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Aims and Scope

Transactions of the Institute of Fluid-Flow Machinery have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

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Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100$^{\text{th}}$ Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery.* At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

Prof. Jarosław Mikielewicz
Editor-in-Chief

Dr Edward Śliwicki
Managing Editor

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PIOTR LAMPART*, MARIUSZ SZYMANIAK and ANNA KARDAŚ

Unsteady forces acting on rotor blades of a large power steam turbine control stage at different levels of partial admission

Institute of Fluid Flow Machinery of the Polish Academy of Sciences, Fiszera 14, 80-231 Gdańsk, Poland

Abstract

Partial admission flow in the control stage of a 200 MW steam turbine is investigated with the help of a RANS solver with the k–ω SST turbulence model in the code Fluent. A 2D model of flow at the mid-span section of the full-annulus is assumed. The results exhibit interesting details of the process of expansion in the control stage. Unsteady forces acting on the rotor blades of the control stage are calculated, and are subject to Fourier analysis.

Keywords: Partial admission turbines; Control stage; CFD; Blade forces

1 Introduction

Partial admission serves as a means of controlling the power output in large power turbines. Steam flow is admitted through a number of nozzle boxes located at the annulus in the form of discrete arcs, see Fig. 1, and each nozzle box is equipped with a control valve. Partial admission increases turbine efficiency, especially at low load operation. On the other hand, partial admission introduces strong circumferential non-symmetry of flow parameters in the control stage and therefore is a cause of additional unsteady loads of the rotor blades. Due to a rapid change of load while entering and leaving the arc of admission, the rotor blades and also blade-fit regions experience higher unsteady mechanical stresses and are more vulnerable to failure [1]. The rotor blades are susceptible to forced

*Corresponding author. E-mail address: lampart@imp.gda.pl

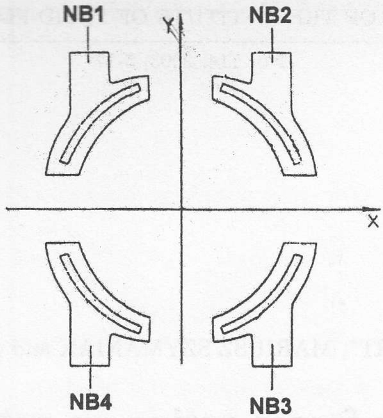

Figure 1. Schematic of partial admission for a control stage.

vibrations. The operation of the control stage gives also rise to dangerous low-frequency (low multiples of rotational frequency) excitations in the system of rotor shaft, bearings and supports.

Partial admission is applied not only in large power turbines. Partial admission stages are also designed in small turbines when at a full-annulus admission the blade height becomes too low and acts to largely increase the tip leakage and secondary flow losses, as well as when the blade height-to-diameter ratio and rotor speed become very low to meet the design value of velocity coefficient u/c_{0t} (where u – rotor speed, c_{0t} – velocity based on isentropic enthalpy drop of the stage). The arc of admission is then decreased, blade height is increased, also the blade height-to-diameter ratio is increased so that the performance (efficiency) of the turbine is improved.

This paper is a CFD study of flow in the control stage of a 200 MW steam turbine. To the best of the authors knowledge, previous CFD studies were made on small turbines with a low number of blades, or considering only a portion of the annulus [2-4]. Although the number of blade-to-blade passages is large in real large power turbines, here the full annulus of the control stage is computed in a 2D flow model. Flow details and unsteady forces acting on the rotor are discussed.

2 Flow conditions

The stator of the investigated control stage (impulse stage) is divided into four nozzle boxes located discretely (not symmetrically, not evenly spaced) at the stage circumference. The total number of guide vanes in all boxes is 45 (10, 14, 12 and 9), the hypothetical number of blade-to-blade passages covering the entire circumference would be 67. The number of rotor blades is 114. Two variants of operation of the investigated control stage are considered here, which refer to the turbine power output of 215 MW - nominal turbine conditions, and 140 MW - a certain low load at which, however, the power output of the control stage itself reaches a maximum value. In order to evaluate thermodynamic conditions for CFD computation of the control stage, a 0D/1D procedure was applied [5] that enables us to determine the distribution of flow into each nozzle box. From this procedure it is found that in the first variant, three valves are fully open and the fourth valve is partly open. In the second variant, only two valves are fully open, the third valve is slightly open and the fourth valve is closed. Thermodynamic conditions, including mass flow rates, pressures behind the control valves and in the control stage chamber, as well as inlet temperatures for the two considered variants of flow are presented in Tab. 1. The flow inlet is assumed normal to the nozzle box front.

Table 1. Two variants of flow conditions for operation of the control stage.

Variant	Turbine power [MW]	Flow rate [kg/s]	Nozzle box inlet pressure [bar]		Inlet temperature [°C]	Exit pressure [bar]
1	215	182	NB1	122	532	95.7
			NB2	103.1		
			NB3	122.3		
			NB4	122.1		
2	140	115.4	NB1	120.9	532	59.8
			NB2	Closed		
			NB3	121.5		
			NB4	60.1		

3 Flow model

The computations are carried out with the help of the code Fluent [6]. The control stage is computed in a 2D flow model at the mid-span section of the stator and rotor. The assumed flow model is based on the RANS approach, supplemented with the turbulence model $k - \omega$ SST. The segregated solver is used with a second order upwind scheme and SIMPLE algorithm for pressure-velocity coupling. An implicit scheme is used for time discretisation and the sliding mesh technique is used to treat the rotor motion. The time step is equal to 0.00003 s, which means that 667 steps are needed to cover one revolution of the rotor. Three to four revolutions are computed for each variant of flow.

The computational domain extends on all blade-to-blade passages of the control stage available for flow, that is on 45 stator passages grouped into four nozzle boxes and 114 rotor passages, plus narrow regions near blocked parts of the annulus. The computational grid is structural of the H-type, containing over 800 000 grid cells in total (each stator and rotor channel contains 4750 cells – 88×54). The computational grid is presented in Fig. 2.

4 Analysis of flow patterns

The pressure drop (p_{ex}/p_{inl}) for the admitted channels is equal to 0.78 for variant 1 and 0.49 for variant 2, and the exit Mach numbers for the admitted channels are 0.4 and 0.8, respectively. Therefore, flow patterns change considerably between the two variants. There are also significant differences in expansion between the blade-to-blade passages located in the arc of admission, outside it, and also at its borders. Even the neighbouring passages can experience considerable differences in flow patterns, which can be observed from Figs. 3 and 4 showing instantaneous contours of static pressure in the stator and rotor cascades for the two variants. The pressure downstream of the nozzle box tends to be the highest where the moving blade enters the arc of admission. Here, the expansion of the nozzle jet is hampered by stagnant fluid in the rotor passage. The high pressure here helps to remove stagnant fluid from the rotor and renew expansion in the rotor passage. The highest pressure, however, is observed downstream of the second, not of the first nozzle. This is due to some expansion of the first nozzle flow in circumferential direction along the wall of the blocked region opposite to the direction of rotation. The lowest value of pressure is downstream of the last nozzle where the moving blade leaves the arc of admission. This helps to retain the fluid in the rotor channels passing through the blocked region. Also further expansion of the nozzle jet in the circumferential direction (direction of rotation) along the wall of the blocked region can be observed.

In the first variant, the static pressure behind the mid-arc nozzles is typically equal to 103 bar, whereas behind the second nozzle rises to 107 bar and behind the last nozzle drops to 95 bar. There is a further pressure drop downstream of this last nozzle in the rotor blade passage and along the wall of the blocked region. The pressure in the front part of the rotor channel at the suction side exhibits a drop to 75 bar. The nozzle exit Mach number changes from 0.4 at the entry to the arc of admission to 0.5 in mid-arc nozzles to 0.6 at the last nozzle. Due to further expansion in the circumferential direction the Mach number near

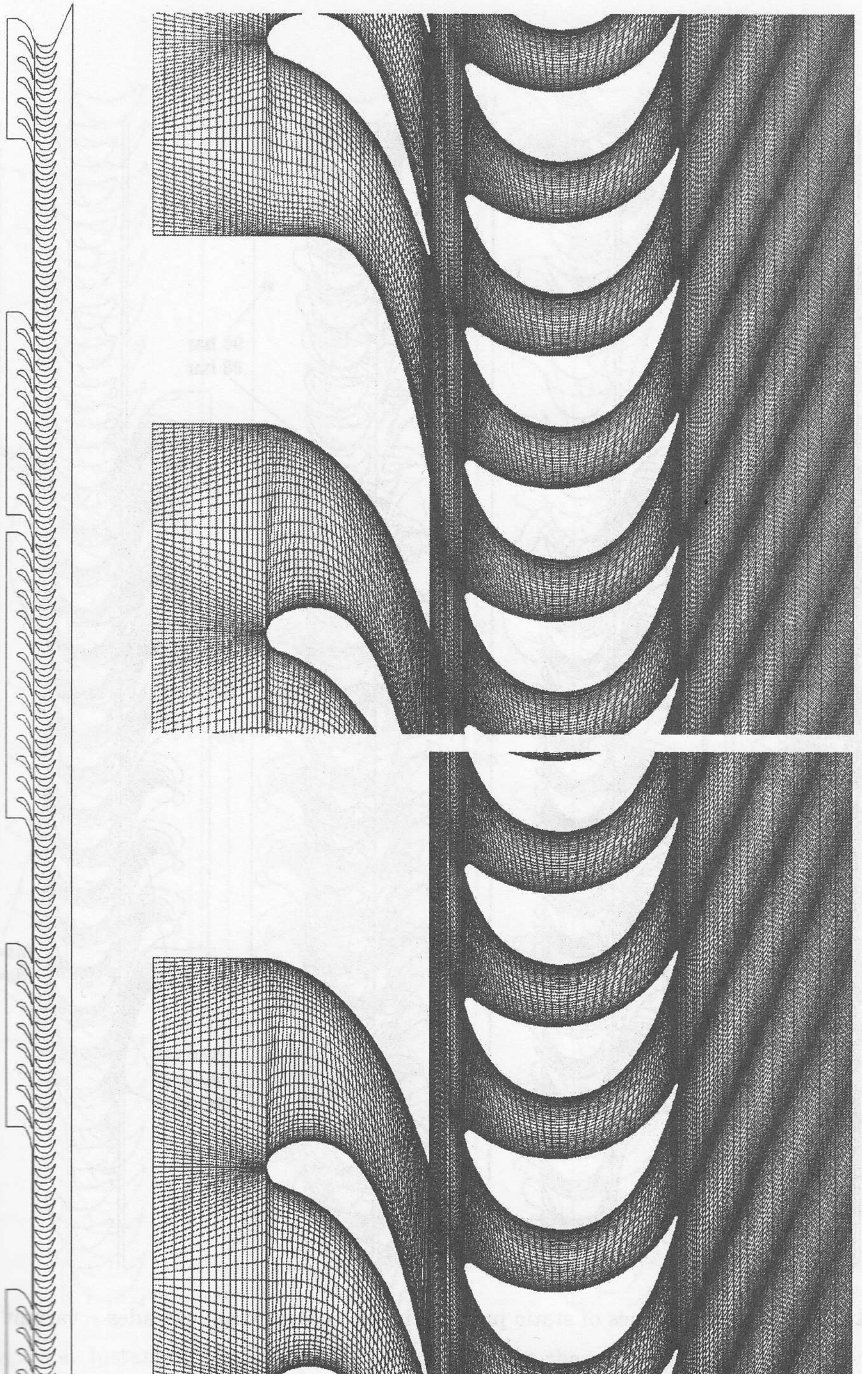

Figure 2. Computational domain (left) with grid details (right).

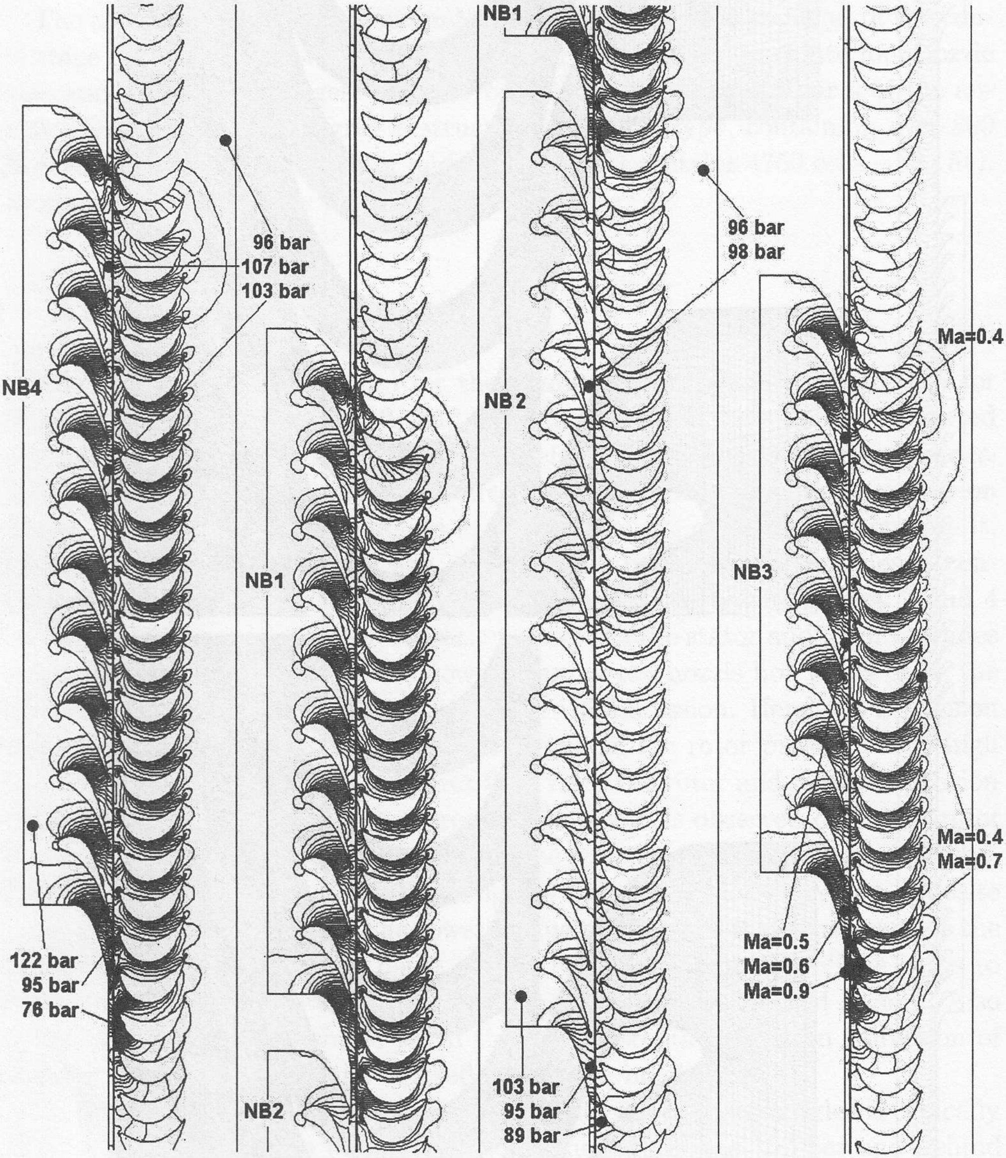

Figure 3. Instantaneous isolines of static pressure in the control stage cascades – variant 1.

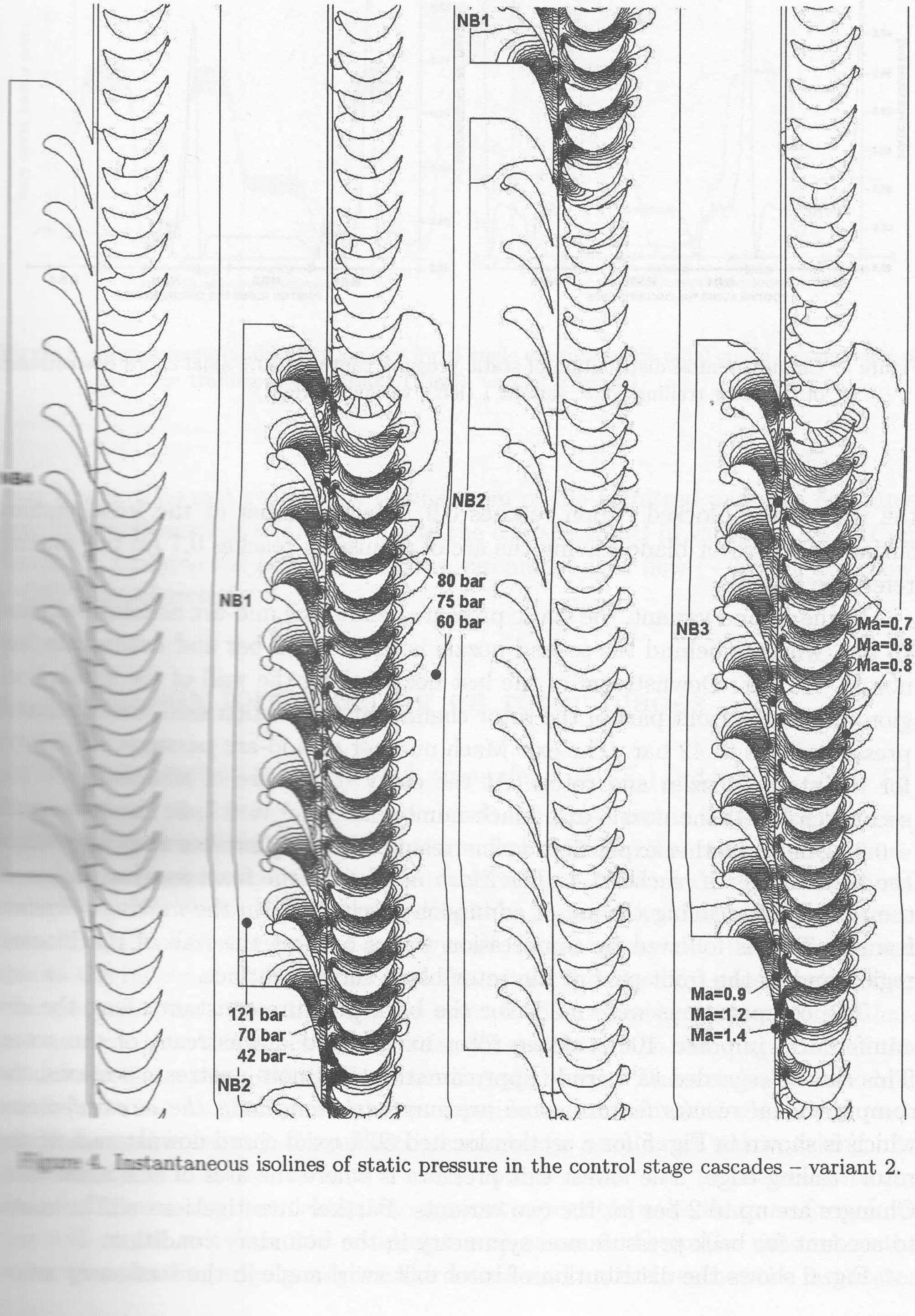

Figure 4. Instantaneous isolines of static pressure in the control stage cascades – variant 2.

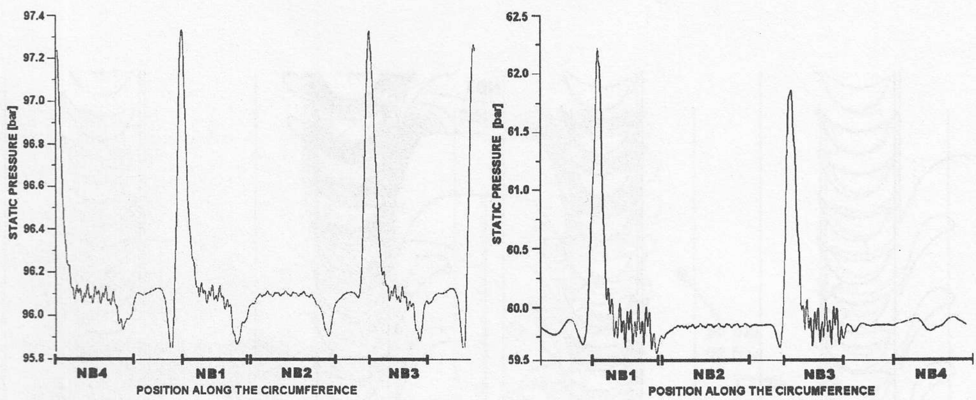

Figure 5. Circumferential distribution of static pressure captured 50% axial chord downstream of the rotor trailing edge: variant 1 (left), variant 2 (right).

the wall of the blocked region reaches 0.9. Mach number at the front suction surface of the rotor blade leaving the arc of admission reaches 0.7 (in the moving reference frame).

In the second variant, the static pressure behind the mid-arc nozzles is around 75 bar, whereas behind the second nozzle is equal to 80 bar and behind the last nozzle - 70 bar. Downstream of the last nozzle along the wall of the blocked region and at the front part of the rotor channel at the suction side, the calculated pressure drops to 42 bar. The exit Mach number at mid-arc passages is 0.8 both for the stator nozzles and rotor. At the entry to the arc of admission (in the second channel) the nozzle exit Mach number is equal to 0.7, in the last nozzle – 0.9. Due to further expansion downstream, the Mach number near the wall of the blocked region reaches 1.4. The Mach number at the front suction surface of the rotor blade leaving the arc of admission reaches 1.2 (in the moving reference frame). This is followed by compression waves both at the wall of the blocked region and at the front part of the rotor blade suction surface.

The computations were made for the back pressure constant along the circumference, imposed 100% of the rotor axial chord downstream of the rotor. This can be regarded as a crude approximation. At more upstream sections, the computational results feature some pressure variation along the circumference, which is shown in Fig. 5 for a section located 50% axial chord downstream of the rotor trailing edge. The lowest exit pressure is where the arcs of admission end. Changes are up to 2 bar for the two variants. Further investigations will be made to account for back pressure non-symmetry in the boundary condition.

Fig. 6 shows the distribution of rotor exit swirl angle in the stationary refer-

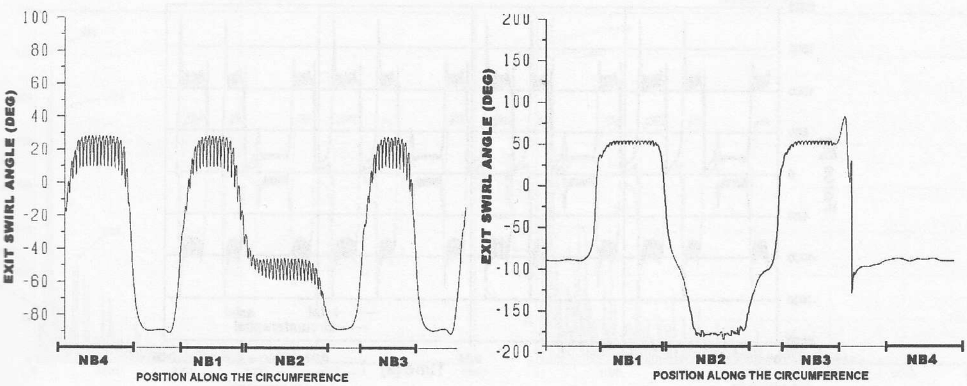

Figure 6. Circumferential distribution exit swirl angle captured 50% axial chord downstream of the rotor trailing edge: variant 1 (left), variant 2 (right).

ence frame. The exit swirl angle downstream of the admitted passages oscillates near 20° (with respect to the normal to the cascade front) in variant 1 and 50° in variant 2. Outside the admitted regions, circumferential flow (−90°) or backflow (−180°) are observed.

5 Unsteady forces acting on rotor blades

Calculated transients of axial (F_x), circumferential (F_y) and total $\left(\sqrt{F_x^2 + F_y^2}\right)$ force acting at the control stage rotor blade are presented in Fig. 7 for the two investigated variants of partial admission flow. When the rotor blade passes through the blocked regions, the forces tend to zero. Well inside the admitted regions, the rotor blades experience loading that is similar to that of full admission flow. "Small" oscillations of the aerodynamic forces observed in these regions are due to the inflow of stator wakes onto the rotor blade leading edges and further transport of these wakes through the rotor blade-to-blade passages. The calculations show that this effect accounts for 3% oscillations in the magnitude of the axial force and up to 10% of the circumferential force. The number of observed small peaks of a similar amplitude corresponds to the number of wakes from each nozzle box. It is exactly the number of nozzles (= number of wakes, taking into account a wake that forms from a sharp corner of the wall at the entry to the arc of admission, but not taking into account a possible wake that can form from the wall at the exit from the admission arc) minus 3. From the time when the rotor blade leading edge meets the third wake and until it reaches the last stator

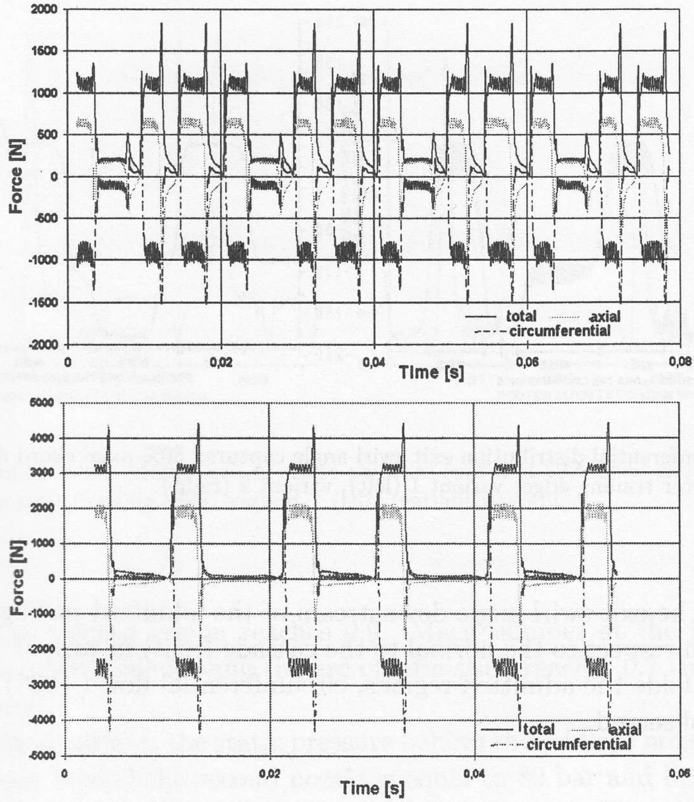

Figure 7. Time variation of axial, circumferential and total force acting on the control stage rotor blade – variant 1 (top), variant 2 (bottom).

blade wake, changes in rotor blade load seem to be determined only by the stator/rotor interaction. In other words, which can also be checked in Figs. 3 and 4, the blade-to-blade passages from the third to the last but one nozzle and the rotor passages downstream experience similar flow conditions and flow patterns, and seem to be only little affected by the borders of the admission region.

The largest peaks in forces acting at the rotor blades are at the entry and exit from the arc of admission, see Fig. 7. The peaks of the total force at the exit from the admission region amount to 50% (variant 1) and 35% (variant 2) of the force value in the arc of admission. The peak amplitudes can even be greater for the force components – the axial and circumferential force, because local changes in pressure at the rotor blades in the cycle of admission, not only change the value of the blade force but also its direction. The presence of these peaks can easily be explained for the circumferential force. This component will

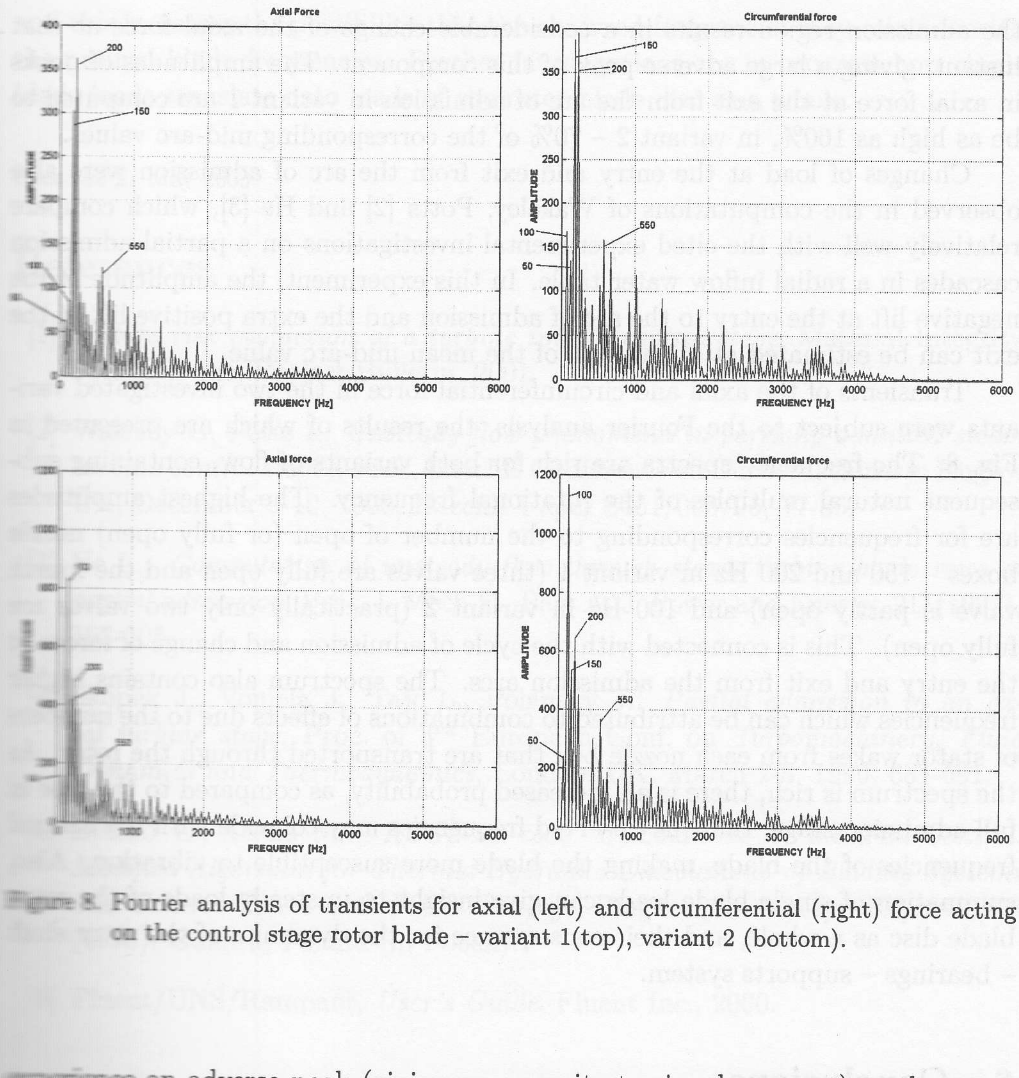

Figure 8. Fourier analysis of transients for axial (left) and circumferential (right) force acting on the control stage rotor blade – variant 1(top), variant 2 (bottom).

experience an adverse peak (giving an opposite torsional moment) at the entry to the arc of admission when the expansion in the passage at the "suction side" of the entering rotor blade is already renewed, whereas at the "pressure side" the fluid still remains stagnant. Conversely, at the exit from the arc of admission the fluid will assume stagnant properties at the "suction side" of the leaving rotor blade, whereas at the "pressure surface" the fluid still expands. This will result in an increased load of the rotor blade in circumferential direction, giving an extra torsional moment. The amplitudes of peaks in circumferential force at the entry to the arc of admission in variant 1 are about 120%, in variant 2 – 70% of the corresponding mid-arc values, at the exit – 70% both in variant 1 and 2. The axial force does not exhibit a peak at the entry to the arc of admission. However, large drops of pressure at the front suction side of the rotor blade leaving the admission region results in a considerable change of the axial force at that instant, giving a large adverse peak of this component. The amplitudes of peaks in axial force at the exit from the arc of admission in variant 1 are computed to be as high as 160%, in variant 2 – 70% of the corresponding mid-arc values.

Changes of load at the entry and exit from the arc of admission were also observed in the computations of Wakeley, Potts [2] and He [3], which compare relatively well with the cited experimental investigations on a partial admission cascades in a radial inflow water table. In this experiment, the amplitude of the negative lift at the entry to the arc of admission and the extra positive lift at the exit can be estimated at about 80% of the mean mid-arc value.

Transients of the axial and circumferential force in the two investigated variants were subject to the Fourier analysis, the results of which are presented in Fig. 8. The frequency spectra are rich for both variants of flow, containing subsequent natural multiples of the rotational frequency. The highest amplitudes are for frequencies corresponding to the number of open (or fully open) nozzle boxes – 150 and 200 Hz in variant 1 (three valves are fully open and the fourth valve is partly open) and 100 Hz in variant 2 (practically only two valves are fully open). This is connected with the cycle of admission and change of forces at the entry and exit from the admission arcs. The spectrum also contains higher frequencies which can be attributed to combinations of effects due to the numbers of stator wakes from each nozzle box that are transported through the rotor. As the spectrum is rich, there is an increased probability, as compared to the case of full admission stage, that the observed frequencies may coincide with the natural frequencies of the blade, making the blade more susceptible to vibration. Also, summation of single blade loads can give insight to unsteady loads of the rotor blade disc as a whole, and their consequence for the dynamics of the rotor shaft – bearings – supports system.

6 Conclusions

CFD analysis of flow in the partial admission control stage of a 200 MW turbine calculated in a full-annulus 2D flow model at the mid-span section of the stator and rotor shows circumferential non-uniformities of the flow fields and exhibits the unsteady character of forces at the rotor blades. The largest changes in forces are when the rotor blade enters or leaves the arc of admission. Well inside the admission region, the oscillations of forces are due to the transport of stator wakes through the rotor. The frequency spectrum of the Fourier transform from force oscillations is rich and contains subsequent natural multiples of the rotational frequency. The largest amplitudes correspond to the number of open nozzle boxes, however, the number of frequencies present in the spectrum is large and this increases the possibility that the observed frequencies may coincide with the natural blade frequencies. Further investigations are in progress how single blade loads translate onto loads of the rotor blade disc as a whole.

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