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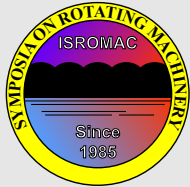
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Commissioning of a Test Stand for Turbocharger Investigations at Constant Turbine Inlet Temperatures

Alexander Esper^{1*}, Christian Lenzen¹, Manfred Wirsum¹



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Abstract

In this paper, the developed turbocharger test stand design at the IKDG facility is evaluated. The specific characteristics of the test stand as well as operational test data is discussed. A test turbocharger is thermodynamically examined in regard of operation at 220°C turbine inlet temperature. An overview of the measurement instrumentation is given and the thermodynamic parameters of the turbocharger necessary for generating a power map, are presented. The key aspect of providing a constant turbine inlet temperature necessary for the acquisition of reliable performance measurements is varied by means of an operation point stability analysis at different rotational speeds across the whole operating range of the test turbocharger. Additionally, an uncertainty analysis of the temperature and pressure measurement instrumentation according to DIN V ENV 13005 and [1] is conducted. Based on these results, a performance map of the compressor as well as the turbine of the turbocharger is generated and discussed. The paper is concluded by providing an outlook regarding the possibilities of upgrading the performance measurement instrumentation in the existing test stand design as well as upgrading certain parts of the test stand.

Keywords

Turbocharger – Test stand – Performance measurement

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INTRODUCTION

Design parameters of state of the art turbochargers evolve towards slender rotating components due to cost and operation mode efficiencies. Slender rotating components increase transient response characteristics due to lower inertia and therefore, improving performance and efficiency. To evaluate the operational characteristics of a turbocharger at different turbine inlet temperatures, a test stand was developed at the Institute for Power Plant Technology, Steam and Gas Turbines (IKDG) to enable investigations of performance maps at selectable constant turbine inlet temperatures. To reproduce real time applications where turbochargers are implemented downstream the exhaust outlet of an engine, a lot of test stands use loop operation where the compressed air of the turbocharger compressor passes a combustion chamber upstream of the turbocharger turbine [2, 3]. By decoupling the mass flows of turbocharger compressor and turbine, a wider range of the performance map can be investigated. Furthermore, this simplifies operating the turbine at constant inlet temperatures. Therefore turbocharger test stands have been developed on which the turbocharger turbine is operated without using the compressed air mass flow from the turbocharger compressor [1, 4]. A significant difference between these test stands and the test stand introduced in this paper is the use of two hot gas generators that run sequentially enabling the adjustment of the turbine inlet temperature with a maximal deviation of about +/- 1.0K during steady state operation. This allows an operation of the turbocharger close to a thermodynamic equilibrium operation to maximize perfor-

mance measurement accuracy compared to conventional test stands using combustion chambers as hot gas generators. Further advantages in the use of an adjustable electrical heater is the simpler control of the heater, the uniform warm-up process as well as the avoidance of thermo shock stress on the measurement instrumentation due to combustion chamber ignition. On the other hand, the implementation of an electrical heater leads to slower warm-up ramps and negates a real time operation with exhaust gas. A unique feature of the IKDG test stand is the possible mass flow range for turbocharger testing since the turbine of the turbocharger is supplied by a six-stage radial compressor with a maximum air mass flow of 12 kg/s. While test stands like [5], [6], and [7] also use sequential hot gas generators, the mass flow range is limited compared to the IKDG test stand setup. The commissioning of the developed test stand and its thermodynamic boundary conditions as well as the chosen operation route, lay the basis for future experimental investigations regarding transient pressure and blade vibration phenomena.

NOMENCLATURE

c	Flow velocity (absolute)
c_p	Specific heat capacity
Ma	Mach number
n	Mechanical exponent
p	Pressure
Pr	Prandtl number
r	Radius
R	Set radius of pipe

Re	Reynolds number
rec	Recovery factor
s	Second
S	Sample
T	Temperature
u	Uncertainty
V	Voltage

Symbols

I	Output signal
\dot{m}	Mass flow
κ	Isentropic coefficient
σ	Input signal

Subscripts

Cbl	Cable
DIN	Deviation compared to DIN
HC	Heat conduction
Inh	Inherent resistance
Lin	Treatment and linearization
Lts	Long term stability
m	Meridional / Measured
Ma	Influence of Mach number
Mt	Transient effects
Mv	Viscosity impact
Mw	Wall influence
Ref	Reference temperature of MAS
s	Static
Shs	Slef heating of sensor
Stab	Stabilization period
t	Total
Tam	Ambient temperature of MAS
Thstr	Thermo-voltage
Vs	Voltage supply of MAS

Abbreviations

CFD	Computational fluid dynamics
DPT	Differential pressure transducer
FSO	Full scale output
MAS	Measuring acquisition system
MP	Measuring plane
TC	Thermocouple

1. EXPERIMENTAL SETUP

1.1 Test stand setup

The test turbocharger is designed for usage in marine engine applications and is provided by Kompressorenbau Bannewitz GmbH (KBB). The turbocharger provides a total compressor pressure ratio of up to 5.5 with a maximum turbine mass flow rate of 5.7 kg/s at a certain inlet temperature of 220°C. This allows the determination of the turbocharger behaviour and efficiency at below usual real time application. The test turbocharger is designed for turbine inlet temperature operation up to 650°C during real time application. The test stand consists of different subsystems composed of air sup-

ply infrastructure, air heater components, test turbocharger, data acquisition system and control unit. Figure 1 shows a simplified flow diagram of the turbocharger test stand at the IKDG facility with the instrumented measurement planes (MP). The design parameters are presented in Table 1.

Table 1. Design parameters of the turbocharger test stand at IKDG

Turbine inlet mass flow	$\leq 6\text{kg/s}$
Turbine inlet pressure	$\leq 6\text{bar}$
Turbine inlet temperature currently	220°C
Turbine inlet temperature possible	500°C
Turbine wheel diameter	$\sim 250\text{mm}$

1.1.1 Compressor facility

The test rig is powered by a high pressure air supply system driving the turbocharger turbine at the IKDG test facility with compressed air. The flow medium is provided by a 6-stage radial compressor (2) which provides a maximum air mass flow of 12kg/s at 30bara. The compressor is controlled by variable inlet guide vanes (1) at the first three stages. A security bypass valve (3) at the outlet of the compressor unit is used during the start-up procedure, for emergency shut-down as well as to adjust the required test stand mass flow. To further increase the adjustment precision a control valve (4) is installed upstream the air preheater.

1.1.2 Air preheater

The air exits the 6th stage of the compressor with a temperature of approximately 150°C. Then, the air temperature is further increased by a 10.5MW natural gas fired air preheater (5)–(7) to a temperature of about 180°C. For investigations requiring higher turbine inlet temperatures the air temperature can be increased up to 500°C.

1.1.3 Turbine section

Before the flow medium enters the turbocharger, a filter (8) cleans out any particles which may be carried along the flow path. The mass flow entering the turbocharger turbine is measured by means of an orifice plate (9) that is designed according to DIN(5167-2, 2004). A control valve (10) is integrated into the turbine piping as a safety measure in case that the lubrication system, providing the turbocharger with oil, fails and therefore prevent damage of any rotating components. Since the NG fired heater provides temperature gradients too wide for steady turbocharger operation, an electrical preheater (11) is directly integrated into the test stand design thus enabling steady state parameters with a maximum turbine inlet temperature deviation of about +/- 1.0K during steady state operation. The fluid enters the turbocharger turbine (12) and exits into the stack. The position of the orifice plate upstream of the electrical preheater was chosen in order to minimize mass flow measurement inaccuracies due to hot gas streaks. The true flow temperature entering the turbine is measured at measurement plane 1.

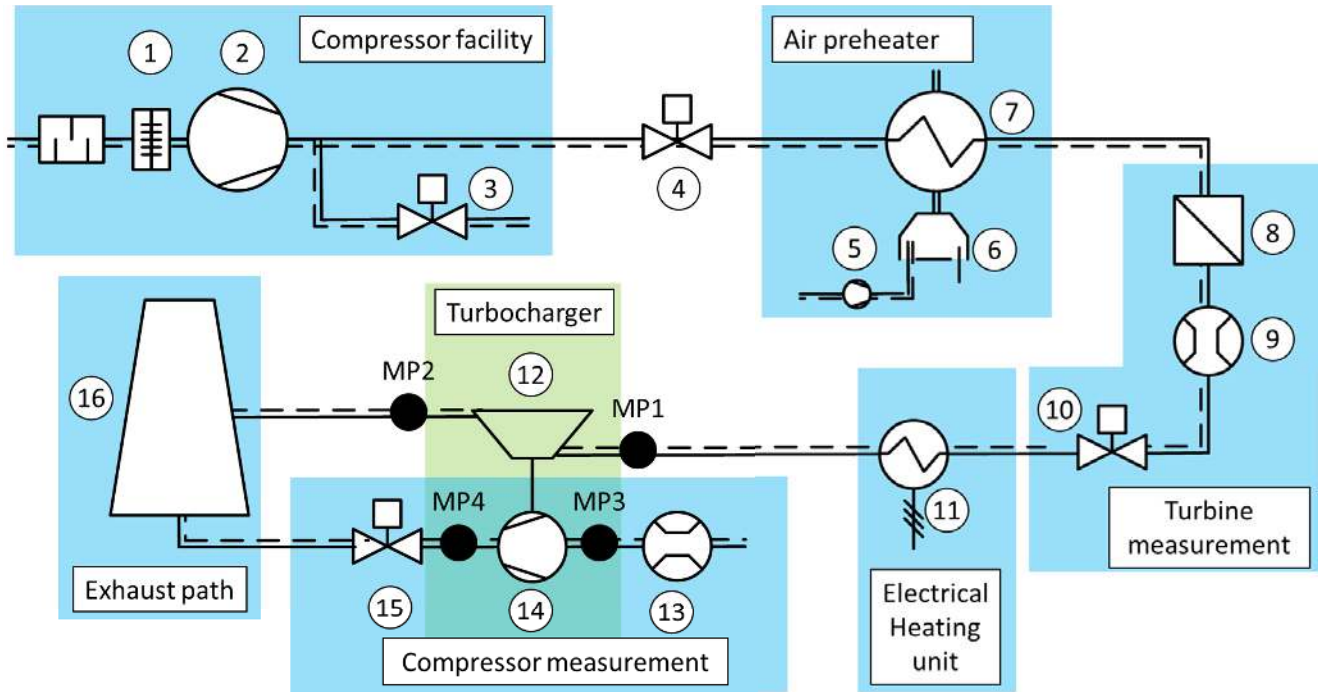


Figure 1. Overview of the test stand at IKDG with instrumented measurement planes

Due to the below real time application no limits are placed on the turbine outlet temperature during test operation.

1.1.4 Compressor section

At the IKDG test stand, the compressor and the turbine side of the turbocharger are operated independently of one another. On the induction side of the compressor the intake mass flow is measured using an orifice plate (13). To acquire all data required to develop a performance map for the turbocharger, a throttle valve (15) is installed at the pressure side of the turbocharger compressor (14) allowing the adjustment of the compressor operation point from stall to surge (discharge volume: 0.008m^3). The compressor is supplied with air at ambient condition and discharges the compressed air into a stack (16).

1.2 Measurement setup

The measurement instrumentation comprises the acquisition of the thermodynamic operating conditions of the turbocharger as well as the rotational speed and the lubricating oil inlet and outlet temperatures. The static pressures are recorded by means of three holes equally distributed over the circumference of the in- and outlet planes of the turbocharger (MP1-MP4). The total inlet and outlet pressure for the turbocharger are calculated from the measured static pressure and the Mach number. Kümmel [8] states, the pressure calculation relies on the average fluid velocity and the average

Mach number (see Equation 1)

$$\frac{p_t}{p_s} = \left(1 + \frac{\kappa - 1}{2} Ma^2\right)^{\frac{\kappa}{\kappa - 1}} \quad (1)$$

The temperature at the turbine side is measured by three type K thermocouples for each measurement plane. The measurement planes are displayed in 1 labeled MP where the flow temperature is measured. The equipped thermocouples feature a sensor diameter of 3 mm and are distributed equally over the circumference of the measuring plane. On the compressor side the temperature at the in- and outlet are measured by means of three Pt100 resistance thermometers with the same circumferential distribution. Since the temperature is not measured by means of total temperature probes, the measured temperature does neither represent the static nor the total temperature of the fluid. The static temperature is calculated using Equation 2.

$$T_s = T_{\text{Probe}} - rec \frac{c(r)^2}{2c_p} \quad (2)$$

The recovery factor was approximated for turbulent flow conditions in reference to [8]. Since the temperature dependency of the Prandtl number is negligible in the examined temperature interval from ambient temperature to 220°C , the static temperature is calculated using a constant recovery factor.

$$rec = \sqrt[3]{Pr} = \sqrt[3]{0.7} = 0.8879 \quad (3)$$

The total temperature is calculated using the velocity at the radial position of the thermocouple sensing head within the fluid flow path in contrast to the total pressure of the fluid that is derived by means of the average fluid velocity. For all measurement planes the temperature was measured at the equivalent same radial position for all three probes. Therefore the insertion depth's influence on measurement uncertainty is not considered throughout the following investigations. The radial velocity profile is attained using Equation 4.

$$\frac{c(r)}{c_m} = \frac{(1+n)(2+n)}{2} \left(1 - \frac{r}{R}\right)^n \quad (4)$$

Kümmel [8] suggests to set $n=1/7$ as in turbulent flow profiles this value is proven to work. Performed comparative measurements with a 6 point pressure measuring rake proved the suggested value [9].

$$\frac{T_t}{T_s} = 1 + \frac{\kappa - 1}{2} Ma(r)^2 \quad (5)$$

2. RESULTS AND DISCUSSION

For the acquisition of turbocharger performance map data it is essential to validate the operating point stability to verify the assumption of steady state operation. Furthermore the operating point stability and the obtained thermodynamic equilibrium state of the turbocharger during measurement procedure ensures comparability of different conducted measurements. Therefore, the tolerances of the steady state definition are analyzed. During data acquisition the following criteria for steady state have been formulated to ensure representative thermodynamic data parameters. The data acquisition procedure entails a sample rate of 1S/s for the measurement of temperature, pressure, mass flow and rotational speed. A total number of 60 sample points is measured and arithmetically averaged to attain the thermodynamic parameters of the current operation point. The analysis of the operation point stability compares each sample point to the arithmetically averaged operation point thus displaying the fluctuation over the data acquisition interval of 60 sample points equivalent with 60 second measurements.

2.1 Operation point stability

During steady state data acquisition, it is essential that neither increasing nor decreasing tendencies in turbine outlet as well as compressor outlet temperature occurs. A maximum deviation of about $\pm 1.0K$ of the turbine inlet temperature during the acquisition time interval at steady state operation was aimed for. This allows a more steadier operation of the turbocharger compared to test stands using combustion chambers as hot gas generators. A stability in rotational speed of 2Hz as well as the stability of the oil inlet temperature of $\pm 0.5K$ during acquisition procedure were also defined as required steady state criteria since fluctuations in the oil inlet temperature directly affect the stability in

rotational speed. The operation point stability was analyzed for three representative rotational speeds of around 25%, 60% and 89% of the turbocharger's maximum speed. Figure 2 and Figure 3 show the stability of the turbine inlet parameters (temperature and pressure) during the acquisition interval of 60 sample points. Maximum peak to peak variation in turbine inlet temperature shows values of about 0.3K, which translates into a relative variation of 0.061%. The maximum turbine inlet pressure peak to peak fluctuation is 4 mbar (0.098%).

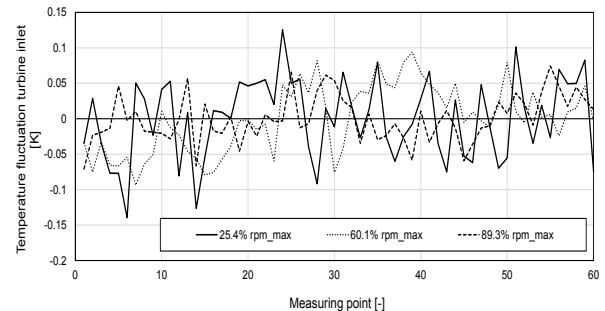


Figure 2. Temperature fluctuation turbine inlet

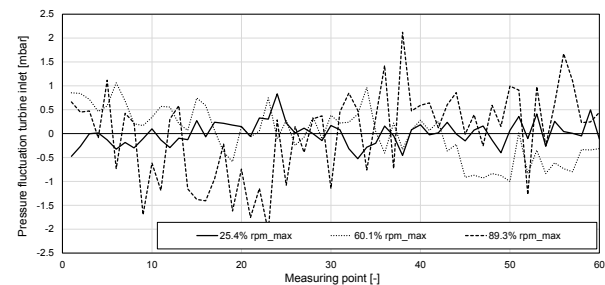


Figure 3. Pressure fluctuation turbine inlet

When adjusting the operation point using the throttle valve at the compressor outlet side, the temperature measured at the compressor outlet indicates, whether thermodynamic equilibrium of the operation point is achieved. The fluctuation in temperature after adjusting the throttle valve and reaching steady state operation is shown in Figure 4. After the turbine and compressor fluid temperatures have reached thermodynamic equilibrium, the acquisition is only started when the oil inlet temperature fluctuation is within the defined limits as shown for three exemplary rotational speeds in Figure 5.

2.2 Uncertainty analysis

In order to evaluate uncertainties of the acquired data, the full measuring chain has to be analyzed, since a certain amount of disturbing influences may cause deviations and errors throughout the measuring interval. In accordance with DIN V ENV 13005 the installed measurement acquisition system

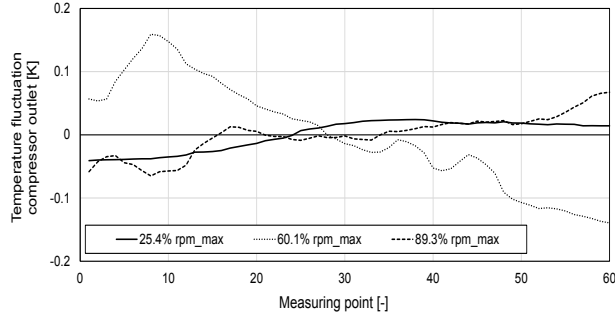


Figure 4. Temperature fluctuation compressor outlet

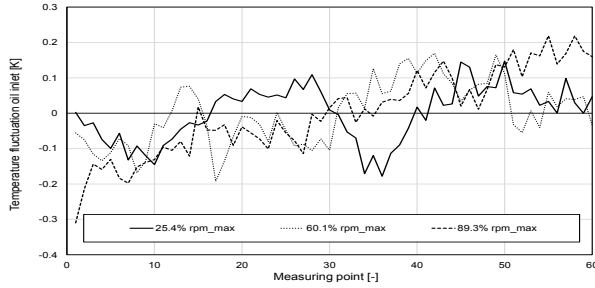


Figure 5. Temperature fluctuation oil inlet

(MAS) and all implemented sensors are assessed. The temperature on the turbine side is measured with thermocouples (TC) of Type K, which are connected to modules with integrated “cold junction compensation”. All TC are connected to the same module and therefore have got the same reference temperature. The temperature at the compressor side is measured with four-wire system Pt100 resistance thermometers. The measurement chain is modeled as suggested in [1].

2.2.1 Uncertainty of temperature measurement

The temperature measurement chain is modeled by means of Equation 6 for the Type K thermocouples implemented on the turbine side. The temperature measurement chain of the compressor side is modeled by means of Equation 7 based on the use of Pt100 resistance thermometers.

$$I_T = T_m + \sigma_{HC} + \sigma_{DIN} + \sigma_{Stab} + \sigma_{Ref} + \sigma_{Cbl} + \sigma_{Vs} + \sigma_{Tam} + \sigma_{Lin} + \sigma_{Inh} + \sigma_{Lts} \quad (6)$$

$$I_T = T_m + \sigma_{HC} + \sigma_{DIN} + \sigma_{Stab} + \sigma_{Shs} + \sigma_{Thstr} + \sigma_{Vs} + \sigma_{Tam} + \sigma_{Lin} + \sigma_{Inh} + \sigma_{Lts} \quad (7)$$

The acquired output signal I_T depends on the measured temperature T_m and a number of uncertainties which are explained and quantified in Figure 6 and Figure 7.

The thermocouples Type K are located directly in the fluid flow path of the turbine by means of compression fittings.

Input parameter	Source of error	Distribution	Uncertainty
σ_{HC}	Heat conduction	Normal	Unattended because TC is insulated to stem
σ_{DIN}	Deviation to DIN	Rectangular	$1.0K/\sqrt{3}=0.577K$
σ_{Stab}	Stabilization period	Cancelled because turbine inlet temperature is kept constant and all operating points are measured in steady state	
σ_{Ref}	Reference temperature		Considered in σ_{DIN}
σ_{Cbl}	Cables		Considered in σ_{DIN}
σ_{Vs}	Voltage supply	Rectangular	Measured temperatures 0°C: 0.49K 100°C: 0.46K 300°C: 0.58K
σ_{Tam}	Ambient temperature of MAS	Rectangular	
σ_{Lin}	Treatment and linearization	Rectangular	$0.58K/\sqrt{3}=0.335K$
σ_{Inh}	Inherent resistance	Rectangular	
σ_{Lts}	Long term stability of electronic components	Rectangular	Cancelled
Combined standard uncertainty with 68.3% confidence interval $u_{TS} = 0.932K$			

Figure 6. Influencing variables for temperature measurement TC Type K

The sensing element is insulated inside the compression fitting thus heat conduction is not considered. Deviations from the characteristic curves are delivered by the manufacturer of the TC. For Type K thermocouples the limiting deviation is given as 1.0K for temperature intervals from 0°C to 1100°C. During data acquisition the test stand runs in steady state keeping the in- and outlet temperatures of the turbocharger constant. Because the TC reacts much faster to transients than the test stand itself, stabilization periods of the TC do not have to be considered. Errors caused by reference temperature, extension cables and linearization are accounted for by σ_{DIN} . Errors due to a change of the ambient temperature of the MAS, inherent resistance and linearization is considered using the MAS manufactures spec sheet. For a measured sensor temperature of 300°C an error of 0.58K has to be considered. The confidence interval of 95.5% is not used because the assumption is made that all errors would occur at the same time. Therefore, the present uncertainty analysis is conducted with a confidence interval of 68.3%. This leads to a combined standard uncertainty of $u_{TS}=0.932K$ for the turbine temperature measurement by means of thermocouples Type K.

The Pt100 resistance thermometers are positioned in the fluid flow path in the same way as the Type K TC thus heat conduction is not considered. Deviations from the characteristic curves are again delivered by the manufacturer. For Pt100 Class A the limiting deviation is set at 0.75K for the maximum occurring temperature of 300°C at the compressor outlet. Although internal calibration studies have shown that the error is below 0.75K, the standard manufacture value of Pt100 class A is used to consider the maximum error influence. In contrast to the Type K TC, errors for the Pt100 may occur due to thermo-voltage and self-heating of the sensor. Since the supply current of 1mA is too low to influence the temperature acquisition, the self-heating of the sensor is neglected. The thermo-voltage is considered according to Scheller. This leads to a combined standard uncertainty of

Input parameter	Source of error	Distribution	Uncertainty
σ_{HC}	Heat conduction	Normal	Unattended because Pt 100 is insulated to stem
σ_{DIN}	Deviation to DIN	Rectangular	$0.75K/\sqrt{3}=0.433K$
σ_{Stab}	Stabilization period	Cancelled	because all operating points are measured in steady state
σ_{ShS}	Self-heating of the sensor	Rectangular	Unattended since supply current of 1mA is used
σ_{Thsr}	Thermo-voltage	Rectangular	<60mK within temperature interval 0 to 500°C [1] $0.06K/\sqrt{3}=0.035K$
σ_{Vs}	Voltage supply	Rectangular	Measured temperatures -200 to 150°C: 0.15K 150 to 850°C: 0.2K Max: $0.2K/\sqrt{3}=0.115K$
σ_{Tam}	Ambient temperature of MAS	Rectangular	
σ_{Lin}	Treatment and linearization	Rectangular	
σ_{Inh}	Inherent resistance	Rectangular	
σ_{Lts}	Long term stability of electronic components	Rectangular	Cancelled
Combined standard uncertainty with 68.3% confidence interval $u_{TS} = 0.583K$			

Figure 7. Influencing variables for temperature measurement Pt100

$$u_{TS}=0.583K.$$

2.2.2 Uncertainty of pressure measurement

Due to the different pressure levels during the operation mode of the turbocharger, a selection of pressure transducers was installed at the in- and outlet measuring planes of the turbocharger. The pressure measurement chain is modeled by means of Equation 8. Likewise to the temperature measurement model, the output signal l_{pi} is composed of the acquired pressure V_{pi} and the accumulated uncertainty variables, which are explained in Figure 8 as a function of the full scale output (FSO) of the sensor measuring interval.

$$l_{pi} = V_{pi} + \sigma_{Mt} + \sigma_{Mw} + \sigma_{Ma} + \sigma_{Mv} + \sigma_{DPT} + \sigma_{MAS} \quad (8)$$

Input parameter	Source of error	Distribution	Uncertainty	
σ_{Mt}	Transient effects		Cancelled	
σ_{Mw}	Wall influence		Unattended	
σ_{Ma}	Influence of Ma-number		Unattended, since $Ma < 0.3$	
σ_{Mv}	Viscosity impact		Unattended, since $Ma < 0.3$	
σ_{DPT}	DPT	Zero pressure offset	Unattended due to calibration	
		Linearity/Hysteresis	Rectangular	0.1% FSO
		Temperature offset	Rectangular	0.1mV
		Temperature span	Rectangular	0.2% FSO
		Long-term stability		Unattended since < 1 year
σ_{MAS}	MAS	Gain Error	Rectangular	160ppm of reading
		Offset Error	Rectangular	156ppm of reading
		Noise Uncertainty	Rectangular	0.62μV

Figure 8. Influencing variables for pressure measurement

Likewise to the temperature measurement, transient effects can be ignored. The wall influence is neglected since only the static pressure is acquired using wall pressure holes. The influence of the Mach number has not been taken into

account because occurring Ma-numbers are less than 0.3 for the measured rotational speeds. This assumption is also applicable for the viscosity impact, since it only has to be considered for $Re \leq 1000$. The accuracies of the differential pressure transducers (DPT) are specified by the manufacturer and reviewed. In addition to the measurement instrumentation uncertainty, the static pressure wall tap diameters need to be included in the pressure uncertainty analysis as well. Bore diameters of 5mm were chosen for all measurement planes. The uncertainty value was calculated from the ratio of manufactured pressure hole diameter to the flow area at each measurement plane based on [10]. Absolute uncertainty values for each measuring plane are given in Table 2 using the diameter ratio of 0.4 to consider the hole diameter uncertainty .

Table 2. Absolute uncertainty values for all turbocharger measuring planes

Measuring plane	Combined uncertainty 68.3%
MP 1	3.10 mbar
MP 2	6.05 mbar
MP 3	6.05 mbar
MP 4	6.40 mbar

2.2.3 Performance map data

The following section is dedicated to show the acquired data from which the performance maps of the turbine and the compressor are generated. The acquired turbocharger turbine performance map is shown in Figure 9. The corresponding compressor performance map is shown in Figure 10. The performance map is acquired by successively closing the throttle valve at the compressor outlet while maintaining a constant turbine mass flow and turbine inlet temperature. As soon as an increase in rpm occurs, the mass flow on the turbine side is readjusted. Thus providing the data for a characteristic curve for a defined turbine speed. The performance data is acquired until reaching the surge line of the compressor limiting the width of the acquired rpm curve. The expected trend of the characteristic turbine rpm curves can be seen in Figure 9.

Due to compressibility effects the reduced mass flow bends towards an asymptote and decreases after peaking at a turbine pressure ratio of approximately 3.5. Since the compressor side of the turbocharger is operated at ambient conditions, the inlet pressure varies depending on the rotational speed due to the orifice plate mass flow measurement upstream of the compressor inlet. The variation of the inlet pressure at compressor inlet in reference to the rotational speed is shown in Figure 11. To further improve performance investigation, the ambient operation should be switched to a closed-loop operation, which can be integrated into the test stand. Since in turbocharger operation the focus lies on the acquisition of the turbine performance where the compressor side is merely used to adjust the load, the closed-loop test stand version will offer a much wider range for obtaining per-

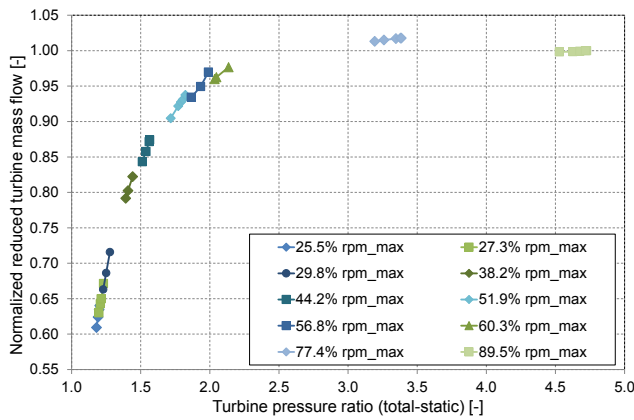


Figure 9. Acquired turbocharger turbine performance map

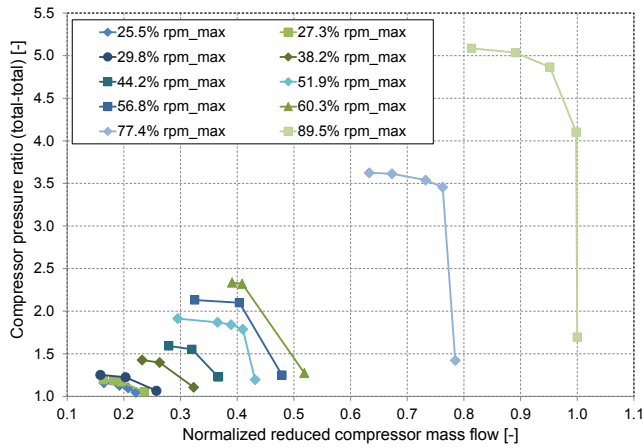


Figure 10. Acquired turbocharger compressor performance map

formance map data. Repeatability investigations were conducted for an individual reference operation point, defined by turbocharger rotational speed and total-static turbine pressure ratio, for certain rotational speeds at two different test days. The results are listed in 3 and show an excellent overall repeatability for turbine performance operation points.

Table 3. Repeatability of turbine performance map data at two different test days

Rotational speed	Error \dot{m}_{red} [%]
44.2% rpm_{max}	0.07
51.9% rpm_{max}	0.27
56.8% rpm_{max}	0.19
60.3% rpm_{max}	0.03
77.4% rpm_{max}	0.09
89.5% rpm_{max}	0.32

When discussing thermal stability and the influence of measurement uncertainty on performance map data, the tur-

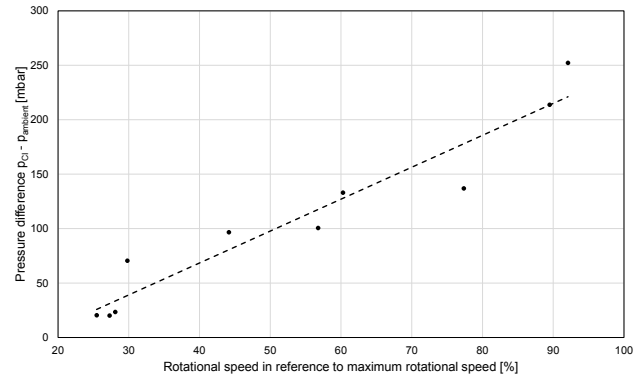


Figure 11. Pressure variation of the compressor inlet in reference to the rotational speed

bine efficiency needs to be assessed. Due to the difficulty in the assesment of turbine efficiency based on measurements at the in- and outlet piping of the turbocharger turbine, it has become common, to introduce the turbine thermomechanical efficiency [11].

$$\eta_{eff} = \frac{P_{pol,compressor}}{P_{is,turbine}} \quad (9)$$

where $P_{pol,compressor}$ is the polytropic power output of the turbocharger compressor and $P_{is,turbine}$ is the isentropic turbine power output calculated on the basis of an isentropic turbine outlet temperature.

$$\Delta h_{is,turbine} = cp_{turbine}(T_{bt,tot} - T_{at,is}) \quad (10)$$

The calculated normalized effective efficiency is displayed in Figure 12 in regard to the total-static turbine pressure ratio. The analysis of the efficiency curves supports the assesment, that good thermal stability was achieved during steady state operation.

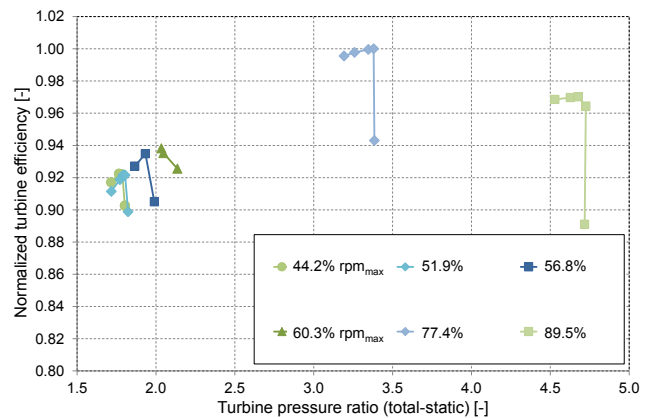


Figure 12. Normalized effective efficiency of the turbocharger turbine at 220°C turbine inlet temperature

3. CONCLUSION AND OUTLOOK

The developed turbocharger test stand provides an opportunity to operate the test turbocharger at stable conditions. Testing has shown, that the requirements of the thermodynamic parameters, like the turbine inlet temperature, are fulfilled. Hence the operation point is stable enough that data for performance map purposes can be acquired at the presented test stand. The operation of the test stand at constant turbine inlet temperatures, especially at below real-time application, can be used for investigations where the implemented measurement instrumentation is limited by maximum operation temperature. The required equilibrium state between compressor and turbine of the turbocharger was proven to be obtained during the performance map data acquisition by comparing the outlet temperatures of both sides. A stability interval of the rotational speed of 2 Hz was accomplished. The uncertainty analysis for the applied measurement instrumentation shows, that the acquired data lies within an uncertainty interval which suffices the required measurement accuracy for obtaining the needed thermodynamic parameters for generating the displayed performance maps as well as CFD validation. A constant oil inlet temperature within tolerances of $\pm 0.5\text{K}$ was also achieved during testing operation of the turbocharger.

In regard of the acquired test data the developed test stand can be modified for further investigations on turbocharger performance and efficiency. The displayed measurement range of the turbocharger compressor can be identified as the limiting factor on the possible measurement range. Since the electrical preheater is less sensitive to low flow turbine conditions, the compressor side needs to be adjusted towards lower mass flow. Internal investigations have shown, that by adjusting the compressor side pressure losses (usage of two parallel pipes equipped with orifice plates) the measurable performance range can be increased. Furthermore, the mentioned variation in compressor inlet pressure can be addressed by implementing a closed-loop compressor cycle with integrated intercooler. Thus providing similar compressor inlet conditions independent from the rotational speed. In a first step Also the installed lubricating oil circuit offers the opportunity to adjust the oil inlet temperature in a wider scope which allows a large spectrum of test turbochargers that can be investigated. Furthermore the compressor facility at IKDG provides enough mass flow to enable testing in a turbocharger scope of application from small automotive up to marine turbocharger applications.

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