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OPERATIONAL BEHAVIOR OF A COMPLEX TRANSONIC TEST TURBINE FACILITY

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ABSTRACT

This paper describes the operation of a continuously operating cold flow test stand for axial turbine stages. The interesting feature of this test stand is the fact that the power generated by the test turbine is used for driving a compressor, which supplies additional air to drive the turbine. The brake compressor was a donation from industry and was a very welcome alternative to a water brake. The advantages of using this compressor lie in the low installation costs and the power savings. The performance of the test turbine stage influences the capabilities of the test stand. A suction blower driven by a helicopter engine can be used to decrease the exhaust pressure allowing to increase the turbine expansion ratio.

The performance of the whole test facility and the complex interaction of the components will be described. In general, design engineers are interested in the performance of their test stages at different operating conditions (i.e. corrected speed, pressure ratio). It will be shown by which control mechanisms different operation points can be obtained and which range of the values can be achieved.

INTRODUCTION

In 1995 work started at Graz University of Technology to design and build a transonic test turbine facility. This facility should be driven by the existing compressor station. So the task was to design a test stand for transonic gas turbine stages in full flow similarity. The test stand should allow access of conventional pressure and temperature probes as well as optical measurement devices such as Laser-Doppler-Velocimetry (LDV) and Particle-Image-Velocimetry (PIV) systems. Temperature and pressure measurement data from the rotor may be transferred to the stationary frame via telemetry. Design and construction details of the plant can be found in Erhard and Gehrer, 2000. Due to severe budget restraints all design work and much construction was done by institute's staff. Changing demands from research projects forced to several modifications. In November 1999 the turbine test facility was run for the first time. Since the commissioning 56 test runs with two different transonic turbine stages were performed. Measurement data were collected during an overall testing time of 64 hours. Together with the results of pre-operational simulations (steady state and transient) this measurement data give information of the operating ranges and the operational behavior of the test stand. Much work was invested in the analysis of the data collected and in the understanding of the control mechanism and the interactions of the components. This forms very valuable information for future testing activities and planning of new component testing.

TEST RIG

Figure 1 shows a schematic diagram of the turbine test rig. Pressurized air delivered by a separate electrically driven compressor station is fed to the turbine stage to be tested. The test turbine drives the brake compressor shown at right. The pressurized air from the brake compressor can be added to the air from the compressor station in a mixing chamber. The air from the compressor station may be cooled to about 50 °C, whereas the temperature of the brake compressor air depends on the pressure ratio. The solution with the brake compressor was chosen, because it was a donation from industry. It was a very efficient alternative to a water brake because it can recover most of the braking power. The mixing chamber is equipped with inserts in order to homogenize the air temperature profile at the turbine inlet. The exhaust air from the turbine normally flows through an exhaust line directly to the silencers in the exhaust tower. Optionally a suction blower driven by a 750 kW helicopter engine may be inserted into the exhaust line to reduce the turbine backpressure and thus increase the turbine pressure ratio.

A turbine stage to be tested must be scaled in such a way that the turbine mass flow does not exceed the maximum air supply of the compressor station. In the case that the air from the brake compressor is added to the compressor station airflow the brake compressor characteristics have to be considered. Additionally, the maximum shaft speed is limited to 11500 rpm by the brake compressor disks.

Three remotely controlled butterfly valves are used for startup, control and shutdown.

These valves will also shut down the machine in case of emergency. Emergency shutdown will be initiated at overspeed, excess of vibration limits, excess of oil temperatures or manually by the operator. A detailed discussion on the emergency shutdown and load changes of the facility can be found in Neumayer, 2001.

At startup the valve between the brake compressor and the mixer is closed, whereas the valve situated in the bypass line from brake compressor to exhaust is open. At a certain speed, the bypass valve is partly closed until the outlet pressure of the brake compressor is equal to the pressure in the mixer. In the case that the air from the brake compressor is added the valve to the mixing chamber is then opened. Then the bypass valve is closed. Afterwards, the speed can be increased by adding more air from the compressor station. Depending on the flow similarity requirements (corrected speed and pressure ratio) of the turbine stage that has to be tested it is possible that the brake compressor does not achieve the prescribed pressure ratio for the given speed. In this case the compressed air from the brake compressor is fed directly to the exhaust line, the bypass valve remains partly open to maintain the braking effect. For this test setup turbine inlet pressure is limited by the compressor station at above 5.0 bar.

This concept allows a continuous operation of the test turbine within a widely adjustable speed range between 7000 and 11550 rpm. The maximum inlet pressure is 5.0 bar, the range of turbine inlet temperatures is from 50 to 185 °C, resulting in a maximum coupling power of 2.5 to 3.0 MW depending on the turbine stage. The modular design of the test turbine allows quick modifications, the meridional flow path, the stage as well as the diffuser can be adapted in a wide range.

In test setups at which the air from the brake compressor is added to the compressor station airflow the maximum turbine inlet pressure is given by the brake compressor pressure limit. This limit is at about 4.6 bar at 11500 rpm.

Most demanding test setups are with stages that require high pressure ratios at high mass flows- for example a second stage that was tested and will be later on referred to as “industrial stage “.

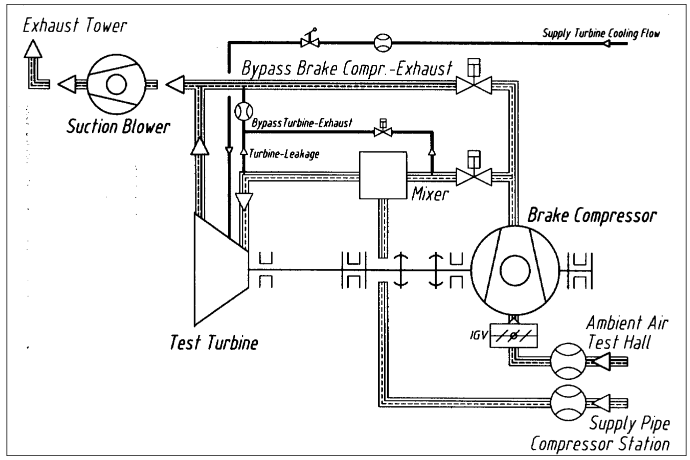

Figure 1: Schematic diagram of the turbine test rig

ROTOR DYNAMICS

The test rig at TTM should be stationary operated in a wide range of speeds and with turbine disks of very different mass and moments of inertia. In order to guarantee safe operation amplitudes should be small. Bending modes 1, 2 and 3 of the shaft assembly with the “TTM” rotor disk installed are shown in Figure 2.

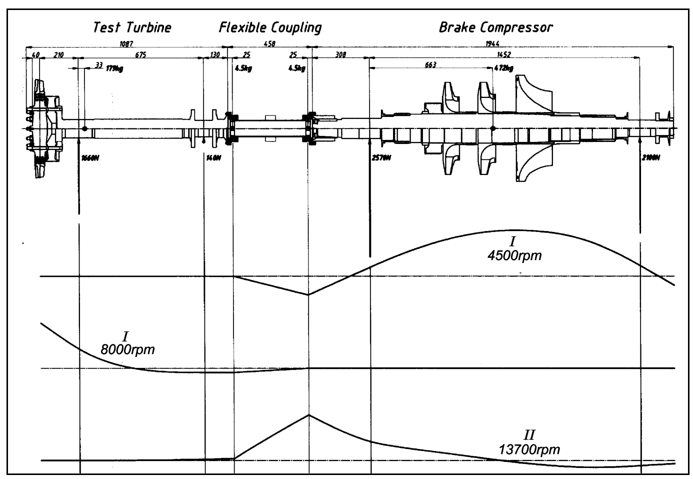

Figure 2: Test turbine and brake compressor bending modes

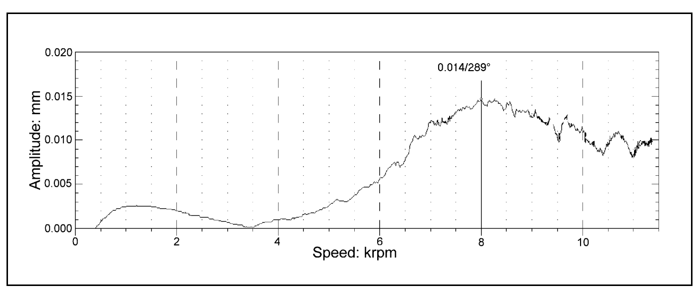

Figure 3: Movable bearing amplitude plot of a typical test run

Figure 3 shows a typical plot of amplitude measured at the moveable bearing (which is next to the turbine disk) versus rotational speed, which was generated during a test run. The amplitude plot does not show sharp peaks, which means that the rotor is very well damped. Safe continuous operation can be guaranteed at any speed.

COMPONENTS

Compressor Station

The compressor station consists of two turbo and two screw compressors. Table 1 gives an overview of the machine types, their characteristics and their maximum power consumption.

Table 1: Compressor Data

Name	Type	Suction Volume [Nm^3/h]	Press. Ratio [-]	Control	Motor Power installed [kW]
SC 20	Single stage radial	27500	2.9	Variable Inlet Guide Vanes	1450
SC 14	Single stage radial	15500	2.9	Variable Exit Guide Vanes, Speed	1250
E1, E2	Screw (both in one housing)	8000	3.1		1600

Two coolers allow the intercooling and/or recooling of the compressed air. In combination with the coolers the compressors may be operated in 14 different operational modes in series and in parallel. Four of the operational modes are used for the operation of the turbine test rig. Figure 4 shows in symbolic form the connection of the compressors in these four different modes, Figure 5 gives the corresponding characteristics. Operational modes “FW1” and “FW2” (single operation of the turbo compressors) are limited to about 2.9 bars. The compressors “SC14” and “SC20” can also be operated in parallel, which adds their volume flows. “FW8” and “FW9” are the high pressure operating modes achieved by operating compressors “SC14” and “SC20” in series. These modes are limited to pressure ratios of 5. The maximum volume flow for "FWF 9" is 32.500 Nm^3/h (10.6 kg/s). The compressor station contains a complex control and monitoring system. Mass flow from the compressor station may be adjusted within certain limits by:

- electrically selecting the speed of the SC14 compressor
- adjusting the inlet guide vanes of the SC14 compressor
- adjusting the variable exit geometry of the SC20 compressor

The exit temperature of the compressor station may be adjusted within certain limits by more or less bypassing recooler “K2”.

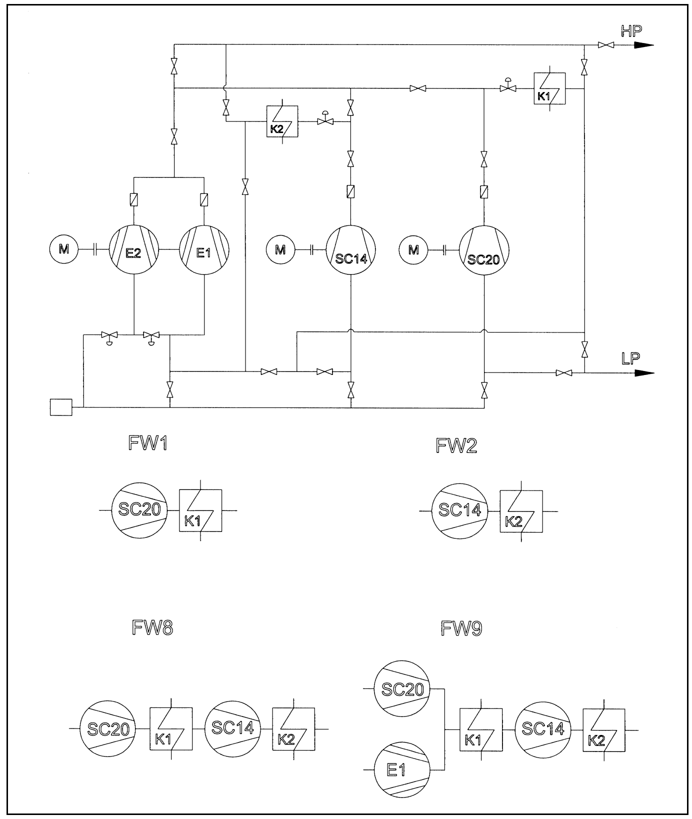

Figure 4: Operational modes of the TTM compressor station

A detailed description of the institute’s compressor station can be found in Pirker et al., 1995.

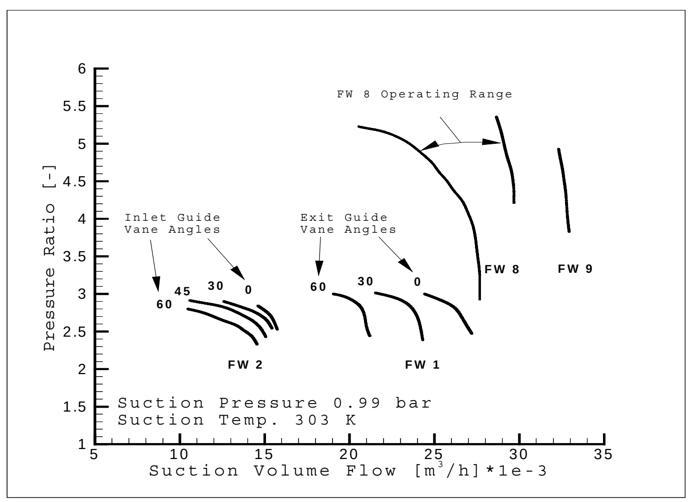

Figure 5: Characteristics of the compressor station for the operational modes relevant to the test turbine

Brake Compressor

The brake compressor is a three-stage radial type industrial compressor and is used for both braking the turbine and supplying additional air. The compressor is equipped with variable inlet guide vanes (IGVs), which are actuated by a stepper motor. Figure 6 shows the compressor characteristics at its nominal speed of 11175 rpm for different inlet guide vane positions. Moving the guide vanes from minimum opening to counter swirl results in higher pressure ratios and mass flows at much higher power consumption. The maximum allowable speed is 11550 rpm.

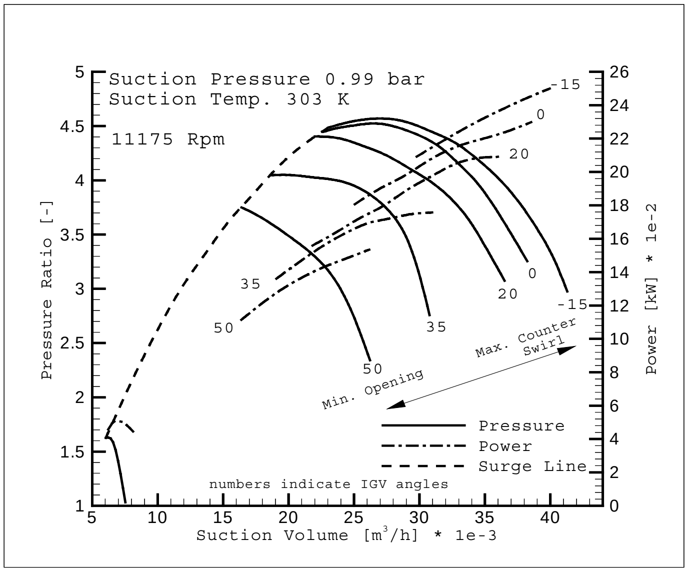

Figure 6: Brake compressor map for constant shaft speed

Turbine

The turbine stages tested influence the operational behavior and the operating range of the test rig because of the power balance between the test turbine and the brake compressor. The flow characteristics of the transonic stage “TTM” which is designed to allow optical access for LDA, PIV and Schlieren flow visualization were obtained from 25 test runs and are shown in Figure 7. For pressure ratios greater than 1.6 the value $\dot{Q} / \sqrt{T}$ remains almost constant due to transonic flow conditions, where $\dot{Q}$ is volume flow at turbine inlet condition and T is turbine inlet temperature. Detailed transonic stage measurement results including a turbine mapping are presented in Seyr et al., 2001.

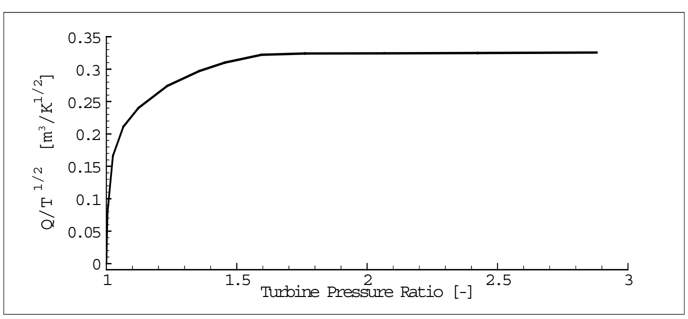

Figure 7: Flow characteristics of TTM transonic turbine stage

Suction Blower

The suction blower, which can be optionally inserted into the exhaust line of the test turbine, may increase the expansion ratio of the turbine above the limit that is given by the brake compressor for a certain corrected speed. Figure 8 shows the characteristics of the blower for constant shaft speeds and an exit pressure of 0.99 bar (ambient pressure). The operating range of the blower is limited to 3000 rpm, which is the maximum shaft speed. The helicopter engine, which drives the blower, delivers enough power to maintain the 3000 rpm at normal test conditions (helicopter engine suction temperature below 35°C, suction temperature of the blower above 14°C) with a volume flow of 80.000 to 100.000 m^3/h. A complete control system for the helicopter gas turbine ensures very accurate adaptation to the demands of the test rig. The minimum exhaust pressure of the turbine is 800 mbar because of the strength limitation of the exhaust piping which has to withstand the difference between the suction and the ambient pressure.

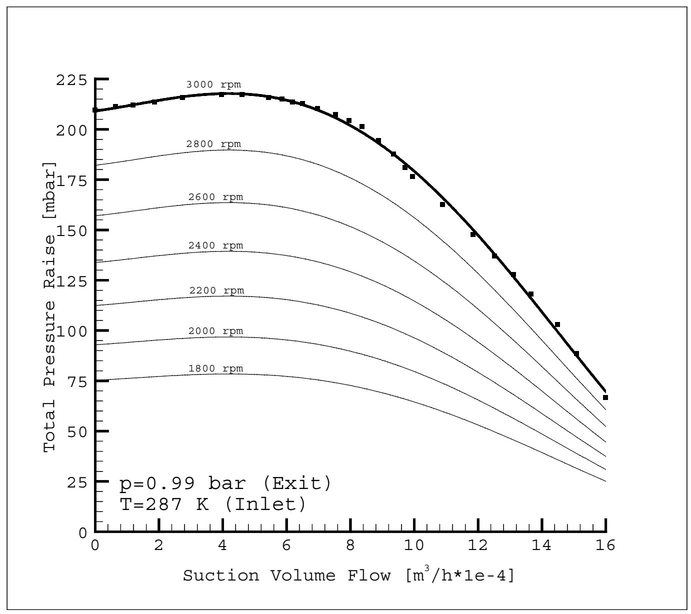

Figure 8: Suction blower characteristics

CONTROL MECHANISMS

Four basic control parameters may be applied in order to achieve a certain operating point:

- Mass flow from the compressor station
- Temperature of the compressed air from the compressor station
- IGV position of the brake compressor
- Turbine expansion ratio set by the suction blower

To determine the efficiency curves of a turbine stage, it is necessary to do measurements at steady operation points defined by certain corrected speeds and expansion ratios. The corrected speed is defined by $N_{corr} = N / \sqrt{T}$ (Volponi, 1999).

In the following it will be shown which operating points can be achieved, how they can be achieved and what are the limits. To do this measurement data of two different turbine stages are analyzed and presented.

Variation of turbine expansion ratio for constant corrected speed (operation without suction blower)

At first the interaction of the test rig components without the suction blower is investigated because in this case the number of influence parameters is smaller. The operation map shown in Figure 9 is based on measurement data of the "TTM" turbine stage. It illustrates the influence of the temperature of the compressor station air and the guide vane position of the brake compressor on the turbine pressure ratio at a constant corrected speed of $N_{corr\%} = 93\%$ of the corrected speed at design point. This condition is chosen because of the availability of data. Following conclusions can be drawn from the diagram:

Reducing the compressor station air temperature at constant IGV position demands a considerable increase of mass flow from the compressor station to maintain the corrected speed at a reduced pressure ratio. This is due to the fact that the physical speed is reduced according to the colder turbine inlet temperature, so that the brake compressor supplies less air at lower pressure to the mixing chamber. On the other hand, a change of the IGV position towards counter swirl allows a strong increase of the pressure ratio. The physical speed stays nearly constant, the additional work of the brake compressor has to be provided by additional air from the compressor station.

So to move from operating point "A" to a higher turbine expansion ratio needs at least the change of two control parameters. One possibility is to increase the compressor station mass-flow and increase the work of the brake compressor by changing the IGV position so that the corrected speed does not rise. This corresponds to a shift from point "A" to point "B". If the bypass ratio of the compressor station recoolers remains the same, the compressor station temperature would slightly rise leading together with the increased brake compressor temperature to a higher mixing temperature. So for a constant corrected speed the physical speed also increases slightly.

The second possibility to increase the turbine pressure ratio is to increase the temperature of the compressor station airflow for a constant IGV position of the brake compressor. This corresponds to a pressure rise along the path from point "A" to point "C". Mass flow from the compressor station will decrease, additional mass is supplied from the brake compressor because of the increased physical speed. The second change of operating point is more economic because it needs less power from the compressor station.

The operating range for a certain corrected speed is limited by the brake compressor as well as the temperature limits of the compressor station. Maximum physical speed is not limiting the operating range at 93% corrected speed because the maximum possible turbine inlet temperature is not that high. The upper temperature limit is steeper than the lower one because of the higher compressor discharge temperature in the uncooled mode. The operating range is also limited by the brake compressor surge line, because for the given corrected speed the highest pressures cannot be achieved.

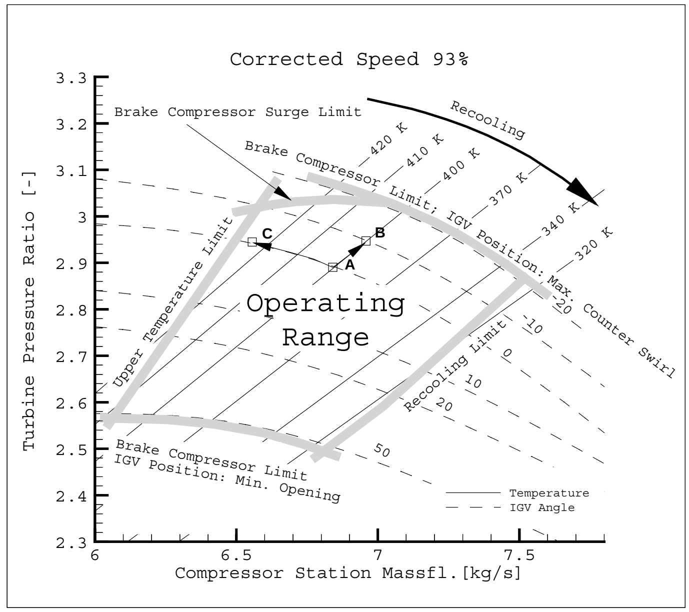

Figure 9: "TTM Stage" operating range for constant corrected speed

Variation of corrected speed for constant turbine expansion ratio (operation with suction blower)

Secondly, the influence of the suction blower on the operation of the test turbine will be investigated. Figure 10 shows the variation of the corrected speed when the turbine expansion ratio and the temperature of the air coming from the compressor station are held constant. This diagram is based on measurements of the second stage of an industrial gas turbine, where the suction blower was employed. Following conclusions can be drawn from the diagram:

Increasing the mass flow from the compressor station at constant IGV position of the brake compressor demands a reduced suction in order to obtain a constant turbine pressure ratio. At the same time the corrected speed increases from point "A" to point "B" in Figure 10. This can be explained by the fact that the additional mass flow from the compressor station leads to a higher inlet pressure to the turbine and to a higher power output. So the brake compressor increases its physical speed supplying also more air at a higher pressure. The corrected speed only slightly rises, because the outlet temperature of the brake compressor also increases, whereas the temperature of the compressor station air is held constant. To maintain the expansion ratio, the backpressure to the turbine has to increase, so that a power shift from the suction blower to the compressor station takes place.

The more efficient way to obtain a higher corrected speed is to close the IGVs of the brake compressor so that the braking effect is reduced. For a constant suction pressure, the mass flow of the brake compressor is reduced despite the higher physical speed, so that the compressor station mass flow has to be raised to maintain the expansion ratio (point “D”). If there is still a power reserve of the suction blower, the mass flow of the compressor station can be kept constant by reducing the turbine backpressure (point “C”). In general the operator will choose a control path which is somewhere in between so that the suction blower and the compressor station compressors will be equally additionally loaded.

The operating range is limited by the brake compressor and the suction blower. Maximum counter swirl means maximum braking effect (see Figure 6) thus forming the lower limit for the corrected speed. The upper limiting line of corrected speed is formed by the minimum IGV opening of the brake compressor. The right limit of the diagram is the ambient pressure line. It is interesting to note that a higher corrected speed can be obtained at a given pressure ratio if the turbine backpressure is increased together with the compressor station mass flow. But this is not economic because of the electricity demand of the compressor station. The limit at the left hand side is set by the exhaust line, which may not be loaded to a higher pressure difference than 200 mbar. At the left limit the volume flow at suction blower inlet does not exceed 80000 m^3/h so that 200 mbar suction pressure (see Figure 8) can be maintained by the blower. For very high pressure ratios, the necessary volume flow can exceed this value, so that the suction blower cannot maintain the 200 mbar pressure rise.

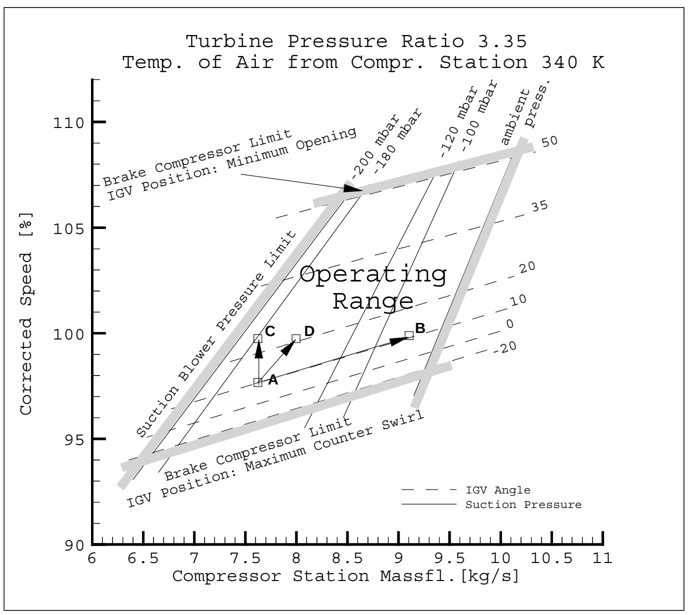

Figure 10: Industrial stage: operating range for a constant turbine expansion ratio and SC air temperature

Variation of turbine inlet pressure for constant corrected speed (operation with suction blower)

To get more insight into the interaction of the test rig components in Figure 11 the variation of the turbine inlet pressure is shown for a constant corrected speed of 93% of design value and a constant compressor station air temperature of 340 K.

At a constant suction pressure the turbine inlet pressure respectively the turbine pressure ratio can be raised by increasing the mass flow of the compressor station. To prevent a rise of corrected speed the braking of the brake compressor is intensified by turning the IGVs to counter swirl. This shift of the load point is indicated by points “A” and “B”.

Increasing the suction at constant IGV angle leads to operating point “C” with a slightly higher inlet pressure (and remarkably higher expansion ratio), whereas the required mass flow of the compressor station is reduced to maintain the corrected speed. The maximum pressure ratio for a given corrected speed can be achieved at maximum suction and maximum counter swirl operation of the brake compressor (point “D” in the scheme). The compressor station mass flow can be kept small.

The upper and lower operating limits are given by the brake compressor IGV positions. The suction blower limit at the left hand side of the map is the 200 mbar suction line, which is maintained at a speed of maximum 3000 rpm, the right hand side limit is the ambient pressure limit. Increasing the turbine backpressure is a way to increase the operating range to lower pressure ratios.

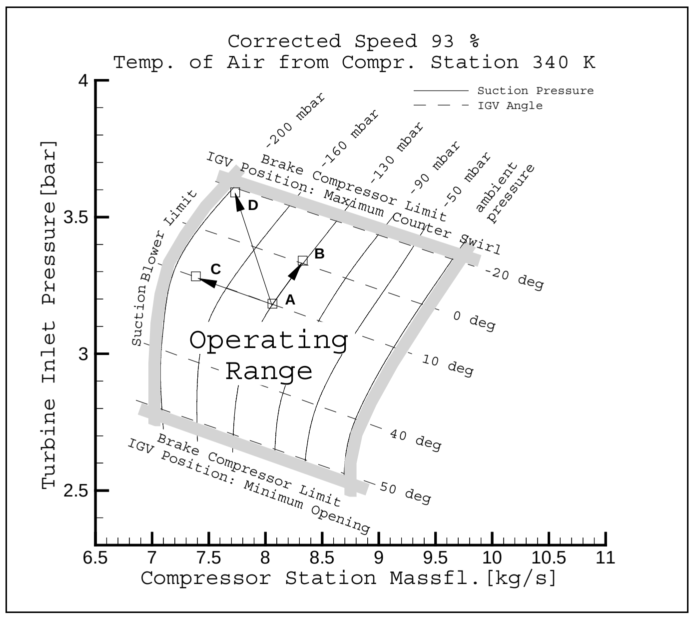

Figure 11: Industrial stage: operating range for constant corrected speed

Maximum operating range for the industrial turbine stage tested

Figure 12 gives an overview showing which turbine expansion ratios can be achieved for a certain value of corrected speed. The dark area at the left of the diagram shows the operating range without the use of the suction blower. The highest corrected speed of 100 % can be reached at minimum IGV opening of the brake compressor, a variation of the pressure ratio is not possible at this operating point. The gray area shows the immense increase in the operating range due to the use of the suction blower. The highest turbine pressure ratios are achievable with maximum suction and maximum counter swirl positioning of the brake compressor and no recooling.

The white area below the dash-dotted line shows operating points with a suction pressure lower than the maximum of 200 mbar.

The suction blower is a very efficient means to enhance the operating range of the test facility.

First, by reducing the backpressure by only 200 mbar, the turbine pressure ratio can be increased by 25 %. Secondly, the same pressure ratio can be achieved with a lower inlet pressure, so that that the same flow functions can be achieved with a reduced mass flow. Third, a lower inlet pressure also allows the brake compressor to supply more airflow. Fourth, a higher turbine expansion ratio leads to a higher driving power for the brake compressor thus increasing its capabilities. All these effects lead to the significantly increased operating range shown in Figure 12.

The two arrows in the diagram indicate in which direction the operating point moves with increased suction (lower arrow) and with increased compressor station mass flow (upper arrow).

The upper thick line indicates the maximum allowable rotational speed of the rotor. The slight slope can be explained by the fact that turbine inlet temperature at higher pressure ratios are higher thus leading to a smaller corrected speed at the same physical speed.

Performance of a test run

To illustrate the practical interaction of the test rig components Figure 13 gives the time history of the most important operational data during a test run. The test run took about 150 minutes, which is required for collecting data at four or five different measurement points. Transient thermal conditions during start-up and load changes make it necessary to wait and stabilize the system for a certain amount of time. Especially at the beginning the massive housing of the heavy industrial brake compressor takes a long time to warm up and in this phase a very high unknown heat flux leaves the system and makes measurements inaccurate. During “Measurement Period 1” data for an operating point at 100% corrected speed are collected. This operating point is achieved with maximum suction, a compressor station mass flow of about 8.3 kg/s and maximum counter swirl of the brake compressor. It takes about 10 minutes to traverse the probes for scanning the measurement planes. During this time interval the turbine inlet condition, the exit pressure as well as the rotational speed should drift very little.

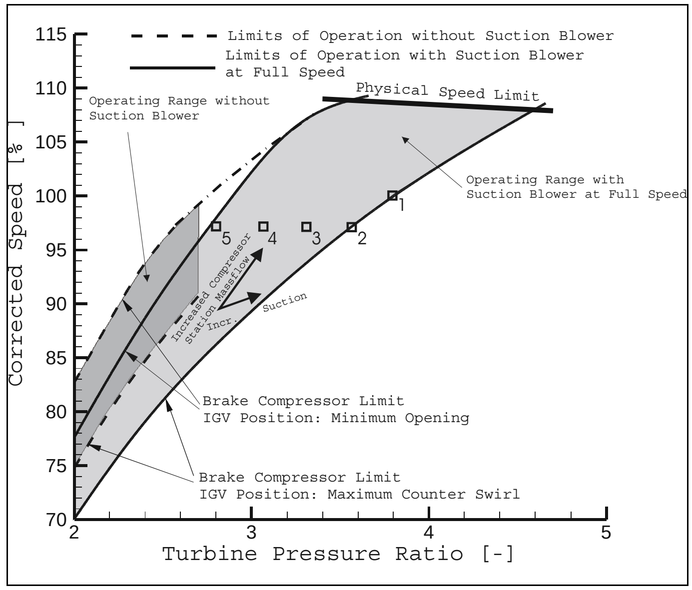

Figure 12: Industrial stage: Operating range

After that measurement phase the compressor station mass flow was reduced to lower the corrected speed to about 97%. Corrected speed should be kept within close limits during the measurements, the step in the IGV curve at 55 minutes indicates a fine adjustment. Ten minutes later the system is stabilized so that “Measurement period 2” could start. Further measurements are performed at the same corrected speed, the pressure ratio is adjusted by reducing the mass flow of the compressor station and turning the IGVs of the brake compressor towards minimum opening. Points “1”, to “5” in Figure 12 correspond to measurement periods “1” to “5” in Figure 13.

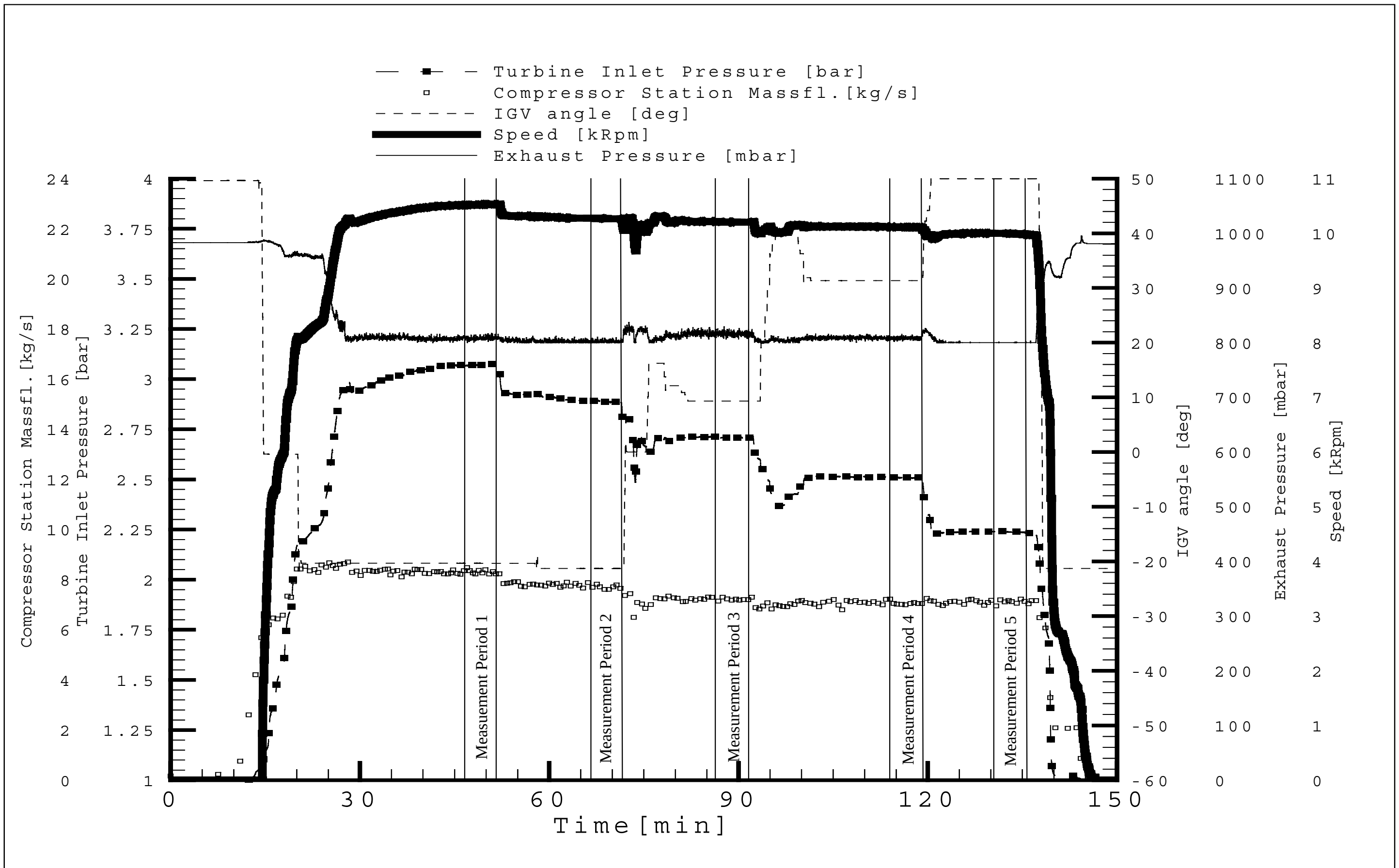

Figure 13: Time history of a typical test run

CONCLUSIONS

A cold-flow transonic test turbine for continuous measurement operation was presented, where many components work together to obtain different operating points. The characteristics of the components were described and maps based on measurement data were displayed to illustrate the very complex interaction of these components. It could be shown that a broad range of operating points can be achieved although the test object (the test turbine) influences its inlet conditions by driving a brake compressor that feeds additional air. Finally a typical test run was presented to give an impression of the time history of the system during start-up, load changes and shutdown and how to obtain different operating points.

ACKNOWLEDGMENTS

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