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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

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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

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Stator blade modification as a method of leakage flow treatment to improve flow efficiency of old-design steam turbine stages

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Abstract

The paper discusses the possibilities of decreasing tip leakage losses in old-design turbines. Two methods of modification – stator blade leading edge cut and uncambered stator blade profile, where the blades of the downstream stator are modified near the tip to better fit to tip leakage flow from the preceding rotor – are presented here and investigated with the help of a CFD code. Changes in flow patterns are described. Also, possible efficiency gains from the applied modifications are estimated in the paper.

Keywords: Turbine stage; Tip leakage loss, CFD

1 Introduction

Leakage flows in turbines result from the presence of technological clearances between stationary and rotating parts of machinery. One form of leakage flows is tip leakage over rotor blades. Tip leakage does not give work to a turbine stage rotor, but its energy is still available for work in the subsequent turbine stage. Tip leakage flow, however, is characterised by parameters other than the main flow in the blade-to-blade passage, which can be viewed from Fig. 1, showing results of span-wise distribution of pressure, velocity and temperature

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measured downstream of an LP stage of a 200 MW turbine, see also Gardzilewicz & Marcinkowski [1]. Especially important here is the difference between the velocities of the tip leakage and main flow, which gives rise to downstream mixing, and resulting enthalpy losses of turbine stages. The paper of Denton [2] shows the determinant effect of leakage mass flow rate and swirl angle of the leakage jet at its re-entry to the blade-to-blade passage (as referred to the flow rate and swirl angle of the main stream) on the level of enthalpy losses due to tip leakage. As the mixing process is hardly accomplished in the stage where it is originated, there will also be an off-design inflow onto the downstream stator blade at the tip, with the possible pressure-side separation.

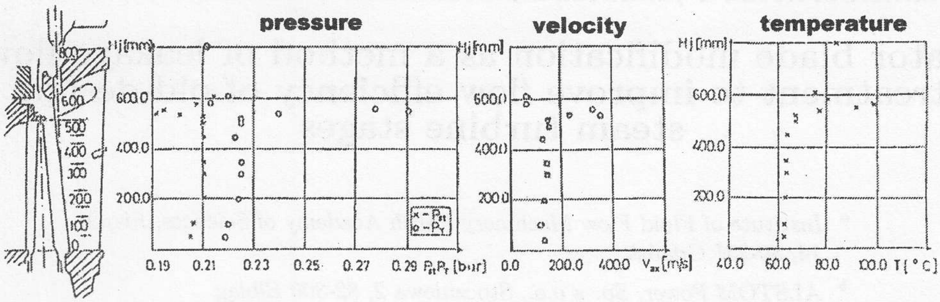
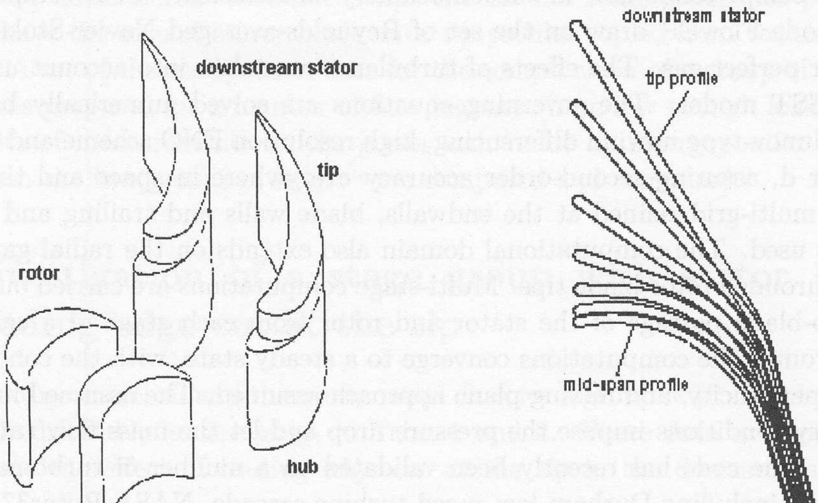


Figure 1. Span-wise distribution of static pressure, total pressure, axial velocity, and temperature measured downstream of an LP stage of a 200 MW turbine.

The most typical way to reduce tip leakage losses is by means of shrouding rotor blade tips and sealing the clearance. The presence of shroud over blade tips changes the mechanism of formation of tip leakage from that driven by a pressure side/suction side pressure difference to the one driven by a leading edge/trailing edge pressure difference, which reduces tip leakage flow rate and losses especially in impulse turbines. The effectiveness of leakage flow reduction is increased with the number of labyrinth seals in the shroud, which can be calculated from the Stodola formula, Perycz [3]. A modern method of reduction of mixing losses due to tip leakage is by means of special turning devices, or bladelets installed in shrouds that enable changing of the leakage jet swirl velocity, decreasing the difference between the direction of the leakage jet and main stream, Wallis et al. [4].

Another idea of leakage flow treatment to increase flow efficiency of an LP turbine stage before the extraction point was patented by Gardzilewicz and Marcinkowski [5], and validated numerically by Gardzilewicz et al. [6]. In this concept, a special stationary deflector installed in the casing downstream of the tip leakage region diverts the tip leakage jet directly to the extraction point, not allowing for its mixing with the main stream.

This paper is devoted to discuss possibilities of decreasing tip leakage losses which arise in the downstream stage due to off-design inflow at the tip. Basically, these possibilities consist in geometric modifications at the tip of the downstream stator blade, by which the stator blade is better adapted to the changing direction of inflow. Two methods of modification are presented here and investigated with the help of a CFD code. In the first method, front part of the blade at the tip is cut, see Fig. 2a, by which the divergence between the direction of the profile camber line at the leading edge and direction of the tip leakage flow is reduced. This idea was patented by Gardzilewicz [7] and described in the paper of Zariankin & Gardzilewicz [8]. In the second method, the downstream stator blade profiles are uncambered towards the tip, also to reduce the difference in directions of the profile camber line and tip leakage flow, see Fig. 2b. This method was applied in a real heat and power station turbine of 25 MW [9].



(a) (b)

Figure 2. Span-wise distribution of static pressure, total pressure, axial velocity, and temperature measured downstream of an LP stage of a 200 MW turbine.

In what follows these two designs are scrutinised how they change flow patterns, as compared to reference geometries. Also, possible efficiency gains from the applied modifications are estimated in the paper. It is expected that the modifications can bring the largest profits in old-design turbines with unshrouded rotors and thin low-load blade profiles. Especially the first method can easily be implemented as part of cheap modernisation programme.

2 Method of investigation

The considered geometric modifications intend to reconstruct the stator blade at the tip to match the direction of tip leakage flow from the preceding stage. The presented ideas are not just recent, they are several years old. Due to moderate expected efficiency gains from application of the ideas, quantitative evaluation of energy profits was very difficult in the past. Experimental investigations on a group of model stages were dismissed here as too expensive. Better possibilities to describe flow physics in the tip leakage region and to evaluate energy profits from application of the discussed improvements appeared with the development of CFD instruments and with the increasing capacity of personal computers.

The analysis presented in this paper is based on numerical investigations carried out with the help of a CFD code FlowER [10,11] – solver of viscous (turbulent), compressible flow in turbomachinery environment. CFD computations in the code FlowER draw on the set of Reynolds-averaged Navier-Stokes equations for perfect gas. The effects of turbulence are taken into account using the $k - w$ SST model. The governing equations are solved numerically based on the Godunov-type upwind differencing, high resolution ENO scheme and implicit operator d, assuring second-order accuracy everywhere in space and time. An H-type multi-grid refined at the endwalls, blade walls and trailing and leading edges is used. The computational domain also extends on the radial gap above the unshrouded rotor blade tips. Multi-stage computations are carried out in one blade-to-blade passage of the stator and rotor from each stage of a calculated stage group. The computations converge to a steady state, with the condition of spatial periodicity, and mixing plane approach assumed. The assumed inlet/exit boundary conditions impose the pressure drop and let the mass flow rate be resultant. The code has recently been validated on a number of turbomachinery test cases, including Durham low speed turbine cascade, NASA Rotor37, NASA low speed centrifugal compressor and model air turbine of ITC Łódź [12,13].

In the course of investigations, results from two groups of two stages are compared - one reference stage group with a geometry typical for the existing turbine blading systems, and a stage group with modification of the second stator near the tip, that is in the region of tip leakage flow from the preceding stage rotor.

The investigations started with a review of old-design steam turbine flow systems (still in operation) of the former turbine manufacturer Zamech in Elbląg. The review extended on large power turbines, heat station turbines and industrial turbines, and aimed at choosing a suitable turbine geometry for beneficial application of the discussed ideas of tip leakage flow treatment. It was decided to investigate the effect of stator blade leading edge cut on a geometry typical for a group of IP stages of 50 to 100 MW steam turbines. These stages have cylindrical blades of height ranging from 100 to 150 mm with thin profiles. Rotor blades are unshrouded with a typical clearance size up to 2% of the blade height. A set of thermodynamic data (inlet and exit pressures and temperatures) for 3D computations of a group of IP stages was prepared with the help of a system of 0D/1D codes GTM [14] and TUR [15] – basically, these codes perform simplified balance heat-and-flow calculations of turbines and are helpful in design of steam turbine stages with cylindrical blading. Then, two groups of stages (without and with the second stator blade modification) were computed and compared. The second idea of tip leakage treatment with downstream stator blade profiles uncambered was decided to be validated on a geometry of the exit from the LP part of a 25 MW turbine where it was actually applied. The rotor blade in the exit stage is 0.43 m long, twisted and unshrouded, the clearance size is equal to 2% of the blade height. Stator and rotor blades are thin. The second stator blade profiles are uncambered towards the tip. A reference stage group was prepared here to have the second stator blade with tip profiles copied from the mid-span section. A set of thermodynamic data for the computation of a group of LP exit stages was prepared for a number of flow regimes (from low to high load) based on results of measurements on the power unit made by Energopomiar Gliwice [16]. Then the two groups of stages were computed and compared.

3 Investigation of a stage group with stator blade leading edge cut at the tip

Sets of geometrical and thermodynamic data for a group of IP impulse stages with cylindrical blades are given in Tabs. 1 and 2. These data are applied to investigate the effect of leading edge cut at the tip in the downstream stator blade.

Geometry of a typical and modified design introduced into the code FlowER is presented in Fig. 3. The picture shows a meridional view with illustrated leading and trailing edges of stators and rotors as well as cascade profiles. A sharp cut of the leading edge shown in Fig. 2 is replaced here by a smooth change of shape where the leading edge is moderately bevelled in the region of tip leakage so as not to generate excessive additional gradients of flow parameters.

The assumed leading edge cut is defined in Fig. 4. In meridional view the leading edge cut is in the form of a straight slanted line changing smoothly into a radial line. A number of cases were investigated changing an non-dimensional blade shape modification parameter Δc defined as a bevel length relative to the axial chord. Another non-dimensional parameter Δl defined as a bevel height relative to the blade height is determined as a function of Δc. In circumferential view the bevelled leading edge is rounded with a circle of diameter equal to

Table 1. Geometrical parameters of a group of IP stages with cylindrical blades

			Stage 1	**Stage 2**
Stator blade	Profile type		N3-70	N3-70
	Blade height	mm	100.7	115.7
	Mid-span diameter	mm	1160.7	1175.7
	Blade angle*	°	42.34	44.51
	Blade number		52	54
	Chord	mm	103.54	103.54
	Relative height**		0.97	1.12
	Relative pitch**		0.68	0.66
Rotor blade	Profile type		P2-35	P2-35
	Blade height***	mm	103.7	118.7
	Mid-span diameter	mm	1163.7	1178.7
	Blade angle*	°	75	77
	Blade number		128	130
	Chord	mm	35.95	35.95
	Relative height**		2.88	3.30
	Relative pitch**		0.79	0.79

* – with respect to the blade cascade front, ** – relative to the blade chord, *** – tip clearance 2%

Table 2. Thermodynamic parameters of a group of IP stages with cylindrical blades

Inlet to stage 1	Static pressure p_0 (bar)	4.35
	Temperature T_0 (°C)	150
	Mass flow rate G (kg/s)	50.5
Exit from stage 2	Static pressure p_{2k} (bar)	2.3
	Gas constant R (J/kgK)	425
	Specific heat ratio γ	1.30

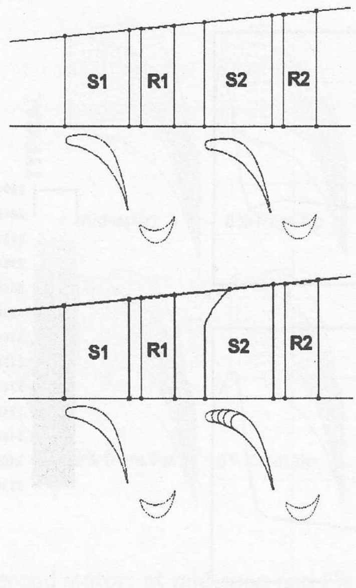


Figure 3. Typical (top) and modified (bottom) geometry introduced into the code FlowER.

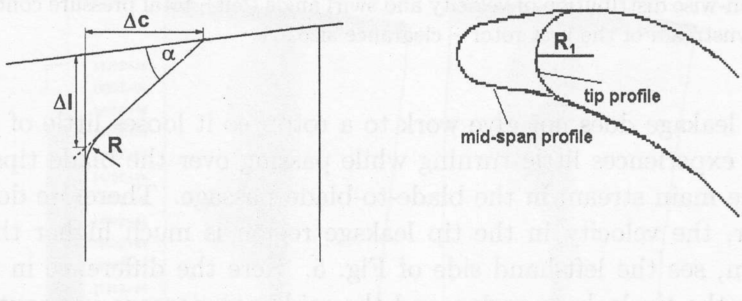


Figure 4. Model of stator blade leading edge cut in meridional and circumferential plane.

the profile thickness at the place of cutting. It was assumed that Δc should not exceed 35%. Larger shape modifications should certainly be subject to mechanical strength investigations. In order not to worsen the process of expansion at mid-span sections it was assumed that the second parameter Δl should not exceed 25%. This parameter was assumed independent of the tip clearance size, as in order to preserve smoothness of modification, the size of cutting may exceed the clearance size even by one order of magnitude. The parameter Δl was defined as $\Delta l = 5/7 \Delta c$. The typical investigated size of tip clearance is 1-2%.

Computations were carried out on H-type grid of approximately 550 000 cells in each blade row. Grid refinements at the walls ($y^+ = 2$) make the grid fulfil the requirements of the assumed turbulence model.

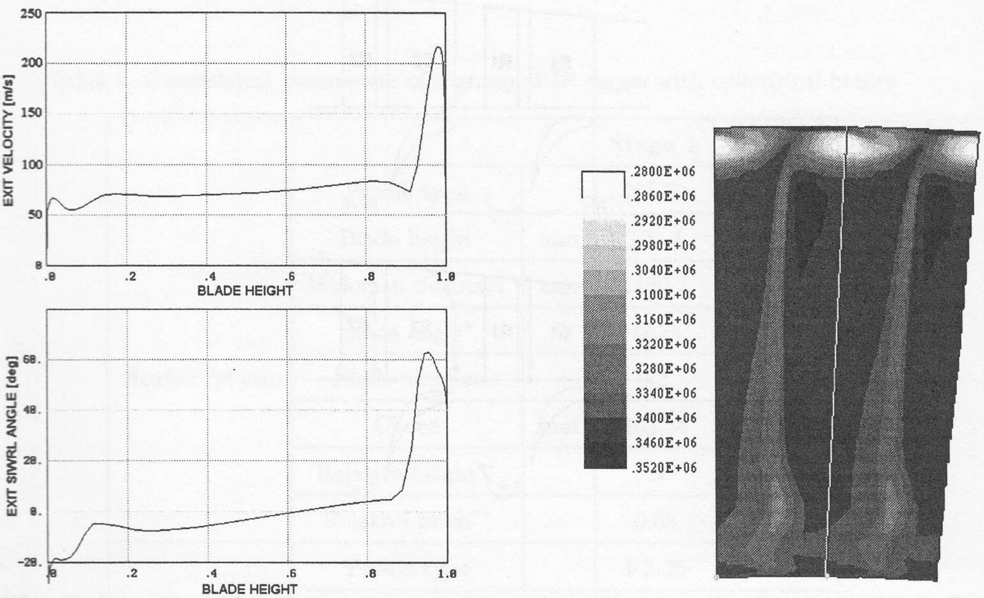


Figure 5. Span-wise distribution of velocity and swirl angle (left), total pressure contours (right) downstream of the first rotor – clearance size 2%.

The tip leakage does not give work to a rotor, so it looses little of its kinetic energy and experiences little turning while passing over the blade tips, as compared to the main stream in the blade-to-blade passage. Therefore downstream of the rotor, the velocity in the tip leakage region is much higher than in the main stream, see the left-hand side of Fig. 5. Here the difference in the velocity between the tip leakage region and the mid-span passage is about 150 m/s, whereas the difference in the swirl angle exceeds 60°. After passing over the blade tip, the tip leakage is subject to an adverse pressure gradient that gives rise to separation of the tip leakage from the tip endwall and formation of a tip leakage vortex. This vortex can be observed from contours of total pressure downstream of the rotor, see the right-hand side of Fig. 5. The tip leakage region seems to extend span-wise up to 10-12% of the channel height. The presented results are for the clearance size equal to 2%.

The distribution of swirl angle presented above implies an off-design inflow onto the subsequent stator blade at the tip. The leading edge cut of the second stator blade makes inflow conditions more favourable near the tip, which can be seen from Fig. 6 showing velocity vectors at selected blade-to-blade sections. The geometric modification changes the interaction between the tip leakage vortex and secondary flow and the main stream in the stator cascade above the mid-

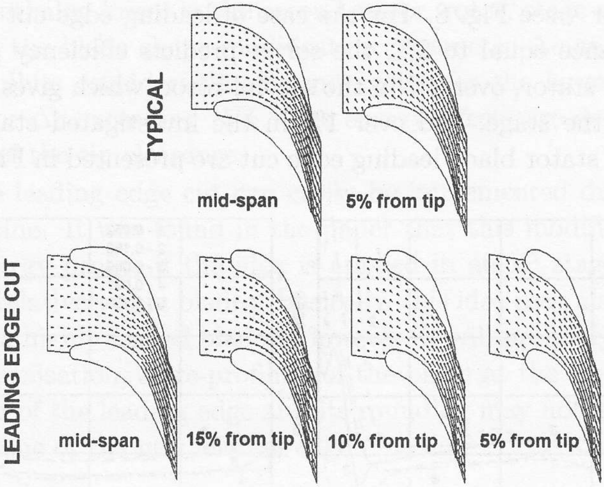


Figure 6. Velocity vectors in second stator: at mid-span and 5% blade height from tip in typical geometry (top row), at mid-span, 15%, 10% and 5% blade height from tip in geometry with leading edge cut $\Delta c = 35\%$ (bottom row) – clearance size 2%.

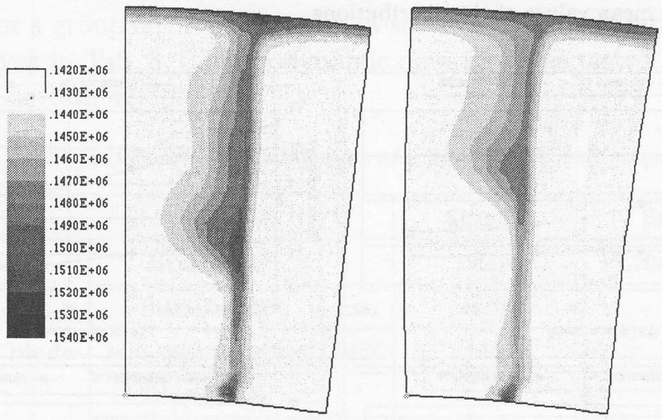


Figure 7. Entropy function contours downstream of second stator: typical geometry (left), geometry with leading edge cut $\Delta c = 35\%$ (right) – clearance size 2%.

span. These forms of vortex flows meet at the suction side of the blade. In the modified geometry they meet more downstream than in the case of a typical geometry. Thus in the modified geometry they have less time to develop in the blade-to-blade passage, resulting in less entropy creation in the second stator. This is confirmed by entropy function contours downstream of the second stator presented in Fig. 7. There are efficiency gains in the second stator and also in the downstream rotor – see Fig. 8. For the case of leading edge cut $\Delta c = 35\%$ and the size of clearance equal to 2%, the solver predicts efficiency gains of almost 2% in the second stator, over 2% in the second rotor, which gives a 2% efficiency improvement in the stage, and over 1% in the investigated stage group. The characteristics of stator blade leading edge cut are presented in Fig. 9 in the form

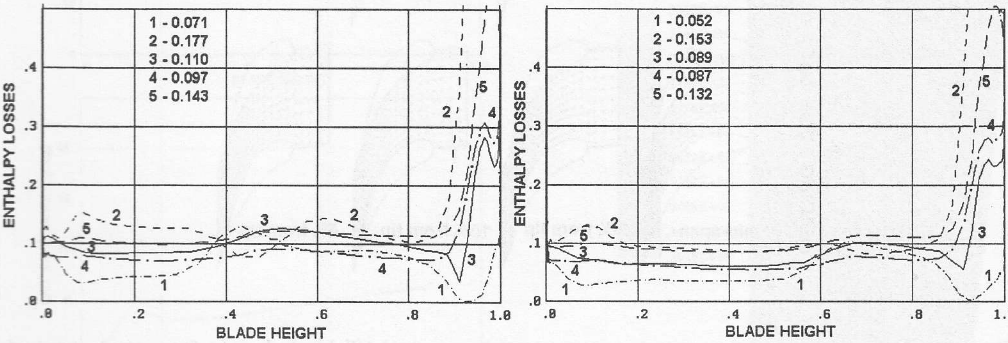


Figure 8. Span-wise distribution of enthalpy losses in second stator (1), second rotor (2), second stage (3), stage group (4), stage group with leaving energy (5): typical geometry (left), geometry with leading edge cut $\Delta c = 35\%$ (right) – clearance size 2%. The numbers indicate mean values of the distributions

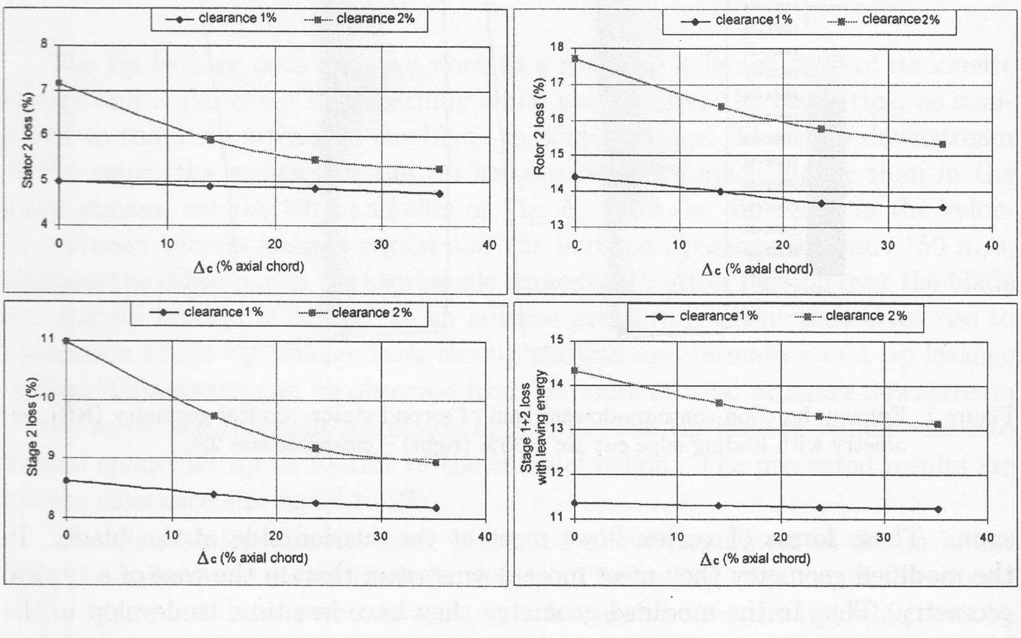


Figure 9. Change of enthalpy losses in second stator (top left), second rotor (top right), second stage (bottom left) and stage group including leaving energy (bottom right) as a function of blade shape modification parameter Δc. For the original blade $\Delta c = 0$.

of variation of enthalpy losses in the second stator, rotor, stage and in a group of two stages with the blade shape modification parameter Δc and clearance size. The picture exhibits considerable efficiency gains for the larger size of the tip clearance and for Δc exceeding 15%. The energy effects are decreased with the decreasing size of the tip clearance.

Stator blade leading edge cut can easily be implemented during modernisation of the turbine. It was found in the paper that this modification can bring considerable energy profits if the idea is applied in an IP stage geometry with cylindrical and relatively thin blades. Basically, this idea may also be effective for stages with thick highly loaded blades. However, it will probably go beyond easy and cheap modernisation, as re-profiling of the blade at the tip will be required if simple cutting of the leading edge and its rounding may not give the adequate aerodynamic shape of the new leading edge.

4 Investigation of a stage group with the second stator profile uncambered at the tip

Investigation of the effect of stator profile uncambered at the tip will be carried out for a group of LP exit stages. A set of geometrical data for this investigation is given in Tab. 3. Thermodynamic data for three tested load conditions are gathered in Tab. 4.

Table 3. Geometrical parameters of a group of LP exit stages

			Stage 1	**Stage 2**
Stator blade	Blade type		PŁ19a	PŁ20a(2503360)
	Blade height	mm	248	391
	Mid-span diameter	mm	1430	1648
	Blade angle	o	–	–
	Blade number		80	70
Rotor blade	Blade type		PŁ07(2717009)	PŁ08(1491961)
	Blade height*	mm	280	432
	Mid-span diameter	mm	1460	1684
	Blade angle	o	–	–
	Blade number		122	152

* – tip clearance 2%

Table 4. Thermodynamic parameters of a group of LP exit stages

		High load	Nominal	Low load
Inlet to stage 1	Static pressure p_0 (bar)	1.04	0.73	0.63
	Temperature T_0 (°C)	100.7	91.1	87.2
	Mass flow rate G (kg/s)	28.97	18.92	13.00
Exit from stage 2	Static pressure p_{2k} (bar)	0.095	0.128	0.128
	Gas constant R (J/kgK)	433.00	438.50	446.20
	Specific heat ratio γ	1.100	1.09	1.10

A typical geometry and a geometry with the second stator profile uncambered at the tip introduced into the code FlowER is presented in Fig. 10. Here, the second stator profiles are changed, without a change of shape of the blades in meridional view. There is about 20° difference in nominal inflow at the tip section of the second stator between the two geometries. Root and mid-span profiles are the same. In the "typical" geometry the mid-span profile is duplicated in the tip section, whereas in the "modified" geometry the stator blade profiles are smoothly uncambered towards the tip.

Similar to the previous investigation of stator blade leading edge cut, the computations were carried out on an H-type grid of approximately 550 000 cells in each blade row with similar grid refinements at the walls ($y^+ = 2$). As this LP turbine operates over a wide range of load, three flow regimes were investigated – a high load, nominal load and low load, which in terms of mass flow rate reads as follows: 38.4 kg/s; 27.0 kg/s and 23.2 kg/s, respectively.

The character of flow in the first stage of the tested group (the last but one in the LP turbine) does not change with load. The exit/inlet static pressure drop is 0.51. The computations exhibit a considerable span-wise gradient of reaction, due to which flow patterns change with channel height. The mean reaction of this old-design stage remains negative (−3.5%). Supersonic velocities are exceeded in the stator throats, especially at the root. The rotor blade is underloaded at the root with a large separation zone at the blade suction surface. The flow efficiency is not impressive, the stage losses exceed 15%. The contours of entropy function downstream of the first rotor exhibit the presence of a tip leakage vortex, rotor wake and a large zone of increased entropy at the root, which can be seen from the left-hand side picture in Fig. 11.

The right-hand side of Fig. 11 shows a span-wise distribution of the exit velocity and swirl angle. Here the effect of tip leakage seems to be restricted to 8% of the channel height. The velocity in the tip leakage region is higher than in

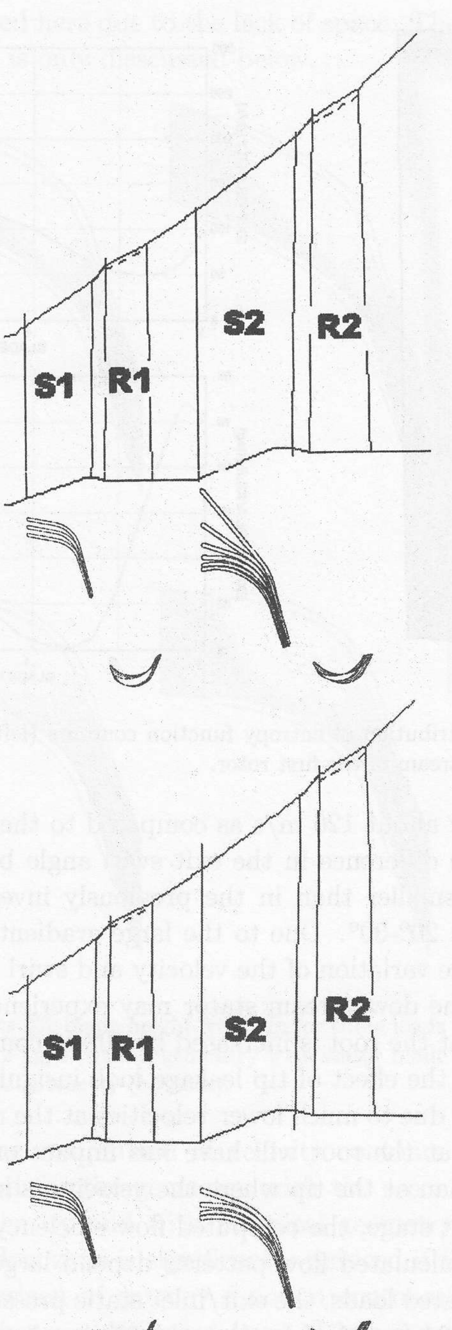


Figure 10. Geometry with stator blade profile uncambered at tip (top) and typical geometry (bottom) introduced into the code FlowER.

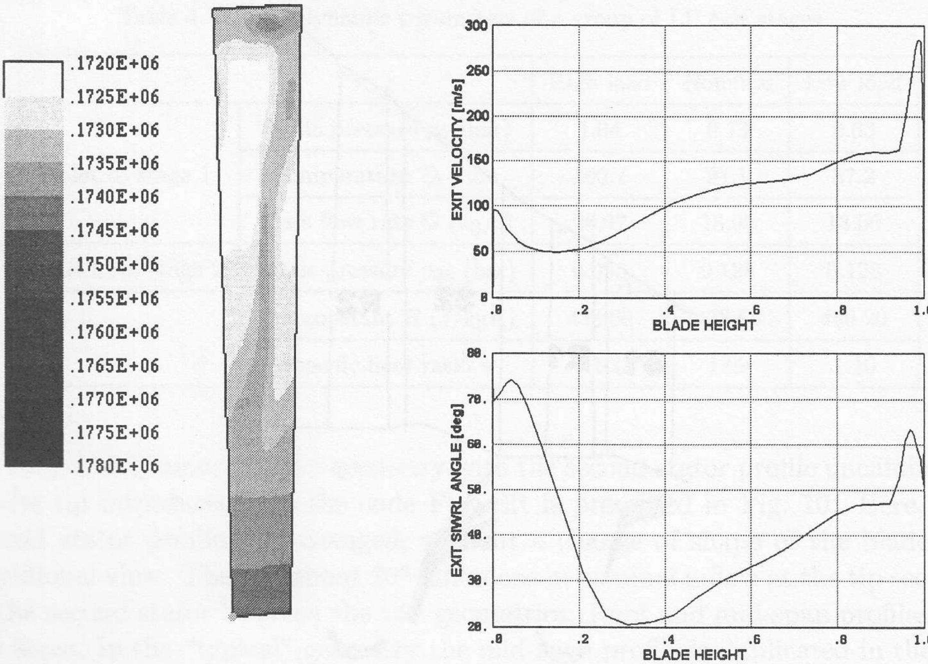


Figure 11. Span-wise distribution of entropy function contours (left), velocity and swirl angle (right) downstream of the first rotor.

the main stream – by about 120 m/s as compared to the passage at 80% of the channel height). The difference in the exit swirl angle between the tip leakage and main stream is smaller than in the previously investigated case of an IP stage group by about 20°-30°. Due to the large gradient of reaction, there is a considerable span-wise variation of the velocity and swirl angle in the main flow, and a large part of the downstream stator may experience an off-design inflow. The exit swirl angle at the root is increased by 50° as compared to the mid-span section, which makes the effect of tip leakage look insignificant at the picture of swirl angle. However, due to much lower velocities at the root, it is expected that the off-design inflow at the root will have less impact on flow efficiency in the downstream stator than at the tip where the velocity is high.

Similar to the first stage, the computed flow efficiency in the second stage is relatively low. The calculated flow patterns depend largely on the flow regime. For the three investigated loads, the exit/inlet static pressure for the second stage is 0.18 (high load), 0.34 (nominal load) and 0.67 (low load). Through this range of load, the flow changes from supersonic with the stator exit Mach number equal to 1.8 (high load) to subsonic - stator exit Mach number 0.7 (low load). There is a considerable span-wise gradient of reaction over some range of load, where the rotor blade is underloaded at the root and overloaded at the tip. These results are not illustrated here due to the lack of space. The effect of stator blade modification at the tip is only disscussed below.

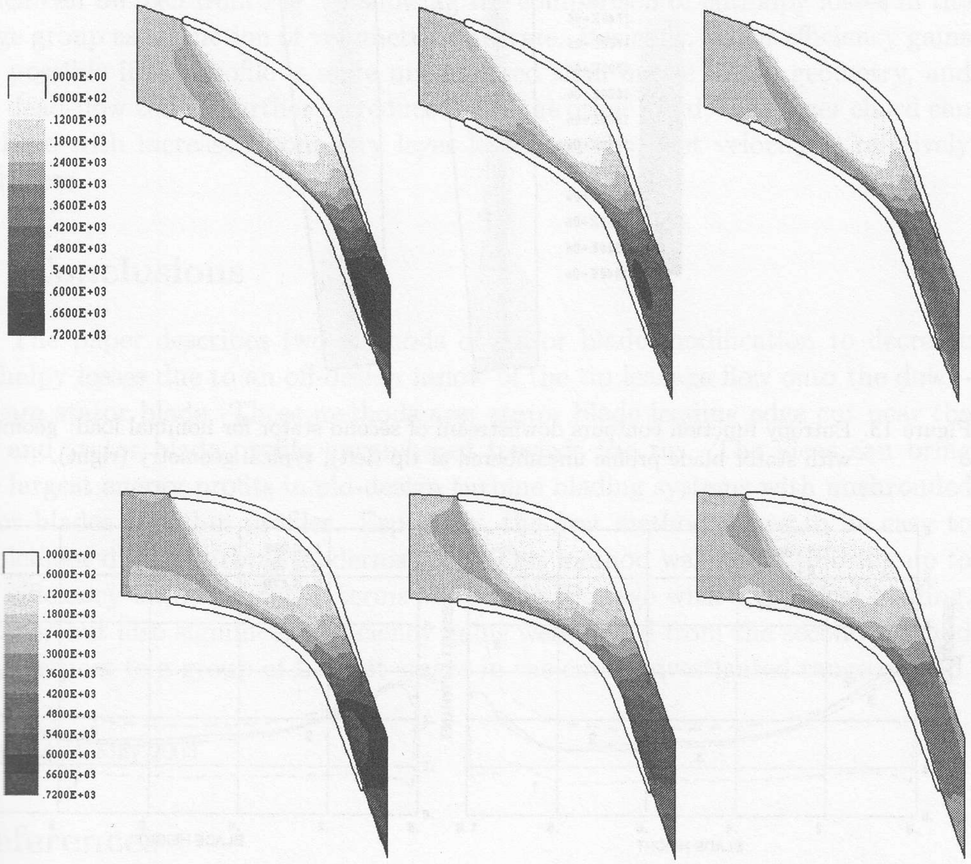


Figure 12. Velocity contours 5% blade height from tip for three loads 38.4 kg/s (left); 27.0 kg/s (centre) and 23.2 kg/s (right): geometry with stator blade profile uncambered at tip (top) and typical geometry (bottom).

Figure 12 shows the comparison of velocity contours at the tip in the investigated range of load for the geometry with the second stator blade profile uncambered towards the tip and with a typical geometry. It is clear that the extension of the dead-flow zone is significantly reduced if the blade profile is uncambered. The comparison of entropy function contours for the nominal load presented in Fig. 13 shows a decreased level of losses near the tip for the geometry with the blade profile uncambered. This can also be observed from the picture of span-wise distribution of enthalpy losses for the nominal load – Fig. 14. The calculated geometry with the second stator blade profile uncambered brings efficiency gains in the second stator as large as 0.7%, in second stage – 0.6%. In

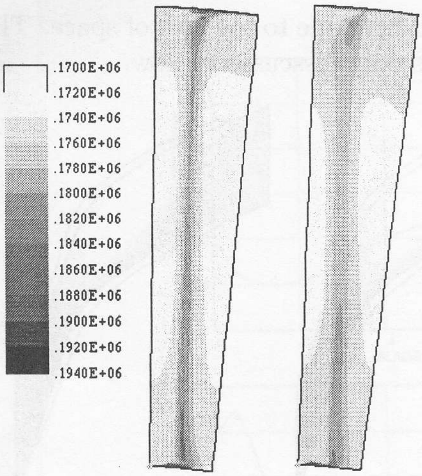


Figure 13. Entropy function contours downstream of second stator for nominal load: geometry with stator blade profile uncambered at tip (left), typical geometry (right).

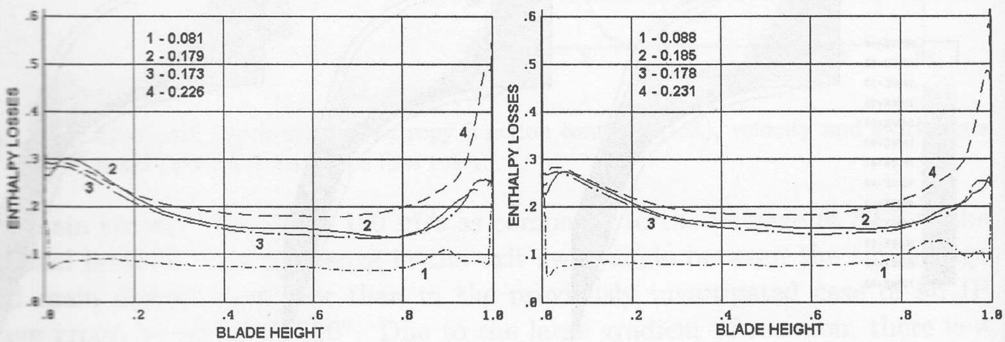


Figure 14. Span-wise distribution of enthalpy losses in second stator (1), second stage (2), stage group (3), stage group with leaving energy (4) for nominal load: geometry with stator blade profile uncambered at tip (left), typical geometry (right). The numbers indicate mean values of the distributions.

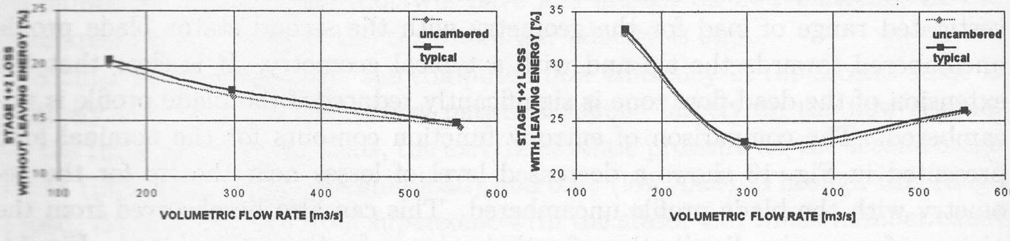


Figure 15. Change of enthalpy losses due to second stator blade uncambered in a group of LP exit stages as a function of volumetric flow rate: enthalpy losses without leaving energy (left) and with leaving energy (right).

terms of the stage group, it gives about 0.5% efficiency improvement (nominal load). Similar efficiency improvements can be expected for high and low loads, which can be seen from Fig. 15 showing the comparison of enthalpy losses in the stage group as a function of volumetric flow rate. Basically, larger efficiency gains are possible if the profile is more uncambered than in the tested geometry, and the dead-flow zone is further be reduced. On the other hand, the longer chord can backfire with increased boundary layer losses, as the inlet velocity is relatively high here.

5 Conclusions

The paper describes two methods of stator blade modification to decrease enthalpy losses due to an off-design inflow of the tip leakage flow onto the downstream stator blade. These methods are: stator blade leading edge cut near the tip and stator blade profile uncambered towards the tip. The ideas can bring the largest energy profits in old-design turbine blading systems with unshrouded rotor blades and thin profiles. Especially, the first method seems to be easy to implement during turbine modernisation. This method was found to bring up to 2% efficiency improvement in terms of a single IP stage with cylindrical blading. Smaller but also siginificant efficiency gains were found from the second method with respect to a group of LP exit stages in the entire investigated range of load.

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