EXPERIMENTAL AND NUMERIC INVESTIGATIONS ON A STEAM TURBINE TEST RIG IN PART LOAD OPERATION

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ABSTRACT
Subject of the paper is to present a purpose-built steam turbine test rig and its prospects for complex analysis of thermodynamic, aerodynamic and mechanical problems. The turbine at a scale of 1:1 was equipped with extensive instrumentation. Pressure and temperature rakes behind the control stage at several circumferential positions, rakes before and behind the HP, IP and LP stages and additional 5-hole probes to measure the complete flow field. Also tip timing measurement of the last LP stages were carried out to analyze their mechanic behavior in critical operating points, for example extreme part load or high condenser pressure. The turbine and its equipment are discussed, different operating points are analyzed and some comparisons of measurement results and numeric calculations are presented. The measurements create an additional database for future design optimizations.

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INTRODUCTION
Various chemical processes operate with fluids at high pressure. To obtain the high pressure levels compressor-trains composed out of several compressors and compressor types are common. These compressor trains are driven by either electric motors or by industrial gas respectively steam turbines. Depending on the chemical process the compressors run at varying rotational speed and thus the industrial steam turbine for mechanical drive applications has to cover a wide range of operation concerning rotational speed and volume flow. Furthermore different rotational speeds in one compressor train are realised via the use of a gear box. Within those trains the power outtake of the turbine can be located at both ends of the machine. To harmonize the dynamic behaviour among the train components especially the length of the steam turbine can be optimized to be as compact as possible.

Driving a compressor train, a single turbine has to run for considerable time at different operating points covering rotational frequencies ranging from 70 to 105 percent of its design point. The rotational speed and high load on the control stage are crucial for the control blade design. In addition the pressure of the saturated steam in the condenser may vary from some ten to several hundred mbar. Thus, a robust three dimensional blade design is crucial for the control blade design. In addition the pressure of the saturated steam in the condenser may vary from some ten to several hundred mbar. Thus, a robust three dimensional blade design is needed in the last stages of the LP section of condensing turbines to guarantee high stability and good efficiency at all conditions. Process dependent life steam is available at a certain pressure and temperature level for expansion through the turbine but may be needed in the process at a different pressure and temperature level and extracted from the turbine or additional steam is available at certain thermodynamic conditions and admitted to the turbine. Hence the setting of an industrial steam turbine is a tailor-made composition of specific turbine components.
Industrial steam turbines generally consist of a partial arc life steam admission module followed by a control stage, a variable flow path through HP, IP and LP blading and a radial (or axial) exhaust hood. Additionally mid-turbine admission and mid-turbine bleeds or extractions are common. To obtain an optimized turbine setting for each application fixed turbine series are less suitable than modular composed machine designs. For this MAN Diesel & Turbo turbine components are clustered in a construction kit out of which each turbine is configured to meet the customers requirements. Abel-Günther et al (2015) present a comparable construction kit for steam turbines in waste to energy power plants.

Concerning the market requirements it is a challenging task to combine a saleable cost-efficient turbine design with a subtle balanced variety of machine components. Within the construction kit each component can be independently optimized concerning thermodynamic, mechanical or aerodynamic issues. To confirm numeric improvements in the design of the turbine components and blading it is desirable to validate them with measured data from steam-driven test rigs. For example Starzmann et al (2011), Eberle et al (2014), Schatz et al (2014) or Bosdas et al (2015) carry out investigations on wet steam effects at scaled experimental LP-turbines. Experimental research steam turbines are investigated by Kolovratnik et al (2014) and Hoznedl et al (2015). Further investigations are carried out directly in power plants (Walters et al (1988), Kleitz et al 1988). Shibukawa et al (2014) describe in their paper a steam turbine test rig for LP stage analysis in different operating points. The mentioned above papers mainly focus on wetness measurements or measurements in single blading parts like HP or IP blading. A full scale steam turbine test-rig, covering all steam turbine components, comparable to the one presented in the present work is not known to the authors.

TEST RIG

Turbine design
The test rig is located at the test facility in Oberhausen, Germany. It is composed out of full scale steam turbine, a gear box and a hydraulic break. For future test setups it is imaginable to replace the hydraulic break by a compressor and to investigate a full scale compressor train test rig. Hence life steam is supplied by a local power plant the design parameters of the turbine are matching the obtainable conditions. For life steam parameters of 30bara and 400°C and exhaust pressure of around 130mbara the current turbine setup has a maximum power output of approximately 12,5MW.

Ranging from high to low mass flows and rotational speed within the specified area, typical operating conditions of industrial steam turbines are points with an extremely part loaded machine. To control the mass flow the life steam passes through a group of valves above nozzle boxes before admitting the turbines control stage.

To examine the influence of extraction on the performance, flow profiles and vibration of the last stage blade, the turbine casing has two holes where extraction pipes can be fixed. The position of both extraction holes within the turbine is located between the IP and LP module which causes a comparatively long area without blades on the runner. The size of the extraction holes is sufficient, that approximately 1/3 of the life steam mass flow could be extracted. The extraction mass flow can be regulated with an adjustable vane. In the setup for the current investigation, no extraction pipes were installed and all steam passes through the LP module and leaves the machine via the radial exhaust hood.

The turbine is mounted on the test facility’s condenser. There is no closed loop for the accruing condenser water. The power outtake is located at the turbines cold end, where it is coupled to the gearbox which is followed by the hydraulic break. To move the turbine from the workshop to the test facility and reverse the turbine is fixed on a base frame. Additionally the secondary systems as oil pipes and emergency oil supply, valves for the sealing steam system, drainage pipes, exhaust hood water injection system as well as access points for all measurements (temperature / pressure / vibration / displacement) are located on the base frame. This means that the whole turbine test rig
package is arranged in a single lift configuration that could easily be connected to other train components as mentioned above.

Fig. 1 shows a 3D CAD model of the steam turbine. The turbine is designed for the testing of different rotors with standard and future blade designs. Therefore, the inner casing parts, i.e., casing parts containing the live steam nozzle and vane carriers, are constructed in separate modules which can be separately replaced. Due to the planned variable blade count and design as well as the need for additional measurement systems, the gap between HP and IP blading is considerably larger than in a customer's machine.

The outer casing is constructed to the company's latest industrial steam turbine design for small to medium turbines (2 – 30 MW). Industrial steam turbines need a robust design to guarantee secure operating under various conditions. On the other hand, to achieve high energy efficiency, the requirement of compactness has to be fulfilled. By integrating both the front and the rear bearing within the casing of the turbine, the overall length of the turbine is considerably reduced compared to a turbine design with external bearing casings. Due to the omission of the external front bearing support, it is possible to mount the turbine casing on front struts. The rear support is constructed with struts supporting directly both sides of the exhaust hood casing.

**Measurement systems**

Inevitable operating measurements as they would be put in a customer's machine are permanently fixed within the turbine. For the most operation points, the control system has standard accesses to these measurements to guarantee secure operation of the machine. Special operation points may exceed standard operation thresholds for example by means of exhaust temperature or blade vibration. To control these points, a secondary control system accesses in part more scientific measurement equipment which is also permanently attached to the turbine. Especially a tip timing measuring system combined with the vibration measurements is installed. Additionally, to evaluate the performance of each of the HP, IP, and LP module separately, various probes can be positioned in the machine. For this and to obtain overall and local flow profiles, two different types of measuring probes are used:

1. Rake probes to measure pressure and temperature across the inlet and outlet area of each module. The rakes are permanently fixed in the machine and not adjustable and accessible.
respective during the testing period. Depending on the blade height the installed rakes contain three to ten measuring points. Six rakes are equally spaced across each cross section to obtain a circumferential resolution of the pressure field. Eight rakes are located behind the control stage. Each of the rakes contains one thermocouple which is placed on different span heights within one cross section. The pressure probe holes within the rakes point in upstream flow direction and the flow may impinge each of the holes with a varying angle of incidence. Consequently the obtained values can range between total and static pressure values, if the flow impinges to tangentially on the probe. To minimize the influence of the flow incidence on the measured values (see e.g. Chue (1975)), the rakes consist of Kiel probes. As the measurement values are recorded simultaneously these measurements give an overall distribution in the cross sections. Though the number of measuring points on one rake is restricted these overall distributions give a good impression of the degree of uniformity of the approaching flow (see Polklas (2013)).

2.) 5-hole probes are used to investigate additionally the velocity vector and Mach number profile in the inlet and outlet cross sections. For this openings are located at several axial and circumferential positions. The probes are placed through the casing at the test facility where the position can be change during the testing period when the turbine is shut down. Contrary to the rake probes, the 5-hole-probes are controlled by a step motor and are moveable in radial direction and can by adjusted to point toward the flow direction. The applied probes are constructed with spherical heads. The thermocouple is place at the lower end the probe. Depending on the height of the flow path two different sizes of the probe heads are available.

Two possible positions for the fixtures carrying a step motor to move the 5-hole-probes are shown in Fig. 1 at the IP and LP outlet cross section. Other axial positions are the appropriate IP and LP inlet cross sections. At these axial positions 4 circumferential locations are accessible from the upper casing. And one position at the HP inlet cross section is available.

To determine the content of wet steam within the LP stages moisture measurements are carried out in the inlet and outlet cross-section of the LP module. For this light scattering probes measure the droplet size spectrum with a high local resolution. The probes are movable and give a radial distribution at a fixed axial and circumferential position. Depending on the probe design the access points of the 5-hole-probes within the LP module can be used. The measurements were carried out from university partners.

Various industrial steam turbine applications require from one turbine a multitude of operating conditions depending on the life steam parameters causing high mechanical strain on the machine parts. Other processes are characterized by a comparable high number of start-ups and shut-downs. Especially for the latter short start-up times are required. The time needed for the starting period is mainly driven by the heating up of the machine parts. Subsequent deformation and displacement of different machine parts cause a reduction of the clearances within the turbine during periods of local temperature imbalance. Additionally, part load conditions cause an asymmetric temperature distribution in the casing carrying the nozzles and the front part of turbine casing. To analyze the thermal behavior the test rig is equipped with several additional instrumentation:

a) The struts carrying the front bearing, the bearing casings and turbine casing are equipped with additional instrumentation. Strain gauges give information on the transient deformation of the struts.

b) Measurements carried out via eddy current displacement sensors give information on the relative movement among the turbine casing components and among turbine casing and runner.

c) Combined with the displacement the surface temperature of the turbine parts and joint screws is measured at various positions.

Thus a 3d temperature distribution is obtained for the casing parts and a 3d temperature and displacement distribution is obtained for the struts. Particularly during the start-up and the cool-down period these values are of great interest. To investigate stimulus on the first HP vane or interaction between the blades and vanes respectively, several vanes in the blading modules are equipped with strain-gauges.
RESULTS
The main focus of the measurements discussed in the following is on part load operating conditions. The main operating points concerning the power out take range from operating conditions with only some 100kW power out take to operating conditions with approximately 80% of the maximum power out take. Concerning the exhaust conditions the pressure levels vary from 100 to 400 mbar. The overall performance of the machine is evaluated with standard measurements of global thermodynamic values and evaluated with the machine layout. Contrary to standard measurements, were wall pressure measurements are typically carried out to control the expansion through the turbine, local probe measurements within the flow path allow a more detailed comparison with numeric solutions.

Thermodynamic Measurements
To take a closer look at the performance of each of the blading modules and local flow losses some representative comparisons of measurements and CFD calculations are shown below. The computational results shown in the following diagrams were carried out with the 3d CFD solver ANSYS CFX. All numeric values shown in this paper result from steady state RANS computations. To account for the variable material properties depending on the state of expansion in the turbine, the properties of steam are modeled with tabulated values according to the implemented IAPWS database. The use of steady state RANS may be in some detail analysis to simple for the complex fluid flow phenomena. Further numerical investigations are going on and will be published later.

Part loaded control stage
Compared to turbines with sliding pressure or throttle control it is possible to reduce pressure losses by mass flow control via a nozzle group. However this kind of turbine control may cause other loss effects around the control stage. To analyze the stimulus on the control stage in part load operating conditions and the flow through the wheel chamber a close look to the circumferential pressure distribution around the control stage is necessary. For this the measurements in the wheel chamber are used to validate a numerical setup of this area. The calculation is carried for all nozzle boxes, all control stage blades and the wheel chamber. Thus the asymmetric pressure distribution upstream of the control stage is included. Furthermore this setup accounts for the possibility of ventilation regions located behind closed nozzle boxes.

Fig. 2: Comparison of pressure measurement and computational results behind control stage

The domain outlet is located at the inlet cross section of the HP module. The pressure as well as the temperature outlet boundary condition correspond the averaged measured values. To meet the
operating point’s mass flow the inlet boundary condition is appropriate adjusted. The computational
domain consist of approximately 28 million nodes. The mesh size of the blade mesh is chosen to
roughly meet the designed downstream flow angle. The results of the computation should carefully
be used to make a statement concerning the performance of the control stage.

In the considered operating point life steam passes through two of four possible flow sections
which are indicated (box 1 and 2) in the sketch of the computational domain on the left side in
Fig. 2. The asymmetric flow distribution can be seen in the contour plot for the computed total and
static pressure distribution. The right side shows a comparison of a measured pressure profile and
computational results behind the control stage. As the measurement probe is not adjusted to the flow
angle in each operating point, the measured profile is no proper static pressure profile. In spite of
the asymmetric flow field, the local computed pressure values are in good agreement with the
measured pressure values. Comparable measurements are carried out via wall pressure
measurements from within the nozzle carrier directly in front of the nozzle to the leakage above the
control stage. For each nozzle section a series of four holes is drilled in the casing parts Thus a
detailed examination of the pressure loss from life steam to the nozzle box and wheel chamber
respectively can be obtained in axial and circumferential direction.

**LP module**

To investigate the uniformity of the flow in the LP inlet and outlet cross section Fig. 3 shows the
total pressure measured with the rake probes in these sections. As merely six rakes are
circumferentially arranged in each cross section, the pressure values between the rake positions are
interpolated for illustration. It is obvious that there exist regions of low pressure and regions of high
pressure. Dark color correlates to low total pressure and light color to high total pressure values.
The absolute pressure differences are relatively low but it can be seen that the circumferential
distribution in the investigated operating point is quite inhomogeneous. It is observed that the
region with low total pressure is slightly counterclockwise shifted from the inlet to the outlet cross
section.

Numerical calculations to determine the performance of blade rows or blade row modules
respectively are carried out on computational meshes with sufficient local resolution. Due to the
requested fine mesh resolution for these investigations the computational domain is usually reduced
to one pitch per blade row. Consequently the simulation does not account for the asymmetric flow
field shown in Fig 3 but preserves a rotational symmetry of the flow.

![Fig. 3: Spline interpolation of measured total pressure ratio values in LP inlet and outlet
cross section and LP module computational domain](image)

Only a simulation of all blades and vanes of a blading module would account for these
phenomena. This kind of analysis would either result in extremely large computational domains
which would demand very large hardware resources and cause very much CPU time to solve or in
too coarse computational meshes to adequately predict the flow effects around the blades.
Consequently the numeric predictability of operating points with non-uniform flow distributions is of great interest. For this a result of a simulation of the test turbines LP module is shown below.

The corresponding computational domain comprises the main flow path through all LP module stages as well as the secondary flow through associated seals and cavities. The domain consists of approximately 8.6 million vertices. The domain outlet is located within the radial diffuser. As inlet boundary condition, the mass flow is adjusted until an averaged rake probe total pressure profile is obtained. The outlet boundary condition is derived from averaged measurement values.

For the outlet cross section of the LP module a comparison of measured and calculated total and static pressure values is shown in Fig. 4 and total temperature and Mach number profiles in Fig. 5. The figures show measured values from both, all the rake probes and the 5-hole-probe. The 5-hole-probe is positioned between rake 1 and rake 2. The other rakes are arranged equally spaced around the circumference as mentioned above.

![Fig. 4: Comparison of measured and calculated total and static pressure behind last LP blade](image1)

![Fig. 5: Comparison of measured and calculated total temperature and Mach number behind last LP blade](image2)

The measured total pressure of the rake probes and the 5-hole-probes are in good agreement in the middle to upper part of the profile. The profile of the computed total pressure distribution
qualitatively corresponds to the measurements for span heights from 0.25 to 0.9. The qualitative good agreement between the profiles of the numeric solution and the 5-hole-probe measurements is found as well for the static pressure and the Mach number. Especially for the static pressure both profiles are almost identical. This may indicate that the flow in this area mostly corresponds a rotational symmetric flow compared to the rest of the profile.

In the tip region the computation predicts a jet flow which is caused by the tip clearance of the blade. This jet is not found in the measurements. Instead the measurements rather suggest a decelerated flow. This is also observed in a slightly increasing measured static pressure and decreasing Mach number profile. Towards the hub, the static pressure measured with the 5-hole-probe increases. The corresponding measured flow angle values (not shown here) indicate a recirculation area in the hub region. This phenomenon cannot be observed in the computation due to the restrictions mentioned above.

Additionally local discrepancies may be cause by the damping wire located between the span heights 0.6 to 0.8. The total and static pressure profiles show the typical wake in this region. It is further assumed that the offset between experiment and calculation is caused by the part load conditions of the investigated operating point. This results in an transient separating diffuser flow that cannot be covered by the used numeric setup. Further investigations on the flow through the whole exhaust hood will be carried out.

Obviously the uncertainty of the 5-hole-probe measurements is relatively high within the LP outlet cross section. To evaluate the 5-hole-probe measurements, index numbers are calculated which depend reciprocally proportional on the measured pressure and pressure differences on the probe head. Due to a comparably low absolute pressure in the exhaust hood region the measurement uncertainty increases.

**Sealing and leakage flow**

The leakage flow through the sealing of the balance piston can be reduced with different types of sealing. Typical sealing types are labyrinth seals or brush seals. The sealing of the test rig turbine is combined of brush seals and labyrinth seals. To further reduce the radial clearance between the runner and the sealing a part of the labyrinth seal is placed on spring back bodies. To investigate the flow through the sealing, several pressure measurements are placed in between the sealing. Additionally the mass flow through the sealing system can be measured.

![Comparison of measured and calculated pressure drop along the seal of the balance piston](image)

**Fig. 6:** Comparison of measured and calculated pressure drop along the seal of the balance piston
A comparison between the measured and calculated pressure drop along the balance piston sealing is shown in Fig 6. The computational domain is simplified to a thin layer with rotational symmetric boundaries. The domain approximately consist of 500,000 vertices with comparable rough local mesh resolution. The boundary conditions are chosen according to averaged measurement values. For the sealing inlet wall pressure measurements are placed in the cavity between the nozzle box and the control stage. Due to poor accessibility only one position is located at a low diameter directly in front of the sealing. Four other wall measurements are place in a cavity above the control stage. For the outlet boundary, values from two static pressure measurements in the balancing pipe close to the turbine casing are available. The outlet temperature is locally measured very close to the pressure. For the wall pressure measurements the accessible locations are nearly in the middle of the sealing. It can be seen that the predicted pressure drop within the sealing very good corresponds to the measured values. As a further result the mass flow through the leakage is obtained.

**Mechanical Measurements**

**Casing Parts**

As mentioned above the displacement of casing parts is very crucial for start-up conditions. To evaluate the clearances within the machine, the temperature distribution and subsequent deformation is numerically calculated. To meet the measured values within and without the casing the setup for these calculations strongly depends on the assumption or knowledge of local heat transfer coefficients depending on the machine section. For example the heat exchange between an uninsulated bearing strut and the surrounding influences the calculated temperature distribution. On the other hand the strain within the strut is caused by the thermal growth of turbine parts which depend on internal heat exchange between steam and casing.

Fig 7 shows a comparison of the measured temperature $T$ at the front strut and computational results of a steady state computation. The measured data shows the transient heating-up of the two struts. The numeric values result from a computation with a standard mechanical setup. The measurements and the computation are in quite good agreement. A further result of the mechanical computation is the displacement of the strut and thus the clearance between rotor and casing due to heating-up. Depending on the temperature difference between the casing components and the rotor, the clearance decreases.

![Fig 7: Comparison of temperature measurement and computational results at front strut](image-url)
In particular operating points the turbine is operated with extreme part load conditions at high exhaust pressure levels. These operating conditions cause a high mechanical load on the turbines last LP stage and thus require a robust blade (see Truckenmüller (2003)). Depending on the LP last stage design, the mechanical robustness of the blade changes. A disadvantage of the most robust blades is the reduced blade length due to a comparable high blade weight. Contrary to the robustness of the blade a maximum exhaust area is desirable to obtain a moderate exhaust velocity and thus obtain high efficiencies and reduce additional losses. To determine the mechanical behavior of the LP stages during extreme conditions tip timing measurements are carried out at two stages within the LP module via eddy current displacement sensors. The vibration of the blades increase while the flow through the turbine is stepwise reduce until the vibrations reach their threshold. This measurements are essential to maximize the operating range for the turbines last stage.

CONCLUSIONS
The request for increasing life steam parameters, higher mass flows and shorter start-up times creates challenging tasks in the layout and design of industrial steam turbines. To optimize the steam turbine layout concerning the customers demand and to standardize the turbine components design requires advanced design tools. MAN’s strategic engineering philosophy for both compressors and turbines leads to highly efficient turbomachinery trains as exemplary the AIRTRAIN - for air separation units for almost every plant size.

The commercial layout of a complete steam turbine or steam turbine train is still carried out with one dimensional design tools. Contrary high sophisticated design tools are commonly used to optimize single components. To combine the results from those advanced methods for the prediction of the complex thermodynamic and loss processes within the flow as well as the mechanical load mechanisms on the blading and casing parts has to be simplified to one dimensional algebraic formulations. To fortify the computational results measurements are carried and subsequently used to optimize the design rules.

The steam turbine test rig was presented in detail and the test rig’s possible measuring options are discussed with main focus on thermodynamic issues. Selected measurements are presented and compared with computational results. Best performance and reliability of tested components could be verified. As part of the technology design process, the measured data is used to validate optimized steam turbine components.

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