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Design and Construction of a Transonic Test-Turbine Facility

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Abstract

This paper describes the design concept and construction of a continuous operating cold flow transonic test turbine facility, which is a unique combination of a 2 MW axial test turbine and a direct coupled brake compressor. To cover the losses additional air is provided through a separate electrically driven compressor station and fed into the turbine inlet casing via special mixer inserts which are also fed by the hot compressed air from the brake compressor. The main test rig dimensions and its capabilities are outlined.

A brief description of the design process with step by step advanced technology design methods is presented from the first ideas up to the latest plant expansion for special testing conditions where two industrial transonic turbine stages will be tested. The aerodynamic and structural design of our transonic single turbine stage (TTM-Stage) is described in detail and the results of the respective flow simulations are presented.

Introduction

The University of Graz had decided many years ago to build new premises for its Institute for Thermal Turbomachinery and Machine Dynamics (TTM). Since the lab completion was finished the design and construction of the Transonic Test Turbine Facility was started from the following considerations:

Industrial turbines with their very expensive hot parts up to now show considerable potential in cost reduction through stage number reduction at comparable efficiencies. This calls for an increase of total enthalpy drop per stage. But to assure the high reliability and long life criteria, blade peripheral speeds could not increase that much and hot part stress and life design indicates the need for higher loading coefficients with considerably higher Mach number levels. This leads to severe supersonic shock activity which has to be optimised by suitable aero design strongly supported by nonsteady flow analysis and testing to provide still acceptable stage efficiencies.

Existing continuous running air test turbines [1] do have their big advantages in scale but in flow similarity condition they are limited to subsonic flows. Test turbines which also could operate in continuous transonic flow e.g. [2] are often limited in the available power for testing highly loaded transonic stages in full flow similarity.

But also the state of the art compression tube turbine facilities at the VKI and MIT which can achieve full flow similarity for the main flow and perhaps even cooling effectiveness have too short continuous operation time to observe all flow details researchers looking for.

To complement the collection of test turbine facilities Prof. Jericha decided well to build up a test rig for continuous operation at full scale and transonic flow ability with blade interaction and pressurised air as flow medium in an open cycle.

Facility Operation Concept

At our institute a compressor station is located in the second basement consisting of two turbo and one double screw compressor with two coolers. The electrically driven machines can be connected in a highly flexible configuration so that air can be provided continuously from 2.5kg/s up to nearly 16.0kg/s with a pressure ratio up to 2.9 bar in parallel operation and 7.0kg/s at 10.0bar in serial operation.

The compressed air temperature can be adjusted from about 150°C down to ambient temperature +15°C in the coolers. Alternatively the test turbine supply pipe (350mm diameter) with its venturi nozzle for mass flow measurement connects the 3bar low pressure or the high pressure line to the test turbine inlet casing at a maximum of 5bar. Mixing with the additional air from the brake compressor and straightened up in the vertical direction through a tandem cascade the flow enters the inlet housing, accelerates further downstream to the nozzle and generates up to about 2.0MW power in the blading of the test turbine, see Fig.2.

A membran type coupling directly transmits this turbine shaft power to our three stage centrifugal compressor GHH G5-6/3L (originally rated at 11175rpm at a pressure ratio from 0.99 to 4.1bar with an airflow of 9.0kg/s to 1.9 MW at our national refinery ÖMV). We got this machine in 1994 and found it suitable as a brake and an additional air supply after recovery of the rusty stator parts and the replacement of the old rotor by a new spare one. This machine sucks air from the test hall through a large filter, a venturi nozzle for mass flow measurement and adjustable inlet guide vanes, which can be manually controlled by a stepping motor to adjust the absorbing brake power. At operating conditions it also can be switched to control the speed of the machine train automatically.

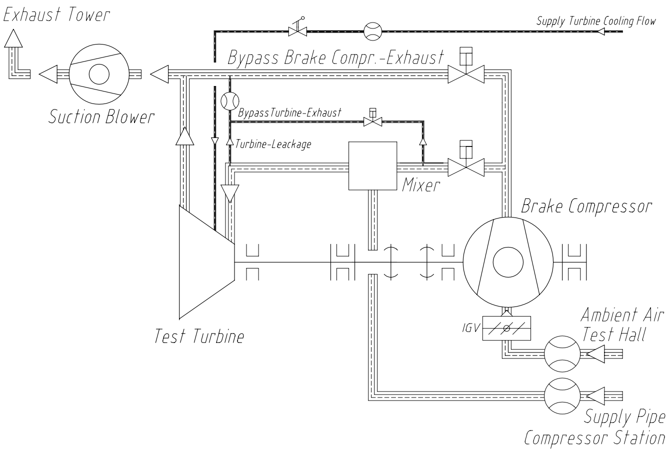

Fig . 1 Transonic Test-Turbine Facility flow scheme

The air compressed through this machine can be used for driving the test turbine if the pressure ratio is lower than approximately 4 and the running speed of the stage to be tested is high enough. Through the overflow pipe (350mm diameter) connecting the GHH brake compressor with the turbine the pressurised air is fed to the mixer in the bottom of the turbine inlet casing to provide the smallest possible temperature distortion in the stage inlet. The use of the compressed air in the test turbine enables us more than doubling the mass flow of our compressor station, which is a big advantage in testing high loaded turbine stages in full similarity.

Because the GHH compressor limits the running speed of the machine train to maximum 11500rpm the brake air has to be bypassed to the exhaust line through a throttling valve if the pressure ratio for the test stage is higher than 4.2 at maximum speed. Since this pressure ratio mainly depends on the machine train speed the throttling mode could also be necessary at lower pressure ratios if the turbine speed is limited regarding similarity conditions. In throttling mode a valve between the brake and the test turbine inlet casing has to be shut to provide pressure independent test turbine operation, see Fig.1.

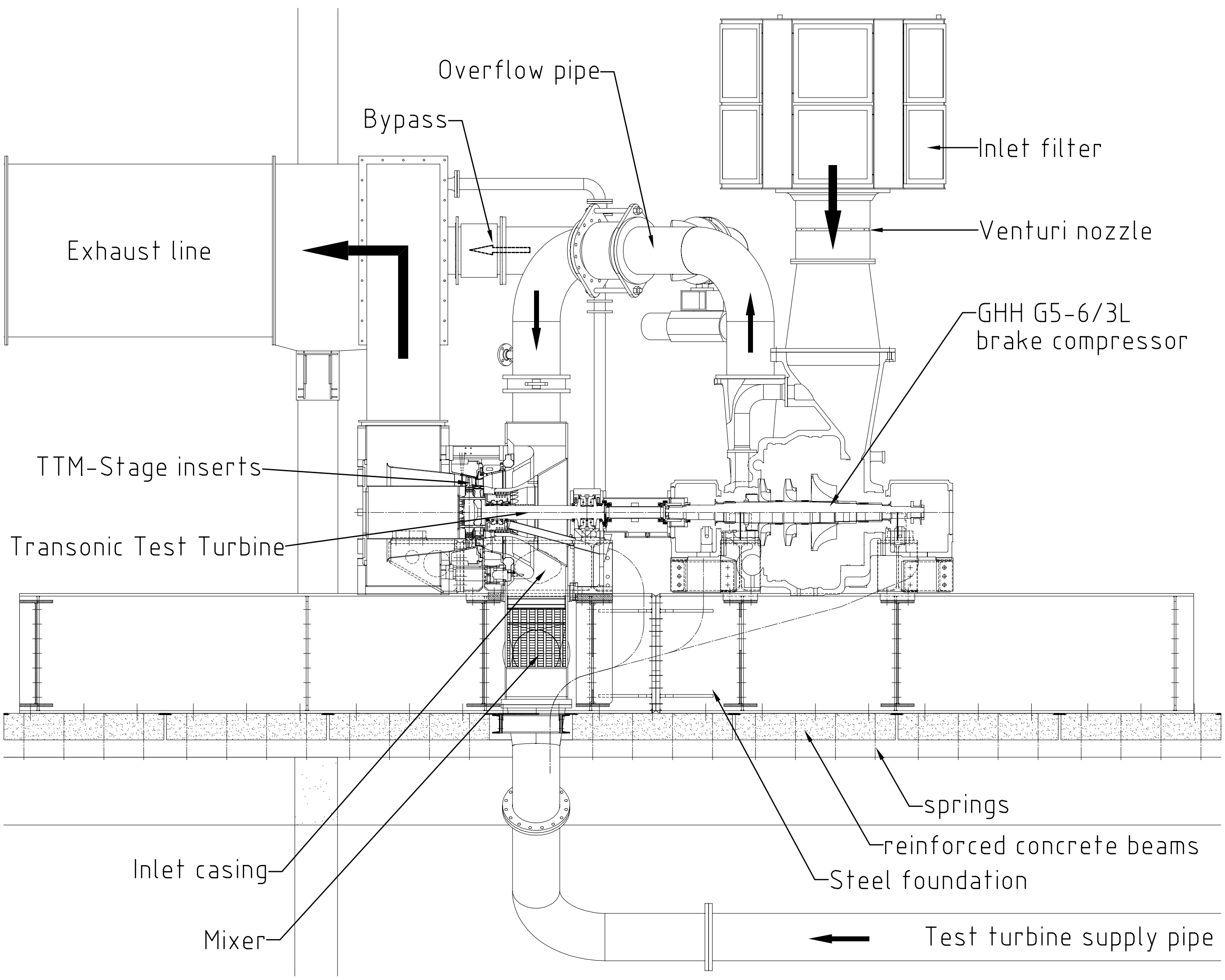

Fig. 2 Transonic Test Turbine Facility section

Normally the exit condition after the diffuser with its 90° flow bend upwards in the diffuser collector is set to ambient pressure. But for some more freedom in pressure ratio and power we have designed and are now building up a suction blower assembly, which can be put into the exhaust line in front of our exhaust tower if necessary.
Besides the stage main flow also leakage and cooling flows can be provided. For this purpose 16 pipes of 15mm inner diameter are connected to the inner inlet section of the test turbine forming groups of four pipes leading to one quadrant each. So if required in some pipes leakage mass flow from the test stage is guided through a measurement orifice to the exit line and at the same time cooling flow can be pressed in parallel through the other pipes [17].

Facility Development History

In 1994 two persons started in building up a state of the art test facility and a computational fluid dynamics (CFD) code development in parallel within the research project "Efficiency improvement and emission reduction of thermal power plants" of the Austrian Science Foundation. The own developed transonic turbine stage (TTM-Stage) (see Fig. 3, 8 and 9) was designed and optimised first on a five blade sections one dimensional calculation with an estimation of losses from [3] and [4] including the machine characteristics of the compressor station operating conditions and the GHH-brake compressor behaviour. With the free parameters speed, meridional path dimensions, turbine inlet parameters, a constant vane exit flow angle and specific work distribution over the blade height I got the geometrical data for the stage inlet and outlet boundaries with the corresponding 3D blading design. The system optimisation for maximum coupling power and high efficiency provides effective design point operation for the GHH-brake compressor and the institutes compressor station which helps in maximising blade height respectively preventing the two end zone loss regions from merging into a single loss structure.

To get the right machine scale without cost overruns a principle decision was made to manufacture a high number of parts our own. Due to cost reasons my first task was an old industrial steam turbine with two Curtis wheels which was planned to be adapted with a new blading but the calculations of the shaft bending modes and the bearing stability of this design did not allow to run it above 9000 rpm. So we decided to build a new test turbine that could satisfy our high demands in a wide adjustable speed range between 7000 and 11500 rpm.

In parallel to this design work the test rig oil supply and large foundation had been built up and the brake compressor had been set on its new place. The shaft, the disks and the blades were ready and the casing was under construction when 1997 a big gasturbine manufacturer met us in Graz and decided for testing turbine stages in Graz.

As TTM had already developed their own TTM-Stage to run the machine and to build up the required measurement techniques in conventional and also optical methods, this stage was now used for commissioning the transonic test turbine facility first in the Brite Euram project "Development of Transonic Industrial Turbine Stages".

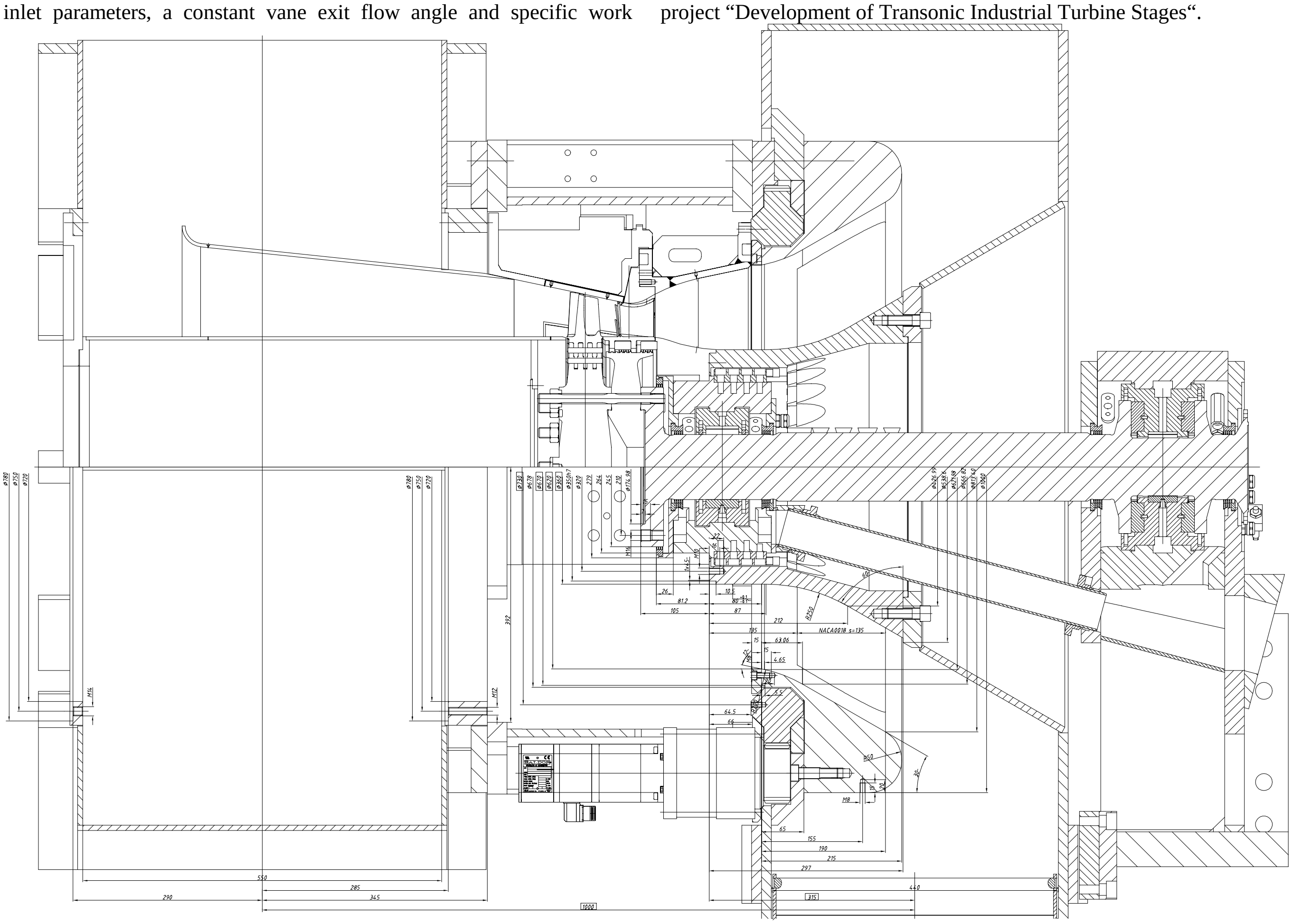

Fig.3 Transonic Test Turbine facility with TTM-Stage (upper half) and fitting dimensions for test stage inserts (bottom half)

Because the new stages needed other boundary conditions in terms of meridional path dimensions and flow parameters for full similarity conditions, considerable changes and also additional parts had to be designed and implemented. The four most important modifications were:

(1) A complete new inlet casing more than doubling the old volume, designed with the help of CFD calculations to give better flow acceleration, an enlarged meridional gas path and the implementation of a mixer.

(2) In comparison to the old test rig concept with approximately 50°C temperature difference between compressor station delivery and uncooled air from the GHH brake compressor now due to flow similarity conditions there can be temperature differences nearly up to 200°C. So we have to build in an effective mixer. The most challenging condition was optimised with a matrix of CFD-calculated mixer configurations generating temperature profiles versus mixing length and one dimensional estimations of pressure loss. The mixing requirements were set to +/-5°C after 300mm mixing length with minimum pressure loss at acceptable design efforts. They have to be fulfilled also when the uncooled brake presses 230°C hot air through seven 19mm wide slits at high velocity and a pressure loss of about 70mbar to mix up with slow velocity streams in 42mm wide registers fed with cooled air coming from the compressor station at about ambient temperature +15°C and a mixer pressure loss of about 30mbar. The resultant distortion of the inlet temperature profile is estimated to be less than +/- 3°C**.**

(3) The overflow pipe first planned very simple for the TTM-Stage was supplemented with a bypass to the exhaust line and additional valves.

(4) If necessary for testing we now will have the opportunity to install a suction blower in the strengthen exhaust line, which is driven through an old helicopter engine. Due to the high electrical power requirements of our compressor station we will drive the blower with an old RR GNOME H.1200 Mk.610 which was used in an Agusta-Bell 204B helicopter rated at 750kW. The 6600rpm power output is now reduced to the 3000rpm subcritical blower shaft assembly by means of a 250mm wide high speed flat belt drive.

All this work led to the following final test rig boundaries (see Fig. 3) and design concepts :

<u>Mechanical & Operational:</u>

- Continuous operating cold flow test facility in open cycle
- Use of a compressor as brake for multiplying mass flow
- Wide adjustable speed range of the test rig with the first bending modes of the two shafts beyond 7000 rpm and the second bending mode sufficiently higher than the maximum speed of 11500 rpm
- Stable tilting pad bearings also at the turbine shaft
- Overhung type turbine shaft for easy disk assembling
- All casing parts split for easy maintenance (except diffuser inserts)
- Modular design for quick modification in test setup
- Test section with high flexibility in meridional path [mm]:
  - Stage inlet adapters starting at D_i = 360, D_a = 620
  - Test section inserts maximum Diameter D_s = 800
  - Diffuser insert flanges D_i = 720
  - Test section length L_{Test} = 406
  - Diffuser length L_{Diff} = 620
- Test turbine stage boundaries (with suction blower) :
  - Mass flow max. about 18kg/s at 2.0MW (22kg/s at 2.5MW)
  - Inlet pressure max. 5bar
  - Inlet temperature max. 180°C
  - Outlet pressure 0.97 (minimum 0.77bar with suction blower)

<u>Measurements :</u>

- Speed, Torque and Power at Membran type coupling
- Mass flow, pressure and temperature for
  - compressor station supply pipe
  - GHH-brake compressor inlet and outlet
  - stage leakage flow outlet
  - stage cooling flow inlet
  - bearing oil supplies inlet and outlet
- Inlet casing pressure and temperature for turbine total inlet parameters
- Diffuser exit condition in pressure and temperature
- Suction blower pressure difference to ambient
- Four measurement planes for more detailed investigation each depending on stage inserts and testing to be performed (see Fig. 8)
  - A in front of the guide vanes for stage inlet
  - B between guide vanes and running blades
  - C behind the blades for stage outlet
  - D axial diffuser exit
- Rotatable stator ring (+/-180°) for simple radial-circumferential traversing access and in the special case of our TTM stage also different vane trailing edge positions
- Best optical access to the TTM-Stage starting from the vane exit to the diffuser inlet (see Fig. 4 and 8).

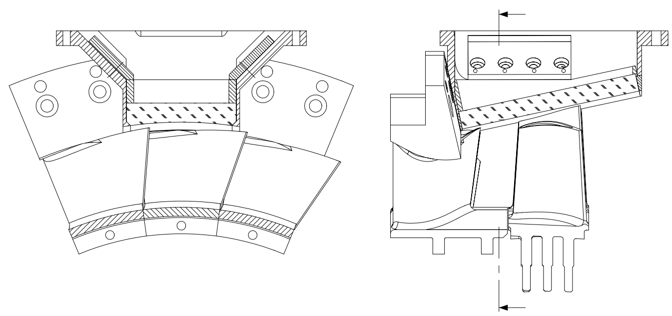

Fig. 4 Particle Image Velocimetry (PIV) window in TTM-Stage

<u>Safety Concept</u>

Exceeding preset danger values each one of the following signals release an emergency shutdown of the machine train through cut-off the compressor station and fast opening (0.4s from 0% to 100%) of the two main valves in the overflow pipe to the bypass into the exhaust line.

- Electric circuit interruption to emergency supply
- 2 channel overspeed system on turbine shaft coupling end
- XY Orbit / relative shaft displacement in the four bearing planes
- Axial displacement at each axial bearing for thrust control
- Temperature measurement of all tilting pad bearings
- Inlet pressure control for all four bearing supply pipes
- Pressure control for the main and the auxiliary electrical oil pump
- Temperature and oil level control of the oil tank
- Overpressure of the Turbine inlet casing

In the worst case the free accelerating turbine (e.g. because of coupling break up due to a severe rub in the brake compressor) should be caught at about 25% overspeed.

The two main valves are opened through spring forces and the most important monitoring systems are supplied from the electrical emergency grid to store the shutdown data for later analysis.

Because the oil pumps are electrically driven only, they are automatically switched to an independent emergency supply in the case of any electric circuit interruption to supply the bearings with oil during shut down, run out and cooling phase.

TTM-Stage Aerodynamic Design

TTM-Stage Design History

To have the opportunity of free design and publishing data it was decided to build an internally developed transonic turbine stage for our test facility (TTM-Stage). In comparison to the high pressure research turbines shown in [5] the aerodynamic loading was quite high first. The loading factor was set to about 2.0, the pressure ratio at 4.1 and the flow coefficient at about 0.65. This led to a hub to tip reaction variation of 0.25 to 0.50 and average Mach numbers in the midspan section of the vane and the blade exit of about 1.1.

In this first approach the Euler calculations showed very high shock losses and we recognised some problems with our constant vane exit angle at the high design pressure ratio of 4.1. Because it was first decided to keep the constant angle we lowered the total inlet pressure to 3.44 bar to reduce these losses. This reduction also provided a better part load condition at 2.9 bar, so that the high pressure ratio at serial operation of the compressor station would not always be necessary.

In the following iterative cycles between the one-dimensional calculations, the 3D Euler flow and the detailed structural analysis of the stress and vibration modes of the blading the new geometry was optimised and the results of the 3D Euler calculations using an solver with implicit approximate factorisation of Beam and Warming were shown in [6].

In a parallel iteration the flow optimised blade was investigated in finite element method (FEM) stress and vibration calculations and especially its connection to the disk was studied in many different ways to predict the operational behaviour. In spite of all efforts the really measured vibration frequencies of the bladed disk modes surprisingly were to close at the nozzle frequency and so we had to reduce the guide vane number from 29 to 24 to overcome this problem.

For this we designed and calculated [7] a completely new vane using a linear projection of the VKI-LS 82-05 [8] for our hub and tip vane section. To improve the stage efficiency and to keep the maximum Mach number on the vane hub suction surface on a still acceptable value we rejected the constant guide vane angle design and opened the vane at the hub to get some higher reaction. Because the mass flow then increased above the required 18.1kg/s we had to turn the vane somewhat closer at the tip and to scale the blade thickness in the hub and tip design sections. Additionally we increased the design point speed from 10500 to 11000 rpm to have acceptable incidence angles at the already existing running blade. The meridional path and the other machine characteristics fitted well with the new vane design and the turbine coupling power was calculated to about 1.9MW at the old total to static pressure ratio of 3.44 to 1.1bar and total inlet temperature of 454.4K.

Fortunately our CFD codes were built up in parallel so that we could examine the stage design in a first full Navier Stokes calculation sometime later. The Navier Stokes investigation of the 24 guide vane nozzle in difference to the beautiful 3D Euler calculation showed severe problems in the hub boundary layer at the trailing edge shock of the guide vane. Although the manufacturing had already started we redesigned the nozzle once more to get the best we could do in a few days.

Navier-Stokes Solver

The flow solver used in the final design and flow simulations is based on a time-marching Euler code [7],[9]. Here only a brief outline of the main features of the Navier-Stokes code is given.

The 3D-Reynolds/Favre averaged Navier Stokes equations with Spalart and Allmaras [10] turbulence closure are treated in conservative form and discretized in time by applying an implicit method leading to a set of non-linear finite difference equations which is solved by applying a Newton-Raphson relaxation technique.

In stationary simulations convergence is optimized by using a local time step based on a local stability criterion and in addition, by applying a multigrid procedure based on [11],[12]. The convective (Euler) parts are discretized using a third-order-accurate, TVD-upwind, cell-centered finite volume scheme, based on Roe's approximate Riemann solver [7],[13]. In order to construct the numerical viscous flux vector at the cell interfaces, a second order accurate central differencing scheme has been adopted. In the present cell-centered scheme, phantom cells are used to handle all boundaries. According to the theory of characteristics, flow angle, total pressure, total temperature and isentropic relations are used at the subsonic axial inlet. At the subsonic axial outlet the average value of the static pressure is prescribed, density and velocity components are extrapolated. On solid walls, the pressure is extrapolated from the interior points and the non-slip adiabatic condition is used to compute density and total energy.

Quasi-3D nonreflecting boundary conditions for sub-/supersonic in-/outlet, based on the work of Giles [14] and modified by Pieringer [15] are implemented and a simple mixing plane approach has been adopted to model the rotor-stator interaction problem for steady flow analysis.

Structured multiblock grids are used to discretize the flow region. All meshes are generated with an algebraic multi-block grid generator based on Bézier curves and Bézier surfaces [16]. The code is also able to handle moving (e.g. rotating) and deforming grids.

In order to meet the desired design goals, CFD had been included in the design phase of guide vane and rotor blade. Starting with an initial design for the blading geometry, based on one dimensional analytical tools, a series of quasi-3D and full 3D-Euler computations led to a refined geometry which then was further optimised by performing Quasi-3D and full 3D-Navier Stokes simulations.

Most of the computations were done on different grids in order to ensure grid independent solutions. To give an example, a typical O-type mesh, used for 3D-viscous flow analysis is displayed in Fig. 5. The boundary layer is resolved down to the viscous sublayer, therefore the grid is exponentially stretched normal to solid walls such that the size of the first cell at the blade surface is below a y^+-value of about 1.

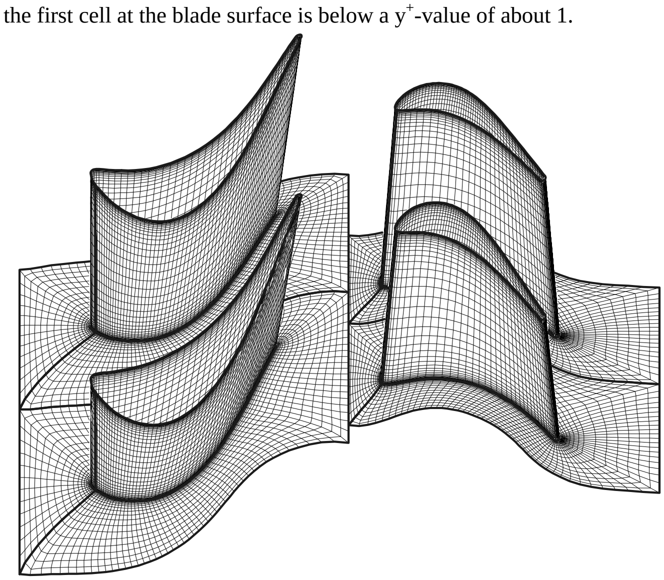

Fig. 5 Structured O-Type grid for 3D-viscous flow analysis (524288 cells in total)

In order to illustrate the flow characteristics at the design point, Mach number contours at midspan predicted by viscous flow simulations are displayed in Fig. 6. Due to the transonic pressure ratio, a strong suction sided trailing edge shock wave at the nozzle exit and a somewhat less pronounced shock at the rotor blade suction side, leading to an strong increase in boundary layer thickness can be observed.

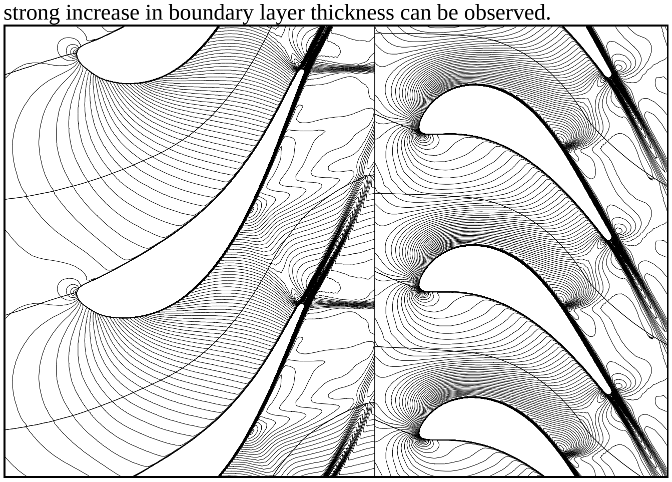

Fig. 6 Mach number contours at midspan predicted by viscous flow simulation (increment 0.02)

It should further be noted that due to the strong trailing edge shock waves, a very big influence of reflecting-nonreflecting boundary conditions has been observed especially at the nozzle exit, as shown by means of pressure contours predicted by quasi 3D inviscid calculations presented in Fig. 7.

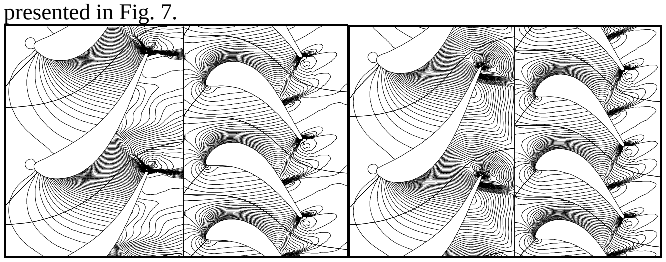

Fig. 7 Q3D-Euler predicted Pressure distribution (101, 0.5:4bar) nonreflecting boundaries left, reflecting outlet condition right

Final Meridional Section

The meridional shape of the gas path (see Fig.8) was strongly dictated from the requirements of the straight optical windows (see Fig.4). To have one window over the guide vane exit to the blade midst promises good optical access to the unsteady rotor stator interaction and to elongate the window over the blade to the exit measurement plane C (see Fig. 8) was not that more expenditure.

The technical challenge was to provide the small clearance of about 0.5mm in proper thermal alignment between the running blade and the glass window also at transient conditions [17]. The next difficulty was the rotation of the vanes to allow for quasi 3D measurements in planes spread through an axial and a radial vector. The rotation of such a traverse ring is state of the art but the free cut guide vanes under the window were a new experiment which has to be proven. The 50mm radius tangentially connects the inlet and the conical gaspath opening over the blade the angle of which was determined through the stage expansion ratio respectively the demand for keeping constant axial velocity. The inner meridional contour was designed to be manufactured cylindrically to have the possibility to use it as optical reflection surface. However this advantage also has the high price of a conical window and clearance at the outer meridional contour (see Fig. 8).

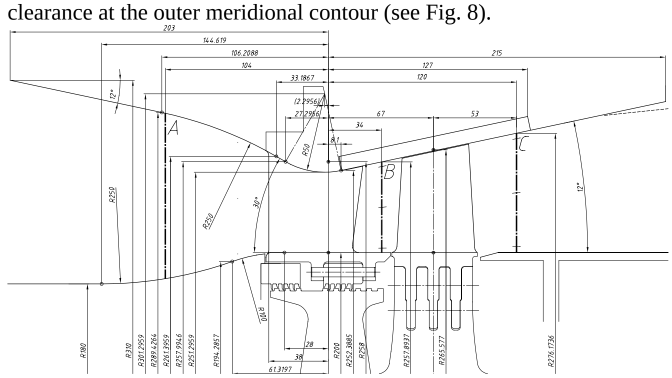

Fig. 8 Meridional Section TTM-Stage

Final TTM-Stage Design

The stator-to-rotor axial gap was designed at about 0.5 stator axial chord that is a representative value used in high loaded stages as a compromise between aerodynamic, mechanical and blade vibration considerations.

Selection of number of airfoils was based on a compromise between optimal loading, minimum trailing edge blockage and secondary losses respectively aspect ratio. Comparing the TTM-Stage to other high loaded stages the guide vane number 24 fits quite well whereas the unconventional small blade number 36 was stressed to give better optical access into a wide blade spacing, but this also led to a high Zweifel coefficient at blade midspan.

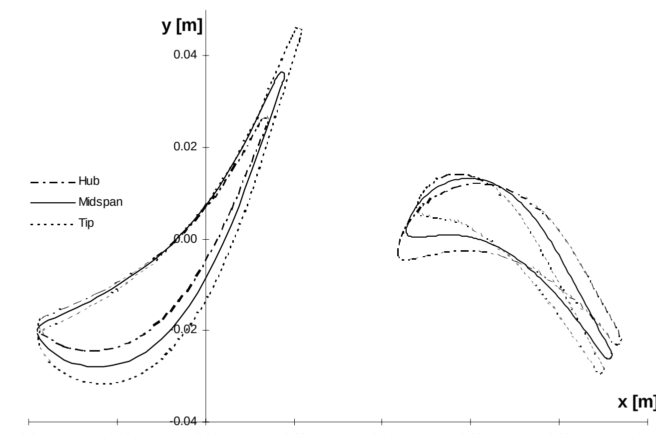

Fig. 9 Hub, Midspan and Tip sections of TTM-Stage

The blading itself (see Fig.9) was designed in only two sections each: the two hub sections of the vane and blade are located at the inner diameter of the stage in R200 whereas the tip section of the vane is defined at R258. The blade tip section is defined on R265.577 (see Fig.8) These sections are wrapped on the corresponding cylinders and then connected with straight lines between the developed hub and tip section points. The surfaces are then cut with the meridional inner and outer contour.

The TTM-Stage design point parameters as predicted from the latest Navier-Stokes flow simulations are as follows (plane A to C see Fig. 8) :

Boundary Conditions			Output	
Inlet (A)	p_{tot} [bar]	3.439	**n [rpm]**	**11000**
	T_{tot} [K]	454.4	**P [MW]**	**1.94**
Outlet (C)	p_{stat} [bar]	1.102	**m [kg/s]**	**18.1**

TTM-Stage Parameters	Hub (5%)	Mean (50%)	Tip (95%)
Pressure Ratio (total to static)		**3.12**	
ψ Loading factor $-\Delta h/u^2$		**1.54**	
ϕ Flow factor c_{ax}/u		**0.54**	
R_{kin} Reaction (hub, mean, tip) [%]	12	**27**	40
Exit swirl (hub, mean, tip) [°]	1	**8**	5
Exit Mach number	0.34	**0.40**	0.37
Reynolds number Vane	2.1E+06	**2.4E+06**	2.6E+06
Reynolds number Blade	1.5E+06	**1.6E+06**	1.6E+06
Airfoil count Vane		**24**	
Airfoil count Blade		**36**	
Aspect ratio Vane		**0.78**	
Aspect ratio Blade		**1.35**	
Zweifel´s Coefficient Vane	0.80	**0.69**	0.67
Zweifel´s Coefficient Blade	0.73	**1.01**	0.94
Trailing Edge Blockage Vane [%]	6.6	**6.9**	7.2
Trailing Edge Blockage Blade [%]	11.6	**9.4**	7.5
Clearance/span [%]		**0.72**	
Stage efficiency (total to static)		**0.85**	

(Reynolds numbers based upon chord length and exit conditions, Stage efficiency total to static isentropic across blading)

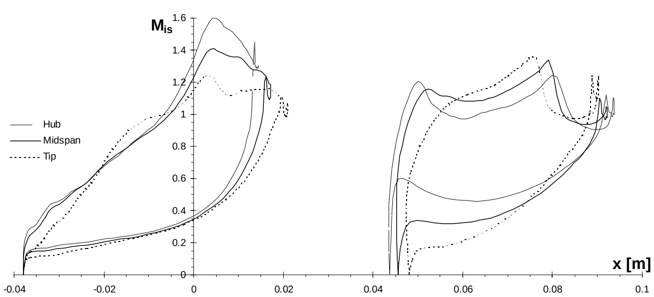

Fig. 10 Vane and blade surface Mach number distributions

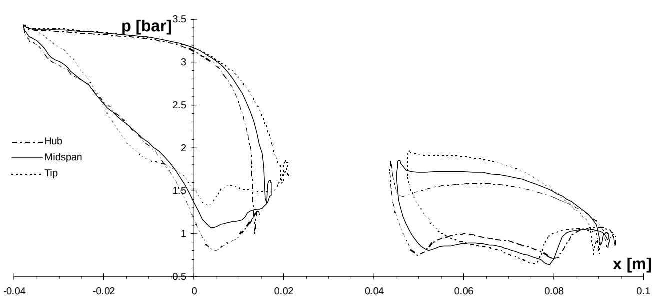

Fig. 11 Vane and blade surface Pressure distributions

Nozzle Design

The 24 vanes in their final design (all values in [mm] at Midspan) have an axial chord of 56.1, a pitch-to-chord ratio of 0.76, a turning angle of 69.7° and a mean exit Mach number of 1.11. The trailing edge thickness is 1.65 and the throat opening 22.30.

Although the original cooled test case profile with 15° exit angle now is moved to 20.3° and the higher loading at exit Mach number 1.11 (see Fig. 12 compared to 0.9 original design) quite changes a lot the flow pattern still has some similarity to the original design goal: There is only one strong trailing edge shock at the suction side end and the one on the pressure side is hardly found.

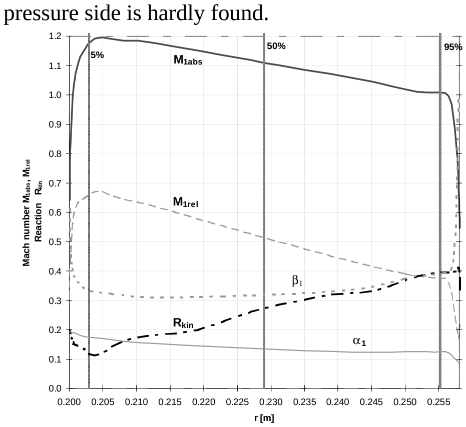

Fig. 12 Vane outlet / Blade inlet conditions

Blade Design

A cooled blade section design implies a larger leading edge for shower head implementation than e.g. a steam turbine profile. In the case of our high inlet relative Mach number of 0.66 at the hub this round leading edge in conjunction with the neighbouring blade suction surface forms a canal which converges so strong that soon after the leading edge the suction surface exceeds Mach number 1.0 from the hub up to the midspan. This high loading can be seen in the Mach number distribution of the hub blade profile in Fig. 10 and further more on the pressure profiles of the hub suction side, see Fig.11. At design point there is a 12 to 15% minimum reaction (see Fig.12) in this region corresponding with low acceleration and a high turning angle of 109°.

The 36 blades (all values in [mm] at Midspan) have an axial chord of 46.8, a pitch-to-chord ratio of 0.75, a turning angle of 106.3°and a mean exit relative Mach number of 0.89, see Fig.13.

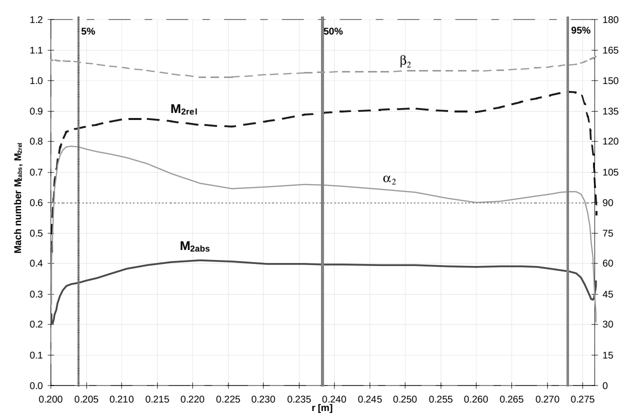

Fig. 13 Blade outlet conditions

TTM-Stage Structural Design

Just to get the highest operational reliability the test turbine design was straightened to the worst case of a free accelerating turbine shaft. As mentioned above in the safety concept the possible maximum speed is about 25% above the operational speed. So in the case of maximum speed at 11500rpm the TTM-Stage would accelerate to about 14300rpm. For the worst case of a further accelerating rotor the machine is caught through breaking off pins in the blade foot to disc fitting between 14300 and 15000rpm (the pan of the blade itself is to short respectively strength), so that it is guaranteed that the rotor disc assembly can not reach critical values.

In case of a blade crash no parts are allowed to leave the casing. The unbalanced mass that could appear after such a scenario was set to about 5% of the 36 blades mass. This worst case hypothesis in case of our TTM-Stage leads to an unbalance force of 450kN rotating at 14300rpm (one blade with its blade foot is equal to 0.55kg at 201mm Radius)!

This utmost high load would exceed the apparent limit of elasticity of the turbine shaft bending in the front bearing at about 300kN, but the bending shaft could be held in the surrounding bearing and inner casing although the screws would be on limit load at 450kN.

For testing the bladed disk it was spin at 12650rpm in vacuum. That corresponds to 15% overspeed (based on operational speed 11000rpm respectively 10% overspeed for 11500rpm maximum). After the spin test the blades could be measured to be moved max. 0.02mm.

The weak point of the casing will be the big PIV-window (see Fig.4) and also perhaps the more narrow Laser Doppler Velocimetry (LDV) window. So the test turbine first is tested with full metal covers mounted instead of the window assemblies. Then step by step first the smaller and then the bigger windows will be used if all works well on remote control and optical measurements are on schedule.

Conclusion

(1) The design concept and the construction of a continuous operating cold flow transonic turbine facility has been presented with its capabilities up to the boundary conditions for other test stage inserts.

(2) The TTM-Stage Design History showed a stressed design process in building up a high loaded “measurement machine“ for optical access to transonic shock activity in rotor stator interaction in parallel to the development of the CFD tools necessary for suitable aerodynamic design.

(3) Nevertheless it can be shown that the TTM Stage is comparable to other transonic turbines [5] in loading and flow coefficients although the design pressure ratio is not so high and the blading design is rather simple.

(4) The low reaction design in a real engine could be good for low blade metal temperature, low stage exit mach number and low root stagger angle for reduced rotor disk fixing stress and ease of platform fit. To get some better blade hub flow acceleration a higher pressure ratio would help moreover this low reaction design will have higher limit loading.

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Finally we would like to dedicate this paper to our institutes chief and adviser of our doctoral thesis Prof. Jericha to his retirement in autumn 1999.

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