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Model tests on the control behaviour of a test air supply system in open or closed-loop operation

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Abstract. The Leibniz Universität Hannover is currently establishing a new mechanical engineering campus which includes a new research building "Dynamics of Energy Conversion" (DEW). This building provides a large compressor station for either steady or dynamic (transient) operation of turbomachinery and power plant test rigs (e.g. air turbine, axial compressor, combustion chamber, planar cascade, acoustic wind tunnel). The test air supply system is designed to enable investigations under high load gradients over wide operating ranges with Reynolds and Mach numbers controlled independently in order to fulfil aerodynamic similarity conditions between reality and model experiments. This is achieved by closed loop operation of the test air supply system which allows independent adjustment of pressure, temperature and volume flow rate as well as independence from environmental influences such as temperature or humidity. The compressor station utilizes as first stage two parallel Roots-type PD compressors and as second stage two parallel screw compressors. The test rigs operate at expansion ratios between 1 and 6. Test rig inlet pressures range from 1 bar(abs) to 8 bar(abs) with a maximum mass flow rate of 25 kg/s. At all conditions temperatures can be regulated between 60°C and 200°C. The test air supply system has a maximum electric power input of approximately 6 MW. As stringent demands on stability and reproducibility have to be met and automatic operation was requested, a scaled and simplified but fully functional model of the test air supply system was built, mainly to enable testing of control methods and devices prior to their final implementation on site. The functional model uses DN150 piping and consists of one Roots-type PD compressor as first stage and one screw compressors as second stage. Both compressors are driven by electric motors regulated by frequency converters. A turbine test rig is represented in the model by an adjustable throttle valve. Precise control of the mass flow rate is provided by a cascaded adjustable bypass around the test rig. The paper describes the test air supply system and the scaled model and presents experimental results on the achievable stability of pressure, temperature and mass flow rate at the test rig inlet in steady operation at several operating conditions of the model.

1. Introduction

The erection of a new mechanical engineering campus (Campus Maschinenbau Garbsen – CMG) for the Leibniz Universität Hannover situated in Garbsen (north of Hanover) started December 2015. Planned for 2019, the campus will provide capacity for about 4500 students and 800 faculty staff members and will concentrate all 18 institutes of the Faculty of Mechanical Engineering in one location.



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A large part of the total investment costs of approximately 145 million Euro is earmarked for the research building “Dynamics of Energy Conversion” (DEW). The building, costing around 42 million Euro, includes a number of test benches for experiments on turbomachinery and power plant components such as engines, generators, turbines, diffusers and compressors.

Research will focus on aerodynamics, aero elasticity and combustion processes in order to better understand and improve the energy conversion processes with the aim to increase efficiency and reduce emissions. Typical investigations are carried out using e.g. a seven stage model turbine [1], a linear cascade wind tunnel [2] or a high-speed axial compressor [3].

Due to the constantly increasing possibilities of simulation methods and the decreasing remaining improvement potential of the components under test, the requirements for test benches are increasing likewise. Today, simulations are used for test pre-prediction which allows for improved positioning and adequate selection of measurement sensors and equipment as well as for optimized overall planning of the experiments. By this, costs are decreased while the quality of the results is enhanced but the requirements for reproducibility, stability and similarity of the experiments increase as well.

As a result, there are stringent requirements for the test air supply system as test air is needed for operation of most of the test benches. Other test benches like the high speed axial compressor that produce their own air flow need to use components of the supply system like e.g. piping, mass flow rate measurement and throttling devices.

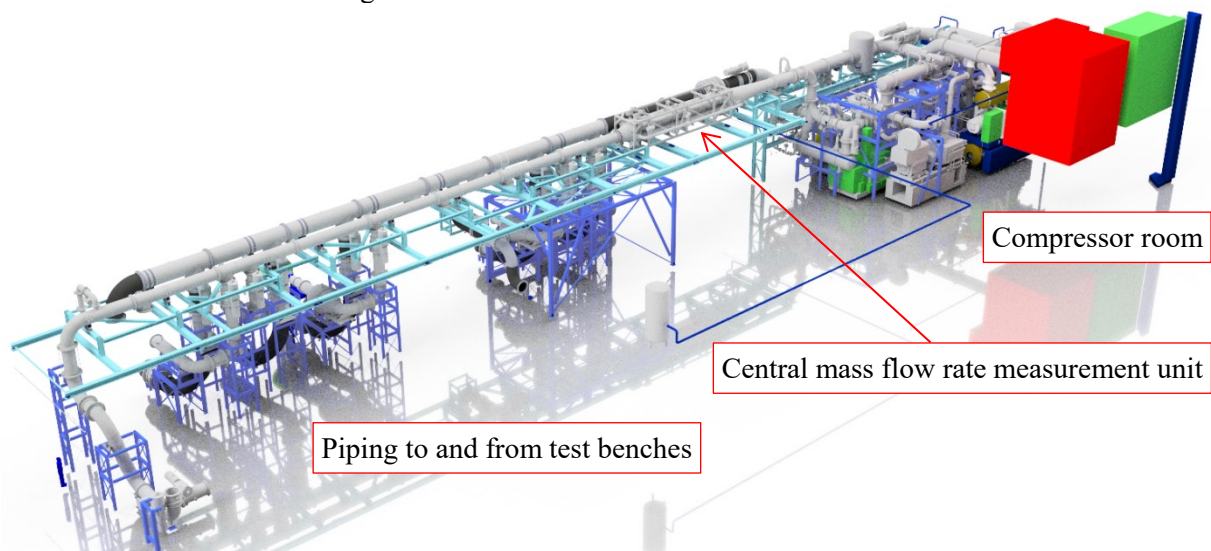


Figure 1. Complete view of the test air supply system (from south).

2. The full-scale test air supply system

Aerzener Maschinenfabrik acquired the order to develop and deliver such a test air supply system. The overall concept foresees a compressor station including temperature control of the test air, a mass flow rate control bypass, a central mass flow rate measurement device, an air distribution system to seven selectable test benches including piping, valves, silencers, coolers etc. and a control system for the selection of different operation modes, -types, -configurations and test bench inlet conditions. The system can be operated in open loop or closed loop mode. The latter allows changing the overall air mass content of the system by either pressurizing or evacuating. Figure 1 depicts a complete view of the test air supply system. The eastern part on the right of the picture shows the compressor room, the green box represents the inlet housing, the red box the outlet housing. In the center, the mass flow rate measurement is situated and in the western part the test air is distributed to the test benches. The overall dimensions are approximately 82 by 15 by 9 m.

The layout of the compressor station shows a symmetrical, modular structure. Two parallel identical lines each comprise a large Roots-type positive displacement compressor (PD blower) as a

first and a dry screw compressor as a second stage. All four compressors are driven by separate 690 V electrical motors with separate frequency converters. Due to the modular structure the compressor station is highly flexible, has a large control range (see figure 6) and runs at high efficiencies even at part load conditions. In addition, a certain redundancy is implemented as the second stage of one line can be connected to the first stage of the other line as well. Operation using only screw compressors or only PD blowers is also possible. In these cases the stage which is not needed is bypassed. The compressor room is visible in figure 2. Differently from figure 1, inlet and outlet housings are not shown. The PD blowers are situated on the right and the screw compressors on the left.

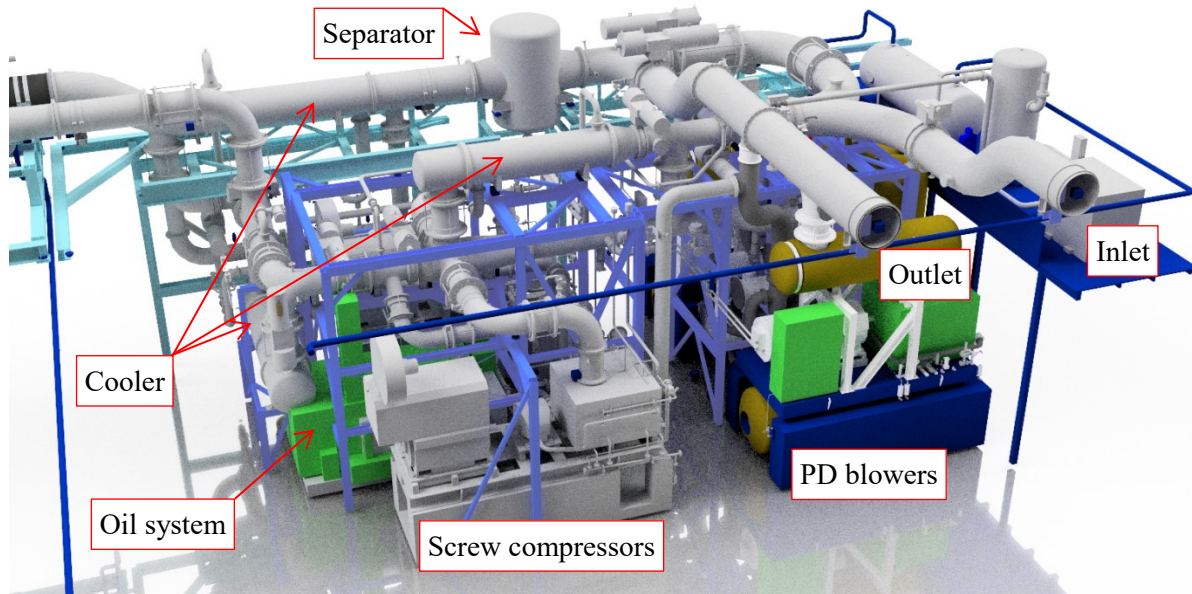


Figure 2. Compressor room.

In order to fulfil aerodynamic similarity conditions between reality and model experiments as good as possible, the ability to independently control Reynolds and Mach numbers is necessary. Considering those values that can be influenced by the test air supply system, the Mach number Ma is proportional to the volume flow rate \dot{V} and inversely proportional to the square root of the absolute test air temperature T

$$Ma = \frac{c}{a} = \frac{\dot{V}}{A} (\kappa RT)^{-0.5} \Rightarrow Ma \sim \frac{\dot{V}}{T^{0.5}} . \quad (1)$$

As air always being the test fluid, the material properties κ and R cannot be significantly altered, but ambient air can still slightly change composition due to variations in absolute humidity. This is the reason why for closed loop operation, a special operating mode to dry the air content is foreseen prior to critical test runs (see section 2.2.).

The Reynolds number Re is proportional to volume flow rate \dot{V} , pressure p and characteristical length L and inversely proportional to the dynamic viscosity η and the absolute test air temperature T

$$Re = c \frac{L}{\nu} = \left(\frac{\dot{V}}{A} \right) L \left(\frac{\rho}{\eta} \right) = \dot{V} L p (A \eta R T)^{-1} . \quad (2)$$

For moderate pressures (the compressor station can deliver an absolute pressure up to 8.0°bar) the dynamic viscosity of air is only slightly influenced by pressure and proportional to approx. the square root of the absolute temperature [4]. Therefore, from equation (2) follows that the Reynolds number is proportional to the volume flow rate, the pressure and the characteristical length and inversely proportional to the absolute test air temperature to the power of 1.5.

$$Re \sim \frac{\dot{V}}{T^{1.5}} p . \quad (3)$$

Comparing equations (1) and (3) shows that the influence of the temperature of the test air delivered by the supply system on the Reynolds number is higher than on the Mach number. Furthermore, pressure can be used to adjust the Reynolds number without a significant change of the Mach number.

2.1. Main components

2.1.1. Positive displacement blowers. The compressor station utilizes as first stage two parallel Roots-type PD compressors GM20.20, each with an intake volume flow rate between 9600 and 48600 m³/h. They can operate with inlet pressures between 0.2 and 3.5 bar (abs). The maximum outlet pressure is 4.3 bar (abs) and the maximum pressure difference is 0.8 bar. The drive shaft sealing system uses oil lubricated mechanical face seals. The oil system is external. The PD blowers are driven by separate variable speed 690 V electrical motors with a nominal power of 1250 kW each.

2.1.2. Screw compressors. As second stage two parallel screw compressors VRa736S, each with an intake volume flow rate between 6900 and 21600 m³/h are installed. They can operate with inlet pressures between 0.25 and 3.5 bar (abs). The maximum outlet pressure is 9 bar (abs) and the maximum pressure difference is 10 bar. The rotors are supported by hydrodynamic sleeve bearings. The screw compressors are driven by separate variable speed 690 V electrical motors with a nominal power of 2700 kW each.

2.1.3. Silencers. All four PD blowers and screw compressors are equipped on their high pressure sides with separate silencers in order to deliver pulsation free air to the test benches and to allow for low sound emission levels of the extended pipework outside the compressor room. Special care had to be taken of the blower silencers to be able to suppress the extremely wide frequency spectrum due to their large operating range. Simulation showed that it is necessary to use a combination of two reactive-type silencers and additionally two quarter-wave tubes to attenuate the very low frequencies created by the PD blowers running at minimum speed.

2.1.4. Coolers and temperature control. Between the PD blowers and the screw compressors a common intercooler is situated. All coolers are of the air – water type. Cooling water is centrally provided and re-cooled in a special building on campus.

A common aftercooler including temperature control of the test air is placed downstream of the screw compressors. At all conditions the inlet temperatures to test benches can be regulated between 60°C and 200°C. This is achieved by cooling the main part of the air flow to a temperature slightly lower than required and mixing this portion with a controlled amount of hot air that is bypassing the aftercooler. In those cases when the compressor outlet temperature is lower than the required inlet temperature of the test bench (because the pressure required at the test bench corresponds to a lower temperature) the air is compressed to a higher pressure (and therefore temperature) by using an additional throttling valve upstream of the aftercooler. Due to the maximum absolute pressure of the compressor station of 8 bar the test air can be treated as an ideal gas and therefore throttling is isothermal.

For closed loop operation it is necessary to keep the inlet temperatures to the PD blowers within their allowable limits wherefore a re-cooler is installed in the low pressure line coming from the test benches. The re-cooler is followed by a cyclone separator in order to protect the compressor station and the whole system from oil drops and solid particles, which may come from the test benches in the case of malfunction.

2.1.5. Mass flow control bypass. In order to precisely control the mass flow rate through a test bench, a combination of two measures is used. At first, the rotational speed of one or more compressors or PD blowers is set in such a way that the delivered mass flow rate is slightly higher than required. Second, the bypass upstream of the central mass flow rate measuring device is opened to achieve fine-tuning.

As the PD blowers and especially the screw compressors (equipped with hydrodynamic sleeve bearings) have a minimum allowable rotational speed, the mass flow rates corresponding to these speeds need to be bypassed almost completely for operation points with very low mass flow rates. Therefore, the bypass is cascaded and consists of four parallel valves ranging from DN 150 to DN 10. An additional DN 700 bypass using a fast-opening relief valve is provided for emergency situations to open when the safety shut-off valves up- and downstream of a test bench are closing.

2.1.6. Central mass flow measurement. The central highly precise mass flow rate measurement unit consists of a diffuser that distributes the

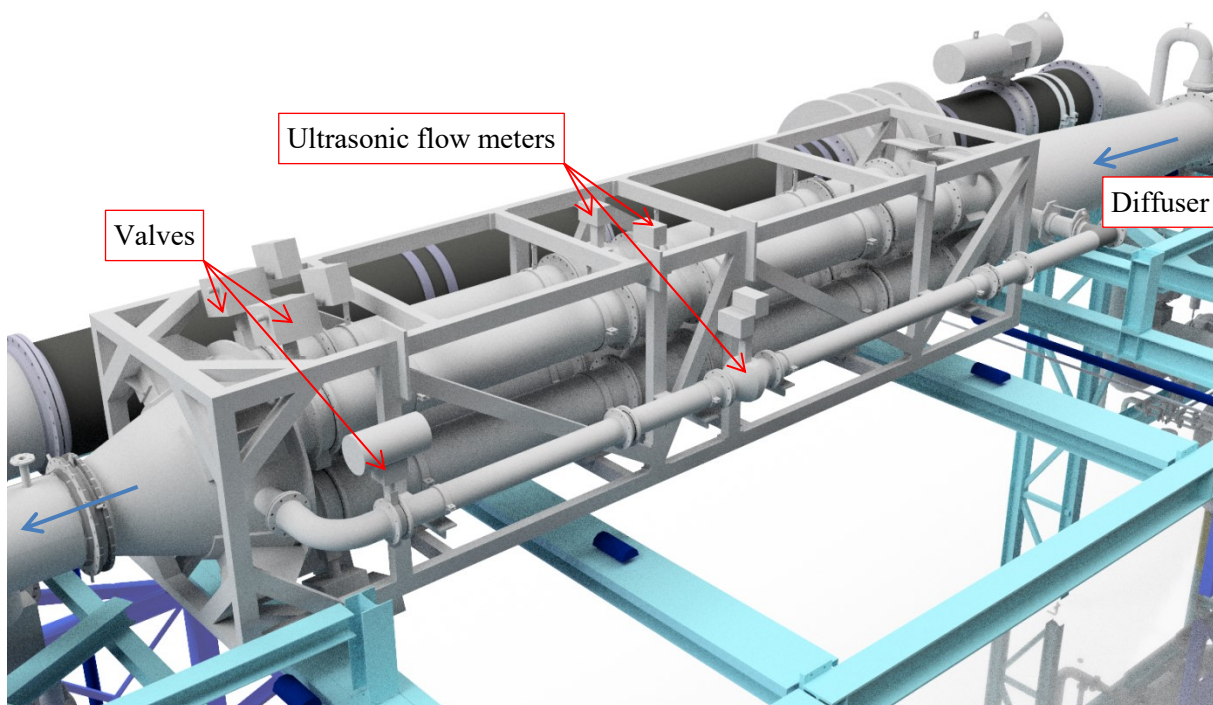


Figure 3. Central mass flow rate measurement unit. Flow from right to left.

flow to five parallel ultrasonic gas meters, four in DN 500 (each with a maximum volume flow rate of 25800 m³/h) and the fifth in DN 200 with a maximum volume flow rate of 4500 m³/h (see figure 3). The number of active flow lines is selected automatically by a control system (using valves) in such a way that all gas meters in use are operated in the turn down regime of lowest uncertainty (10...100 % of maximum volume flow rate). Each line is equipped with a flow straightener of the Etoile-type followed by a fine wire mesh. All ultrasonic gas meters use four measurement paths.

Inside the gas meters the static pressure measurements use high precision sensors with 0.01% IS-50-accuracy. This means that measurement uncertainty is 0.01 % of half of full scale when the measurement value is in the range between 0...50 % of the full scale, and 0.01 % of reading when the measurement value is in the range between 50...100 % of full scale. Temperature is measured using the average value of three precision-type PT100 sensors downstream of the nozzle where the separate lines merge again. The maximum combined uncertainty of the sensors and the 24 bit transmitter occurs at the lowest temperature (60°C) with 0.183 %. A total combined uncertainty of the mass flow

rate measurement of 0.55 % can be anticipated including all influencing measurements (averaged squared uncertainty).

In order to achieve an even distribution of the flow into the four DN 500 branches and even velocity profiles at the gas meters, the diffuser including the upstream bends was CFD-simulated. As visible in figure 4 (flow from left to right), the bends can produce significant swirl and hence wall mounted vortex generators at the diffuser inlet and built-in components to reduce the diffuser outlet area were found to be necessary.

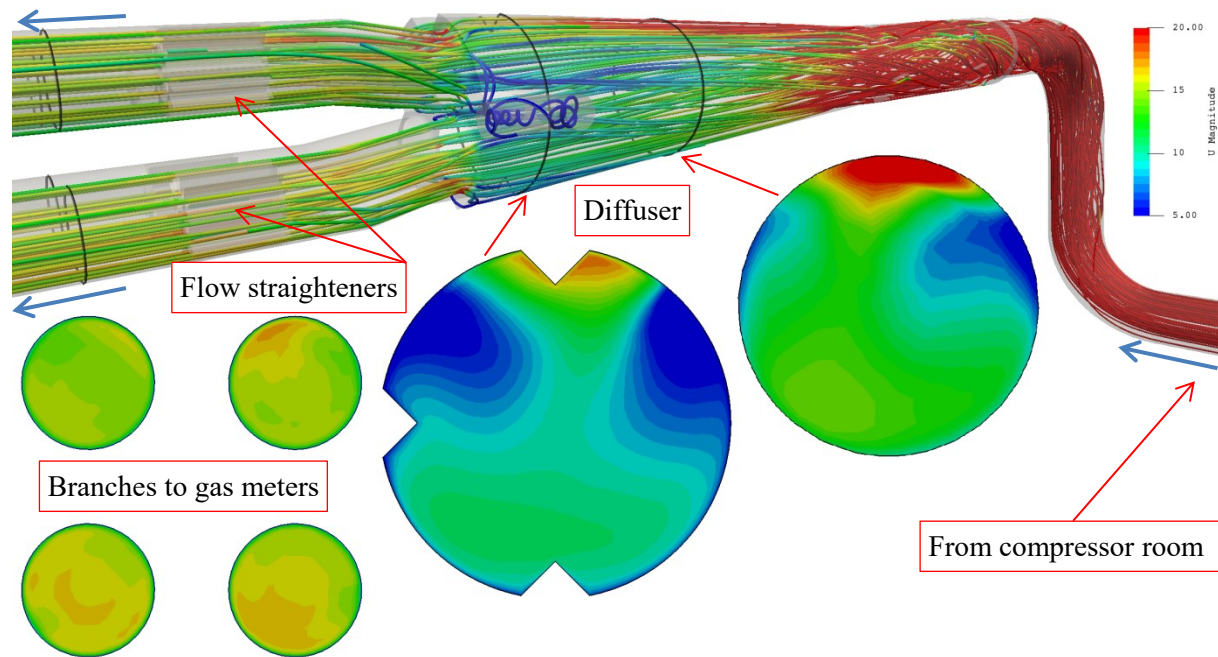


Figure 4. CFD-simulation of the distribution diffuser. Streamlines coloured by velocity magnitude.

For plausibility checks and detection of substantial leakages at the test benches, a second simple mass flow rate measurement device is positioned downstream of the test benches. The thermal flow sensor measures directly a flow velocity at reference conditions. The cross-sectional area and a velocity profile factor allow calculation of the mass flow rate. The profile factor can be derived by comparing with the readings from the central mass flow measurement.

2.1.7. Pipework and settling chambers. Piping between compressor station and test benches are sized in DN 700 on the high pressure side and DN 1000 downstream of the test benches. All pipework is designed for a nominal pressure of 10 bar(g) and implemented with combined sound and heat insulation, which allows reaching shorter times until stable operation with constant temperatures.

For sensitive test benches additional settling chambers are provided. CFD simulations were used to develop the design of the chambers (see figure 5) which incorporate multiple guide vanes in the upstream bends, a flow straightener followed by a wire mesh and an outlet nozzle. To allow future modifications, the settling chambers can be opened and built-in components can be changed.

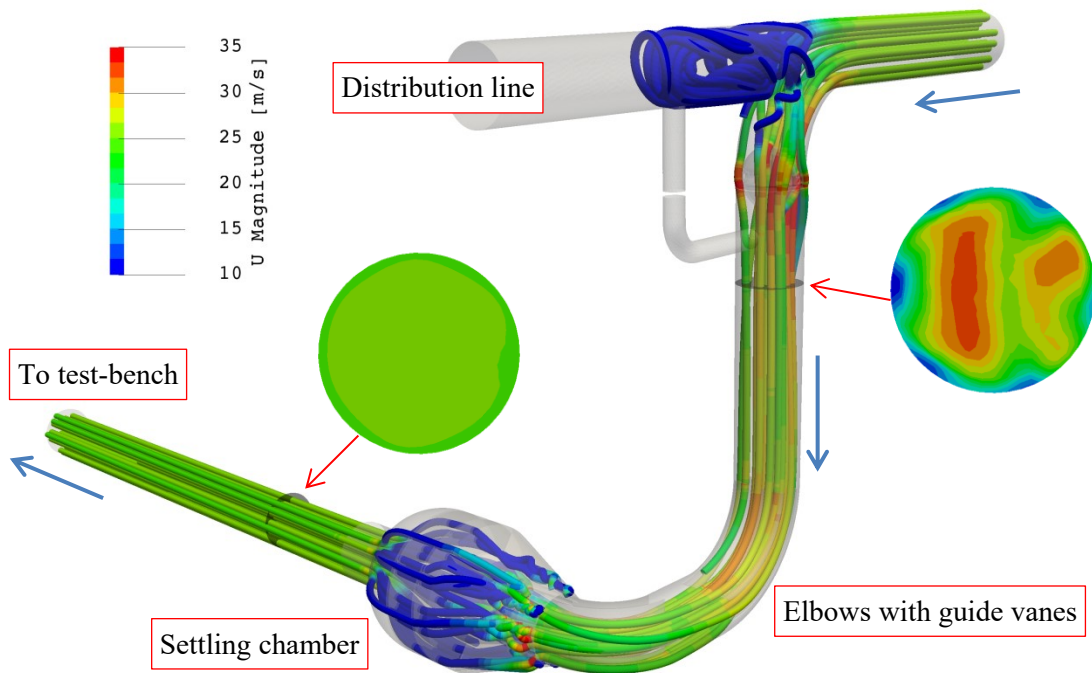


Figure 5. CFD-simulation of a settling chamber. Colours by velocity magnitude.

2.2. Operation

In open loop mode the test rigs operate at expansion ratios between 1 and 6. Test rig inlet pressures range from 1 bar (abs) to 6 bar (abs) with a maximum mass flow rate of 22 kg/s (79200 kg/h). In closed loop mode the test rigs operate at expansion ratios between 1 and 4.

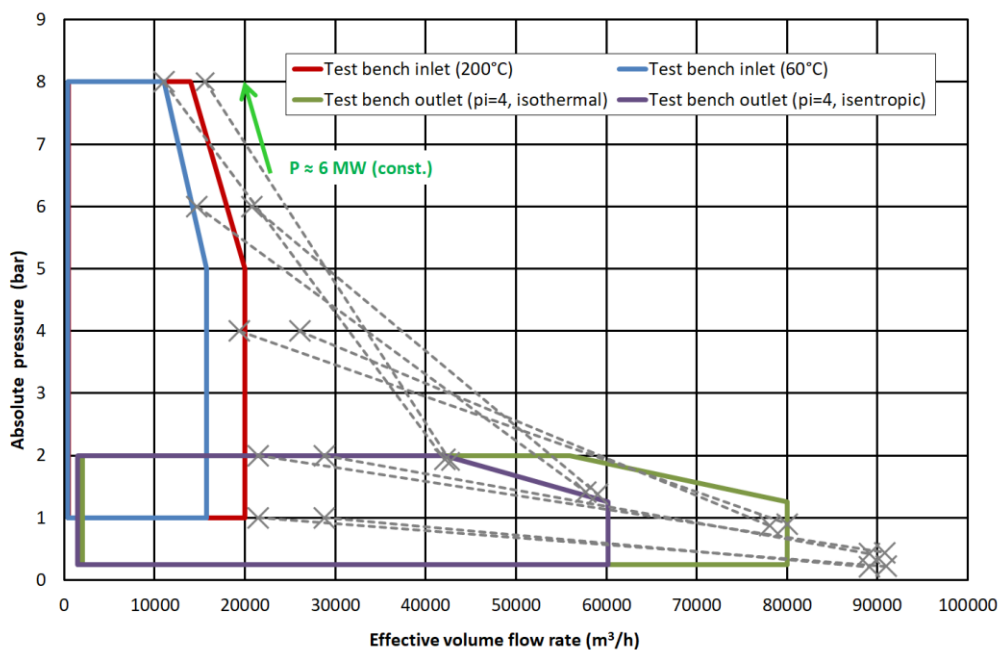


Figure 6. Operational area of the test air supply system.

The inlet pressures in closed loop mode range from 1 bar (abs) to 8 bar (abs) with a maximum mass flow rate of 25 kg/s (90000 kg/h). At all conditions temperatures can be regulated between 60°C and 200°C. The test air supply system has a maximum electric power input of approximately 6 MW. Figure 6 depicts the required operational area of the test air supply system in a p over \dot{V} diagram. The dotted lines represent examples for the change of state over test benches with an expansion ratio of 4.

For the description of the different ways to operate the test air supply system it is helpful to distinguish between different operation modes (open loop or closed loop), operation types (steady or transient) and operation configurations (which of the four compressors are involved). In addition, the operation can be either pressure controlled or mass flow rate controlled. Table 1 lists the aerodynamic inlet conditions that different test benches require.

Table 1. Aerodynamic inlet conditions at different test benches.

Test bench	Pressure [bar]	Temperature [°C]	Max. mass flow rate [kg/h]
Wind tunnel + planar cascade	1...2	60...100	80 000
Combustion chambers	2...6	60	28 000
Turbines open loop (OL)	1...4	60...200	80 000
Turbines closed loop (CL)	1...8	60...200	92 000

If tests with elevated temperatures are necessary directly after starting of the air supply system, the special operating mode “warming up” helps to accelerate the heating-up of pipes and other components. An additional bypass downstream of the test bench with the largest distance from the compressor room is then used to run the system at high pressure and therefore high temperature before a certain test bench is connected.

In closed loop operation, it is reasonable to extend the “warming up” mode into a “drying” mode. In this mode, the system runs at highest outlet pressure (8 bar (abs)) and the aftercooler is adjusted to maximum capacity, so that most of the water content is extracted by condensation.

3. The scaled functional model

As stringent demands on stability and reproducibility have to be met and automatic operation has been requested, it was decided to build a scaled and simplified but fully functional model of the test air supply system. The model enables testing of control methods and devices prior to their final implementation on site which saves commissioning time and provides useful insights into the operational behavior of such a closed system where mass flow rate (or pressure) and temperature have to be precisely controlled over a very wide operational range. All investigations presented in this paper are carried out on the scaled functional model.

Design and construction of the model are described in [5] and depicted in the perspective sketch of figure 7. The main simplification of the model is that only one flow line is implemented which consists of one Roots-type PD compressor GM13.6 (enforced for elevated inlet pressures) as first stage and one screw compressor VM10 as second stage. The GM13.6 provides an intake volume flow rate between 150 and 1400 m³/h. It can operate with inlet pressures between 0.25 and 2.2 bar (abs). The maximum outlet pressure is 3 bar (abs) and the maximum pressure difference is 0.8 bar. The VM10 provides an intake volume flow rate between 122 and 600 m³/h. It can operate with inlet pressures between 0.25 and 3 bar (abs). The maximum outlet pressure is 8.5 bar (abs) and the maximum pressure difference is 6.5 bar. Both compressors are driven by electric motors regulated by frequency converters. Their respective turn-down ratios are limited to the same values as those of the full scale compressors and the ratios of the maximum inlet volume flows of the PD-blowers and the screw compressors are approximately the same in the full scale system (48600/21600 = 2.25) and in the model (1400/600 = 2.33). DN150 piping is mainly used on the low pressure side and DN100 on the high pressure side. The DN200 ultrasonic gas meter requires inlet and outlet tubing of the same

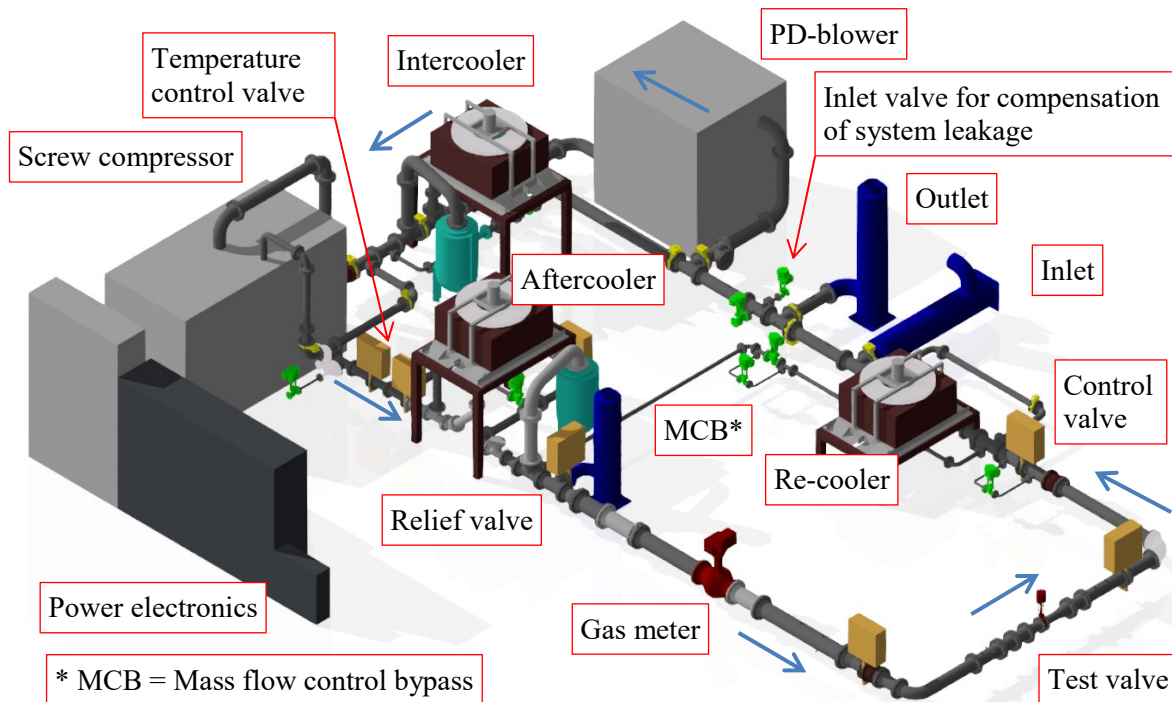


Figure 7. Outline of the functional model.

diameter. As these instruments are quite expensive, it was decided to use only one instead of five parallel meters and the DN200 meter will later on be re-used as the fifth meter in the full scale mass flow rate measurement unit. Only one test rig is present in the model which consists of an adjustable throttle valve representing a turbine. A settling chamber upstream of the test rig is not contained in the model. Precise control of the mass flow rate is provided by a cascaded adjustable bypass around the test rig consisting of only two instead of four parallel valves. In figure 7 the main components of the model are addressed. All coolers are of the air – air type with the possibility to control the cooling capacity via VSD-driven fans and bypass valves.

3.1. Control system

The operational limits of blower and screw compressor are monitored separately by standard compressor controls incorporated in each of the units. Alarms and trips of the compressors and all measurement values from the functional model are reported to a central Programmable Logic Controller (PLC). The PLC is connected with a Human-Machine Interface (HMI) panel to enable manual control and with a process control computer for automatic control. This computer is operated remotely by a separate PC which is using the specialized test stand automation software PAtools. Commands from the HMI or the PAtools software are transferred to the different actuators of the functional model via the PLC.

3.2. Experiments and results

First experiments were carried out manually using the HMI panel and have been successively replaced by programmed sequences and controls. As a starting point, simple proportional–integral–derivative controllers (PID-controllers) as provided by the PAtools software have been implemented. At the present state, only experimental data of steady operation conditions is available but transient tests are scheduled as well. All experiments described below are carried out using both stages. All figures that visualize stability of certain values represent stable operation points that develop some time after a target value of a control process was set.

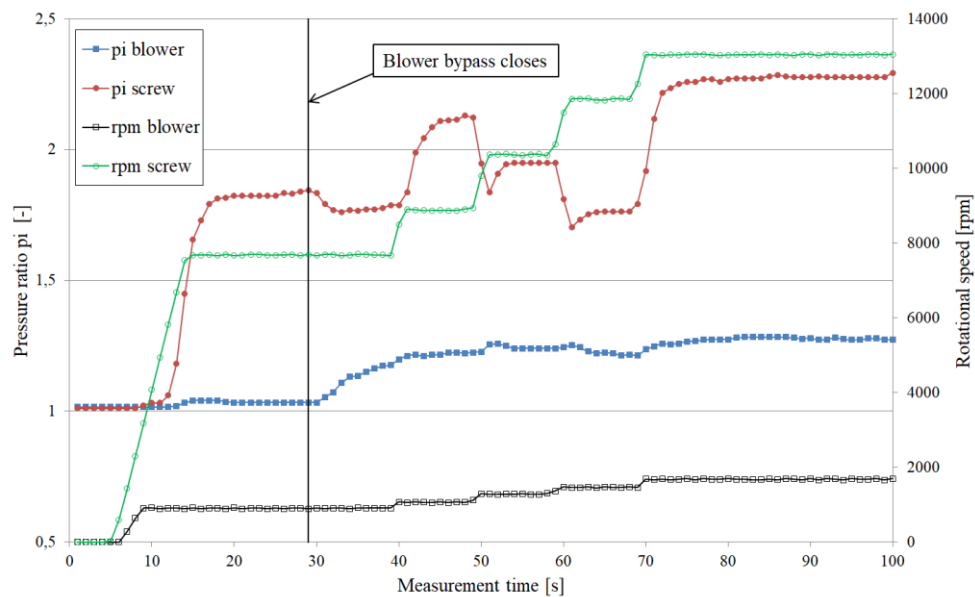


Figure 8. Start-up sequence - closed loop (CL), two stages.

3.2.1. Start-up procedure. The start-up procedure depicted in figure 8 is created in analogy to the full scale system where the rotors of the screw compressor are supported by hydrodynamic sleeve bearings. These bearings need a minimum rotational speed in order to develop their load capacity. It is therefore important to start with low load (low pressure difference) and accelerate quickly to the minimum rotational speed. Figure 8 shows that this is done within the first 10 s. After that, at 29 s, the bypass around the blower is closed and both machines accelerate (here in steps) until another stable operation point is reached at approximately 80 s with a screw compressor outlet pressure at 2.8 bar (abs) and a mass flow rate at 760 kg/h.

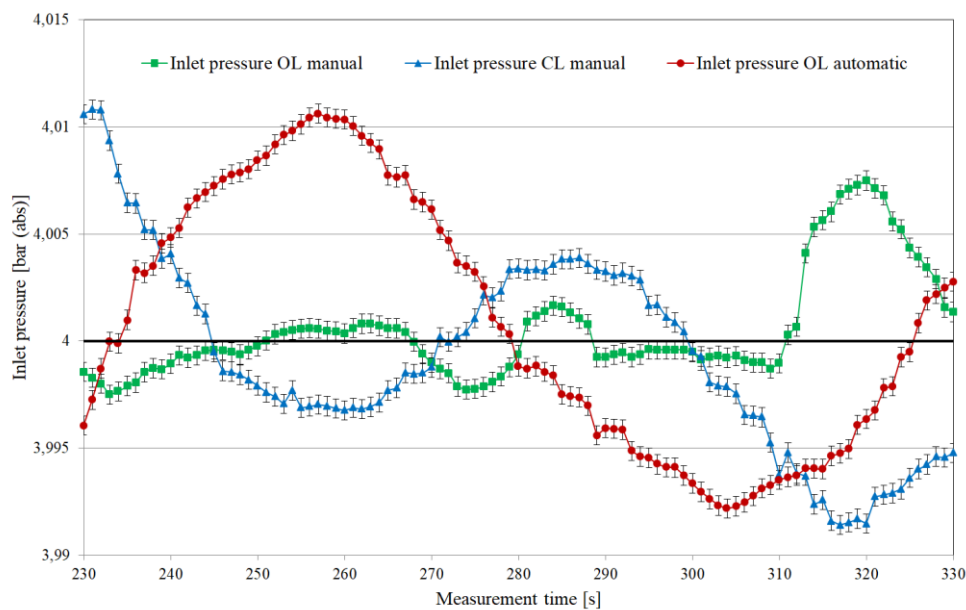


Figure 9. Stability of test rig inlet pressure.

3.2.2. Pressure control and back-pressure control. An example how precisely a constant pressure of 4.0 bar (abs) upstream of the test valve (representing e.g. an air turbine) can be held while the valve settings are unchanged is displayed in figure 9. The expansion ratio over the test valve is 2.0 while the mass flow rate has been adjusted in order to reach and hold the target pressure upstream of the test valve. The system is operated with both stages at a mass flow rate of approximately 350 kg/h. In the diagram, vertical bars represent the measurement uncertainty of the pressure measurement.

Comparison of the three graphs reveals that manual control gives superior results and that therefore the control settings still have to be improved. Best results are achieved in open loop (OL) operation mode because then no compensation of system leakages is needed in order to keep the PD blower inlet pressure (see figure 7) constant which simplifies the overall control task.

During the same tests as described above, the outlet pressure of the test valve had to be controlled as well in order to keep the expansion ratio of 2.0 constant using the control valve downstream of the test valve as depicted in figure 7. Measurement of this pressure is done with a sensor having a significantly higher uncertainty than on the pressure side. In addition, an AD-converter with only 12 bit resolution is used, which results in the stepped curves as visible in figure 10. The target pressure of 2.0 bar (abs) is depicted together with the upper and lower values resulting from the measurement uncertainty as given by the sensor manufacturer. As visible in figure 9, at this stage of the control system development, manual control still gives better results than automatic control.

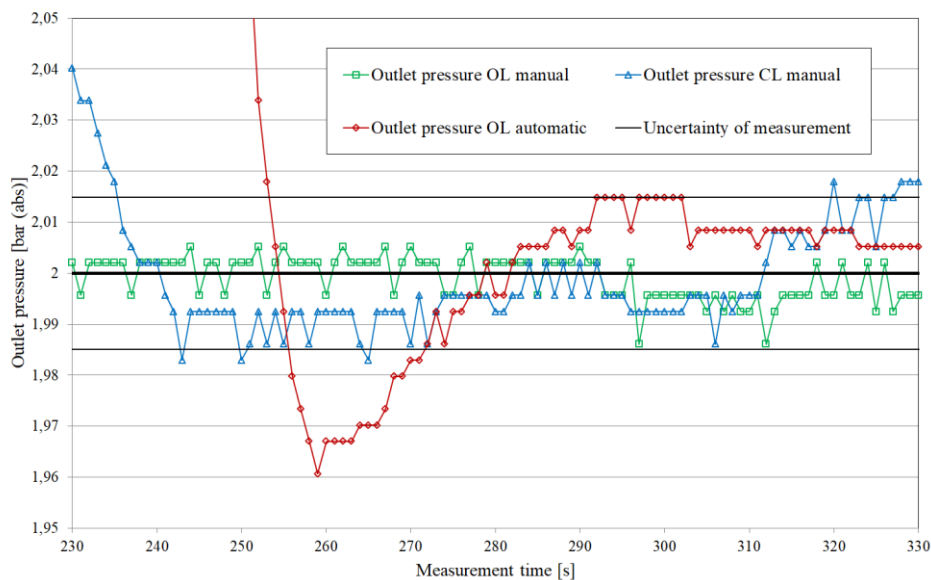


Figure 10. Stability of test rig outlet pressure.

3.2.3. Mass flow rate control. As the test valve setting was not changed during the pressure control tests of the section above, the inlet pressure variation (see figure 9) is a measure for the mass flow rate variation as well. As for certain applications it is important to run the test air supply system in mass flow rate controlled mode, the stability of the directly measured mass flow rate is investigated in this section. Figure 11 shows mass flow rate variations and deviations from the target value 400 kg/h in different control and operation modes. The system is operated with both stages and an expansion ratio of 2 over the test valve and a test valve inlet pressure of 2.5 bar(abs). In automatically controlled CL mode, the standard deviation from the target value is the highest (1.18 %) of the three curves. This is to be expected as in this case the highest number of control circuits is simultaneously active.

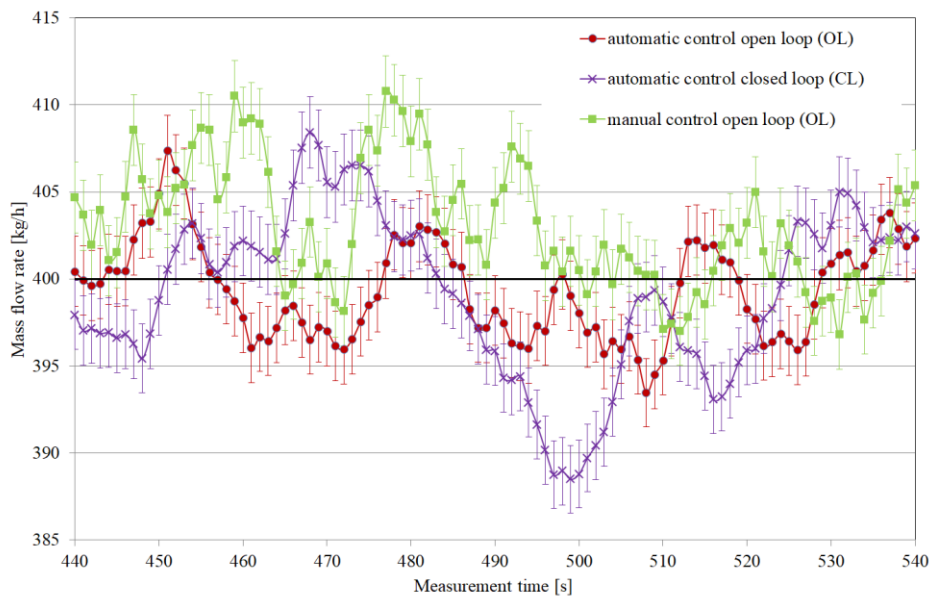


Figure 11. Mass flow rate control.

3.2.4. Inlet pressure repeatability. Figure 12 proves how precisely the same test valve inlet pressure 4 bar(abs) can be adjusted in open loop operation with both stages and the same settings for the test valve on two different test days at 365 kg/h. PD blower speed and screw compressor speed are set to the same values and manual fine tuning using the small mass flow bypass valve is applied. The pressure variation achieved in this example is on both days within $\pm 0.11\%$ of the target value with a standard deviation of less than 0.06%. The repeatability of the average values is within 0.006%.

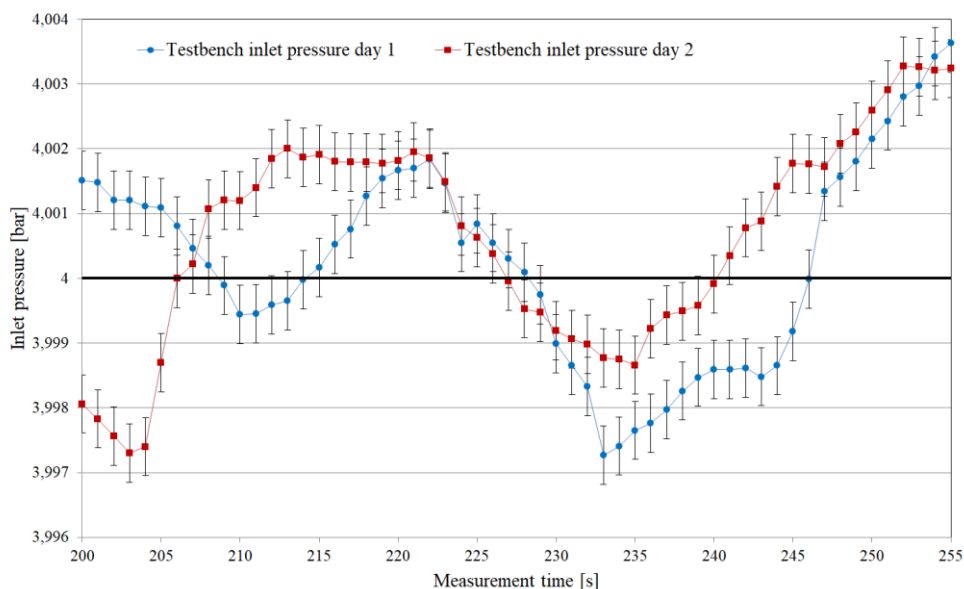


Figure 12. Inlet pressure repeatability in open loop manual control operation.

3.2.5. Temperature control. The temperature at the test valve is influenced by the compressor outlet temperature (and therefore by the compressor pressure ratio) but has to be controllable independently. If the compressor outlet temperature is significantly higher than the temperature which is desired at the

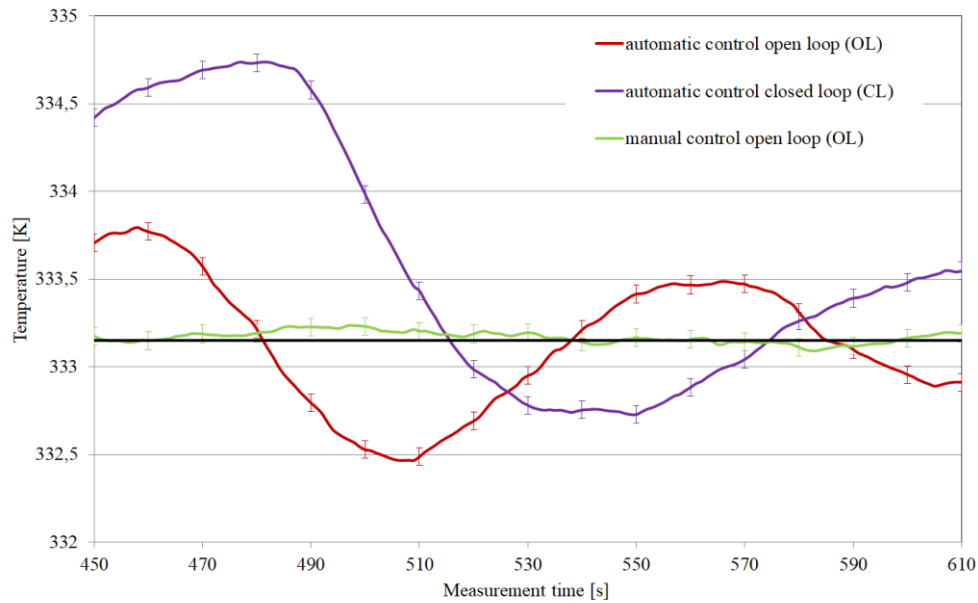


Figure 13. Stability of temperature.

test valve, control is possible by using the aftercooler. Its cooling performance can be controlled by the cooler fan speed and the settings of the bypass valves parallel to the cooler. If the compressor outlet temperature is too low (due to a low pressure desired at the test valve) than the screw compressor outlet has to be throttled additionally using the temperature control valve (see figure 7). Both cases are shown in the following figures for a test valve pressure of 2.8 bar (abs). Temperature stability at a target value of 333.15 K without the need to use the temperature control valve is depicted in figure 13 and for a target value of 342.15 K and the use of the temperature control valve in figure 14. In both cases the required stability (max. +/- 3K deviation from the target value) is reached.

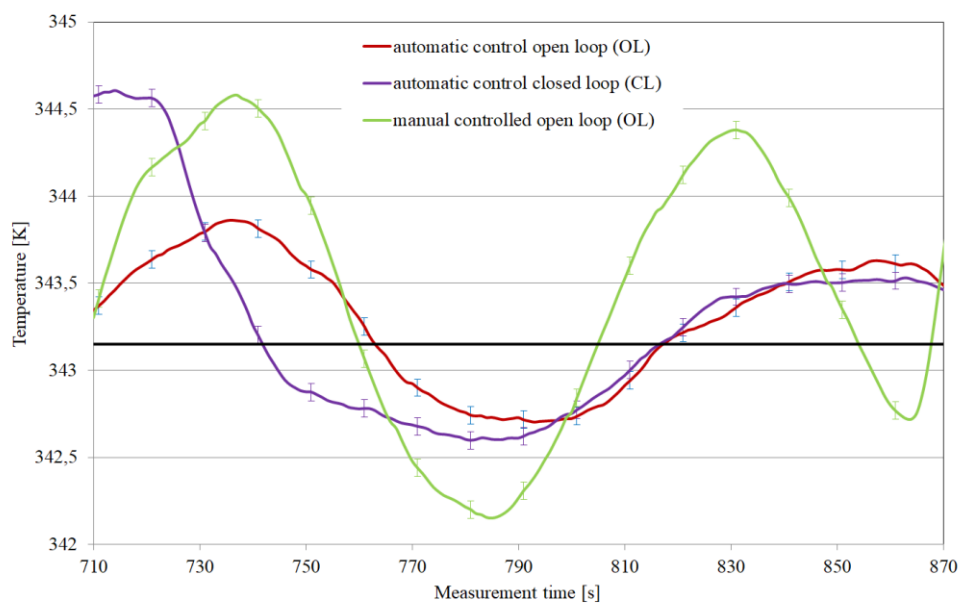


Figure 14. Stability of temperature using temperature control valve

Figure 14 reveals a maximum deviation of 1.6 K (standard deviation of 0.72 K) in automatically controlled closed loop operation. After a certain settling time, automatic control in closed loop (CL) as well as in open loop (OL) operation mode show similar deviations from the target value. Obviously, the automatic compensation of system leakages which is needed in CL in order to keep the PD blower inlet pressure constant has no significant influence. Again the performance of manual control is superior, which shows the potential for fine tuning of the control parameters.

Figure 14 shows that the additional use of the temperature control throttle does not significantly change the performance in automatically controlled closed loop operation. But in open loop operation this time manual control shows oscillations that are much less pronounced in automatic control.

4. Conclusion and outlook

The control system for a large scale test air supply unit was developed and successfully tested using a scaled functional model. At all operational modes that were tested, the required stability and precision criteria were fulfilled, although a significant improvement potential was shown as well. In many cases manual control gave superior results over automatic control which reveals that fine tuning of the controller parameters has a potential to decrease deviations from target values. Additional performance can be expected from using PID-controllers with adaptive values of the controller parameters. In a next step, further operating points distributed over the large operational area and transient behavior should be investigated. For the experiments in this paper, pre-selection of compressor operating conditions was done manually (see figure 9). The implementation of the two-stage performance calculation into the software is planned.

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