterioration of apparent turbine efficiency. This refers to the fact that the real compressor work input is smaller than the apparent work input. Figure 12 displays the reevaluated efficiency curves (cross symbols).

- Heat loss which decreases the working capacity of the exhaust gas prior to expansion at the rotor. This means reduced energy input in turbine rotor for the measured expansion ratio. The total temperature at rotor inlet and the measured expansion ratio determine the isentropic state at turbine outlet and explicitly characterize ideal turbine work output. As a result, the real efficiency attains higher values according to definition.
Figure 12 compares the apparent turbine efficiency with the real turbine efficiency based on the considerations above. We detect the effects of both the corrected compressor efficiency and heat loss on the turbine efficiency. Taking into account the reduced energy input (and turbine ideal work output respectively) due to heat loss there is a general advance of turbine real efficiency compared to the apparent efficiency. The effect is slightly greater at low speed. However, the order of magnitude of efficiency revaluation in Fig.12 bears no relation to the magnitude of the normalized heat loss in Fig.8 comparing over the entire speed range. This arises from the curvature of the isobar $p_4$. The h-s-diagram in Fig.13 describes the expansion process for constant input $h_{t3}$ while at high speed $u > u_{ref}$ the expansion occurs from a higher pressure level (marked with filled circles). We point out that there is a difference in enthalpy level after expansion $\Delta h_4 = h_{4s} - h_{4s}$ if comparing the case with and without heat loss. This difference is larger at low speed and qualifies the effect of heat loss on turbine work if comparing low speed and high speed. Hence, the revaluation of turbine efficiency at low speed does not reach the order of magnitude of heat loss in Fig.8.

The general message is that the turbine efficiency will be significantly underestimated if heat loss is disregarded. We propose the following relation for the real turbine efficiency:

$$\eta_{TM} := \frac{m_C(h_{t2} - q_C - h_{t1})}{m_T(h_{t3} - q_T)(1 - \frac{\Pi_T^{\gamma}}{\gamma})}. \quad (21)$$

Here, $\Pi_T = p_{t3}/p_4$ stands for the turbine expansion ratio. The denominator should be computed according to the h-s-diagram in Fig.11, i.e. isobaric change of state $3' \rightarrow 3$ followed by isentropic change of state $3 \rightarrow 4s$. The nominator takes into account the real compressor work input. Further, the real turbocharger overall efficiency should be redefined in the same way:

$$\eta_o := \frac{m_C(h_{t2} - q_C - h_{t1})}{m_T(h_{t3} - q_T)(1 - \frac{\Pi_T^{\gamma}}{\gamma})}. \quad (22)$$

4. Conclusion

Turbochargers are generally measured under non-adiabatic conditions on hot gas test stands. However, the efficiency definitions do not allow for heat flow phenomena with regard to the control volume of the test stand. This leads to inappropriate computation of efficiency for both compressor and turbine which we refer to as apparent efficiency. Manufacturers’ performance maps generally reflect pessimistic valuation of efficiency. In this work it is shown that
the real compressor efficiency is the adiabatic efficiency. At low speed compressor efficiency in manufacturers’ maps is significantly underestimated. A fast and reliable guideline is introduced to determine the real compressor efficiency at low speed based on manufacturers’ map and without any a priori knowledge. The approach bases on an isobaric change of state due to heat addition which is analytically proved based on test data.

Further, it is shown that heat loss attains significant values on the turbine side. Heat loss is modeled isobaric prior to expansion in the rotor. The measured expansion ratio is referred to a revised total enthalpy difference. This results in turbine efficiencies which differ from the apparent efficiencies in manufacturers’ maps. We draw the following conclusion:

- At high speed where the effect of heat loss dominates, the real turbine efficiency is of a higher level than given in manufacturers’ maps.
- At low speed the effects of both heat loss and real compressor efficiency are significant. Here, the real turbine efficiency can attain lower values than given in manufacturers’ maps.

References


Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( a )</td>
<td>sound speed</td>
</tr>
<tr>
<td>( c )</td>
<td>velocity</td>
</tr>
<tr>
<td>( c_d )</td>
<td>drag coefficient</td>
</tr>
<tr>
<td>( c_p )</td>
<td>specific heat capacity</td>
</tr>
<tr>
<td>( e )</td>
<td>specific internal energy</td>
</tr>
<tr>
<td>( D )</td>
<td>diameter</td>
</tr>
<tr>
<td>( h )</td>
<td>specific enthalpy</td>
</tr>
<tr>
<td>( \dot{m} )</td>
<td>mass flow rate</td>
</tr>
<tr>
<td>( \text{Ma} )</td>
<td>Mach number</td>
</tr>
<tr>
<td>( P, p )</td>
<td>shaft power, pressure</td>
</tr>
<tr>
<td>( \dot{Q}, q )</td>
<td>heat flow, specific heat</td>
</tr>
<tr>
<td>( T )</td>
<td>temperature</td>
</tr>
<tr>
<td>( u )</td>
<td>rotational speed</td>
</tr>
<tr>
<td>( w )</td>
<td>specific work</td>
</tr>
</tbody>
</table>

Greek Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>( \gamma )</td>
<td>isentropic expansion factor</td>
</tr>
<tr>
<td>( \eta )</td>
<td>efficiency</td>
</tr>
<tr>
<td>( \lambda )</td>
<td>power coefficient</td>
</tr>
<tr>
<td>( \rho )</td>
<td>density</td>
</tr>
</tbody>
</table>


\[ \Pi \quad \text{pressure ratio} \\
\varphi \quad \text{flow coefficient} \\
\psi \quad \text{pressure coefficient} \\

\text{Subscripts} \\
1 \quad \text{inflow compressor} \\
2 \quad \text{outflow compressor} \\
3 \quad \text{inflow turbine} \\
4 \quad \text{outflow turbine} \\
ad \quad \text{adiabatic} \\
app \quad \text{apparent} \\
C \quad \text{compressor} \\
\text{max} \quad \text{maximum} \\
o \quad \text{overall} \\
\text{opt} \quad \text{optimal} \\
par \quad \text{parasitic} \\
ref \quad \text{reference value} \\
S, s \quad \text{shaft, isentropic} \\
T, t \quad \text{turbine, total} \\
\text{th} \quad \text{theoretical} \\
TM \quad \text{combined turbine mechanical} \\

\textbf{Appendix A} \\

In the following, the correction approach on the basis of adiabatic power coefficient is applied to another turbocharger. We recognize the linear shift of peak efficiency with outlet flow coefficient which we also obtain from simulation results using the compressor model proposed in Nakhjiri et al. [10].

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figA1}
\caption{Apparent compressor efficiency.}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figA2}
\caption{Real (corrected) compressor efficiency.}
\end{figure}

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figA3}
\caption{Real compressor efficiency (simulation results).}
\end{figure}