VIRTUALIZATION OF THE DLR TURBINE TEST FACILITY NG-TURB

B. Schneider, German Aerospace Center (DLR), Institute of Propulsion Technology, Linder Höhe, 51147 Cologne, Germany

Abstract

The gas turbine technology has already reached a high level of development and maturation. Further improvement requires a better understanding of the individual components and the effects of implementing new technology. For this purpose, the German Aerospace Center (DLR) operates a turbine test facility, called Next Generation Turbine Test Facility (NG-Turb). The NG-Turb comprises of a two shaft configuration with up to 2.8 MW total shaft power to allow investigations on high-, intermediate- or low-pressure turbines. The test facility works in a closed loop cycle with dried air as working fluid. This cycle is driven by a four stage radial compressor with pre-, inter- and post-cooler. Numerous test bench control variables enable a largely independent setting of Mach and Reynolds number at the turbine rig.

For the optimization of the operational performance and the assessment of cycle modifications a virtualization of the test facility is required. Therefore, a model based engineering approach was established and digital representations of the thermodynamic behaviour for the main components were developed. To perform test facility simulation a thermodynamic synthesis of the components was evolved. This implies a procedure for the correct setting of control variables and an numerical setup to ensure a realistic interaction of the component models. With that digital representation predictions of operating states and feasible operating conditions with respect to test bench constraints, like pipe temperatures, compressor power and shaft speeds, can be conducted. First studies with this digital system have shown that the convergence of the compressor unit represents a challenging task. Therefore, a reduced thermodynamic model was set up to improve the simulation stability and the understanding of the multi-stage compressor behaviour. With knowledge gained, an approach was developed that allows the simulation of demanded turbine operating points. Finally, a successful validation of the compressor unit was carried out by using data from manufacture simulations and commissioning measurements. As an use case, different test facility adaptations were simulated to reach a defined temperature and pressure combination at turbine test rig inlet. The simulation results led to a successfully adaption of the test bench, the extension of the compressor spool speed range.

Keywords

Thermodynamic Simulation; Performance Prediction; Test Bench Simulation; NG-Turb; Next Generation Turbine Test Facility; Transition; CleanSky2; GTlab; Virtualization; Digitalization

1 INTRODUCTION

The gas turbine technology has reached a high development and maturation level. For further improvements a better understanding of the individual components is needed. Therefore, the German Aerospace Center (DLR) operates a large turbine test facility in Göttingen, called Next Generation Turbine Test Facility (NG-Turb). In this facility novel turbine stages are tested with dried air as working fluid. Some parts of the test bench are operating below the atmospheric pressure.

Figure 1 shows a simplified circuit diagram of the NG-Turb. The working fluid is compressed by a four stage radial compressor unit. Downstream of the compressor unit the flow is divided into a so-called hot stream (orange line), which is equipped with an electrical heater, and a so-called cold stream (upper dark-blue line) wherein a water cooler (aftercooler) is integrated. By controlling the mass flow ratio of the streams and the power of the electrical heater the turbine inlet temperature can be adjusted with high accuracy up to a maximum temperature of $540\,$K. In the test section two independently rotating turbine rigs can be measured with a total shaft power of up to $2.8\,$MW. Downstream of the turbine test section the fluid flows through a throttle and another cooler (precooler) before it enters the compressor unit inlet again. This precooler prevents the exceeding of the compressor inlet temperature design limit of $300\,$K. Another important pipeline section is the bypass which is connected to the cold stream behind the water cooler. The bypass leads the fluid back to the compressor unit inlet. In this section also a tapping point for cooling air is installed. A detailed description of the facility can be found in [1].

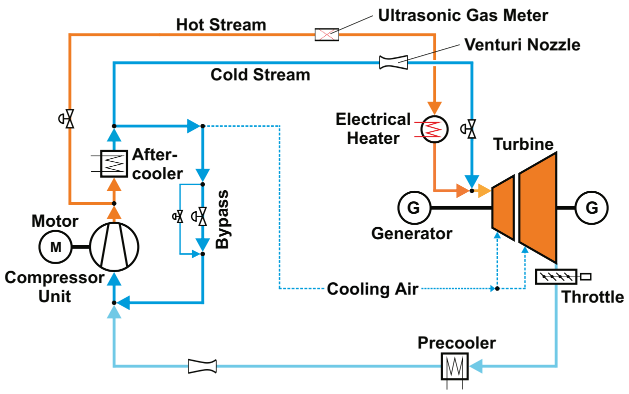

Figure 1: Simplified test bench cycle

For measurement campaigns it is necessary that the test facility provides the working fluid with desired temperature, mass flow rate and pressure to reach similarity parameters. In the planning phase it is difficult to estimate whether an operating point can be reached and in which order the required operating points should be approached. Furthermore, it is not easy to find a set of control values that leads to the requested turbine inlet conditions. These circumstances resulted in the decision to develop a virtual representation of the NG-Turb, initially as a thermodynamic model.

As simulation environment the framework GTlab is chosen. This program was developed within the DLR with primary focus on the preliminary design of aero engines and gas turbines. GTlab provides, among others, a graphical user interface, libraries for thermodynamic components and process elements. The highly modular structure allows an easy extension of the existing libraries and afterwards an easy synthesis with the already implemented components and processes. Moreover the framework contains interfaces that allows simulations with different levels of detail which can be used for future considerations [2–4].

2 MAIN COMPONENTS

For the creation of a thermodynamic test bench model a digital representation of the main components is necessary. In this section the main cycle elements are described. This description also includes the practical occurring boundary conditions, control variables and limits.

2.1 Compressor Unit

The compressor unit consists of four radial stages, two parallel low pressure (LPC), an intermediate (IPC) and a high pressure compressor stage (HPC). Between the LPC and IPC an intercooler is located. All stages are attached to planetary wheels of a planetary gear system so that the stages are driven at same rotational speed. The available power depends on the spool speed and is provided by an electrical motor that generates up to $3.7\,$MW. The operator has the possibility to vary the inlet guide vane (IGV) angles of the compressor stages which are simultaneously varied by stage based adjustment laws. Another way to influence the operating point is the variation of the compressor spool speed. The intercooler has no control variable that can be used during operation.

2.2 Bypass

The bypass is connected to the cold stream and is equipped with a control valve and a fast opening flap. The control valve is mainly used to adjust the mass flow rate of the test section independently from the compressor unit operating point. It should be noted that the valve position has also a slightly influence on the fluid inlet conditions of the compressor unit and thus on the overall behaviour. Parallel to the bypass valve the fast opening flap is installed for emergency procedures.

2.3 Electrical Heater

Upstream of the turbine test rig an electrical heater is installed with a controllable heat output of up to 300 kW. The heater allows a prompt temperature rise or a permanent heat up of the working fluid. In operation it is frequently used to compensate the thermal inertia of the test bench during ramp up and to influence the turbine rig inlet temperature.

2.4 Turbine

The turbine is the measuring section of the test bench. It is possible to install two turbine rigs with 2.5 stages that can be rotated at different rotational speeds. The two turbine spools are connected to generators that regain a part of the electrical input energy. The control variables of the rigs, for example the spool speed, the IGV-positions, the coolant mass flow rate etc., can be seen as boundary conditions. An important aim of the NG-Turb virtualization is the prediction of the turbine inlet conditions.

2.5 Throttle

Downstream of the turbine a throttle is installed, which can rise the pressure ratio over the cycle. In practice this control value is often used because compared to other control variables this valve has a minor impact of the overall system and thus of test bench stability.

2.6 Water Cooler

There are three main water coolers in the circuit that are affecting the temperature of the working fluid. A cooler is placed between the LPC and IPC (intercooler), behind the compressor unit in the cold stream (aftercooler) and between the turbine and the compressor in the return flow (precooler). All of these coolers are not adjustable. The cooling water mass flow rate is fixed and the temperature is set by the connected water reservoir conditions. This reservoir is cooled by a cooling tower whereby the cooling power depends on the ambient conditions and the operational time.

2.7 Piping

The Mach numbers in the main test facility piping are mainly low and the pipe surfaces are well isolated. As a result, for a warmed-up test bench only moderate pressure and temperature losses are expected. But for cold starts the cooling effect that results from the thermal inertia of the piping has a non-negligible influence on the operation.

The test bench contains further components, which are classified as less important for the demanded level of detail. As a result, the GTlab performance code was extended by the necessary functions to simulate the described main components.

For example, a new compressor component that determines outlet conditions from a three dimensional performance map was implemented. In these maps the pressure ratio and the efficiency are assigned to the current reduced mass flow rate, the relative reduced spool speed and the IGV-position. The map enables inter- and extrapolation between the given map values, which are adapted from compressor manufacturer data-sheets. Moreover, a digital representation for the water cooler including the fluid water, the electrical heater and the bypass valve were implemented.

3 SYNTHESIS

The synthesis of the performance components to the whole test bench is carried out stepwise. This allows identifying errors in the physical component description and in the created equation systems which are used to add further constrains and component dependencies.

In the following performance model illustrations, the circuit is starting at so-called FlowStart-elements (source of fluid) and ending at FlowEnd-elements (sink of fluid). Between the source and sink the cycle is modelled with further performance elements that are connected with each other. Thereby the blue lines represent working fluid connections, the red lines shaft connections and yellow lines connections of the secondary air system (SAS). The numbers that are shown between the elements represent stations wherein fluid conditions are calculated and stored to the results.

3.1 Detached Compressor Model

This performance model is set up to create the compressor stage adjustment laws, to check the implemented compressor representation in combination with the compressor maps and to gain initial experience with the compressor unit behaviour.

The new compressor component uses a performance map that needs the stage IGV-angle for the calculation. In practice these angles are set for the whole compressor unit by a single parameter, the IGV-position. In reality, this value is transformed by an internal control software of the compressor according to fixed adjustment laws. For the thermodynamic model, stage specific adjustment laws are deviated from compressor data sheets and manufacturer simulation results that were re-calculated with the current compressor performance model that is shown in figure 2. The used manufacturer simulation results

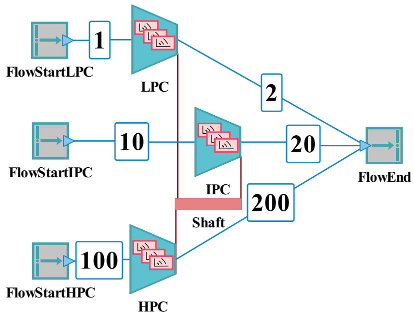

Figure 2: Detached compressor performance model

contain the stage in- and outlet fluid conditions (total pressure, total temperature, mass flow rate, fluid composition), the compressor spool speed and the power consumption of the whole compressor unit. In the compressor model the inlet conditions for every stage (FlowStartLPC, FlowStartIPC, FlowStartHPC) and the shaft speed were adapted from the manufacturer simulation results. The equation system is used to reach the stage outlet pressures by varying the IGV-angles. The calculated solutions were verified with the amount of the stage outlet temperature deviation. This process was repeated for every given operating point. The found IGV-angles are combined with additional information from manufacturer data sheets that includes the basic shape of the adjustment law and the border of the IGV-angles. In figure 3 the IGV-angles of the IPC and the HPC are plotted over the LPC-angle. In this chart the points represent the calculated positions and the lines the created adjustment laws. The deviations in this plot are mainly caused by the roughly rounded manufacturer simulation results

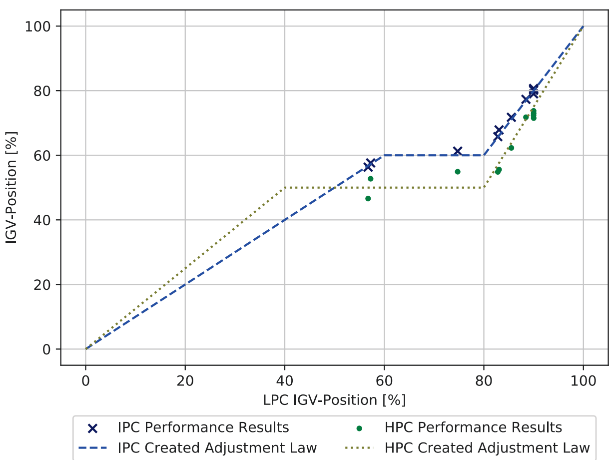

Figure 3: Compressor stage adjustment laws

and possibly a further slight deviation can be caused by differ map creation as well as access processes.

3.2 Compressor Unit Model

In the next step a representation of the compressor unit is modelled whereto the compressor stages were directly connected (see figure 4). In the compressor unit performance model the two parallel low pressure stages are summarised in one component, the LPC. Hereby the computational time and the convergence behaviour of the system could be improved. Between the LPC and IPC a water cooler (Intercooler) is placed. This component influences the temperature and reduces the pressure of the working fluid. For the following simulations the heat losses in piping are disregarded and therefore the pipes are simulated as ducts that only model pressure losses. Moreover, the adjustment laws derived in section 3.1 are applied to the compressor stages.

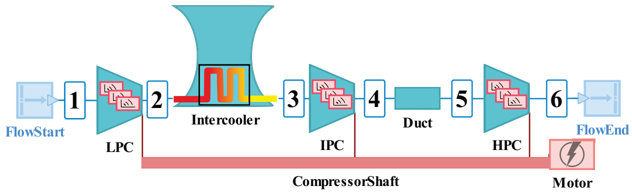

Figure 4: Compressor unit performance model

This compressor unit representation is verified with the manufacturer simulation data that were already used for the adjustment law creation. Therefore, the inlet fluid properties, the intercooler fluid outlet temperature and the compressor spool speed are set as boundary conditions. The stage IGV-angles are simultaneously varied by a single independent, the IGV-position, in order to reach the demanded outlet pressure. For the validation the remaining fluid properties and the compressor unit power consumption are used.

The differences between the simulation results are converted to percentage variance whereby the manufacturer data are used as reference values. The obtained deviation for the pressure is less than $2\,\%$ ($\Delta p_{max} = 990\,$Pa), for the temperature less than $0.3\,\%$ ($\Delta T_{max} = 1.3\,$K) and for the total power consumption less than $0.55\,\%$ ($\Delta P_{max} = 26.9\,$kW). Therefore, the performance setup within the applied components and the implemented adjustment laws can be seen as sufficient modelled. A detailed analysis of the distribution can be found in the appendix A.1.

3.2.1 Influence Of The Control Variables

For a better understanding of the compressor unit control variables a parametric study was carried for a specific operating point. For this study the inlet temperature and the inlet pressure of the compressor unit model are set as fixed boundary conditions. To reach the required outlet pressure the mass flow rate and either the mechanical shaft speed or the IGV-opening position are varied.

The study results have shown that a decreasing IGV-opening position reduces the mass flow rate and the power consumption but increases the compressor unit outlet temperature. In the case of reduced shaft speed all considered parameters are decreasing (see table 1). Because of the complex interaction

constant	shaft speed	IGV-opening
changed	IGV-opening ↓	shaft speed ↓
temperature	↑	↓
mass flow rate	↓	↓
power consumption	↓	↓

Table 1: Influence of the compressor control variables

of the compressor maps and the decoupling of the stages by the intercooler the presented trends cannot be generally transferred to all combinations of fluid inlet conditions and control parameter settings. However, further simulations have shown that the trend can be used for the surrounding operating area and thus for the upcoming measurement campaign.

3.2.2 Convergence

The main challenge for the convergence of the compressor unit model is the interaction of the three compressor stages which map positions are decoupled by the intercooler and which are all driven with the same mechanical spool speed. Moreover, the shape of the performance map is depending on the given IGV-angles. During a calculation compressor components are interpolating the pressure ratio and the efficiency based on the relative reduced spool speed and the reduced and standardised mass flow rate (equation 1 [5]). This means that for a given spool speed line a limited mass flow rate range can lead to valid results that are placed between the surge margin line and the choke line. If a set of boundary conditions leads to a converged LPC result, then the temperature and the pressure of the fluid is decreased by the intercooler. Afterwards the compressor map determination repeats for the IPC and the HPC. Thereby, small variations in the parameter setting may lead to massive changes on the compressor unit behaviour. For example, a small mass flow rate offset leads to strong changes in the predicted pressure ratio, if the compressor operating point is near the choke line. As a consequence of this, parametric studies must be carried out with small changes in the control variables and the inlet fluid conditions to find enough converged results. Practical experience has shown, that the IGV-opening position should be varied by 1.0%-steps between $24 \ldots 100\%$, the initial relative mechanical shaft speed by 5%-steps between $85 \ldots 105\%$ and the natural mass flow rate by 0.5 kg/s-steps between $1 \ldots 20$ kg/s. This means that for a given pressure ratio 15015 calculations are necessary.

$$n_{red-rel} = \frac{n_{red}}{n_{red-ref}} \quad \text{with} \quad n_{red} = \frac{n}{\sqrt{T}}$$
$$w_{red-std} = w \cdot \frac{p_{ref}}{p} \cdot \sqrt{\frac{T}{T_{ref}}} \tag{1}$$

3.2.3 Validation

To ensure that the thermodynamic model is suitable for modelling the test bench behaviour a validation with measured data is carried out. The validation data is gained from measurements during the compressor unit initial run. In this initial run the working fluid was led from the compressor unit outlet through the aftercooler and the bypass back to the compressor unit inlet. This setting enables the re-calculation of the measurement points with the compressor unit performance model.

During certification of the compressor, various compressor data was collected for 13 different operating points. The results contain the compressor unit inlet and outlet fluid temperature and pressure, the mass flow rate, the mechanical compressor spool speed and the electrical power consumption of the drive motor. These results cover a pressure ratio range from 3.1 to 12.8 with reduced and standardised LPC mass flow rates from 24.7 kg/s to 75.4 kg/s.

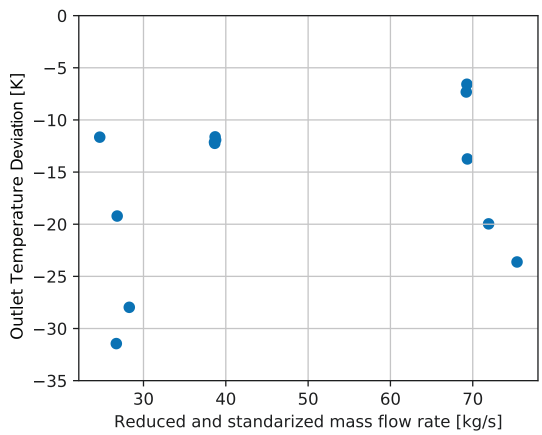

Figure 5: Temperature deviation over reduced and standardised LPC mass flow rate

Figure 5 shows the absolute temperature differences between the calculated and measured data over the reduced and standardised compressor unit inlet mass flow rate. It can be seen that all calculated temperatures are higher than the measured ones which leads to the negative values. This systematic deviation reflects the disregarding of the pipe heat losses. The observed deviations are in an offset range from $-31.45\,\mathrm{K}$ to $-6.6\,\mathrm{K}$. The power consumption differ-

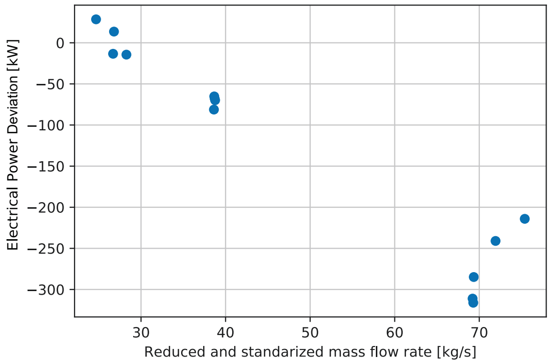

Figure 6: Power consumption deviation over reduced and standardised LPC mass flow rate

ences are shown in figure 6. The predicted values tend to higher deviations by increasing mass flow rates. If the results are corrected with a linear polynomial (see equation 2), that is created via orthogonal regression, the deviation is in a range of $-56\ldots83\,\mathrm{kW}$. If the measured data are used as reference the, range corresponds to $-2.3\ldots3.1\%$.

$$f_{cor}(w_{red-std}) = -6120.14\,\frac{\mathrm{W}}{\mathrm{kg/s}} \cdot w_{red-std} + 164279.7\,\mathrm{W} \quad (2)$$

The correction of the observed deviation by equation 2 for power consumption and by the averaged offset of $-16.1\,\mathrm{K}$ for the temperature is the simplest way but it can lead to extrapolation issues. Therefore a physical modulation should be preferred.

3.2.4 Model Improvement

For the reduction of the observed simulation errors (see section 3.2.3) a piping representation that predict the heat loss based on the working fluid conditions was implemented. Equation 3 shows the applied heat flow equation within the approach for the thermal conductivity k. For this, the piping conductivities k_i from the inner pipe surface, the pipe material, the isolation material and the outer surface are important, whereby all material conductivities are assumed as constant and were adapted from [6]. The outer conductivity is also presumed as constant due to the nearly steady air conditions in the test facility building.

$$\dot{Q} = k \cdot A \cdot \Delta T \quad \text{with} \quad \frac{1}{k} = \sum_{n}^{i=0} \frac{1}{k_i} \quad (3)$$

With equation 4 the inner conductivity is determined by the convective heat transfer coefficient h, the diameter d and the current Nusselt number Nu. The Nusselt number is calculated by using correlations taken from [6]. The diameter is adapted from engineering drawings and the heat transfer coefficient is taken from property tables for current fluid conditions.

$$k_{inner} = \frac{Nu \cdot h}{d} \quad (4)$$

This method is applied between the intercooler and the IPC between the IPC and the HPC. A further implementation is not necessary because of the intercooler and the measurement position in front of the LPC. In the first calculation run the fixed parts of the heat conductivity are adapted to the averaged temperature deviation of $-16.1\,\mathrm{K}$ for a randomly selected operating point. The determined conductivity is used as boundary condition for the following calculations.

The reworked performance model leads to a worse convergence behaviour, the most operating points did not converge. The successfully calculated results are plotted in figure 7. It can be seen, that the temper-

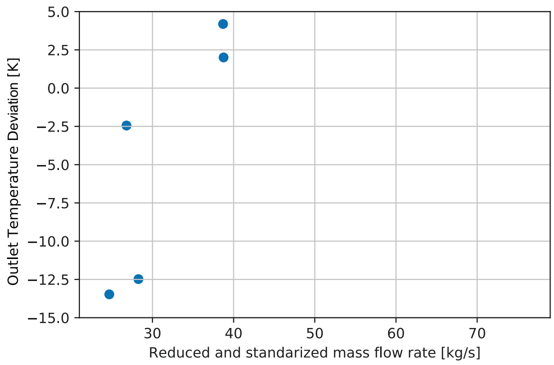

Figure 7: Temperature deviation over reduced and standardised LPC mass flow rate for the piping performance model

ature offset range could not be narrowed but the absolute simulation results are moved toward the measured values. The power consumption deviation has increased by the implementation of the piping elements (see figure 8).

The reason for the deterioration of the convergence behaviour and the predicted power consumption could not be fully clarified, but one cause could be that the used compressor maps already include the temperature losses. This assumption based on the fact that in the manufacturer simulations the heat losses are not explicit considered. This could lead to the incompatibility of the compressor stage maps and thus to the observed reduced convergence and increased power consumption deviation. Hence the averaged temperature offset and the created power correction polynomial are used for the following calculations.

3.3 Test Bench Model

Figure 9 shows the test bench performance model that is based on the previous described compressor

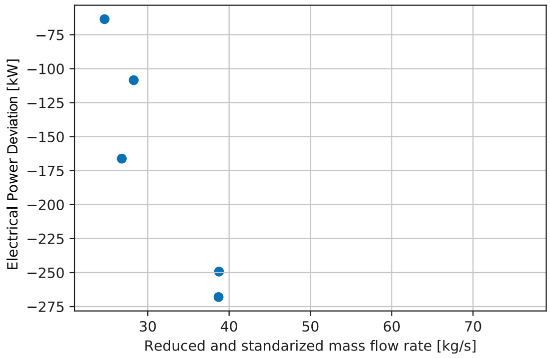

Figure 8: Power consumption deviation over reduced and standardised LPC mass flow rate for the piping performance model

unit model. Downstream of the compressor unit the stream is split (Splitter-Hot-Cold) into the hot stream (lower part) and the cold stream (upper part). The reunion of these streams in front of the turbine is realised by a second splitter (Splitter-Main-Cold-Bypass) and a flow merger (FlowMerger-Hot-Cold). The remaining mass flow rate of the cold stream is used for the turbine cooling (SAS) or flows through the bypass line (Bypass). Before the working fluid exit the cycle by the FlowEnd all streams are merged together, whereby the turbine outlet flow is led through a water cooler (Precooler). The equation system is finally used to close the cycle by link the fluid exit conditions to the fluid entry (FlowStart).

Initial simulation tests have shown that it is difficult to find input parameter sets that lead to convergent results. Already the compressor unit model has small control variable areas in which the simulations converge (see section 3.2.2). The extension to the test bench model leads to further deterioration of the stability. This makes it necessary to carry out prior compressor unit studies. From the gained results the most promising parameter sets are used as initial values for the overall simulation.

To improve the process stability and raise the automation the test bench model is split into three calculation steps. As first step the compressor unit is calculated as described in section 3.2. In the case that the used input settings lead to a converged result the outlet conditions of the compressor unit are transferred to the second system that contains the remaining test bench (starting on station 6 in figure 9). If this second performance model is also converged, the outlet conditions are set back to the inlet of the compressor unit. This process is repeated until the changes are under a certain residue or the calculation failed. Through this procedure the preliminary studies could be avoid and the calculation split reduces also the equation system, whereby the amount of converged results increases.

4 APPLICATION

A practical application arose from an ongoing measurement campaign. Some demanded turbine inlet conditions could not be reached. For requested measurement points low temperatures in combination with small pressure ratios and small mass flow rates are required. In the considered case the cold stream could not be used because of the lower achievable accuracy of the installed mass flow rate measurement system, a Venturi nozzle. As a result, the turbine inlet temperature is similar to the compressor unit outlet, what leads to the requirement to reduce the compressor unit outlet temperature. Therefore, simulations of possible technical solutions were carried out to estimate the potential of the modifications.

4.1 Assumptions And Simplifications

For the described test bench prediction, a complex calculation procedure that consumes a lot of time per operating point is necessary. Through the closing of the loop cycle the inlet conditions of the compressor unit are depending on his own behaviour in combination with the remaining test bench. Additional, the test rig is often an unknown element. For the prediction of the rig behaviour a valid performance map is necessary that is typically not available at early stage of a measurement campaign. Moreover, the simulation experience has shown, that already the convergence behaviour of the compressor unit is a challenging task. Hence some assumptions and simplifications are introduced for the following studies:

- All cooler outlet temperatures are considered as equal because the installed components are dimensioned sufficiently large and as a result the temperature difference is insignificant.
- The given operating points are stationary. This means that no time dependent changes are considered.
- The compressor unit inlet conditions are regarded as independent from the calculated compressor outlet properties. This assumption will still yield to realistic results because the throttle downstream of the turbine in combination with the three main cooler in the cycle and an installed pressure control system are leading to widely independent compressor unit inlet conditions.
- For the following study the digital representation is reduced to the compressor unit model. This is possible because only the hot stream is used. Therefore, the turbine inlet conditions can be equated with the compressor unit outlet conditions.

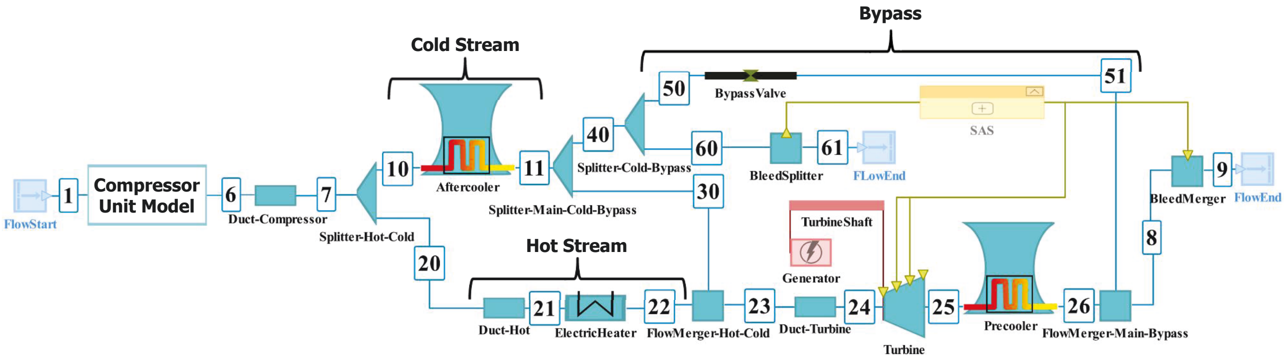

Figure 9: Test bench performance model

These simplifications allow parametric studies over the full control variable range within a reasonable time span and a satisfying convergence behaviour.

4.2 Simulation Setup And Results

The following calculations are based on the compressor unit performance model (figure 4). For the prediction the whole test bench operating area is covered. This means that the mass flow rate, the IGV-opening and the initial compressor spool speed are set as input values and are varied as described in section 3.2.2. The equation system varies the spool speed until the demanded outlet pressure is met. Additionally, the inlet pressure is varied between $20\,\mathrm{kPa}$ and $80\,\mathrm{kPa}$ in $5\,\mathrm{kPa}$-steps to take possible pressure ratio changes into account. If a dataset does not lead to a converged result, the operating point is considered as not reachable.

Not all converged results from the parameter studies fulfil the physical boundary conditions of the test bench. There are limitations at the compressor input power, the piping temperature, the mechanical spool speed and the surge margin distance. Therefore, the calculated results are filtered in a post process.

4.2.1 Flexible Coolant Heat Flux

In this parametric study the intercooler outlet temperature is set as an additional boundary condition and is varied in the range of $283.15 \ldots 563.15\,\mathrm{K}$. The other coolers are further considered as not flexible and consequently the compressor unit inlet temperature is set to the reference temperature of $315.15\,\mathrm{K}$. The results of this study show the influence of a flexible intercooler performance whereby the temperatures below the reference temperature equals an increased heat flux and the area above $315.15\,\mathrm{K}$ equals a decreased cooling power. In figure 10 the compressor unit outlet temperature is plotted over the intercooler outlet temperature. The vertical red line shows the reference cooler outlet temperature of $315.15\,\mathrm{K}$ and the different symbols represent the used compressor unit inlet pressure. It can be seen that an increased intercooler temperature results in a rising compressor unit outlet temperature. This shows that the compressor map position change for the IPC and HPC could not

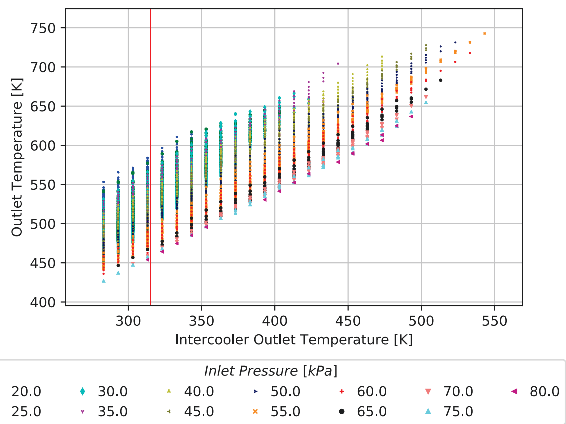

Figure 10: The influence of a flexible intercooler operation on the compressor unit outlet temperature

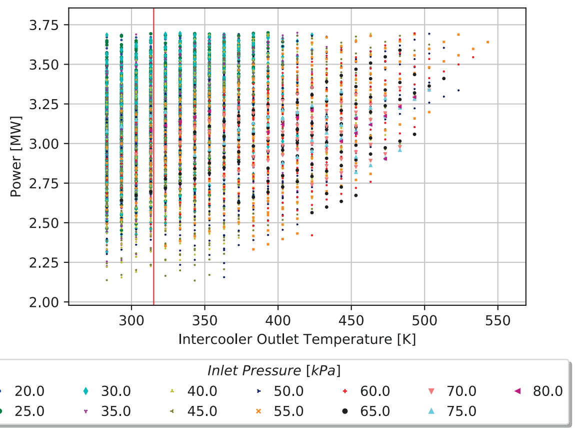

Figure 11: The influence of a flexible intercooler operation on the compressor unit power consumption

lead to an overall efficiency rise that reduces the outlet temperature. Moreover, it is not possible to lower the power consumption due to the changed mass flow rates that are resulting from reduced intercooler performance. The power consumption even tends to rise with the intercooler temperature (see figure 11). Also, the upper power limit of the drive motor by 3.7 MW can be seen in this chart.

4.2.2 Spool Speed Range Extension

In this study the extension of the mechanical compressor spool speed range is investigated. A prior vibration analysis has shown that the spool speed can be reduced from current lower limit of the relative reduced spool speed $n_{rel} = 0.85$ to $n_{rel} = 0.7$ without the danger of running into a drive system resonant frequency. From manufacturer side an adjustment on the compressor control software to avoid operation above the surge limit and an adjustment on the installed frequency converter is necessary. For the current study the compressor unit initial spool speed range has been extended to $n_{rel} = 0.7 \ldots 1.05$. For this purpose, also the post procedure has been adapted.

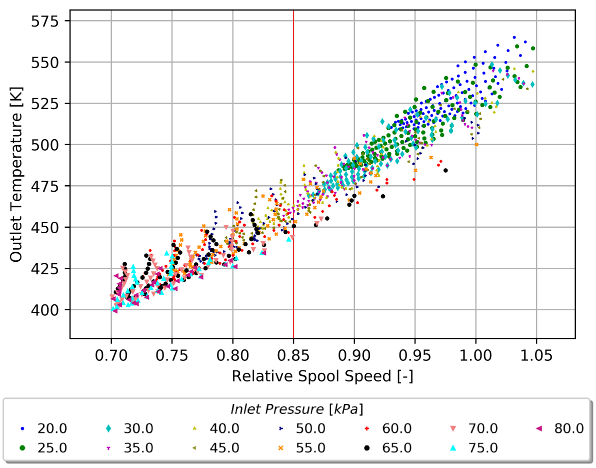

Figure 12: The influence of a compressor spool speed range extension on the compressor unit outlet temperature

In figure 12 the predicted compressor outlet temperature is plotted over the relative mechanical spool speed. The different symbols represent the current inlet pressure. It can be seen that the spool speed range extension leads to possible operating points that provide the needed combination of reduced temperature and pressure ratio. Moreover, the study indicates that the reduced spool speed will lead to lower electrical power consumption (see figure 13). Moreover, it can be seen that the maximum power consumption drops with decreasing spool speed. The reason for this is that the drive motor limit with 3.7 MW is not

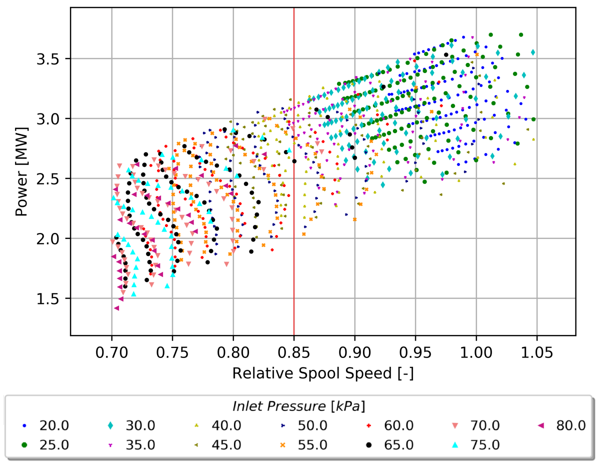

Figure 13: The influence of a compressor spool speed range extension on the compressor unit power consumption

the only limitation. For relative spool speeds below one the compressor shafts are limiting the transmitted power with an additional torque limit whereby the linear decrease is caused.

4.3 Realisation

The simulation results have shown that the compressor spool speed range extension enables the achievement of lower turbine inlet temperatures in combination with lower pressure ratios and with lower power consumptions. The prediction of the flexible intercooler heat flux indicates that with a higher cooler outlet temperature no improvement for the considered operational point can be reached. Only an intercooler outlet temperature reduction can lower the compressor unit outlet temperature. This is achievable through a higher cooling efficiency that is not easy to implement. Moreover, the gained effect is minor in comparison to the spool speed range extension.

Consequently, this leads to the decision to realise the compressor spool speed range extension. The necessary adaptions and tests were carried out within in three days. Afterwards the required operating points were successfully approached.

5 CONCLUSION AND OUTLOOK

The main components of the turbine test bench NG-Turb were identified and physical representations were developed and implemented in the used GTlab performance code. A detached compressor model was set up to create adjustment laws which describe the relative inlet guide vane angles of the compressor stages. Afterwards the performance components were synthesised to a compressor unit and a test bench performance model.

The compressor unit model was validated with manufacturer simulation results and measurement data. Moreover, this model was successfully used for the prediction of test bench adaptions, the controllable intercooler performance and the compressor spool speed range extension.

The analysis of the results led to the realisation of the compressor spool speed range extension that enables the facility to reduce the outlet temperature, the pressure ratio and the power consumption. Through this modification the required turbine inlet conditions could be achieved.

To improve accuracy of the model the warm up phase of the test bench should be considered. Therefore, the instationary thermal effects of the test bench have to take into account. New measurement results have also shown that the accuracy of the simulation result decreases with a growing distance to the compressor design point. Especially in the extended spool speed range increasing discrepancies were detected. A further investigation indicates that the used compressor maps could cause this deterioration. These performance maps are based on measurement points that are record with the relative design spool speed for different inlet guide vane angles. The other spool speed lines were determined by a simplified correlations, the fan laws. Consequently, the compressor representation should be improved by applying a method that can be dynamically enhanced through the continuously growing measurement data base.

ACKNOWLEDGEMENT

This project has received funding from the Clean Sky 2 Joint Undertaking (JU) under grant agreement No 945541. The JU receives support from the European Union's Horizon 2020 research and innovation programme and the Clean Sky 2 JU members other than the Union.

Nomenclature

Abbreviations

DLR .... German Aerospace Center
GTlab ...... gas turbine laboratory
HPC .... high pressure compressor
IGV .... inlet guide vane
IPC .... intermediate pressure compressor
LPC .... low pressure compressor
NG-Turb .... Next Generation Turbine Test Facility
SAS .... secondary air system

Symbols

Δ .... difference between two states
$\dot{Q}$ .... heat flux
A .... area
d .... diameter
h .... convective heat transfer coefficient
k .... heat conductivity
n .... rotational speed
Nu .... Nusselt number
P .... power
p .... pressure
T .... temperature
w .... mass flow rate

Indices

i .... part size value
$inner$ .... inner side value
max .... maximum value
red .... reduced value
ref .... reference value
rel .... relative value
std .... standardised value

References

[1] Hans-Jürgen Rehder et al. "Next generation turbine testing at DLR". In: *Turbo Expo: Power for Land, Sea, and Air*. Vol. 50787.

[2] Richard Becker et al. "Development of a Gas Turbine Performance Code and its Application to Preliminary Design". In: *Deutscher Luft- und Raumfahrtkongress DLRK* (2011).

[3] Stanislaus Reitenbach et al. "Collaborative Aircraft Engine Preliminary Design using a Virtual Engine Platform, Part A: Architecture and Methodology". In: *AIAA SciTech Forum* (2020).

[4] Maximilian Vieweg et al. "Collaborative Aircraft Engine Preliminary Design using a Virtual Engine Platform, Part B: Application". In: *AIAA SciTech Forum* (2020).

[5] Jack D., Mattingly, and William H. Heiser. *Aircraft Engine Design*. American Institute of Aeronautics & Astronautics, 2002. ISBN: 1-56347-538-3.

[6] Stephan Kabelac, Verein Deutscher Ingenieure, and Gesellschaft Verfahrenstechnik und Chemieingenieurwesen. *VDI-Wärmeatlas*. 10, bearbeitete und erweiterete Auflage. Springer, 2005. ISBN: 978-3-540-25504-8.

A Appendix

A.1 Manufacture Simulation

Figure 14 and 15 show the deviation between the manufacturer simulation results and the thermodynamic compressor unit performance model for the station temperature and pressure values (see figure 4). The plotted deviation is calculated by using the manufacture values as reference.

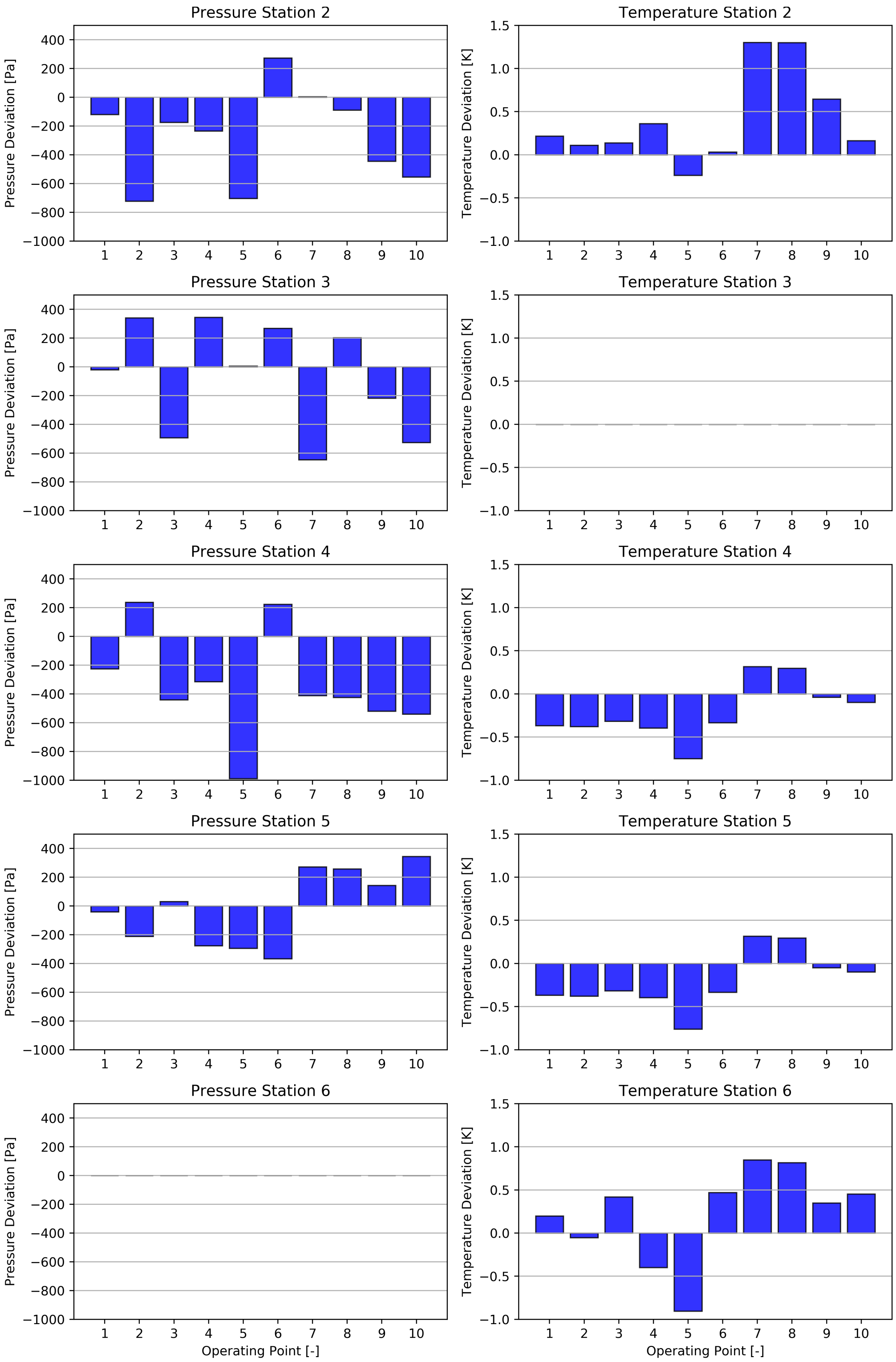

Figure 14: Deviation between the performance model and the manufacture simulation results

The shown differences correspondent to less than two percentage deviation for the pressure values and less than one percentage deviation for the temperature values.

It should be noted, that the temperature for the third station is a boundary condition and the compressor unit outlet pressure (Station 6) is a target value of the equation system. Hence there are no deviations for the station fluid properties detected. Moreover, the heat losses in the piping are not modelled in both simulations and so the temperature deviation of the stations 4 and 5 have the same values.

Figure 15 shows the deviations of the calculated compressor drive engine power consumption. For the considered operating points the maximum difference is 27 kW that is corresponding to less than 0.55%.

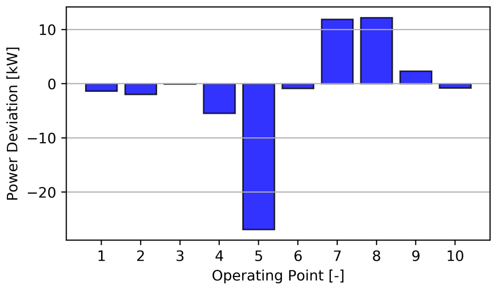

Figure 15: Compressor unit power consumption deviation between the performance model and the manufacture simulation results