NEW PHENOMENOLOGICAL AND POWER-BASED APPROACH FOR DETERMINING THE HEAT FLOWS OF A TURBOCHARGER DIRECTLY FROM HOT GAS TEST DATA

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ABSTRACT

Heat transfer and inhomogeneous flows in the turbine inhibit the calculation of the isentropic efficiencies from hot gas test data using conventional methods. In previous works the basis for such a calculation has been built such as special measurement devices and a power-based approach for calculating the heat flows. The main objective of the present work is the calculation of the isentropic turbine efficiencies and further development of the approach for obtaining isentropic compressor efficiencies from hot gas test data. The turbocharger is described aerodynamically and thereby the friction power can be calculated by setting up the power balance. For validation of the compressor, adiabatic measurements are used as well as CFD simulations for the turbine. Reasons for the latter are changing inlet conditions that do not allow the transfer of adiabatic data to diabatic or hot conditions.

KEYWORDS

Turbocharger, heat modeling, power balance, isentropic efficiencies

NOMENCLATURE

CFD	Computational Fluid Dynamics
CHT	Conjugate Heat Transfer
ECP	Effective Compressor Power
ETP	Effective Turbine Power
Δh	Enthalpy drop/rise
η	Efficiency
ICE	Isentropic Compressor Efficiency
ICP	Isentropic Compressor Power
ITE	Isentropic Turbine Efficiency
$\dot{m}$	Massflow
P	Power
T	Temperature
TC	Turbocharger
THP	Turbine Heat Power

Subindex

0	Ambiance
1	Compressor inlet

2	Compressor outlet
3	Turbine inlet
4	Turbine outlet
C	Compressor
CW	cooling water
is	isentropic
T	Turbine
corr	corrected

INTRODUCTION

For the evaluation of powertrain concepts concerning energy efficiency and emissions, 1D simulation is the state of the art tool. It is necessary to create highly accurate and easy to build models of components like internal combustion engines, turbochargers and exhaust after-treatment systems. Since most internal combustion engines are now turbocharged, the modeling of turbochargers is becoming increasingly important, because of the influence the behavior of turbochargers has on the engine process. Detailed physical or semiphysical submodels are available for heat transfer, friction and combustion. For turbochargers, the model consists of measured standard maps without any information about heat transfer or friction, which shows that there is a huge gap between these two submodels and their depths.

The conventional way of the measurement of turbochargers on hot gas test benches is described in SAE J922 and J1826. Thermodynamically, it is orientated on realizable options for temperature measurement accuracy, flow influence, the experimental effort and capabilities, as well as a representative exhaust temperature in internal combustion engine applications. The heat transfer within the turbocharger as a result of high temperature gradients falsifies the isentropic efficiencies for both the compressor and the turbine by increasing the total enthalpy change. The basic assumption of adiabacity for calculating the turbocharger efficiencies in standard maps has to be reconsidered. For these reasons, the state of the art measurement procedures and the standard characteristic maps are not reliable. Especially at low speeds, the diabatic influence grows because the aerodynamic enthalpy change decreases while the transfered heat stays almost the same. This also applies to partial load operation of passenger cars. It corresponds to the relevant load spectrum for certification cycles and city traffic.

One way to decrease the influence of heat transfers is measurements under adiabatic conditions. This procedure is thermodynamically feasible on the compressor so that the isentropic efficiencies can be transferred to the hot gas measurements. On the turbine side, the operating points change significantly between these two experimental modes so that there is no possibility to assign and transfer the isentropic efficiencies. Another problem is the inhomogeneous flow at the turbine outlet, which needs special measurement equipment for reliable T_4 (temperature at turbine outlet) data described in section Bibliography/Prework. This is the critical value for determining the turbine isentropic efficiency directly from measured data. Only in adiabatic measurements does this procedure lead to real isentropic efficiencies. In diabatic or hot gas tests, the results are superimposed by heat flows and without the possibility to transfer the real isentropic efficiencies from adiabatic tests there is no knowledge about the quantity of outgoing heat. Another way to obtain the real turbine isentropic efficiencies is a CFD/CHT simulation, requiring geometrical knowledge and a high simulation effort. Other possibilities for heat flow determination are specific empirical approaches for the compressor, the turbine or the friction, to calculate the power balance. But these empirical approaches are commonly validated on spe-

cific turbochargers and have a limited prediction behavior on new ones.
To improve turbocharger modeling, this study introduces a method to calculate heat transfer of turbochargers using standard SAE maps. Thereby it is possible to calculate heat which derives out of the turbine and flows into the compressor group without knowledge of either the geometry or the material of the turbocharger parts. It is based on a reliable turbine exit temperature measurement, furthermore on a power-based calculation of the turbine heat flows. Its universality was shown on different turbochargers and turbine inlet temperatures in previous investigations (Baar et al. (2013) & (2014)). In this paper, the approach will be developed further. When applied additionally on the compressor, the approach is able to determine the heat flows from experimental data and from this the isentropic efficiency. So the turbine and the compressor are described aerodynamically. As a result, the friction power can be calculated. The advantage of the new approach is its simplicity and the determination of heat flows and friction power directly from experimental data without any specific empirical approaches, CFD simulations, or adiabatic measurements. A CFD simulation for the turbine and an adiabatic measurement for the compressor are carried out only for demonstrating the validity.

Bibliography/Prework

The objective of the paper is to develop an approach for the characterization of heat flows directly from test bench data without design knowledge and further without 3D simulation or adiabatic experiments. As a result, isentropic efficiencies are obtained and can be used as a starting point for further calculations. Other unknown variables can be determined afterwards. Together with transient experiments and calculated thermal inertias, thermal network models shall be created for use in engine process simulations.
In recent years various investigations have been carried out to determine the isentropic efficiencies for the compressor and the turbine. Cormerais et al. (2006) used adiabatic test data to determine the compressor isentropic efficiencies. These were transfered to the hot gas tests. Deviations in measured enthalpy flow or effective power were then explained by heat flows. For the turbine, this procedure does not work out because the inlet states change significantly. The operating conditions of the turbine inlet is tremendously important. Therefore another method was applied by extrapolating the turbine isentropic efficiencies. The exact approach was not described. With this information a power balance was set up to determine friction. Heat flows were then calculated with some assumptions. External heat flows were neglected as a consequence of insulated housings. Cormerais et al. (2009) took another approach a few years later by determining the friction using the hydrodynamic bearing theory and optimizing its correlations. For the compressor, the procedure was the same, so the turbine isentropic efficiencies are finally calculated.
Baines et al. (2009) compared hot and adiabatic test results concerning the lubricating oil temperatures. They assumed that the enthalpy drop in adiabatic tests was driven only by friction. A correlation was then applied to the speed dependent Reynolds and Strouhalnumbers. These correlations were adopted for hot gas tests and deviations explained by heat flows. The procedure for the compressor was similar to Cormerais et al. (2006, 2009). External turbine heat flows were determined by taking into account the internal oil and the compressor heat flows as well as the total turbine enthalpy drop. The resulting discrepancy was then represented by external turbine heat flows. External compressor and bearing housing heat flows were neglected and explained with the lower range of temperatures.
Burke et al. (2014) parameterized a thermal network model for the application in engine pro-

Figure 1: Three devices to record T_4; left: mixing device; middle: spider web sensor; right: TC ring (Baar et al. (2013))

cess simulation. Usually isentropic efficiencies are the starting point to calculate all missing heat flows. Afterwards, the specific model parameters can be calculated. But they used another procedure to obtain the parameterization data. Correlations of thermal parameters found on previous investigations on different sized turbochargers were scaled for the current device.
Serrano et al. (2015) utilized a special test bench with incompressible fluids instead of air and exhaust gases. With that procedure, they separated heat flows and expansion/compression effects related to the temperature change. Internal convection heat transfer coefficients were collected on standard hot gas tests. Stationary and transient experiments served as a basis for material parameters which were used for a thermal network model. A physical friction model was calibrated on the basis of adiabatic tests and the enthalpy drop in lubricating oil with the assumption that it is created by friction power. Isentropic compressor efficiencies were obtained with adiabatic data transfered to diabatic hot gas data.
In recent research projects and publications, Baar et al. focused on a new method to evaluate the turbine heat flows. The first work was published by Baar et al. in 2013. It focused on possibilities to record plausible and reliable values for the turbine outlet temperature T_4, because this is the only way to enable the determination of the turbine isentropic efficiencies directly from measured data. In the paper three methods were introduced. The mixing device was employed to homogenize the flow field behind the turbine to allow the recording of a plausible value of T_4 with three standard thermocouples. The second device was a newly created concept called the spider web sensor. It was based on a resistance thermometer with a resistor consisting of a spun platinum wireweb. Since the flow affected the wire over the whole length it was possible to derive a representative mean value for turbine outlet temperatures. The TC ring was installed behind the turbine outlet and consisted of six thermocouples with different insertion depths. It was rotated during the measurement through 360° using 60° steps. Thus virtually 36 thermocouples delivered information of the flow filed downstream from the turbine. This procedure gave a detailed temperature field of the turbine exit intersection and therefore a representative mean value for T_4. All three devices were used in the same setup in close quarters after the turbine under adiabatic and hot gas conditions. Additionally, the insulation of the setup was part of the investigations. It was changed in three steps from no insulation, to insulating only the turbocharger, up to an insulation for the turbocharger as well as for the measurement pipes, to

decrease heat transfer as much as possible. The results were very promising, because all three devices delivered similar T_4 results.
The second step was published by Baar et al. in 2014. The mixing device and the spider web sensor made it possible to determine the isentropic turbine efficiencies under adiabatic conditions. Two sets of conditions were considered in this investigation. The first was the classic idea of eliminating the internal heat flows of the turbocharger by setting the temperatures of the compressor, the turbine and the lubricating oil to equal values. The second idea was employed after evaluating the first data and recognizing that despite the massive insulation, the downstream region of the turbine still showed a significant heat transfer to the environment. Therefore the second idea focused more on the turbine, since the heat flow of the compressor seemed not to be that big. It resulted in the equilibrium condition of T_4 and T_0. During the experiments, despite some operation points at low turbocharger speeds and at the max flow compressor point throughout the whole map, both criteria could be met. The results were compared to an adiabatic CFD simulation and showed similar behavior.
Further investigations considered the impact of slightly violating the adiabatic conditions and the effect on the measured isentropic turbine efficiency. With the possibility to directly record the isentropic efficiency, the missing link was the possibility to compare measurements under different boundary conditions. Therefore, a new way to evaluate turbochargers was developed and was presented by Baar et al. (2015) and Zimmermann et al. (2016). Part of the approach was the comparison of adiabatic and hot test data with a common parameter, the isentropic compressor power. It separated the power absorbed from the turbine shaft and the amount of power going into heat transfer. The approach is described in detail in the New Approach section because the revision is necessary to prepare for the next steps presented in the section Further Development. This approach gave the almost unique possibility to directly compare turbine adiabatic and hot gas measurements. The approach was tested on a couple of different turbochargers of different sizes and it was employed on wastegate turbochargers as well as on VGT turbochargers (Zimmermann et al (2016)). Furthermore, it allows calculating the real isentropic turbine efficiency from hot gas measurements by referring to the adiabatic measurements.

EXPERIMENTAL PROCEDURE

Turbocharger Test Bench & Setup

The turbocharger test bench used for the investigation belongs to the Department of Internal Combustion Engines at the Technical University Berlin. The layout of the figure is attached in section Illustrations and Tables. On this test bench it is possible to measure over a wide range for temperatures (T3: adiabatic/cold to 1050 °C), massflow (up to 1400 kg/h) and maximum mechanical compressor pressure (up to 4 bar) which provides the turbine massflow. The instrumentation consists of thermocouples for exhaust and Pt100 for air temperature measurements as well as for other fluids like lubricating oil and cooling water. Temperatures and pressures are generally measured before and after turbo machinery with three averaged azimuthal thermocouples and four azimuthally arranged pressure pipes to take into account the mixed flow field. Because of the special mixing device for homogenizing the turbine outlet flow there is an additional pressure sensor before and after it for detecting the streaming losses. All sections of measurement were insulated to minimize the heat losses between the turbocharger and sensors.
A wastegate turbocharger was fully instrumented with surface thermocouples. Five sensors were positioned on the turbine housing, four on the bearing housing, three on the compressor backplate, and four on the compressor housing. An additional thermocouple was placed bet-

ween the turbine and the bearing housing because of the necessity as a boundary condition for a future CHT analysis which will be performed for validation purposes. Furthermore the turbine was painted black for the emissivity coefficient to become independent from temperature influences (ideal black body). This is a tribute to CHT boundary conditions.

Tests under Adiabatic Conditions

An important procedure to determine compressor isentropic efficiencies is the measurement with near adiabatic boundary conditions. Several ways to realize these conditions have been discussed (Baar et al. (2014) & (2015), Zimmermann et al. (2016)). In this paper, the proposal is followed to adjust the turbine inlet temperature to the value of the mean oil temperature (T_3=$T_{Oil\ mean}$). Another condition is the alignment of the turbine outlet and the compressor inlet or the ambient temperature (T_4=T_0). Other approaches involved alignment of turbine inlet, oil and compressor outlet temperatures (T_3=$T_{Oil\ mean}$=T_2). All conditions are restricted by the maximum oil temperature of 150°C, so the range of possible turbine inlet temperatures is restricted. With primary results it turned out that the operating range was rather small, which led to the first procedure. There is also the possibility to fulfill all the prescribed conditions but this procedure needs a closed loop operation on the compressor side to control its outlet temperature. Since the experimental effort was already very high and the adiabatic data is needed only for validation purposes, it was decided to apply only the first condition (T_3=$T_{Oil\ mean}$=T_2).

Tests under Diabatic Conditions

The standard measurements on the hot gas test bench were performed with various turbine inlet temperatures as well as standard measurements according to SAE J922 and J1826. Furthermore variations of oil and cooling water temperatures were measured to study the influences on heat flows. Baar et al. (2014) showed that insulations did not prevent heat flows in hot gas tests. There is no reliable way to create near adiabatic conditions in hot gas tests with insulation. Nevertheless the measurement pipes were insulated for reducing the heat flows in the intersections. The pipes for providing the lubricating oil were also insulated because the enthalpy change is affected by heat transfer to the ambiance. In this paper the new approach is focused and discussed in detail on a standard SAE map. The measured variations of T_3, T_{Oil} and T_{CW} will be part of future investigations.

NEW APPROACH

The first idea was it to find a criterion which makes it possible to evaluate measurements under adiabatic conditions in terms of real adiabacity, an ideal state. An abscissa was needed which is unaffected by heat flows. Since heat transfers occur after expansion at the turbine and before compression at the compressor, effective power is not convenient. Isentropic power is more appropriate because only inlet temperatures are needed for the calculation. Turbine isentropic power is dependent on the inlet temperatures, which change drastically under different operating conditions. For that reason, the only acceptable parameter was the isentropic compressor power. The work was motivated by turbine related considerations because the difficulty was to compare reduced values (similarity of Mach) with physical ones. This led to a power-based approach, since the effective turbine power is a purely physical parameter in contrast to the reduced massflow or the reduced speed. The effective turbine power was then chosen for the axis of ordinates. It revealed interesting characteristics which can be seen in Figure 2. The new approach was previously described by Zimmermann et al. (2016). They found a characte-

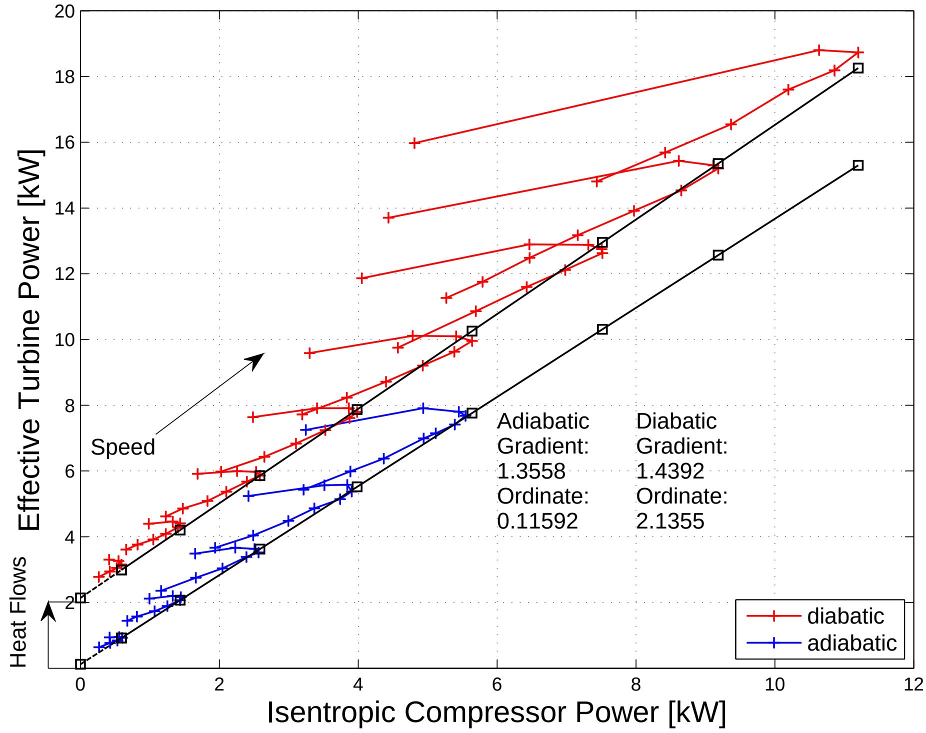

Figure 2: Turbine power-based consideration

ristic phenomenon for effective turbine power which can be made visible with the power based approach. The red lines represent experimental data under diabatic conditions (hot gas tests). The speedlines rise from bottom left to top right. The adiabatic lines are in blue. The reason for the small number of adiabatic speedlines is the way the enthalpy arises. For experiments at adiabatic conditions, the turbine inlet temperatures are very low. Hence the needed turbine effective power to supply the compressor is provided by a higher massflow in contrast to tests under diabatic conditions and high gas temperatures. The restriction for the maximum turbine massflow is limited by the compressor maximum pressure of 4 bar (compressor 2 in Figure 7). In comparison, the blue and red curve progressions show a remarkable similarity. This phenomenon can be used to recalculate the measured effective turbine power to aerodynamic power without heat influence. The information gained by comparison of the different conditions is important for understanding the characteristics of that phenomenon. The approach was investigated on four different turbochargers and showed its validity in terms of the prescribed similarity between adiabatic and diabatic turbine speedlines (Zimmermann et al. (2016)). When considering the points of maximum isentropic power for each speedline, the appearance lead to the assumption of near linearity. The black lines emphasize this property. The motivation for regarding the maximum isentropic power is the conspicuous position in the graph and the fact that these values are unaffected by different test benches, because they lie somewhat near the middle area of the maps. The restricting factor is the compressor map with its boundaries being represented by the surge and resistance line, which can differ between test benches.

Another similarity occurs between the two lines connecting the respective maxima. The gradient of the lines is quite in the same range. The combination of the similar characteristics makes it possible to correct the diabatic values. When extending the black lines until the axis

of ordinates (dashed line) it conforms to an isentropic compressor power (ICP) of zero. That also means a speed of zero and therefore theoretically no friction power. Every deviation from zero at the axis of ordinates must be induced by heat flows. The value of the ordinate for the adiabatic test is 0.116 kW. This is ideally considered the deviation from real adiabatic conditions. The turbine heat power (THP) of the diabatic measurement is 2,14 kW. Hence the criterion was originally thought to estimate the quality of the adiabatic measurement to the ideal state, it was able to do so likewise with other test data. At 11.5 kW of ICP the turbine heat power THP has a value of approximately 3 kW. This difference to 0 kW ICP results from a different number of speedlines and the linear interpolation of the points of maximum ICP. An increasing number of speedlines increases the gradient of the interpolated line and thereby the difference between adiabatic and diabatic data with rising ICP. If it were possible to measure the missing three adiabatic speedlines the gradient of the interpolating line would be larger and the difference smaller. With a smaller number of diabatic speedlines the difference would also decrease. The influence of the number of speedlines on the THP, can be explained with the ETP rising slightly greater than linear with rising speed. That means that the difference of adiabatic and diabatic data is increasing with rising numbers of speedlines. An important question is how this behavior affects the correction of the diabatic data. The ordinate gets smaller with more speedlines because the gradient of the interpolation line gets larger. It leads to the conclusion that the interpolation method should change from linear to a higher degree polynomial. This will be part of future investigations.
Furthermore, heat losses in the mixing device downstream the turbine are considered in the power-based approach. The directly measured effective power from hot gas test increases with these heat losses. The adiabatic data does not change with the mixing device although perfect adiabacity is not possible. The speedlines and the ordinate of the diabatic ETP rises with mixing device heat losses and therefore the amount for the correction as well. It is not possible to quantify the heat losses because it is not possible to measure the temperature before the mixing device. The only possibility is a CHT simulation or an experimental heat flow investigation of the test bench without a turbocharger. It would be then possible to get values for temperatures before the mixing device and to quantify the heat losses.

FURTHER DEVELOPMENT

Is the approach applicable on the compressor? In Figure 3 the same kind of investigation is carried out on the measurement values of the compressor. The abscissa is the ICP. For the axis of ordinates the measured ECP is chosen. The diabatic and adiabatic test data shows in analogy to the turbine a quite similar curve progression for identical speedlines. On the compressor side, the issue with a limited number of speedlines occurs also because the need high massflows lead to the pressure limit for the mechanical compressor, which provides the combustion chamber and the turbine with exhaust gases. The ordinate intercept values from compressor diabatic data are very small compared to the turbine values. This is quite reasonable because just a small amount of the heat from the turbine reaches the compressor. Nevertheless the compressor diabatic power has a tenfold higher ordinate intercept than the diabatic data. The gradients of the black lines which connect the maxima of the isentropic compressor power are likely at the same range, although the diabatic data has a slightly larger gradient. Altogether the characteristics observed for the turbine also apply for the compressor. The value for the incoming heat flows is 0.025 kW in the adiabatic case and 0.24 kW for the diabatic data. The positive value on the compressor side describes incoming heat because the higher enthalpy rise affects the measured

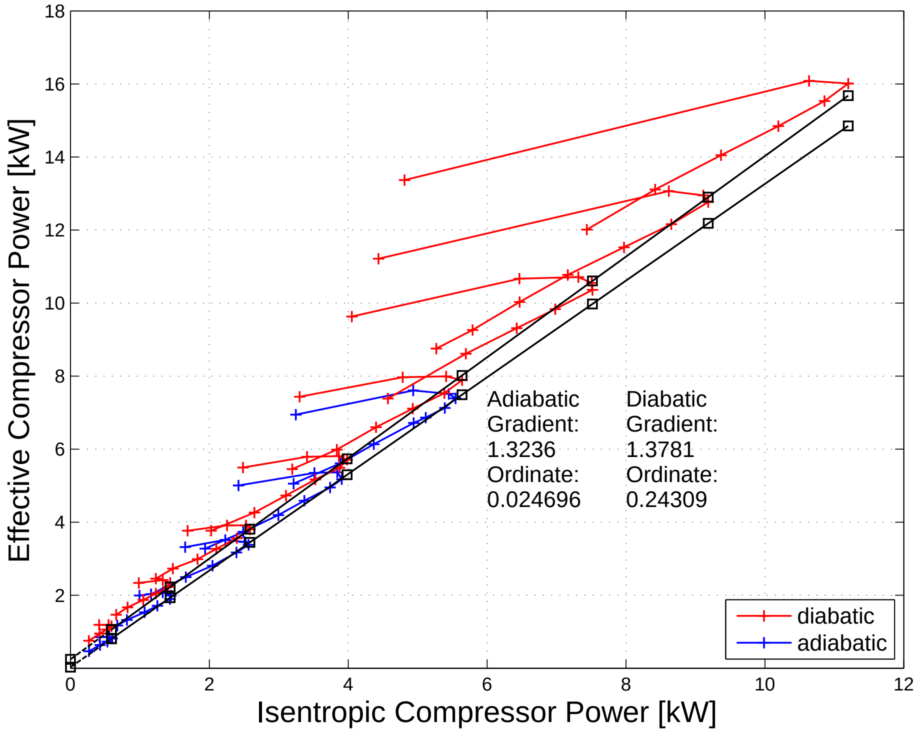

Figure 3: Compressor power-based consideration

ECP. Heat flows which occur upstream or downstream the compressor can not be separately quantified. They just increases the total enthalpy change so that the ordinate intercept increases as well and it leads to a larger correction value. For the turbine the opposite is true. Outgoing heat flows increase the measured ETP because it has a higher enthalpy drop. With all that information collected, the powers can be recalculated. With a quite simple shift of the diabatic data the black lines of maximum isentropic powers goes through the ordinate of zero. By means of adiabacity, this should theoretically be the ideal state. For this purpose we actually do not need measured adiabatic data for calculation. The knowledge about the phenomena is sufficient to calculate the heat flows. The measured adiabatic data is only used for validation purposes. The difference between the interpolating lines of diabatic and adiabatic data is showing the same phenomena with rising ICP. At 0 kW of ICP the THP is 0.24 kW and at 11.5 kW it is about 1 kW. The linear interpolation method of points of maximum ICP underlies the dependency between the number of speedlines and the ordinate intercept as well as the gradient. In future Investigations a higher degree polynomial will be applied.

Nevertheless we have at this point effective powers which are corrected in terms of heat flows. These values theoretically describe the aerodynamic behavior of the compressor and the turbine. The corrected powers can be used directly to recalculate the isentropic efficiencies using

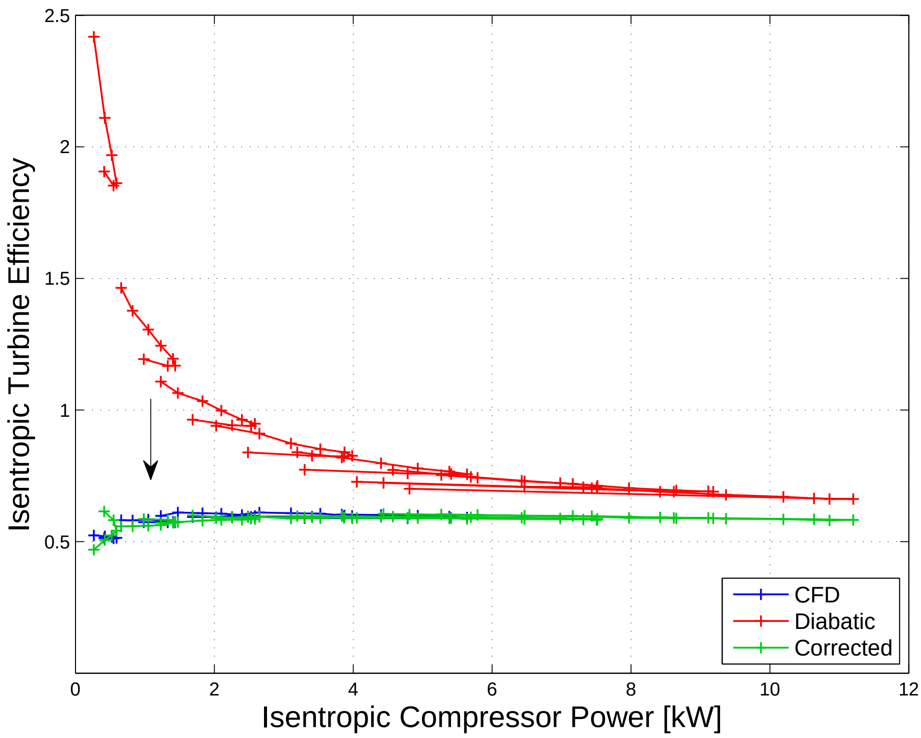

Figure 4: Isentropic turbine efficiencies

equations (1) and (2).

$$\eta_{Cis_{corr}} = \frac{\Delta h_{Cis}}{\Delta h_{C_{corr}}} = \frac{\frac{P_{Cis}}{\dot{m}_C}}{\frac{P_{C_{corr}}}{\dot{m}_C}} = \frac{P_{Cis}}{P_{C_{corr}}} \tag{1}$$

$$\eta_{Tis_{corr}} = \frac{\Delta h_{T_{corr}}}{\Delta h_{Tis}} = \frac{\frac{P_{T_{corr}}}{\dot{m}_T}}{\frac{P_{Tis}}{\dot{m}_T}} = \frac{P_{T_{corr}}}{P_{Tis}} \tag{2}$$

In Figure 4, isentropic turbine efficiencies are plotted against isentropic compressor power. The red line represents the diabatic data from standard measurement and shows the typical efficiency rise at low speeds where heat flows become increasingly influent. The enthalpy drop which is without correction superimposed by heat flows, does not show reasonable values. In Figure 4 it is clear that the heat flows are rather constant and at low effective powers the relative influence grows. The corrected efficiencies are represented by the green line. The effect of the recalculation of the effective powers is obvious. For validation purposes a CFD simulation was carried out on the wastegate turbocharger, for which design data was available. The applied Solver was Ansys CFX 16. To allow conclusions about deviations caused by heat flows or aerodynamics between simulation and experiment, the validation was performed with adiabatic experimental data. The validated CFD model was then used to calculate the hot gas case. For modeling the turbulence the Shear Stress Transport (SST) model was used. Since the condition y$^+$<1 is set, the wall layer is relatively finely resolved. The turbine (housing, wheel) and the measurement

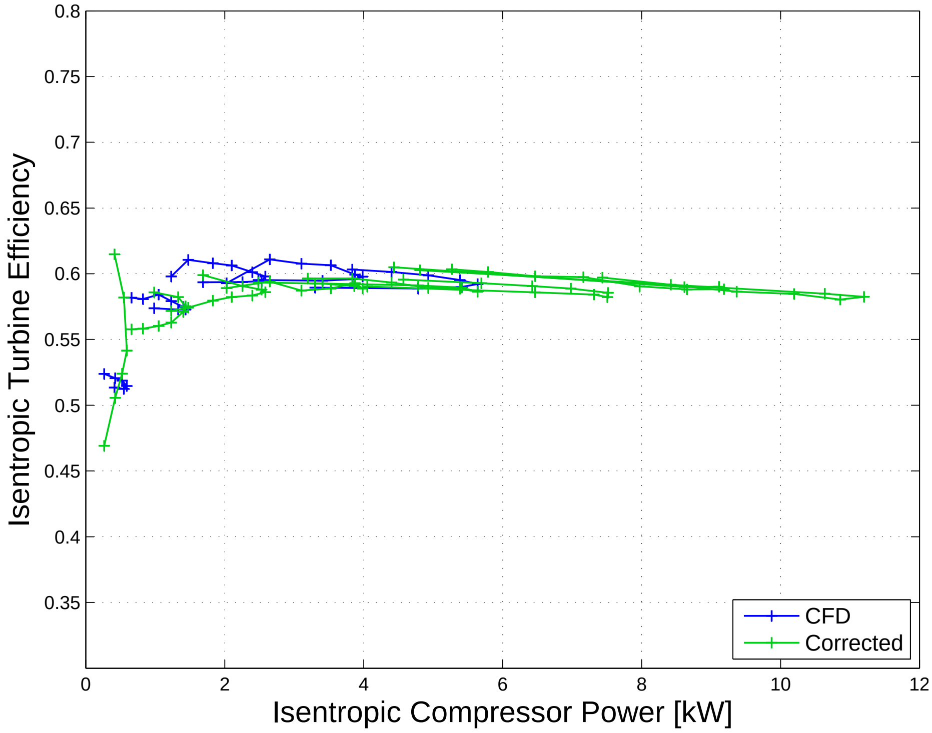

Figure 5: Isentropic turbine efficiencies zoomed

pipes with the mixing device are modeled with 14.3 million elements. For these reasons, the grid independency test was not performed. The blue line shows the simulated values. In comparison with the CFD data, the corrected isentropic efficiencies are located in the right range. For low speeds there are some deviations because of the sensor accuracy as well as a very high sensitivity to isentropic efficiencies. In Figure 5 the deviations are zoomed (section Illustrations and Tables). Nevertheless this is a quite good result for such a simple correction from standard hot gas tests. The only special equipment needed is the mixing device for turbine exit temperature measurement.
In Figure 6, the isentropic compressor efficiencies are plotted against the isentropic compressor power. The diabatic data represented by red lines is affected by heat flows at low loads so that the efficiencies have to be corrected. The peak compressor efficiencies stay almost constant over the speedlines. Compared to the adiabatic test data, the corrected values fit rather well. Only at the lowest speed are there deviations, for the same reason of sensor accuracy as for the turbine. The measurements contain deviations because of the inevitable error propagation. Future investigations will therefore include considerations. The results show that the new approach can be used on turbines and compressors.

CONCLUSIONS

To improve turbocharger modeling by increasing the detail level, this study introduces and develops a method based on previous work to calculate heat transfers of turbochargers using standard SAE maps. Additional special equipment for direct measurement of the turbine exit temperatures was discussed in previous papers (Baar et al. (2013)). It is possible then to calculate heat from the turbine which flows partly into the compressor group without know-

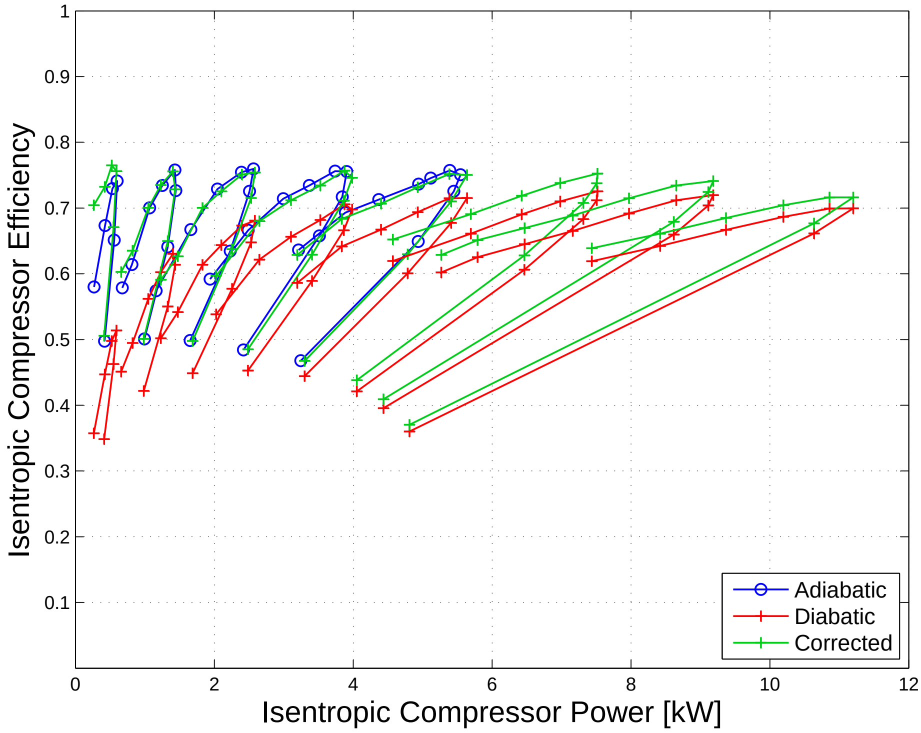

Figure 6: Isentropic compressor efficiencies

ledge either of the geometry or the material of the turbocharger parts. In recent research it was shown that this approach works for the turbine power on different turbochargers (Zimmermann et al. (2016)). In the present work, this approach is developed further by applying it to the compressor which showed a similar characteristic that is the basis of the procedure. The diabatic standard data was corrected by means of heat flows. Furthermore, isentropic efficiencies were determined and validated by simulation and adiabatic tests. The corrected compressor values showed good fitting with adiabatic data as well as the corrected turbine values with CFD simulation results beside the values from lowest speed. The new approach is independent from design data and uses physical power-based characteristics so that it can be universally applied to other turbochargers. There are no calibration factors that have to be fitted on other devices only measurement data. Moreover high speedlines can be corrected which usually can't be measured in adiabatic experiments. The simple procedure can be easily applied and produces results with a good fit in comparison with methods which have a much higher experimental or simulative effort.

The next steps will include the evaluation of the surface thermocouples. By combining the results of the present work and friction powers, the remaining heat flows can be calculated. Strain gauges in combination with telemetry systems will be used to validate the determined friction powers. With transient experiments, thermal capacities can be determined for the application of a thermal network model in engine process simulation. The objective is a new procedure for setting up a turbocharger model that is able to represent the aerodynamic and the thermal behavior with higher accuracy than the state of the art procedures in greater depth and with less effort. For this purpose another objective is the on-engine investigation of the turbocharger, which is a challenge with regard to the measurement. The small and rarely straight pipes do not allow such measurements in comparison to a turbocharger test bench.

Furthermore, the approach showed that the interpolated power gradients increase with more speedlines. Especially the comparison with adiabatic data showed this trend, which suggested that it could be better to use a higher degree polynomial for interpolating the points of maximum isentropic compressor powers. Another important topic is the consideration of the error propagation. Each sensor itself has quite a good accuracy but the propagation of the errors can be rather large especially at low speeds. It is very important to take this effect into account to decide on the range of accuracy for the investigations.

The use of the mixing device for homogenizing the turbulent flow has the disadvantage of heat losses because of its size. In the present work these heat losses are included in the measured power but cannot be quantified why an experimental investigation will be part of future work as well as a CHT simulation. The first procedure is not a big experimental effort, since the mixer can almost be used for most of the passenger car turbochargers, so the map only has to be created once. This would have an additional positive effect on the results because deviations from measured and true isentropic efficiencies would be reduced.

ILLUSTRATIONS AND TABLES

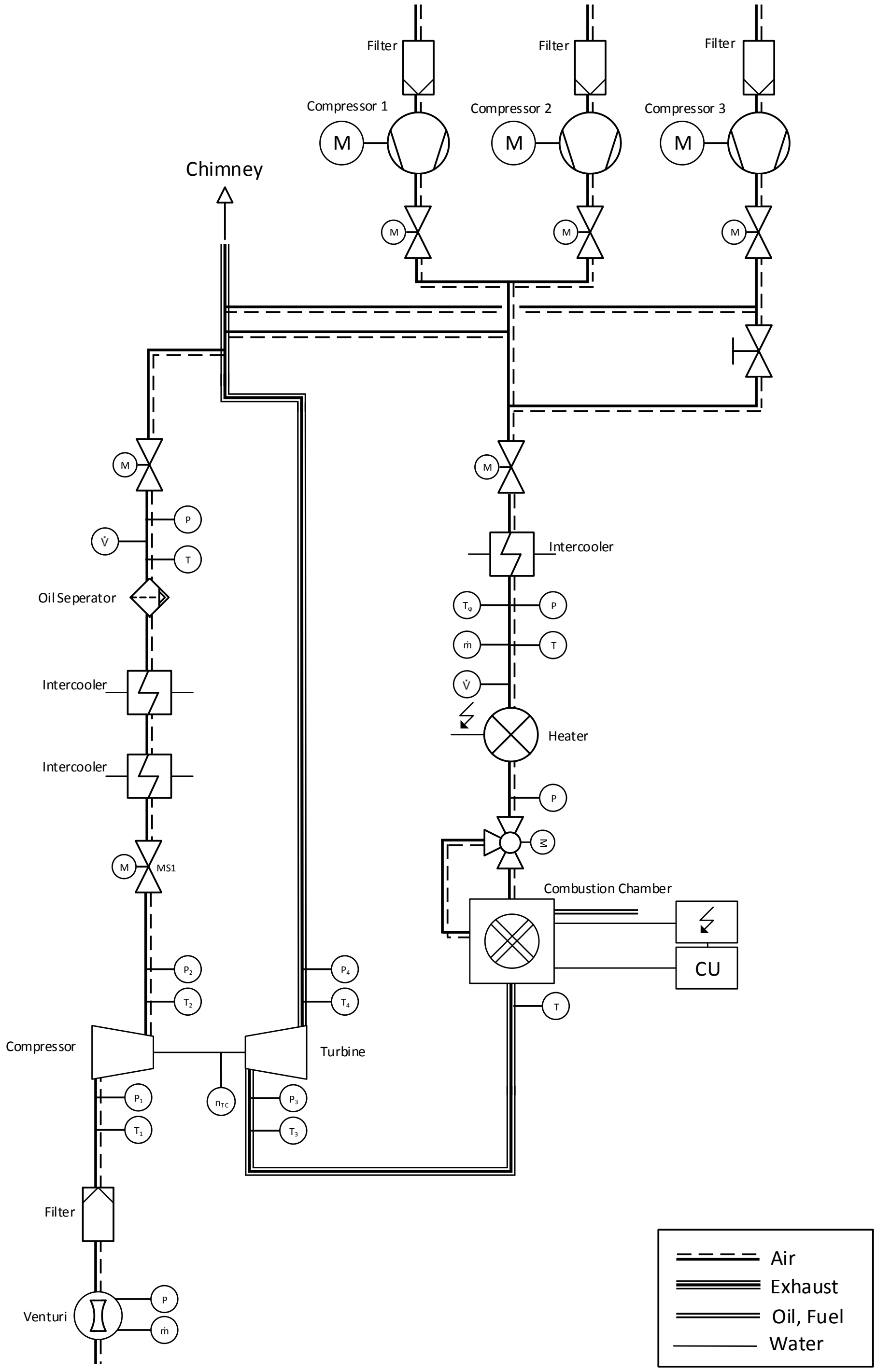

Figure 7: Turbocharger test bench Technical University Berlin

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