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INVESTIGATION OF LOSSES AT THE INLET TO THE SHROUDED BLADE ROTOR

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Abstract

Numerical investigations of the flow at the entrance to the labyrinth sealing over the shroud of the axial turbine rotor blade were carried out. An influence of the gap width on losses in the rotor as well as the flow structure are described and discussed for four values of the labyrinth clearances which correspond to the mass flow rates of leakages from 0.5 to 2% of the main flow. For narrow gaps, the fluid enters the labyrinth cavity only, however, for wider gaps, reverse flow regions appear. The fluid return to the main channel at the rotor inlet disrupts the flow structure in the blade passage and intensifies the loss generation. The amount of the reverse flow increases with gap widths. The border value of the gap width which assures the one-direction flow inside the labyrinth cavity and a reduction of the loss level was determined for the turbine under study.

Key words: axial turbine, shrouded blades, labyrinth sealing, leakages, secondary flows

INTRODUCTION

The main flow in turbomachinery is accompanied with leakages through the necessary structural gaps between rotational and stationary elements. The most obvious effect of leakages is a reduction of the mass flow rate through the blade passage and, in consequence, a decrease of the energy transfer between the fluid and blades. One of leakages, which is the object of this study, is the leakage over shrouded rotor blades of the axial turbine. In this case, the fluid is removed from the main flow path in front of rotor blades and directed to the channel over the shroud. It undergoes numerous changes in the flow direction, contraction and expansion processes in labyrinth sealing elements and, finally, returns to the main passage downstream of the rotor blades. All these phenomena are responsible for fluid energy dissipation in a turbine stage as well as in subsequent stages to some extent. An appropriate design of the labyrinth sealing assures the minimization of the leakage mass flow rate and leads to the minimization of energy dissipation.

Mixing of the leakage with the main flow at the rotor outlet is widely recognized as the most serious dissipative process due to a significant difference in velocity values and flow directions of these two streams (Denton, 1993). During the recent years, great efforts have been devoted to experimental investigations of the mixing zone downstream of the rotor blade and methods of a loss reduction in this region (Pfau et al., 2001; Pfau et al., 2007; Rosic and Denton, 2008).

Dissipative processes at the entrance to the labyrinth sealing are not so intensive (Denton, 1993) and probably therefore, investigations of the flow at the entrance to the labyrinth sealing over the rotor shroud are limited (Pfau et al., 2007). What is obvious, uniform removal of a part of the boundary layer in front of the rotor can lead to improvement of the flow structure in the blade passage and reduce the secondary flow intensity (Gundlach, 2008). Unfortunately, axial distances between stators and rotors are limited and the labyrinth channel entrance is located just downstream of the stator blade trailing edges and just upstream of the rotor blade leading edges. Thus, the forward influence of the stator and the backward influence of the rotor blades are very important.

The pressure and velocity distributions change in this clearance along the circumferential direction. In the case of a wide clearance, in front of the leading edge and at the pressure side of the blade, the fluid enters the gap. However, in front of the blade suction side, a part of this flow returns to the main passage (Pfau et al., 2007). This reverse flow changes inflow conditions of the rotor and has a significant influence on the loss generation in its blade passages by means of the secondary flow intensification. It modifies the development of both legs of the horseshoe vortex but due to the fact that its rotation direction is the same as the passage vortex this structure is strengthened (Pfau et al., 2007).

The flow behavior in the gap depends on its width (Sobczak et al., 2011; Sobczak, 2011). The amount of the reverse flow decreases with the gap width. For wide gaps, reverse flow regions which spread over the whole gap width appear, whereas for narrow gaps, the fluid enters the labyrinth cavity only. In such a case, even though the boundary layer suction is not perfectly uniform the secondary flow reduction can be observed. Thus, it can reduce a negative effect of the shroud flow to some degree (Gundlach, 2008).

An impact of the gap width at the entrance to the labyrinth sealing on the axial turbine rotor performance is presented in this paper. A series of numerical simulations were carried out for a wide range of gap widths and for various labyrinth sealing clearances which resulted in different leakage mass flow rates.

An increase in the computational power and progress in Computational Fluid Dynamics (CFD) methods allows one to investigate flow structures in turbomachinery with higher and higher precision. The main flow path with complex secondary flow structures can be properly predicted (Langston, 2001; Kumar and Govardhana, 2010). However, due to complex and highly dissipative character of labyrinth flows, CFD simulations of leakages are more challenging. Therefore, in turbomachinery investigations, leakages are processed numerically with various level of precision. A few years ago the main flow and the side flow were considered separately. The leakage mass flow rate and parameter changes were usually determined on the basis of empirical data or a simplified numerical method. An influence of the side flow on the main flow in turbine simulations were taken into account by means of removal and delivery (sink-source) of the appropriate mass flow of the leakage (Chodkiewicz et al., 2005; Hunter and Orkwis, 2000; Lampart, 2006). Unfortunately, this approach neglects the parameter variation at the entrance and the exit from the labyrinth sealing and decreases the approximation quality. Detailed numerical investigations of the loss mechanism requires solutions of the full three-dimensional object, i.e., the main flow channel and the side flow gaps. In such a case, they can adequately represent the reality (Rosic et al., 2006; Rosic and Denton 2008). Therefore, in the current study such a procedure is applied.

NUMERICAL PROCEDURE

The simulations presented in this paper were carried out for a computational model of the two-stage axial turbine with highly loaded rotor blades installed in the Institute of Turbomachinery, Lodz University of Technology. Stators consist of 22 and rotors of 33 prismatic blades. In the paper, the flow in the first rotor is analyzed in detail. Its pitch to chord ratio is equal to 0.75, the blade height to the diameter – 0.188, the aspect ratio – 1.48, and the rotor blade stagger angle – 140°. The Mach number reaches maximally 0.35 and the Reynolds number is $2.2 \cdot 10^5$. A more detailed description of the turbine can be found in (Krysinski et al., 2011; Smolny et al., 2011).

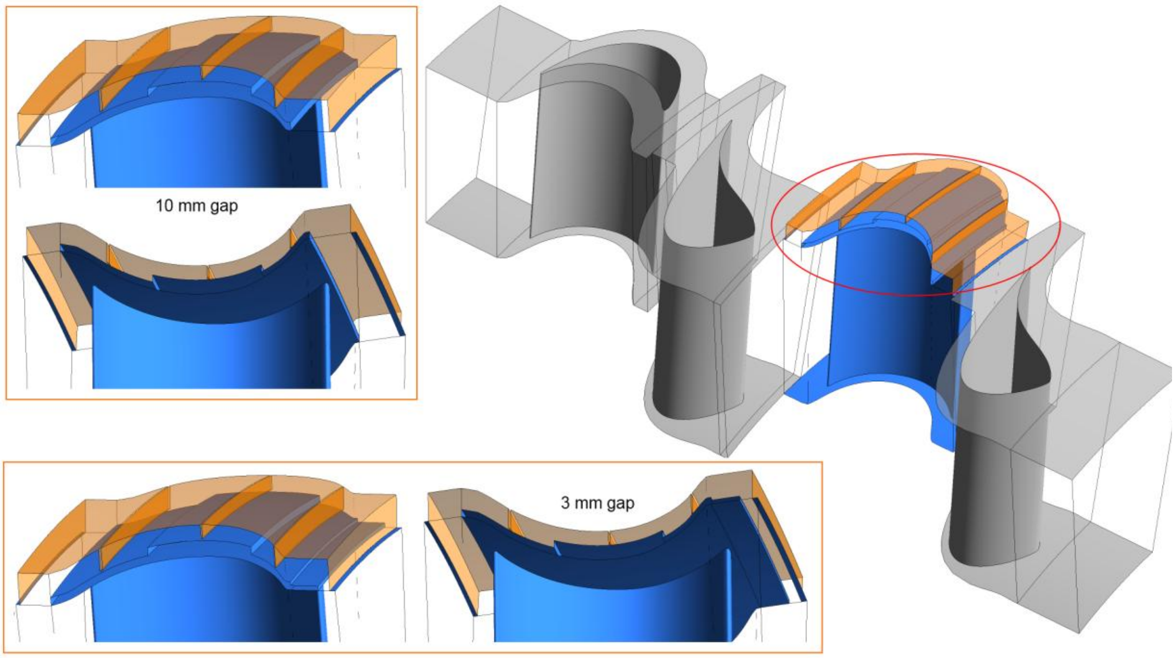

Fig. 1. Two-stage turbine with the first rotor shrouded blade and the sealing for two entrance gap width configurations

For this numerical study, the original design of the model turbine with gaps at the rotor blade tip was modified by an introduction of shrouds and a labyrinth sealing for the first rotor. A general view of the task configuration with two different widths of the gap at the entrance to the labyrinth sealing is shown in Fig. 1. Detailed dimensions of the shroud region are presented in Fig. 2. Due to a significant ratio of the labyrinth length to the height, the plot is divided along the axial direction. Blue arrows indicate the place of division. Red lines represent the position of the leading and trailing edge of the rotor blade. The investigations were carried out for gap widths at the labyrinth sealing entrance from 1 to 10 mm. Four values of the labyrinth sealing clearances were studied: 0.25, 0.5, 0.75 and 1.0 mm.

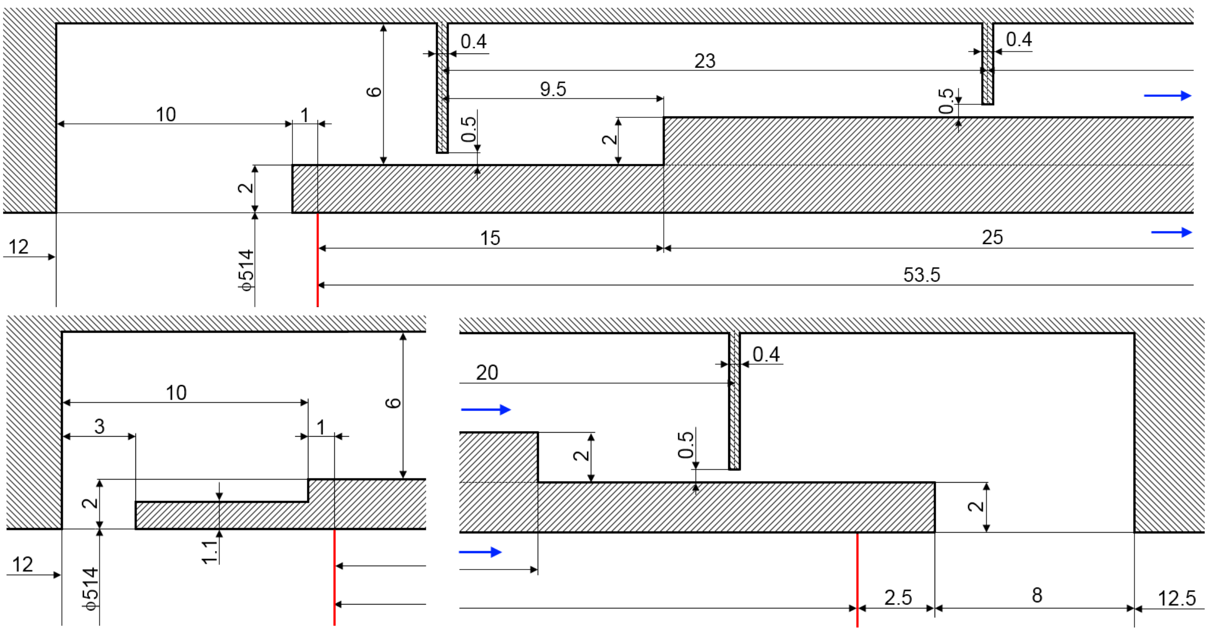

Fig. 2. Dimensions of the labyrinth sealing for the first rotor for 10 mm and 3 mm entrance gap widths for 0.5 mm labyrinth sealing clearances

Computational meshes for stators and rotors were generated by means of the ANSYS TurgoGrid software, which provides easy generation of high quality hexahedral meshes for blade passages in turbomachinery. However, for the shrouded rotor, this software did not guarantee a high mesh quality in the regions of the labyrinth entrance and outlet. Thus, in this configuration, a hexahedral mesh of the first rotor was generated with ANSYS ICEM CFD. As far as the solver requirements are concerned, good qualities of meshes were obtained in all cases. Approximately 1.0, 3.5, 0.7 and 0.6 million nodes were used for the first stator, the first rotor, the second stator and the second rotor, respectively. ANSYS ICEM CFD was also used for meshes generation in the side flow channels over the rotor bandage (a separate mesh for each clearance height and entrance gap width). In these cases, the meshes were composed of approximately 2.0 million nodes. Significant refinements of the mesh in the hub, blade and, especially, shroud regions were done to reveal properly the secondary and side flow fields in those regions. The maximal value of the y^+ parameter of the first mesh layer from the wall is equal to 16 and its averaged value in the whole machine is below 3 (below 2 in the shroud region of the first rotor). This mesh configuration is sufficient to solve fully the boundary layer in the shroud regions and guarantees a proper application of the wall function in other zones where strong flow separations are not present.

The ANSYS CFX code was used in the simulations. The second order space discretization was employed. Three-dimensional, compressible flows of air in single passages of the turbine wheels were considered. Due to limitations of the available computers, steady state simulations were conducted with Stage (circumferential averaging) interfaces between the stationary and rotational wheels. Despite its limitation, this approach is widely applied in numerical simulations of turbomachinery (Gier et al., 2002; Rosic and Denton, 2008). In the case of the turbine under study, due to a significant distance between the stator blade trailing edge and the labyrinth entrance gap (12 mm), the influence of the circumferential pressure variation downstream of the stator is limited.

The total pressure equal to 130 kPa as well as the static temperature of 325 K and low turbulence intensity (Tu = 1%) were imposed as the boundary conditions at the turbine inlet. The averaged static pressure equal to 99.7 kPa was applied at the outlet. Channel walls were smooth and adiabatic. The rotational speed of the turbine rotor was equal to 3300 rpm. The Shear Stress Transport turbulence model was applied with an automatic wall function in the regions of insufficient refinement of the mesh. An adequate level of convergence was obtained in all the cases under analysis. Root Mean Square (RMS) residuals did not exceed 5.0e-5. Mass, momentum and energy flows were properly balanced in the computational domain (well below 0.01% of the mean flows).

SIMULATION RESULTS

Numerical investigations of an influence of the gap width at the labyrinth sealing entrance on the first rotor performance were carried out for different values of the labyrinth sealing clearances.

The relative mass flow rate of leakages: referred to the mass flow rate in the main passage – m_L/m and referred to the mass flow rate of leakages for the labyrinth sealing entrance gap $h = 10$ mm – $m_L/m_{L=10}$ are presented in Fig. 3. The mass flow rate for the labyrinth sealing gap heights of 0.25 mm corresponds approximately to the leakage of 0.5%, 0.5 mm to 1%, 0.75 to 1.5% and 1.0 mm to 2%, respectively. For the sake of clarity, in the further descriptions instead of the labyrinth sealing clearance heights, the relative mass flow rate will be used.

Some reductions of the leakage mass flow rate are observed for narrow entrance gaps (up to 4 mm) due to the throttling effect. What is obvious, this effect is more significant for higher leakages (1.5 and 2%). For higher entrance gap widths (above 4 mm), this effect disappears and leakage changes are negligible.

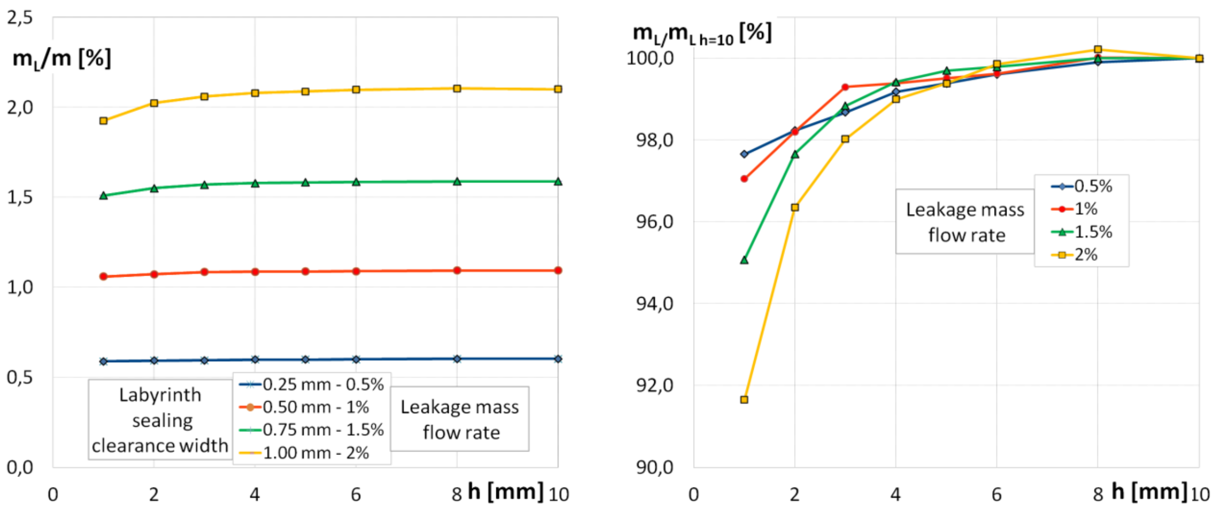

Fig. 3. Relative values of the mass flow rate in the labyrinth channel over the rotor bandage

The entropy loss coefficient ζ_S is defined according to (Denton, 1993):

$$\zeta_S = \frac{T(s - s_1)}{(h_{t2} - h_2)}$$

where the entropy loss coefficient of the whole first rotor domain $\zeta_{S\,1\text{-}2}$ and the entropy loss coefficient between the rotor inlet plane and the plane downstream of the rotor trailing edge but before the mixing zone (leakage returns to the main passage) $\zeta^*_{S\,1\text{-}2}$ used in the further analysis are defined as follows:

$$\zeta_{S\,1\text{-}2} = \frac{T_2(s_2 - s_1)}{(h_{t2} - h_2)}; \qquad \zeta^*_{S\,1\text{-}2} = \frac{T_2^*(s_2^* - s_1)}{(h_{t2} - h_2)}$$

where T, s, h_t and h are mass-averaged temperatures, entropies, total enthalpies (in the stationary frame of reference) and enthalpies at the rotor inlet (subscript 1), outlet (2) and at the plane downstream of the rotor trailing edge but upstream of the mixing zone (2*), respectively.

The data obtained from the numerical investigations of the turbine first rotor with different labyrinth sealing configurations were compared to the case of the ideal first rotor channel, i.e., without a leakage. The relative entropy loss coefficients of the first rotor (referred to the loss coefficients for geometry without leakages – subscript 0):

$$\Delta\zeta_{S\,1\text{-}2} = \frac{\zeta_{S\,1\text{-}2} - \zeta_{S0\,1\text{-}2}}{\zeta_{S0\,1\text{-}2}}; \qquad \Delta\zeta^*_{S\,1\text{-}2} = \frac{\zeta^*_{S\,1\text{-}2} - \zeta^*_{S0\,1\text{-}2}}{\zeta^*_{S0\,1\text{-}2}}$$

are presented in Fig. 4 for different mass flow rates of leakages (heights of the labyrinth sealing clearances) and different widths of the entrance gap.

The entropy loss coefficient determined for the whole rotor domain $\Delta\zeta_{S\,1\text{-}2}$ depends strongly on the mass flow rate of the leakage. The energy dissipation in the mixing zone downstream of the rotor trailing edge increases for higher leakages. The loss coefficient depends also on the widths of the entrance gaps. In all the cases under analysis, the relation is non-linear. Generally, the loss growth rate as a function of the gap width is higher for narrow gaps rather than for gaps above 6 mm. However, the relation is more complex for the leakages of 1 and 1.5%. Small changes of the entropy loss coefficient $\Delta\zeta_{S\,1\text{-}2}$ are observed up to the gap width of 3 mm (approximately 5% of the rotor blade chord). A rapid growth of the coefficient takes place for a gap of 4 mm. A further increase of the gap leads to a steady but much smaller increase in losses. A similar relation can be distinguished for a leakage of 2%, but in a much lower scale.

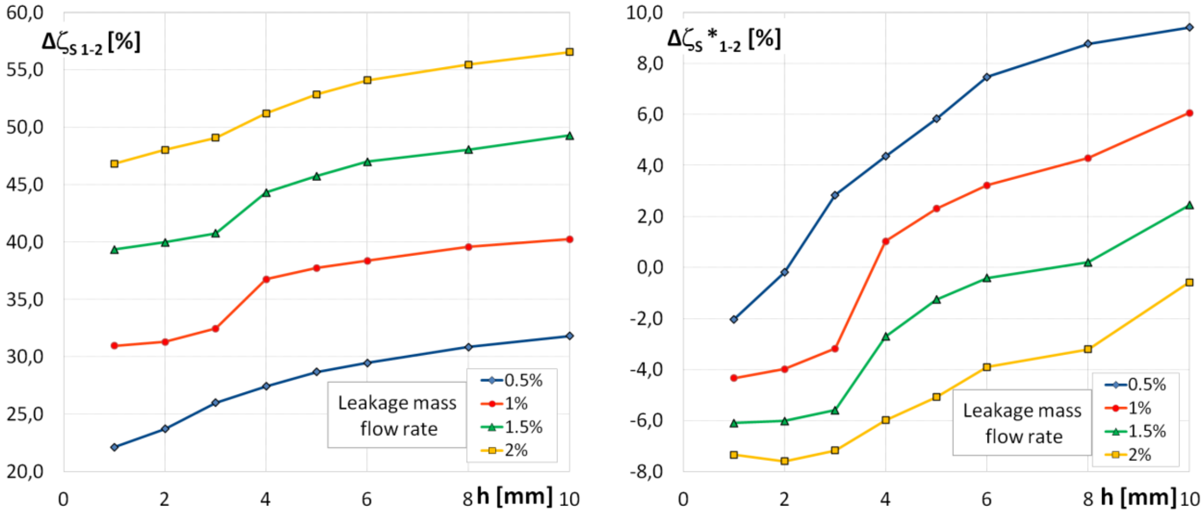

Fig. 4. Loss coefficients in the rotor domain $\Delta\zeta_{S\ 1\text{-}2}$ and between the rotor inlet and the blade trailing edge $\Delta\zeta_{S}{}^{*}{}_{1\text{-}2}$ for different widths of the entrance gaps and different leakages

A similar distribution is observed for the entropy loss coefficient determined for conditions between the rotor inlet and the blade trailing edge $\Delta\zeta_{S}{}^{*}{}_{1\text{-}2}$, i.e., in the case when the mixing losses are not taken into account. Negative values of this parameter indicate that the losses in the blade passage are reduced in comparison to the reference case (simulation without leakages). It takes place due to removal of a part of the boundary layer with the lowest kinetic energy, which decreases the intensity of the secondary flows generated in the rotor. A reduction of losses below the reference case level (below 0) is observed within the whole entrance gap range for 2% of the leakage and for narrow gaps for other ones. If the loss coefficients for the narrowest and the widest gap are compared, one can see that the dissipation growth rate decreases with the mass flow rates of leakages. Once again, a rapid growth of the coefficient takes place between gaps widths of 3 and 4 mm for the leakages of 1 and 1.5%.

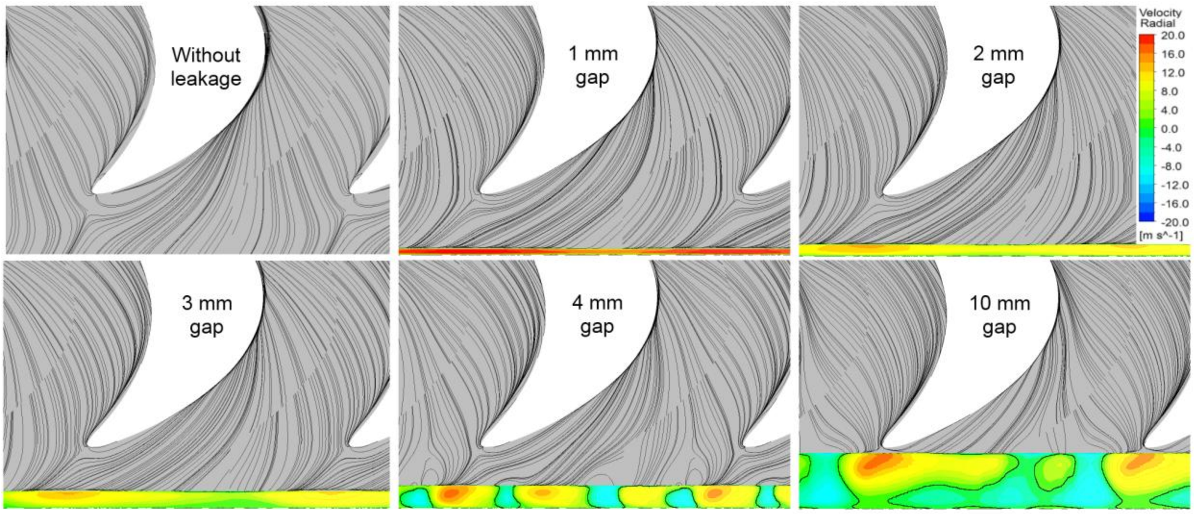

Fig. 5. Distributions of the radial velocity component in the entrance gap and streamlines in the shroud region for 1% of the leakage mass flow rate

Radial velocity distributions at the inlet to the gap presented in Fig. 5 for 1% of the leakage mass flow rate reveal the reason of these differences. Up to the gap width of 3 mm, despite the fact that some non-uniformities are observed, the fluid flows only in one direction, inside the labyrinth channel. A pressure distribution in the circumferential direction intensifies the inflow to the labyrinth sealing (higher values of the radial velocity component) in front of the leading edge of the rotor blade and reduces in front of the blade suction side. Up to the gap width of 3 mm, despite the fact that these non-uniformities are observed, the fluid flows only in one direction, inside the labyrinth channel. Above this gap width, the flow character changes. Zones of the reverse flow (separated by black lines) appear in front of the blade suction side as it was observed by (Pfau et al., 2007) and they extend across the whole gap. Their size increases with a further growth in the gap width. The fluid return to the main channel disrupts the flow structure in the blade passage, which can be seen on the streamlines distribution downstream of the reversal zones.

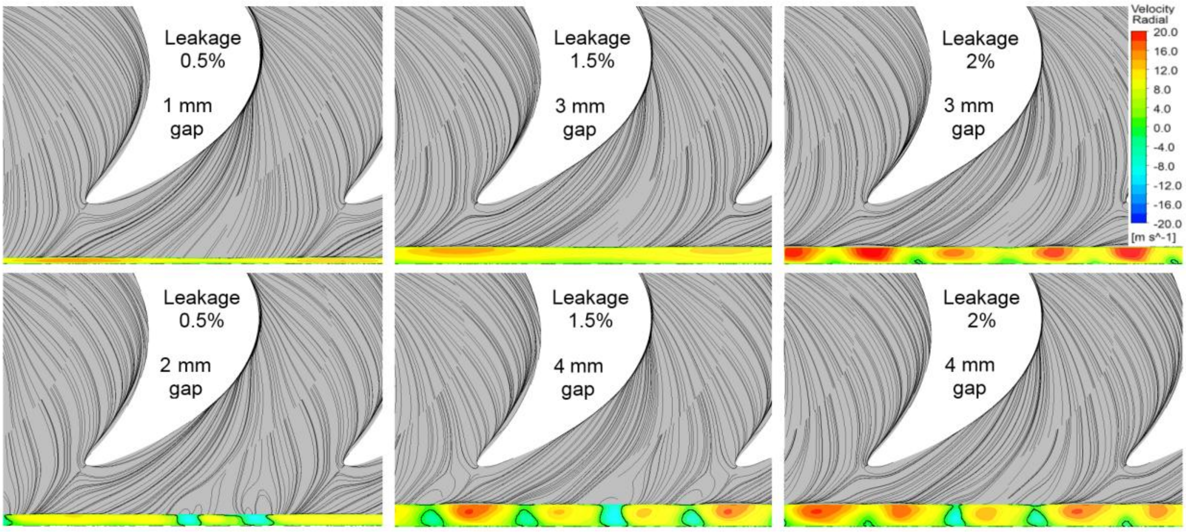

Fig. 6. Distributions of the radial velocity component in the entrance gap and streamlines in the shroud region for 0.5, 1.5 and 2% of the leakage mass flow rate

In Fig. 6 radial velocity distributions are presented for the remaining mass flow rates of leakages. Two gaps widths are chosen: the maximal one for which one direction flow is observed and a 1 mm wider one, where reversal zones appear. Excluding the case of the lowest leakage, the flow character transition takes place for gap widths between 3 and 4 mm. A similar number of the reverse flow zones are observed but their size depends on the leakage intensity.

The boundary layer at the shroud, 0.5 mm downstream of the blade leading edge, is observed for the case without leakages as the region of a high value of the entropy loss coefficient ζ_S (Fig. 7). For 1% of the leakage mass flow rate and for gaps up to 3 mm, it is almost completely removed. Some small residuals observed at the pressure side of the channel grow in size with an increase of the gap width. When the flow returns from the labyrinth entrance cavity to the blade passage (for gaps higher than 3 mm), a mixing process takes place, which is a source of intensive dissipation. It modifies the flow structure in the main channel and results in modification of the secondary flow field. Significant differences in the dissipation level are observed at the rotor trailing edge (before the leakage return to the main channel). The removal of the boundary layer diminishes the size of the passage vortex for gaps up to 3 mm. Mixing at the rotor inlet intensifies the secondary flow structures as can be observed for gaps of 4 and 10 mm.

In Fig. 8 loss coefficient ζ_S distributions are presented for the remaining mass flow rates of leakages. For 0.5% of leakages (1 gap width), not whole boundary layer upstream of the rotor is sucked and the region of a high value of the entropy loss coefficient remains. For 1.5 and 2% of mass flow rates (3 mm gap widths), small regions of losses are observed at the pressure side of the channel. The flow returns from the labyrinth entrance cavity to the blade passage for higher gaps and a mixing process takes place. However, for 2% of the leakage, the mass flow rate it is much less intensive than for lower leakages.

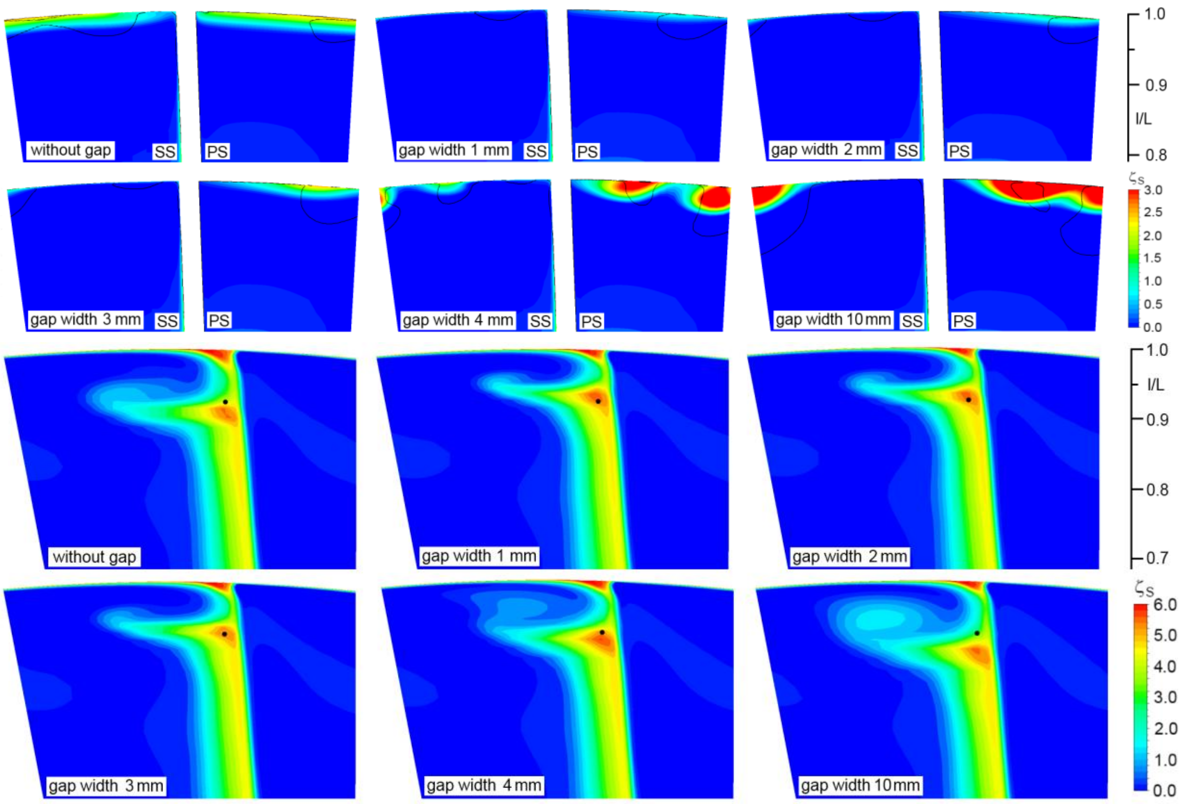

Fig. 7. Loss coefficient distribution in the shroud region at the blade leading (top rows) and trailing edge (bottom rows) cross-sections for 1% of the leakage mass flow rate; image order as in Fig. 5, black dots represent the same location

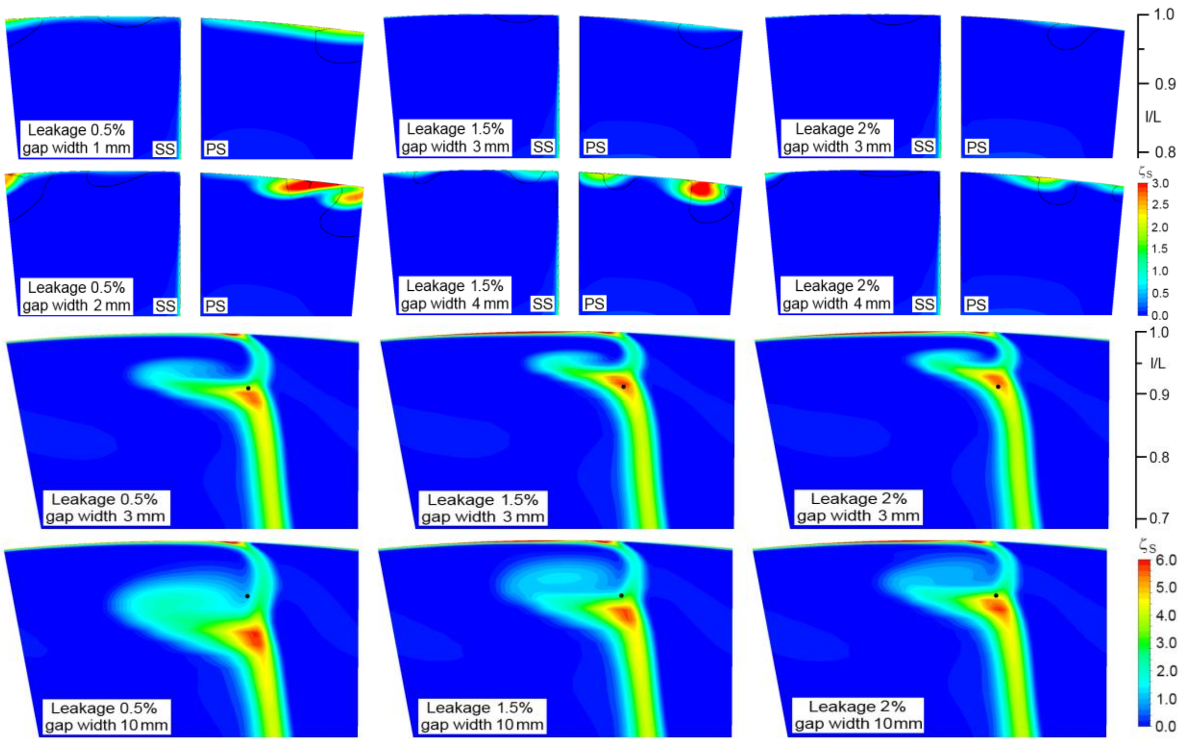

Fig. 8. Loss coefficient distribution in the shroud region at the blade leading (top rows) and trailing edge (bottom rows) cross-sections for 0.5, 1.5 and 2% of the leakage mass flow rate; imagee order as in Fig. 6, black dots represent the same location

At the rotor trailing edge, the loss coefficients for all other leakages are shown in Fig. 8 for 3 and 10 mm gap widths. Higher mixing at the rotor inlet, observed for smaller leakages, intensifies the passage vortexes. They grow in size and propagate deeper in the radial direction. At the same time, more intensive secondary flows cause that the dissipation rate in the boundary layer developed at the shroud, at the suction side of the channel, is reduced.

CONCLUSIONS

The numerical investigations of an influence of the gap width at the entrance to the labyrinth sealing over the shroud of the axial turbine rotor blade are presented. A wide range of gap widths (from 1 to 10 mm) was analyzed for four values of the labyrinth clearances which correspond to the mass flow rates of leakages from 0.5 to 2%. Some reductions of the leakage mass flow rate are observed for narrow entrance gaps (up to 4 mm) due to the throttling effect. This effect is negligible for higher entrance gap widths.

For narrow gaps (up to 3 mm), the fluid enters the labyrinth cavity only, excluding the case of the lowest leakage, where such a behavior is observed for 1 mm gap only. For wider gaps, reverse flow regions which spread over the whole gap widths appear. The amount of the reverse flow increases with the gap widths. The fluid return to the main channel disrupts the flow structure in the blade passage. Secondary flow structures increase their intensity and, in consequence, the entropy loss coefficient value in the rotor increases significantly. The reverse flow for the same entrance gap width decreases for higher mass flow rates of leakages. Consequently, the secondary flows intensity and losses in the rotor blade passage decrease. However, this beneficial phenomenon does not compensate higher mixing losses downstream of the rotor blade, where the side flow returns to the main channel.

In the investigated range of leakages, the border value of the gap which assures mostly the one-direction flow (3 mm) is approximately equal to 5% of the rotor blade chord. It is relatively small, however can be achieved in some turbine designs.

Further studies are to be conducted for different blade types to work out general guidelines of this zone design. There is a significant distance between the stator blade trailing edge and the labyrinth entrance gap in the case of the turbine under study. Thus, the influence of the circumferential pressure variation due to stator blades is limited but not negligible. Therefore, the influence of the relative movement of the rotor and stator blades should be taken into account in further investigations. Time consuming unsteady simulations are to be conducted for selected mass flow rates of leackages and entrance gap widths.

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