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EXPERIMENTAL INVESTIGATIONS INTO THE NONUNIFORM FLOW IN A 4-STAGE TURBINE WITH SPECIAL FOCUS ON THE FLOW EQUALIZATION IN THE FIRST TURBINE STAGE

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ABSTRACT

Control stage turbines are in widespread use to control turbine power output in power generation and industrial applications. Depending on the point of operation and the turbine design the inflow to the multi-stage turbine is highly non-uniform. It is important to know how the inhomogeneities in mass flow and temperature distribution develop and attenuate inside a multistage flow passage.

The object of this study is to experimentally measure and report the inhomogeneous flow structure and attenuation in a multistage turbine, especially in the first turbine stage. In a scaled down original turbine consisting of a control stage with cross-over channel and a 4-stage turbine velocities, pressures and temperatures are extensively measured at the 360° circumference at the inlet, within the first stage and at the outlet of the multistage turbine part. By closing a 20% sector of the control stage a circumferential flow inhomogeneity with significant pressure, velocity and temperature gradients is created at the inlet of the multistage turbine. For analyzing the equalization of the mass flow distribution in the first stage, the experimental results are normalized with a reference case at full admission in order to separate flow phenomena created by the turbine geometry.

The results show that the flow inhomogeneity at the inlet of the multistage part is significant, but the flow is basically equalized after the third stage. Most of the flow equalization takes place within the first turbine stage while the guide vane is the main driver for this process. In opposite to this the temperature inhomogeneity does not attenuate significantly within the multistage turbine part.

KEYWORDS

Axial multi-stage turbine, flow equalization, inhomogeneous flow, partial admission

NOMENCLATURE

ψ_y	-	work parameter	$\psi_y = \frac{y}{u^2/2}$
y	[kJ/kg]	aerodynamic work	
u	[m/s]	rotational speed	
p	[Pa]	pressure	
$\bar{p}$	[Pa]	averaged pressure	
x	[m]	radial position	
h	[m]	channel height	
c	[m/s]	velocity	
$\bar{c}_{radial}$	[m/s]	radial averaged velocity	
$\bar{c}$	[m/s]	averaged velocity	
T	[K]	temperature	
$\bar{T}$	[K]	averaged temperature	
$\frac{d\dot{m}_{i,j}}{dA_{i,j}}$	[kg/m^{2}s]	mass flux density	

INTRODUCTION

In turbomachines the design is usually based on an uniform inflow into the blade rows. Due to the design and point of operation the flow inside a turbomachine is not always axisymmetric and uniformly distributed in the radial and circumferential direction [1-4]. In steam turbines with nozzle section control especially the flow downstream of the control stage towards the first cascade of the multistage turbine is highly irregular. This increases the danger of blade and bearing damages [5, 6] as well as mixing losses

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within the multistage turbine part. Hence power output and turbine efficiency are reduced [7-9].

From a design standpoint it is interesting to know besides the magnitude of partial admission losses how the inhomogeneities in mass flow and temperature distribution propagate and attenuate in the subsequent blade rows of the turbine.

EXPERIMENTAL SETUP

The test facility is composed of a test turbine with a control stage and four reaction stages operating with compressed air. It consists of two separately suspended shafts carrying the rotor disk of the control stage on one shaft and the four rotating blade rows of the fully admitted stages on the other. Details about the turbine aerodynamic design and geometric parameters are given in [9]. Both rotors independently dissipate their power at corresponding water brakes (Fig. 1). Each water brake and its rotor bearing are integrated into a swing frame, which runs in hydrostatic bearings enabling a very accurate angular momentum determination.

The inlet casing of the control stage of the turbine consists of four nozzle sections, separately controlled and supplied with air, covering 2x20% and 2x30% of the entire cross-section of the torus-channel. These sections are separated by sector separation bars.

The casing of the fully admitted part can be turned 90 degrees by a worm gear and an electric motor. For a complete assessment of the flow field cylindrical probes and multi-thermocouple probes are installed. These probes measurre the flow spanwise at seven radial positions in four circumferential sections (Fig. 3a) at the inlet (MP03), behind the first guide vane (MP11) and first blade row (MP12) and at the outlet (MP04) of the multistage part, fig. 2. Thus, the setup allows to determine 2D flow and temperature field on the entire circumference. The flow and temperature field is measured in circumferential direction every 2.25° and at 7 radial positions, resulting in a grid of 1120 measuring points for each measuring plane. In addition to the probes, there are several pressure taps for the determination of the static wall pressure distributions on the casing within all turbine stages.

A pair of two 3-hole cylindrical probes with pressure taps at different radial positions (Fig. 3b) allows measurements at seven radial positions of the flow channel. Eight cylindrical probes and four multi-thermocouple probes are fixed in each measuring plane of the inner casing of the turbine. Additionally the probes in MP11 are installed on a traversable ring which allows the assessment of the flow field behind the first guide vane for more than one pitch (±6°) relative to the traversable casing, fig. 3a.

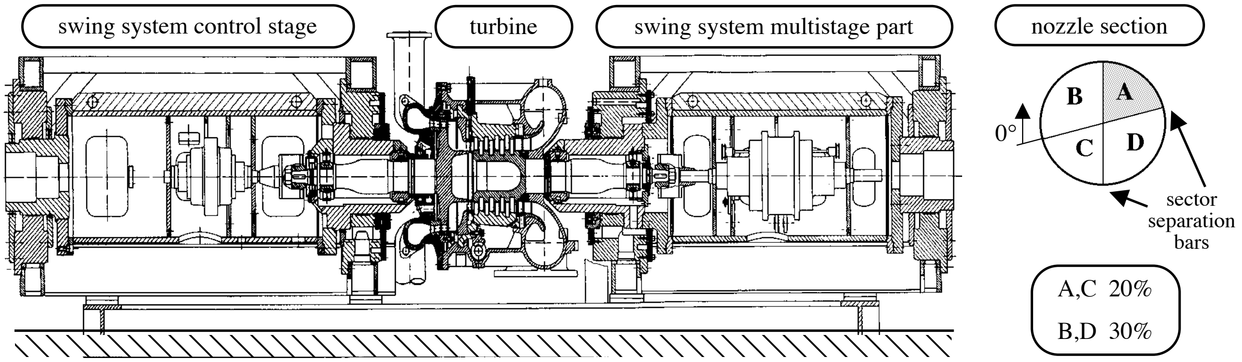


Fig. 1: Turbine test rig with water brake system

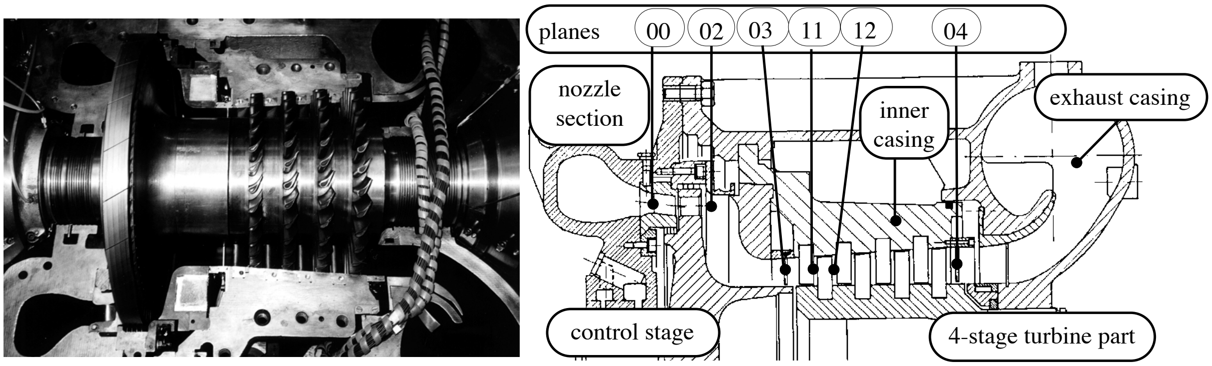


Fig. 2: Photo and cross section of test turbine with definition of measuring planes

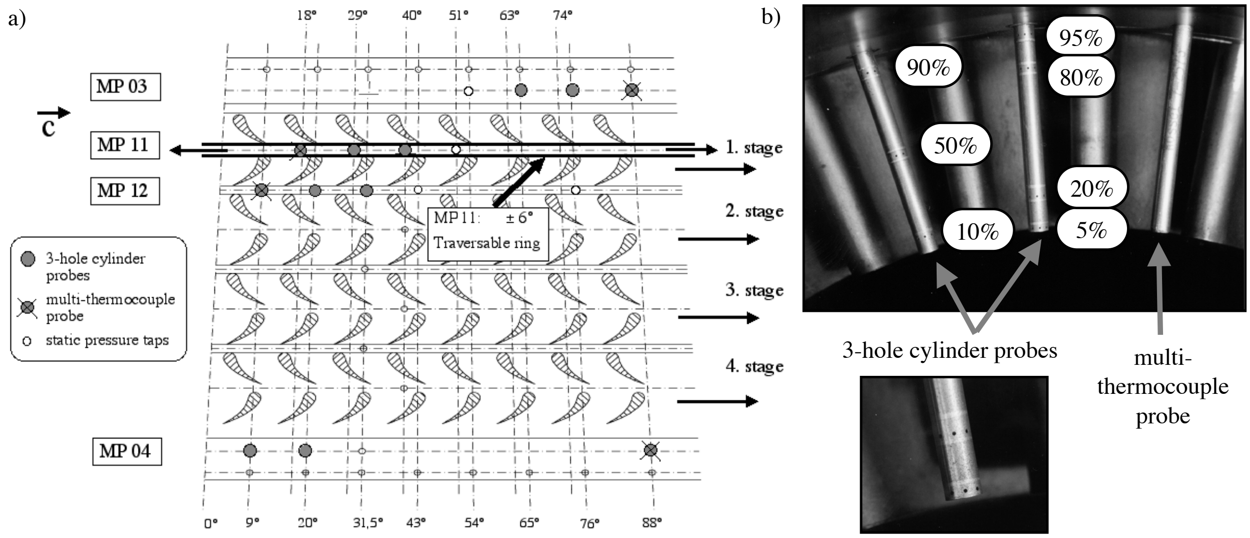


Fig. 3: a) View of the multistage turbine with position of the instrumentation in a single quadrant
b) 3-hole cylinder probes and multi-thermocouple probe with seven radial measuring planes

Extensive experimental investigations with miniaturized cylindrical probes inside this turbine test facility concerning probe blockage effects, probe-endwall interference and corrections for measurements in flows with severe velocity gradients allow the reliable analysis of the results measured behind the first guide vane [10].

Table 1: Investigated experimental configurations

Test Cases	R	M
Parameters		
ψ_y	2.3	2.3
ε	100%	80%

The experimental data presented in this paper are measured at full admission (ε = 100%) as a reference case (R) and at partial admission (M) by closing one 20% section of the control stage (ε = 80%).

The point of operation of the control stage is free of swirl at a constant pressure ratio. For full and partial admission the static pressure at the inlet of the 4-stage turbine is constant. The multistage turbine runs in the design point at ψ_y= -2,3 which represents a swirl-free repeating condition with its corresponding pressure ratio for the 4-stage turbine part at full and partial admission, table 1.

ANALYSIS OF THE NON-UNIFORM FLOW AT THE INLET OF THE MULTISTAGE TURBINE

Due to the design of the turbine there are flow inhomogeneities in radial and circumferential direction at the inlet of the multistage turbine part even at full admission. These inhomogeneities are created by the inlet geometry of the control stage and the equalization process inside the cross-over channel [1]. Furthermore, the non-symmetrical exhaust casing influences the flow and temperature field in circumferential direction inside the turbine and especially at the outlet of the multistage turbine part [11].

Fig. 4 shows the total pressure and temperature distribution of the entire 360° circumference at three radial channel heights (mid channel, hub and tip region) for full and partial admission. In the open sectors the total pressure decreases from the beginning to the end of the sector in rotational direction (Fig. 4a and 4b, labeled 4.1). The pressure and velocity distribution at full and partial admission is governed by the shape of the inlet guide tubes of the nozzle section and the sector separation bars as well as by the circumferentially inhomogeneous expansion inside the control stage. The wakes of the sector separation bars lowering the work output in the control stage also cause small peaks in the temperature distribution (Fig. 4a and 4b, labeled 4.2).

At partial admission (Fig. 4b) a significant drop in total pressure occurs in the area of the closed sector A due to the reduced mass flow. The two edges of sector A have a quite different influence on equalization of the temperature and pressure distribution. At the sector separation B-A the flow is directed towards the closed sector A by the guide vanes. Here the flow can participate at the work extraction. This results in a better flow equalization and therefore lowered total temperature (Fig. 4b, labeled 4.3). Additional the temperature distribution is significantly influenced in

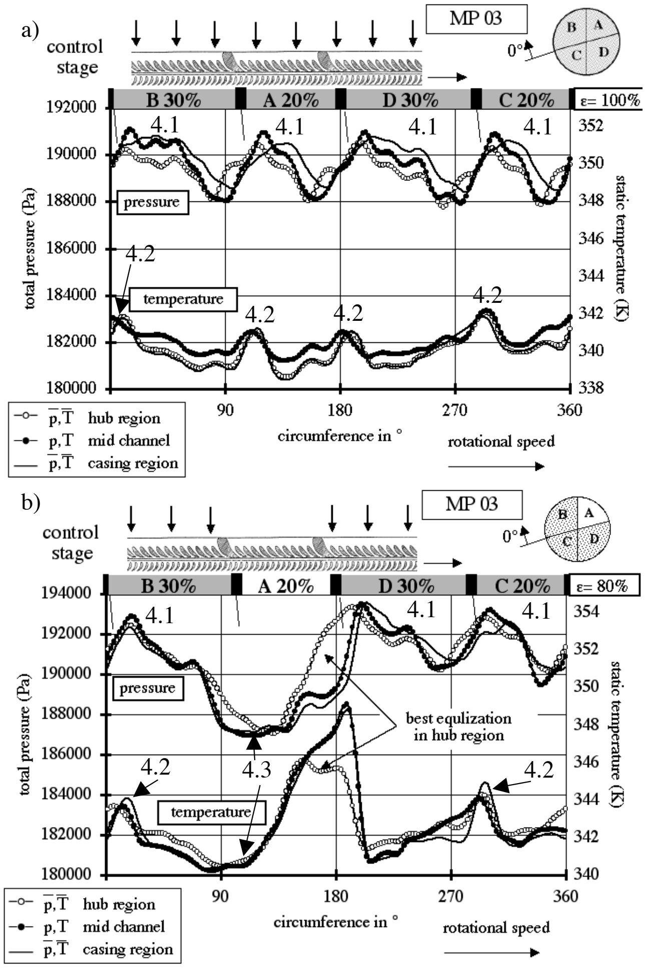


Fig. 4: Distribution of total pressure and static temperature
a) at full admission (ε = 100%)
b) at partial admission (ε = 80%)

rotating direction by windage losses and the lowered work output in the closed sector A resulting in an increase in temperature towards the sector separation A-D. Here the opposite effect occurs. The flow is directed away from the closed sector A and therefore a sudden drop in temperature and increase in total pressure can be detected.

The flow is hardly equalized inside the cross-over channel. Only in hub region the pressure and temperature is equalized somewhat due to the complex flow phenomena in the cross-over channel [12].

The flow phenomena created by the turbine geometry and effects due to partial admission can be visualized with the 360° circumferential velocity distribution. Fig. 5 shows the measured 2-d flow vector distribution at the inlet of the multistage turbine part at full and partial admission.

At full admission (Fig. 5a) the flow intends to equalize the lowered mass flux created by the sector separation bars. In casing region (80% to 95% of the channel height) the flow is quite homogeneous due to the high velocities. In hub region significant variations of the flow vectors exist due to the sector separation bars driven by the lowered mass flux and flow phenomena in the cross-over channel.

At partial admission (Fig. 5b) a similar behavior of the flow in the open sectors can be detected (e.g. sector C). The reduced mass flow in the closed sector A is visible in the velocity distribution across the entire channel height. As already discussed the two edges of the closed sector A show a different flow structure. In general the flow vector is directed towards the closed sector A to equalize the reduced mass flux in that region. At the sector separation A-D the velocity is locally increased with high gradients in the flow angle. At the sector separation B-A velocity gradients are smoother and the flow is more equalized. This equalization influences the velocity distribution up to half of the sector B. In hub region again the strongest velocity gradients can be detected due to lowered mass flux and flow phenomena inside the cross-over channel.

With the measured data of static pressure, temperature and meridional velocity the local mass flux density can be calculated for all 1120 positions in the measuring grid.

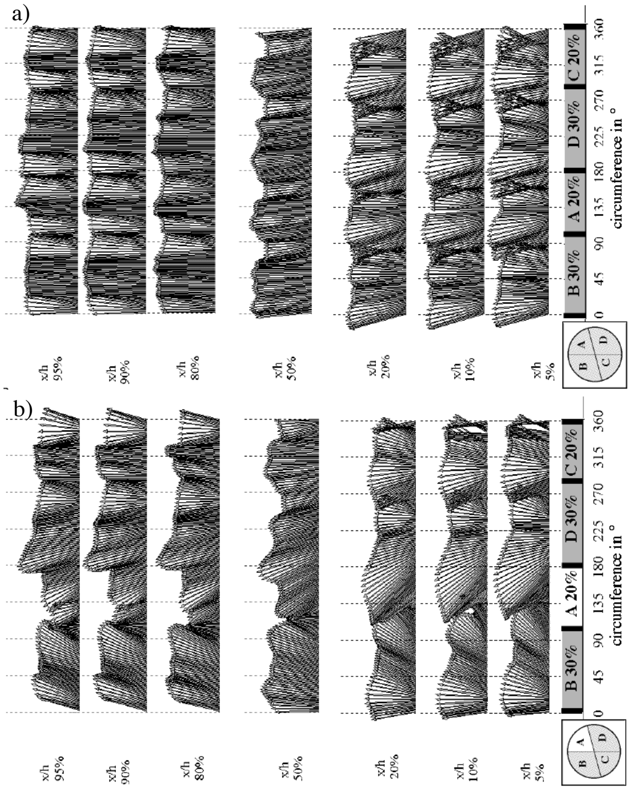
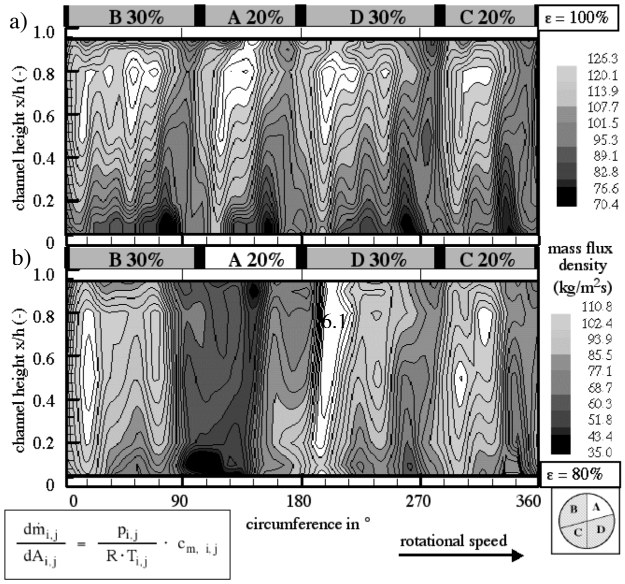


Fig. 5: Two-dimensional velocity vectors (MP 03)
a) at full admission (ε = 100%)
b) at partial admission (ε = 80%)



$$\frac{d\dot{m}_{i,j}}{dA_{i,j}} = \frac{p_{i,j}}{R \cdot T_{i,j}} \cdot c_{m,\ i,j}$$

Fig. 6: Mass flux density distribution (MP03) at a) full admission (ε = 100%) b) partial admission (ε = 80%)

Fig. 6 shows the mass flux density at full and partial admission. The flow equalization in radial and circumferential direction inside the cross-over channel is neihter perfect at partial admission nor at full admission.

At full admission (Fig. 6a) the mass flux in the outer region is larger than near the hub. The effect of the sector separation bars separating the 4 single nozzle section can still be seen as mass flow minima extended across the entire channel height. At the beginning of each sector (in rotation direction) the mass flux is higher decreasing in rotating direction. This is created by already discussed non-uniform inflow to the open sectors. The maximum radial mass flux occurs at approximately 85% of the channel height. In hub region a more equalized mass flow can be detected.

At partial admission (Fig. 6b) the mass flux in area of sector A is nearly half that of the open sectors. The maximum mass flow with the severest gradients occurs at sector separation A-D (Fig 6b, labeled 6.1) while in region of sector separation B-C the mass flow is more equalized. The maximum radial mass flux is shifted to the mid channel region due to the reduced mass flow in a constant cross sectional area inside the cross-over channel.

This radial and circumferential non-uniform flow enters the 4-stage turbine part. It should be noted that flow non-uniformities in circumferential direction have a much stronger impact on the mechanical stresses in the blades through cyclic loading than the radial velocity profile. Therefore the attenuation and development of this flow inhomogeneity is of high interest.

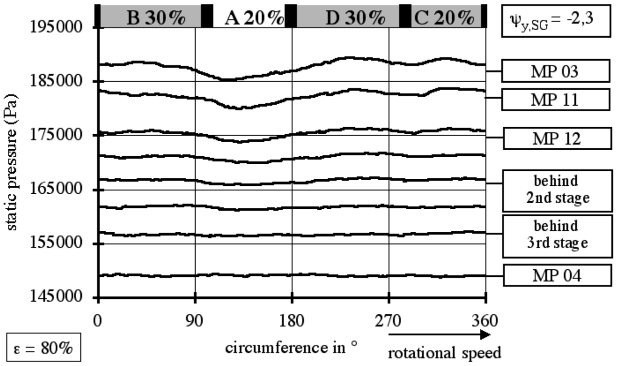


Fig. 7: Static wall pressures at the casing of the 4-stage turbine (ε = 80%; $\Psi_{y,SG}$ = 2,3)

FLOW EQUALIZATION IN THE 4-STAGE TURBINE

The flow non-uniformity decreases further during the expansion inside the 4-stage turbine part. By analyzing the static wall pressures at the casing behind each vane and blade row the equalization of the inflow inhomogeneity can be valued.

Fig. 7 shows the static wall pressure distribution at the casing for partial admission (ε = 80%) on the entire circumference. In region of the closed sector A the static pressure is lowered at the inlet (MP03). Also the influence of the sector separation bars and the goemetry of the control stage can be detected. In all vane and blade rows nearly the same pressure ratio is expanded due to the 50% reaction blading. The non-uniformity in the static wall pressure at the inlet created by closing the 20% sector A is basically equalized behind the third stage.

Behind the first cascade (MP11) the non-uniformity is shifted by appoximately 20° in direction of the rotational speed. In opposite to this no further significant shift of the circumferential non-uniformity occurs in the subsequent blade rows. The static pressure equalizes quickly in circumferential direction driven by impulse exchange effects. The flow inhomogeneity moves axially through the 4-stage turbine.

Fig. 8 shows the distributions of the static wall pressure behind each turbine stage normalized with its mean value. So the equalization effects driven by the work extraction inside the turbine are eliminated and the effects driven by impulse effects can be analyzed comparable for full and partial admission.

At full admission (ε = 100%) the fluctuation due to the sector separation bars is even equalized behind the first stage. In the following turbine stages the upstream influence of the non-symmetric exhaust case creates a maximum in static pressure at 165° circumference. Despite this the flow is basically equalized.

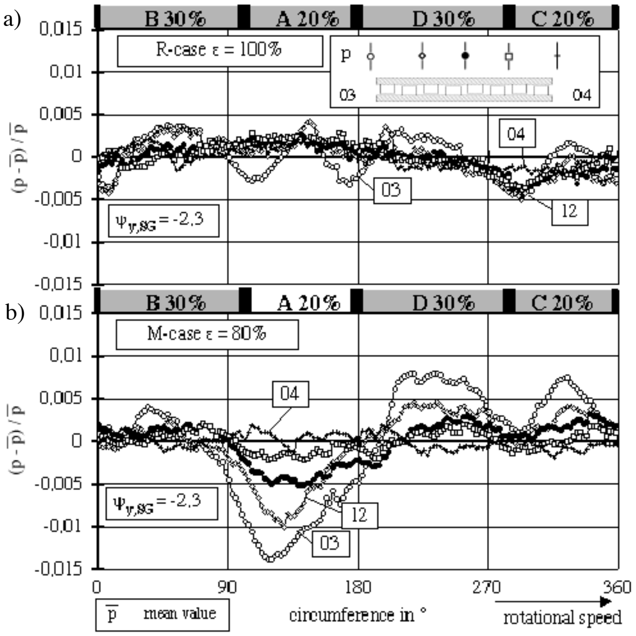


Fig. 8: Normalized static casing pressure at a) full admission (ε = 100%; $\Psi_{y,SG}$ = 2,3) b) partial admission (ε = 80%; $\Psi_{y,SG}$ = 2,3)

At partial admission (ε = 80%) the static wall pressure distribution at the inlet of the turbine is influenced by similar effects as already discussed in Fig. 4. During the expansion inside the first stage a significant equalization of the flow in area of sector A from 2,3% to 1,5% of the static mean pressure occurs. After the 3rd stage the flow is equalized with a maximum variation of 0,5%.

For further quantification the standard deviation of the static wall pressures at the casing for full and partial admission is used. The inhomogeneity attenuates quickly within the first stage by nearly 50%. Behind the second stage nearly the level of the reference case is reached. Behind the third stage the flow is basically equalized.

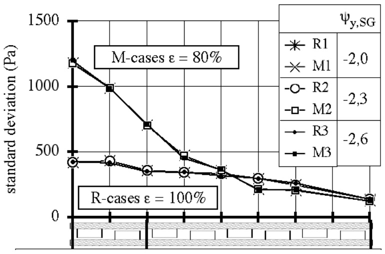


Fig. 9: Standard deviation of static wall pressures at full and partial admission

In order to investigate the influence of small swirl ratio variations inside the multistage turbine part, the point of operation is varied from a co-rotating swirl at Ψ_y= -2,0 to a counter-rotating swirl at Ψ_y= -2,6. Only slight differences in flow equalization can be detected in the standard deviation and therefore this influence is not further analyzed in this paper.

It is to state that the main equalization process takes place within the first stage. Therefore the flow inside the first stage has to be analyzed more in detail.

FLOW EQUALIZATION WITHIN THE FIRST STAGE

The fluctuations of the spanwise mass-averaged velocity normalized with its mean value judge the non-uniformity of the flow across the circumference. Fig. 10 shows these fluctuations at full and partial admission at the inlet of the 4-stage turbine and behind the first turbine stage.

At the inlet the already discussed variations in velocity due to geometry of the control stage and the cross-over channel result in a maximum non-uniformity of ±10% at full admission (ε = 100%) and more than ±30% at partial admission (ε = 80%). At the outlet of the first stage the fluctuations due to the sector separation bars are still visible but somewhat equalized. The fluctuation range of ±8% is nearly the same for full and partial admission. In circumferential direction the upstream effect of the non symmetric exhaust casing influences the flow significantly. Besides this the flow on the entire circumference (and especially in the region of sector A) is quite homogeneous.

The flow field behind the first guide vane is of high interest in order to evaluate the influence of vane and blade on the equalization within the first turbine stage.

Inside the turbine the flow distribution is influenced by the geometry of the turbine and the presence of the non-uniformity itself. To separate these equalization effects from general effects inside the turbine, the velocity distributions of the partial admission case is normalized with the reference case at full admission. By applying this method, the influence of the exhaust casing and general turbine

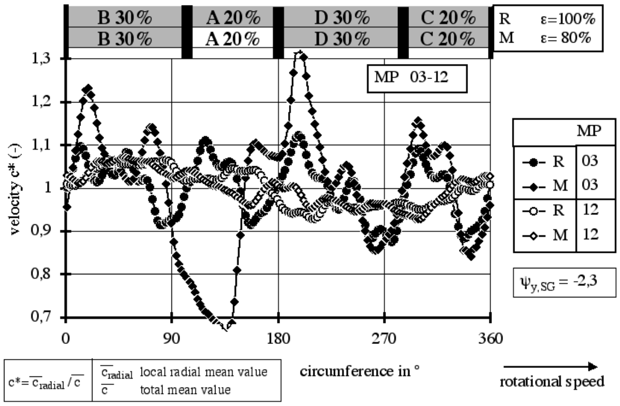


Fig. 10: Velocity fluctuations at the inlet and behind the first turbine stage at full and partial admission.

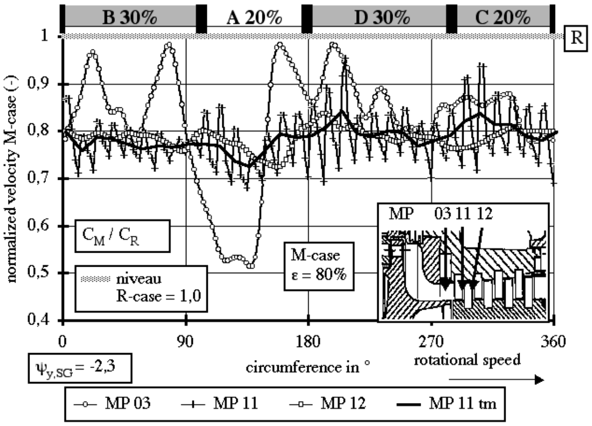


Fig. 11: Velocity ratios within the first stage (MP03-11-12) at partial admission

effects is eliminated. It is possible to analyze the attenuation effects of the flow non-uniformity at partial admission compared to full admission.

Fig. 11 shows the normalized mass-averaged velocity ratios at the inlet of the first stage (MP03), behind the first guide vane (MP11) and at the outlet of the first stage (MP12). In order to smooth out the pitchwise fluctuations of the 32 guide vanes and judge the equalization behind the entire cascade also a pitchwise-averaged circumferential velocity ratio (MP11 tm) is given.

The level of the velocity ratios in average is about 80% of the reference case corresponding to the partial admission ratio of ε = 80%. The circumferential non-uniformity at the inlet is significantly smoothed behind the first guide vane (MP11 tm). Especially the reduced velocity ratio in region of the closed sector A is equalized quite well behind the first guide vane and further attenuated during the expansion inside the first rotor.

The flow equalization takes place inside and in front of the first guide vane. A possible theory is that the upstream influence of the guide vane in combination with the velocity gradients at the inlet and the circumferential pitchwise work extraction within the first stage equalizes the flow quickly by impulse exchange effects.

With the experimental data an analysis of the mass flow redistribution in the entire cross-section within the first turbine stage is possible. In Fig. 12 the local mass flux density at partial admission is normalized with the mass flux density at full admission at the inlet (MP03), behind the first guide vane pitchwise-averaged (MP11) and at the outlet of the first stage (MP12).

The overall averaged value of the mass flux density ratio for all three measuring planes is 80% of the mass flux density at full admission. The closed 20% sector A causes a mass flux density ratio minimum down to 40% at the inlet.

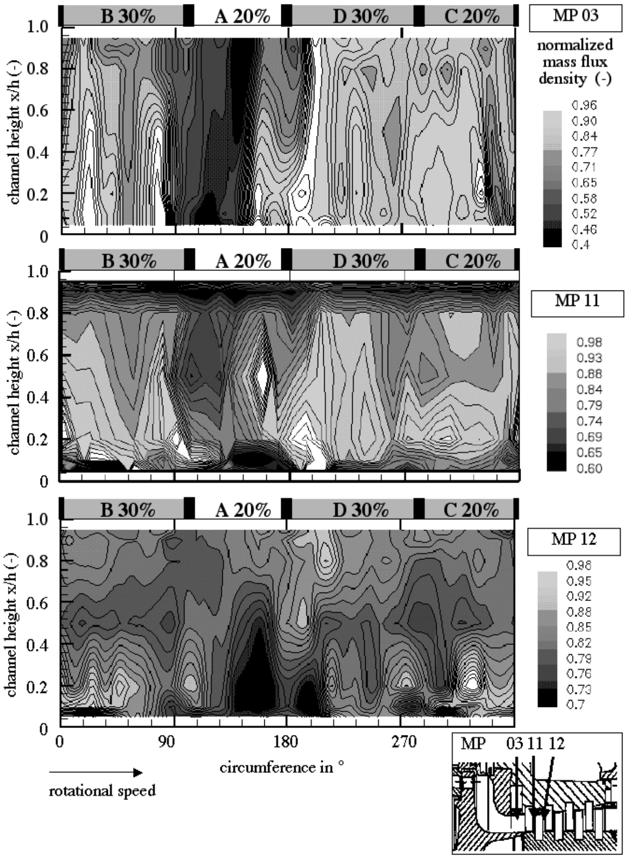


Fig. 12: Mass flux density ratios in the first stage (MP03-11-12) at partial admission

The increased mass flux at the sector separation A-D can be detected again across the entire channel height as well as the better-equalized region at sector separation B-A. In sector C the level of the mass flux density ratio is already about 80% of the mass flux density at full admission.

The normalized pitchwise-averaged mass flux density ratio behind the first guide vane shows a quite well equalized flow field with a minimum of 60% of the mass flux density at full admission in region of sector A. This minimum is shifted by approximately 20° in rotational direction. The increase in mass flux ratio at the sector separation A-D is slightly equalized. The radial mass flux from mid channel to hub region is increased compared to the inlet. Thus, most of the flow equalization takes place in front and inside the first guide vane.

Behind the first stage the flow inhomogeneity is further equalized. Only between mid channel and hub region in sector A a minimum in mass flux density of about 70% of full admission can be detected. The radial mass flux density ratio is redistributed with a minimum in mid channel region increasing towards the hub and casing due to the expansion inside the first rotor.

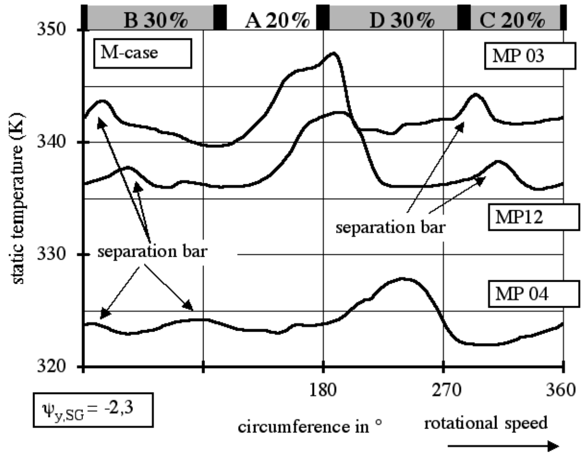


Fig. 13: **Static Temperature distribution at partial admission in the 4-stage turbine part (MP03-12-04)**

The mass flux inside the first stage is equalized starting from a variation of 40% to 100% at the inlet down to 70% to 95% behind the first stage compared to full admission.

Further details about the mixing process in the first stage can only be gathered by extensive numerical studies to judge this phenomenon in detail on an analytical basis.

Fig. 13 shows the spanwise averaged temperature distributions at 80% partial admission at the inlet (MP03) behind the first stage (MP12) and at the outlet of the 4-stage turbine part (MP04).

During the expansion inside the 4-stage turbine the temperature non-uniformity discussed in Fig. 4 moves circumferentially 90° in rotating direction and attenuates by changing its form and peakiness. The small temperature peaks created by the sector separation bars can even be detected behind the first stage (MP12) and as weak peak behind the multistage turbine part (MP04). The temperature streak is still present at the outlet of the 4-stage turbine smoothed out to a wake function-like shape. The influence of the non-symmetrical exhaust casing is visible as an additional non-uniformity in circumferential direction.

In opposite to the significant equalization of the inhomogeneous mass flow the non-uniform temperature distribution does not attenuate significantly inside the 4-stage turbine part.

CONCLUSIONS

In the present paper the development of a circumferential and radial mass flow and temperature inhomogeneity is analyzed by an experimental investigation.

In a scaled down turbine consisting of a control stage with cross-over channel and a 4-stage turbine a significant flow and temperature non-uniformity was created by 80% partial admission of the turbine. This flow phenomena influencing this inlet inhomogeneity are analyzed in detail.

The analysis of the flow equalization inside the 4-stage turbine part using static wall pressures at the casing shows that the flow inhomogeneity is basically equalized behind the third stage and moves mainly in axial direction. Most of the flow equalization takes place in the first turbine stage. This result is confirmed by extensive flow and temperature measurements and analysis of the complete 360° flow field in the first stage. A detailed flow analysis shows that the guide vane is the main driver of the equalization process.

In opposite to the complete equalization of the flow inhomogeneity the temperature non-uniformity attenuates only slightly and is shifted in circumferential direction.

A possible theory is that the equalization of the pressure/velocity non-uniformity is driven by impulse exchange effects, the equalization of the mass flux by changes in density and velocity. The mass particles do not change their position significantly and therefore no extensive mixing takes place on the circumference. The guide vanes and the rotor shift the particles in rotating direction. Therefore the static energy related to the mass particle with its temperature can only slightly be equalized by heat transfer. But heat transfer effects are not that strong due to the short dwell time of the particles during the expansion inside the turbine.

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