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Determination of the isentropic turbine efficiency due to adiabatic measurements and the validation of the conditions via a new criterion

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Abstract
The determination of the isentropic turbine efficiency under adiabatic and SAE boundary conditions is studied in this paper. The study is structured into two parts. The first part describes the possibility of measuring the isentropic turbine efficiency directly. Normally this is not possible in measurements conducted following the SAE J922 guidelines. Therefore, the experiments have been carried out under adiabatic conditions, and combined with improved measuring equipment. The results were compared with adiabatic computational fluid dynamics simulations of this turbocharger. In the second part, a new criterion is defined in order to evaluate the quality of the adiabatic measurements and compare them with standard measurements. The investigation has been carried out with multiple turbochargers ranging from very small to medium passenger car size turbochargers. In the end, a possible application for the criterion is given.

Keywords
Turbocharger, turbine, experimental

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Introduction

The testing procedure for turbochargers according to SAE J922 recommends a turbine entry temperature of 873 K.[1] This represents a good average value referring to actual operation temperatures of turbochargers, if applied to an engine. On the downside, it leads to several problems during the testing procedure. Due to the high turbine inlet temperature, the results are superposed by the influence of heat transfer. This effect has the biggest impact on low turbocharger speeds. The heat transfer causes difficulties to identify the aerodynamic behavior of the turbocharger at the test bench. Especially, the isentropic turbine efficiency cannot be obtained under those conditions.

There are mainly two reasons. First, the already mentioned heat transfer, which influences the actual value of the turbine outlet temperature. And second the complex flow field downstream the turbine wheel, where there is a turbulent flow field with high spatial temperature gradients. They cover a width of 50 K in computational fluid dynamics (CFD) simulations.[4]

There are a number of possibilities to look into this. One option is to determine the heat flows that occur during the process. Approaches using this method are shown for example by Bohn et al.[2] and Lückmann et al.[3] In addition to the heat transfer information, the friction power has to be determined as well. With operating models for friction and heat transfer, the isentropic efficiency can be calculated from the measurement values.

The approach is to minimize the heat flows during the actual measurement in order to achieve almost adiabatic conditions. Approaches that follow this idea are investigated by Baar et al.[4] and Baines et al.[5]

Adiabatic conditions and setup

SAE J922 and J1826 are today's standard to record turbocharger characteristic maps.[6] The agreement on this standard, including the recommended turbine inlet temperature of a constant 873 K, resembles the real-world application to a proper degree. It allows the comparison between different turbocharger test benches. But on the downside, the isentropic turbine

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efficiency (equation (1)) cannot be obtained using this measurement setup.

$$\eta_{Tis} = \frac{T_3 - T_4}{T_3 - T_{4is}} \tag{1}$$

The first reason is the occurrence of heat flow within the turbocharger and the heat flow towards the environment. This can be reduced by proper insulation up to a certain point, but, the issue still remains. Due to the significant heat loss, between the measurement planes of turbine inlet and turbine outlet, the measured T_4 drops low and this increases the isentropic turbine efficiency up to implausible values high above 1. This effect occurs at almost every turbocharger speed.

The heat flow can be minimized by applying adiabatic conditions. Therefore, the adiabatic conditions suggested by Baines et al.[5] were used for the initial investigations. The approach is set atop of two columns. The first aims to minimize heat transfer within the turbocharger by setting all dominant temperatures to equal values at every operation point. The key temperatures for the turbocharger are the compressor outlet temperature, which best represents the compressor housing temperature, and the turbine entry temperature, which dominates the turbine side and the oil inlet temperature that governs the bearing housing. During the experiment, those temperatures were adjusted at every operation point to equal values to fulfil the second equation. The turbocharger test bench offers the possibility to control the turbine inlet temperature, as well as the oil inlet temperature. Therefore, the governing temperature is the compressor outlet temperature, which is depending on the actual operation point of the turbocharger and cannot be controlled. All other temperatures are set to match this value. In fulfilment of the second equation, the temperature difference that drives every heat transfer becomes zero and, thereby, the heat transfer diminishes according to equation (3).

$$T_2 = T_3 = T_{Oil,mean} \tag{2}$$

$$\dot{Q}_i = \alpha_i \cdot A_i \cdot \left(T_i - T_j\right) \tag{3}$$

The second column is a thick insulation to minimize the external heat flow to the environment. To verify the functionality of the setup, additional temperature probes have been installed at the surfaces of turbine housing, bearing housing, and compressor housing. All in all, eight probes have been installed. Three of them are mounted on the turbine housing, another three on the compressor housing. The remaining two are installed on the bearing housing. The employed probes are small PT100 resistance thermometers, which have a sufficient temperature range for the experiment and provide a high resolution as well as a high precision. The setup for the adiabatic measurement is shown in Figure 1. The complete setup is insulated within the measurement planes that correspond to the system boundaries.

After taking care of the heat flow, only one obstacle of measuring the isentropic turbine efficiency remains—the unknown flow structure of the fluid downstream the turbine wheel. Due to the expansion of the hot gas and the rotation of the impeller, the flow field downstream shows a complex pattern of temperature gradients as well as a high swirl component. This makes it practically impossible to

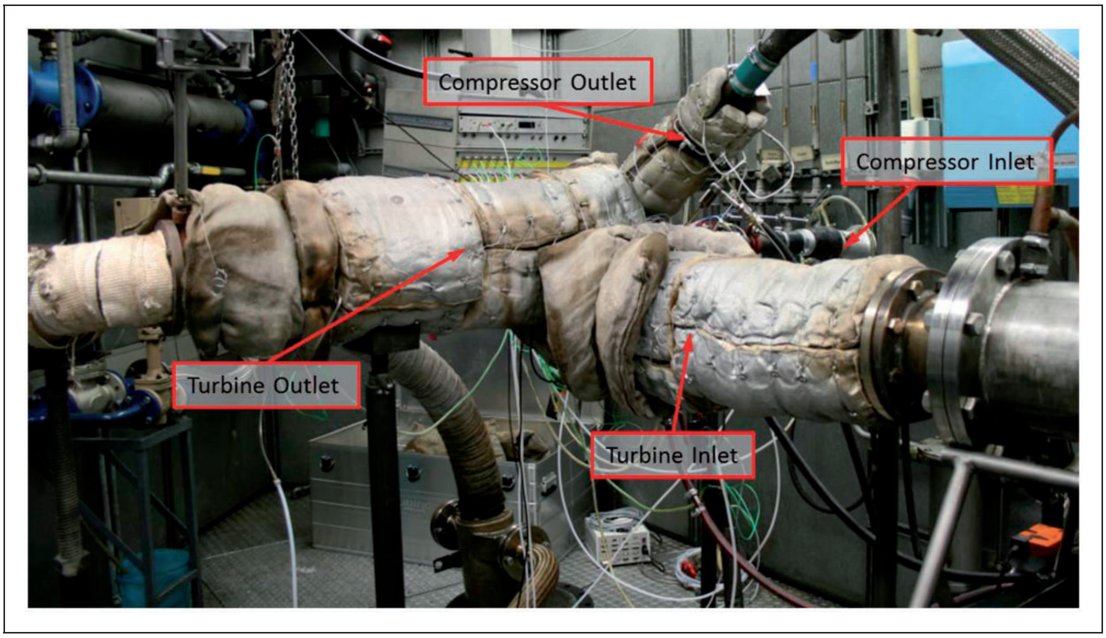

Figure 1. Turbocharger setup at the test bench including insulation.

determine the right turbine exit temperature with the standard three or more thermocouple approach.

It is necessary to employ some additional equipment into the setup to obtain a reliable value for the turbine outlet temperature. There is a trade-off. It is possible to measure the temperature as close as possible behind the turbine. This minimizes the heat loss problem, but the measurement equipment has to be able to deliver a reliable result despite of the complex flow field. A successful approach has been presented in the 2014[7] where a new sensor device, capable of measuring the mean value of a complete tube cross section, has been introduced.[4,7]

To improve the reliability of the standard temperature sensors, a mixing device has been installed in the measurement pipe following the turbine outlet. The effect of this device has been topic of an intense CFD investigation as well as an experimental validation, which is also published by Baar et al.[4] and the main results as well as a schematic drawing of the mixing device are displayed in Figure 2. The CFD results show how efficiently this mixing device reduces the temperature gradient in the flow field and enables a valid temperature measurement at its exit. Starting from a difference of 10 K across the flow field without the device, the remaining temperature difference decreases down to 1 K with the mixer. The experimental validation displayed similar results, which are presented in detail in Baar et al.[4] The device is installed in the measurement tube at a distance of 40 cm behind the turbine and homogenizes the flow field. On the downside, the mixer generates a small pressure loss. The pressure loss value ranges from 10 mbar at small flow rates to 120 mbar at max mass flow rate.[4] To compensate the pressure loss in the calculations, an additional pressure sensor has been installed in front of the mixing device.

One last aspect that has to be mentioned is that due to the temperature change across each speed line, the speed parameter (equation (4)) differs from operation point to operation point. The usual range observed during experiments is a change of about 100 $r/min/\sqrt{K}$ per speed parameter line. Finally, due to the change of oil temperature, the friction situation changes from operation point to operation point as well.

But since for this investigation the goal is to focus on the isentropic efficiency, this aspect does not affect the results.

$$n_{TC_{red}} = \frac{n_{TC}}{\sqrt{T_3}} \tag{4}$$

Experimental results

With all requirements meeting, the experiments were carried out at the turbocharger test bench of the Technical University of Berlin. An electric heating system with a maximum power output of 24 kW has been used to control the turbine inlet temperature during the adiabatic measurements. During the campaign one parameter had to be changed. The surface temperature probes delivered values, which showed that the match of the oil inlet temperature onto the compressor outlet temperature, led to mismatching bearing housing temperatures. Due to the heat from the friction power, the bearing was warmer than the housings of turbine and compressor. To rematch the surface temperatures, the mean oil temperatures was set equal to the compressor outlet. Figure 3 shows the final results for the fluid temperatures. The temperatures are displayed above the corrected compressor mass flow to transfer information into a known chart. The corrected compressor mass flow is

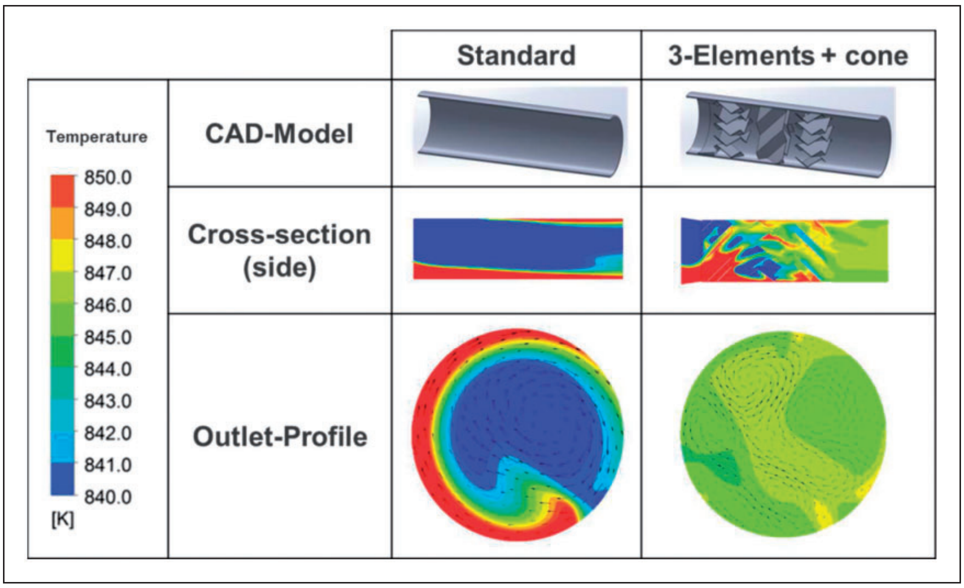

Figure 2. Scheme of mixing device and CFD results of the influence towards the distribution of turbine outlet temperature.

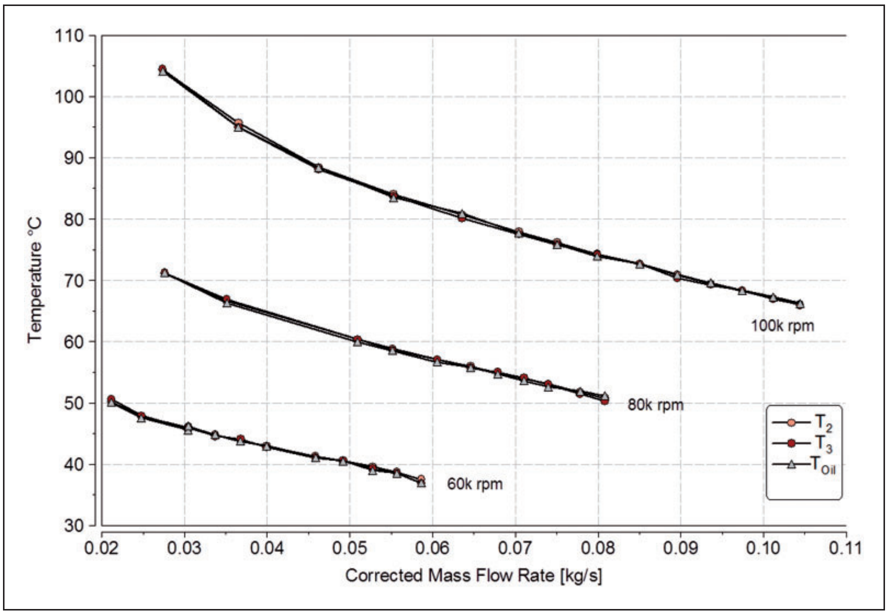

Figure 3. Mean values of fluid temperatures under adiabatic conditions for turbocharger speeds between from 60 k r/min to 100 k r/min.

calculated by using the equation recommended in SAE J922. The results underline that throughout the complete test the adiabatic conditions have been kept well. As mentioned before, additional temperature probes have been installed in the housing to ensure equal surface temperatures.

These results are shown in Figure 4. Compared to the fluid temperatures, the overall values are a few Kelvin lower. The difference is negligible at 60 k r/min, but increases with higher compressor temperatures to about 5 K at 100 k r/min at the surge line. The wall temperatures fit quite well with each other. Figure 5 shows the values for the overall differences in housing temperatures throughout the complete measurement campaign. Displayed are the minimal, the mean, and the maximal values for all probes for each operating point. The figure illustrates that the maximal difference between the hottest and the coldest point reaches 6.5 K in the worst operating point.

This indicates that despite the massive insulation there still remains a minor heat transfer into the environment as well as a minor internal heat transfer. Nevertheless, all the surface temperatures are in the same range, indicating that the approach overall does work well.

After considering the temperatures and surveying the functionality of the approach the more interesting objective, the determination of the isentropic turbine efficiency, can be pursued. The diagram in Figure 6 shows the different turbine efficiencies. To calculate the isentropic turbine efficiency equation (1) is employed. The effective efficiency (equation (6)) is calculated in accordance with the SAE standard.

$$\eta_{Cis} = \frac{\mathrm{T}_2 - T_1}{\mathrm{T}_{2\mathrm{is}} - T_1} \tag{5}$$

$$\eta_T = \eta_{Tis} \cdot \eta_{Tm} = \frac{1}{\eta_{Cis}} \cdot \frac{\dot{m}_C}{\dot{m}_T} \cdot \frac{\Delta h_{Cis}}{\Delta h_{Tis}} \tag{6}$$

The first thing that is apparent is the rise of the effective efficiency for the SAE measurement for the small turbocharger speed at 60 k r/min. This can be explained by the impact of heat transfer from the turbine towards the compressor. Thereby, the compressor outlet temperature rises and decreases the measured isentropic compressor efficiency calculated by the use of equation (5) decreases. This change leads to the rise of the effective turbine efficiency in equation (6), where the isentropic compressor efficiency is used in the denominator. This is a measurement error resulting from the assumption that the turbocharger can be treated as an adiabatic system. During the measurement under adiabatic conditions, this trend is not present, simply because there is no significant heat transfer into the compressor. Above the effective efficiency are the results for the isentropic efficiency. These values look plausible; they are about 10% points higher than the effective values. This is where the values are estimated to be, because they do not include friction. Furthermore, the course of the values for all turbocharger speeds indicates that the measurement must have been very close to the adiabatic level.

If there still had been a significant heat transfer on the turbine side, the isentropic turbine efficiency values were supposed to be much higher. Very assuring was the final comparison of the measured data with the adiabatic CFD simulation. The results of the simulation are almost exactly at the same level as the measurement results, and underline the approach.

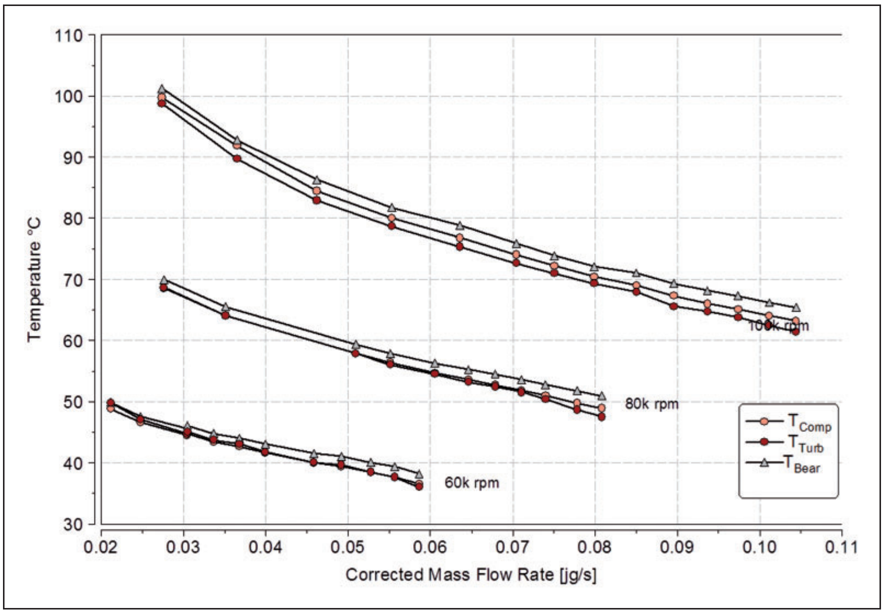

Figure 4. Mean values of surface temperatures under adiabatic conditions for turbocharger speeds between 60 k r/min and 100 k r/min.

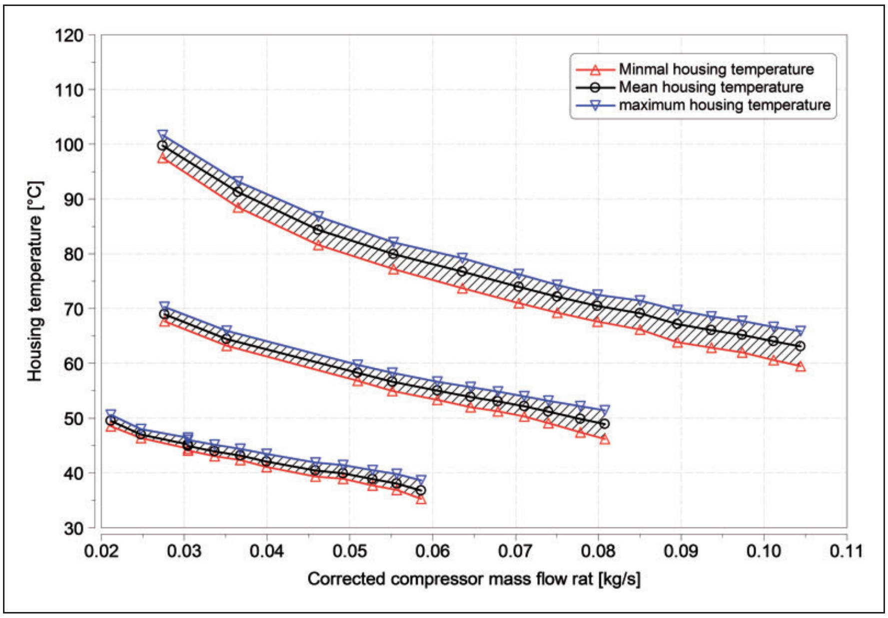

Figure 5. Comparison of overall housing temperatures.

Definition of the new criterion

The measurement campaign under adiabatic conditions led to reasonable isentropic turbine efficiencies. But there is still missing a criterion to determine, if the given measurement fulfils adiabatic conditions to a sufficient level or not. There exist already some approaches to define an adiabatic criterion e.g. the χ-criterion developed by IAV Berlin,[8] but to develop a simple criterion, without detailed knowledge of the construction of the turbocharger, to determine whether a measurement is adiabatic or not, it is important to decide for a measured variable that can be used, to compare different measurement conditions with one other. In order to achieve this, at first it is necessary to look into the measurement signals, which are given by the turbocharger test bench. The primary objective is to find a base solely pillowed upon signals that are reliable. In this case, reliable means those quantities must not be affected by the heat transfer that innately occurs in every operating turbocharger. This leads to the quantities on the system inlet boundaries. Because on both system entries there is no heat transfer that influences the measurement values. Any prior heat transfer may happen, but is not affecting any equation regarding

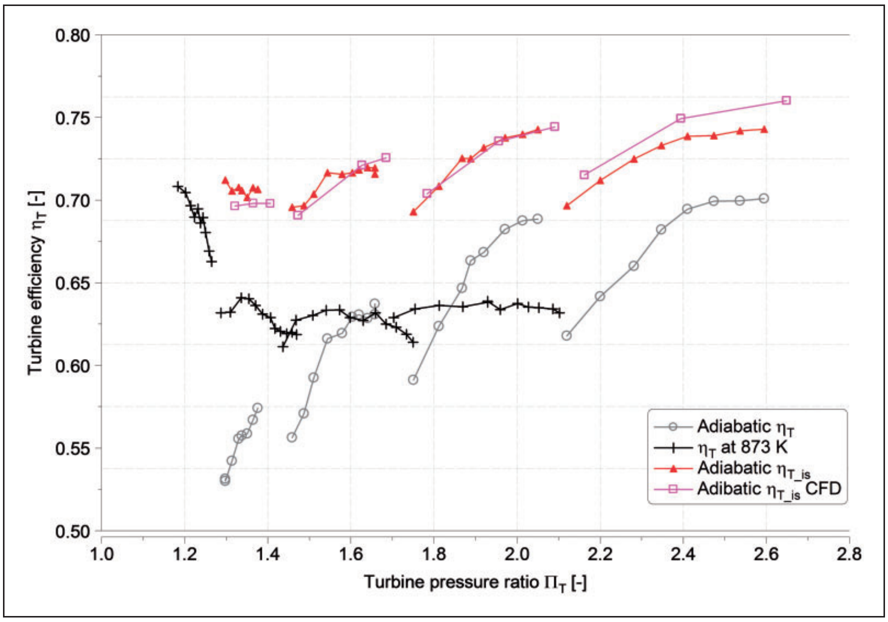

Figure 6. Characteristic turbocharger maps for adiabatic conditions and at 600°C for 60 k r/min to 120 k r/min.

the observed system. To compare properties of the turbocharger turbine under different conditions, usually the reduced turbocharger speed and other reduced quantities are employed. This decision is based upon the equivalent theorem of Mach.[9] But observing the compressor side on different turbine temperatures and taking reduced turbocharger speeds into account is difficult, because the power consumption of the compressor is strictly related to the real, the physical turbocharger speed. Therefore, a power-based observation seems more effective than the observation of reduced quantities.

In order to pursuit the idea of a power-based approach, first of all a reliable x-coordinate is needed. The measured, or calculated, variable is the isentropic compressor power. The big advantage is the prior demanded independency from all occurring heat transfers. The isentropic compressor power, as shown in equation (7), uses the compressor inlet temperature as the only temperature and this is the most reliable temperature available at the system boundaries of a turbocharger. Since it is only dependent on the ambient temperature

$$P_{Cis} = \dot{m}_c \cdot c_{pC} \cdot T_1 \cdot \left(\Pi_{C_{tt}}^{\frac{\kappa-1}{\kappa}} - 1\right) \tag{7}$$

The next reason for choosing it is based on the fundamental character of the compressor. Neither its isentropic compressor power nor its true isentropic compressor efficiency is affected by the state of the fluid. Both features are also completely independent from any heat transfer. The measured isentropic compressor efficiency, however, is affected by the occurring heat transfer from the turbine and the housing into the compressor, and the heat transfer from the compressor into the environment. But this is only a measurement error, because the temperatures employed to calculate the isentropic efficiency are affected by the heat transfer. As afore said, the true isentropic compressor efficiency remains unaffected. This is a minor simplification, since in reality the air inside the compressor heats by a few Kelvin. But this has only influence onto the specific heat capacity. The heat capacity is temperature dependent. But the highest difference of compressor outlet temperature between adiabatic and hot measurements found during the experiments is about 20 K. This rise changes the heat capacity only by about then 0.1%.

At one certain compressor operating point with one dedicated mass flow rate, a defined speed, and pressure ratio, the compressor will always need the same amount of power. This is valid regardless of the state of the turbine side. Due to this, always the same turbine shaft power is needed. This provides a basis to compare turbocharger measurements under different thermal conditions with each other.

To determine whether the measurement is sufficiently adiabatic on the turbine side, the total enthalpy flow over the turbine can be observed. As mentioned before the identical compressor operating point needs always exactly the same amount of turbine shaft power. In other publications it is already mentioned that the measured turbine power can be divided into two parts (equation (8)). One part is the aerodynamic power P_T, which goes into the turbine shaft, and the rest is energy, which is transferred in the form of heat transfer $\dot{Q}_T$ into the environment or adjacent turbocharger components. Concluding this, the total turbine power is used as axis of ordinates for this criterion.

$$\Delta\dot{H}_T = \dot{m}_T \cdot c_{pT} \cdot (T_3 - T_4) = \dot{Q}_T + P_T \tag{8}$$

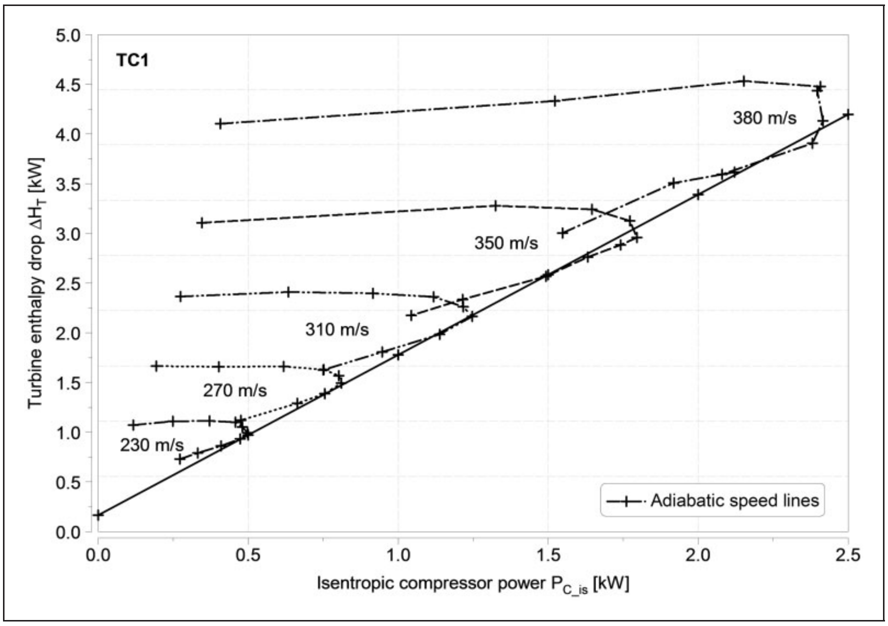

Figure 7. Isentropic compressor power and turbine enthalpy flow under adiabatic conditions.

The idea behind this is quite simple. In Figure 7, the plots of total turbine enthalpy drop over the isentropic compressor power are shown for a small automotive turbo charger. Additionally, the figure also shows the connection between the measurement points at maximum isentropic compressor power at each speed line as interpolated straight line (solid). This approach follows the idea that the point of maximum compressor power is part of all compressor characteristic maps. Furthermore, contrarily to the resistance line, or the choke line it is rarely affected by the test bench setup. If assumed that all the turbine power is used to be converted into compressor power, as soon as the required compressor power reaches zero, the turbine power has to be down to zero as well.

As mentioned before, the chosen comparison point itself is not important, it is only important, to use the same point for all speed lines. Investigations on using different compressor operation points will be discussed later. The interpolated line cuts the x-coordinate almost at zero, missing zero by 0.165 kW.

This indicates an almost adiabatic measurement. It is also remarkable, how well all the points of maximum compressor power fit into the straight line. To illustrate the impact of heat transfer in this criterion, the adiabatic measurements are compared to the standard measurements under SAE conditions. The results are listed and described in the following section.

Application onto multiple turbochargers

During recent experiments, multiple turbochargers have been tested under the adiabatic conditions defined in equation (2). All the turbochargers have been tested at the same test bench with equal adjacent geometry, equal instrumentation, and equal insulation. These facts ensure good comparison between the single measurement campaigns. The results for those investigations are shown in Figure 8. The red lines represent the hot gas measurements at a turbine entry temperature of 873 K according to the SAE J922. The black lines represent the adiabatic measurement recorded under the adiabatic definition introduced above. The solid lines show the connection between the points of maximum isentropic compressor power and the extrapolation towards zero isentropic compressor power. The remaining offset from the x-axis of the charts can be regarded as heat loss at zero compressor power. The adiabatic measurements show almost no remaining heat loss at zero compressor power. This can be observed for all four investigated turbochargers. The slopes of all four straights appear to be the same; in fact, they only differ by about 5%.

In the same manner, as for all adiabatic measurements the turbine power at zero isentropic compressor decreases to zero. It can be observed that for the SAE conditioned measurements, a certain amount of turbine power remains at zero isentropic compressor power. This can be interpreted as minimal heat transfer into the environment during the measurement and is a clear sign that the measurement does not fulfil adiabatic conditions. The remaining heat transfer power differs from 2 to 3 kW. The exact values and their comparison to the adiabatic measurements are shown in Figure 9. This figure underlines the huge difference in heat transfer and, thereby, the resulting increase in turbine power, or heat energy, needed to result in the same amount of power on the turbine shaft.

To solidify the approach, deeper investigations have been performed on those turbochargers, where additional measurement data had been available.

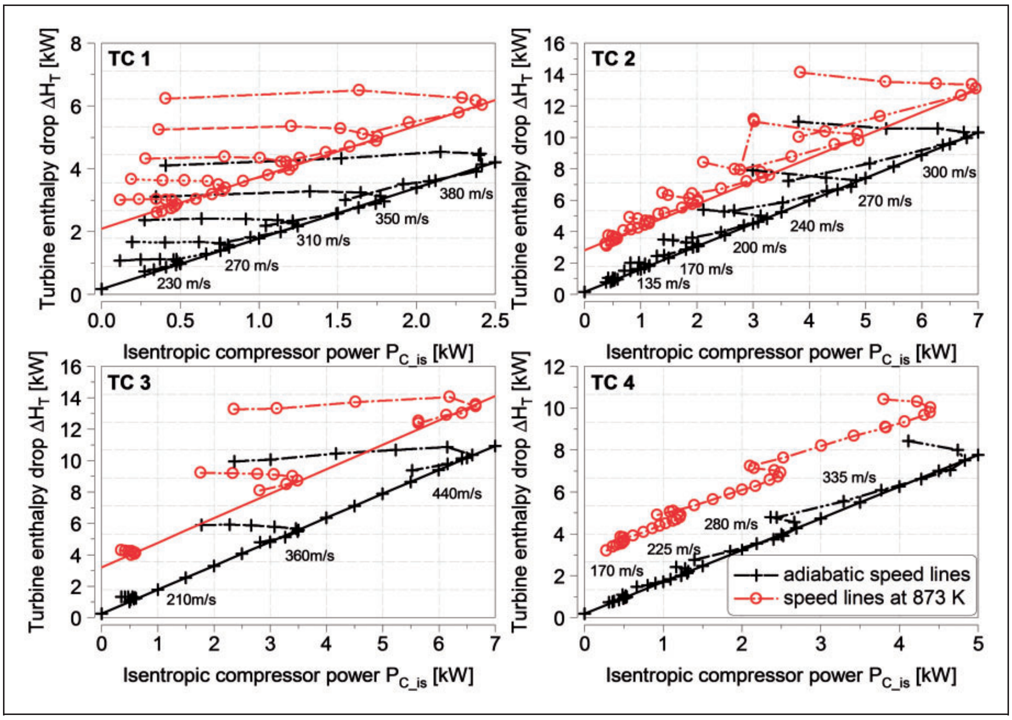

Figure 8. Adiabatic criterion for adiabatic conditions and for hot conditions applied on four different turbochargers.

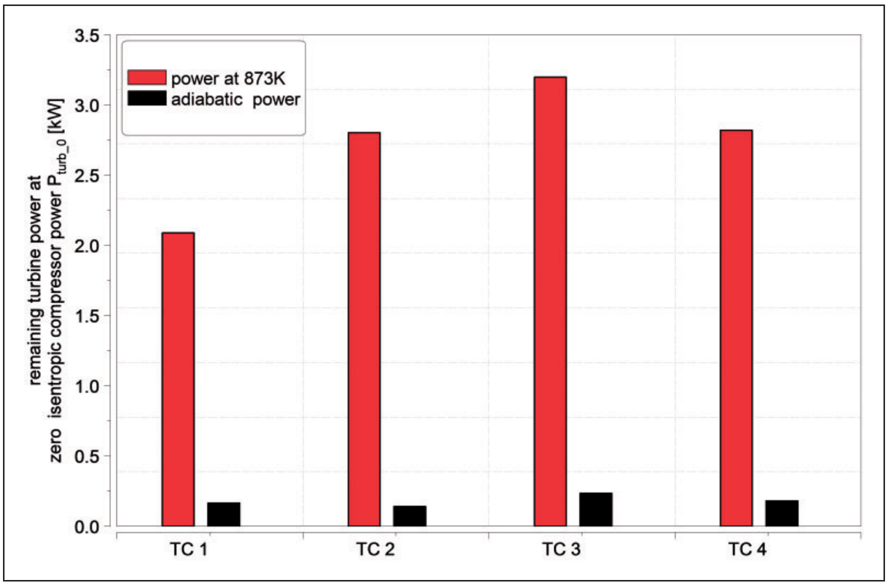

Figure 9. Heat transfer at zero isentropic compressor power for all investigated turbochargers.

The first variation discussed is a variation of turbine entry temperature. For two of the four turbochargers (TC1 and TC3), additional characteristic maps at different turbine entry temperatures had been recorded. For TC1, the additional turbine temperature was 473 K, and for TC3 the additional temperature was 1173 K. The charts of those additional maps have been added to the prior shown charts. Both variations are presented in Figure 10. For the smaller turbocharger TC1 the temperature being lower than 873 K but still higher than temperature needed for adiabatic conditions, the reference line can be found in between both lines and closer to the adiabatic line. This matches with our approach, since the turbine inlet temperature is closer to the adiabatic turbine inlet conditions. For TC3, the highest turbine entry temperature also induces the highest heat losses and, therefore, the highest straight line. For both temperature variations the criterion fits. This variation stresses that the established criterion works on a wide range of turbine entry temperatures and that it can also be applied to a wide range of turbochargers.

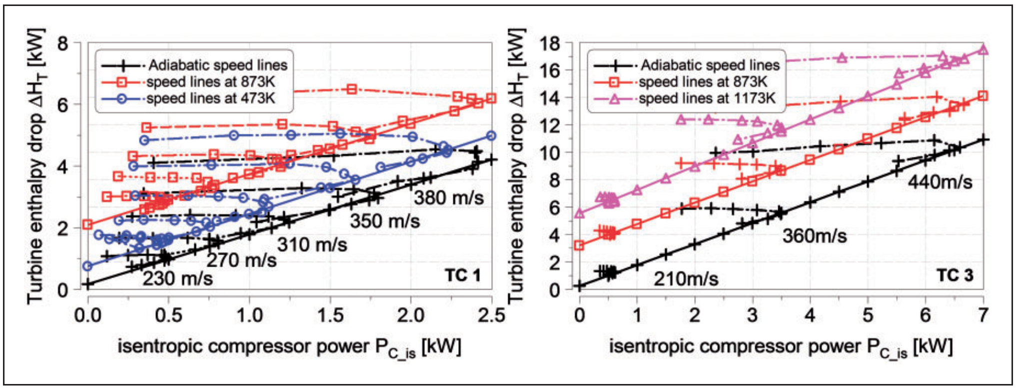

Figure 10. Turbine inlet temperature variation for two turbochargers (TC1 left with additional 473 K and TC3 right with additional 1173 K).

Conclusions

In its first part, this publication delivers the possibility to directly measure the isentropic turbine efficiency. The employed setup gives a direct and unhindered way to investigate the aerodynamic properties of a given turbocharger. The comparison with the numerical investigations shows that the results are plausible.

In the second part, the new criterion has been introduced and the functionality of the criterion has been shown for multiple turbochargers and with multiple turbine entry temperatures. The approach is completely power based and, therefore, offers the opportunity to easily compare turbine measurements under different boundary conditions. The approach reduces the turbocharger into a black box model.

Outlook

Indeed the criterion offers a lot more. It is a pure power-based approach and will therefore work at all turbine temperatures, as already shown. The next idea can be to use this criterion to recalculate simple hot gas measurements and, thereby, to obtain the isentropic turbine efficiency without having to do adiabatic measurements at all. A possible approach would use the characteristic line drawn by the connection of the maximum isentropic compressor power operation points. It is possible to completely solve equation (8) by knowing the total enthalpy drop and the adiabatic need power and transform it into the following form (example for 873 K)

$$\Delta\dot{H}_{T873} = \dot{m}_T \cdot c_{pT} \cdot (T_3 - T_4) = \dot{Q}_T + P_{Tadiab} \quad (9)$$

$$\Delta\dot{H}_{T873} - P_{T\,adiab} = \dot{Q}_T \quad (10)$$

Now with $\dot{Q}_T$ available, a quasi-adiabatic temperature can be determined, which would occur, if the measurement at 873 K would have been adiabatic. This temperature is called T_{Q0} and is indicated as a heat transfer free turbine temperature. With this temperature, it should be possible to determine the isentropic turbine efficiency directly from hot gas measurements (equation (11)). This shall be the topic in our upcoming research works.

$$\eta_{Tis} = \frac{T_{Q0} - T_4}{T_3 - T_{4is}} \quad (11)$$

Furthermore, the slopes which are all about the same will be taken into consideration to find the reason behind this property.

Finally, if the information about the heat losses from the turbine under given boundary conditions can be determined from the data, this might be used to parameterize turbocharger heat transfer models such as the lumped capacity heat transfer model described in the work of Burke et al. This will also be a part of future work, because this is a unique possibility to set up a process that can be used to estimate the heat transfer of a given turbocharger, without the gigantic effort normally needed to gather all the information for the heat transfer models.

Declaration of Conflicting Interests

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Appendix

Notation

A	surface
c_p	isobaric heat capacity
h	specific enthalpy
H	enthalpy
$\dot{m}$	mass flow
n	turbocharger speed
P	power
$\dot{Q}$	heat flow
T	temperature
α	heat transfer coefficient
η	efficiency
κ	isentropic exponent
Π	pressure ratio

Subscript

C	compressor
is	isentropic
m	mechanic
$Q0$	heat flow free
red	reduced
s	static
t	total
T	turbine
TC	turbocharger
1	compressor inlet
2	compressor outlet
3	turbine inlet
4	turbine outlet