PROPOSAL OF MODERNISATION OF A STEAM TURBINE STAGE BEFORE EXTRACTION, CFD AND CSD ANALYSIS

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Abstract: The paper presents an idea and analysis of application of a new solution of a stage before extraction in the modernised LP part of 200MW turbines with ND37 exhaust. The idea of the patent is to introduce to the turbine steam path a ring directing high energy tip leakage steam from the stage to the regenerative extraction. Owing to this it is possible to cheaply achieve some energy benefits from the operation of first heat exchanger which in the modernised 200MW turbine is continuously underheated. Ring design has been checked using CFD and CSD numerical codes.

Keywords: steam turbine stage, flow and stress optimisation

1. Introduction

In the steam turbine design it is necessary, for operating reasons, to keep clearances over the blades. The steam flow through the clearances has higher energy as compared to the main stream, as well as the flow direction is different. These facts are the origin of losses in this part of the turbine duct resulting from swirl zones generation, intensive mixing processes and blockage of the flow to heat exchangers. In particular, the intensity of the dissipative processes is largest in the area above the unshrouded rotor blades of LP part last stages, where the steam flowing out the clearances has transonic velocity. Based on analyses, a new very efficient solution has been proposed, patented and practically applied. It turned out that such a construction can also bring economical effects in new steam paths of large power output steam turbines.

2. New turbine stage before extraction point

The idea of the new solution of steam turbine stage before regenerative extraction is shown in Figure 1.

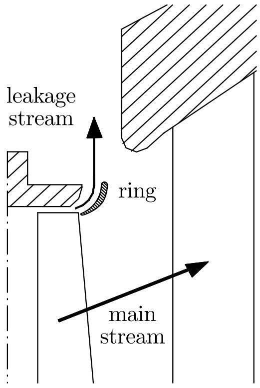

Figure 1. New design of the stage before extraction point

A properly shaped ring installed in the area of tip clearance of unshrouded rotor blade is directing the leakage stream to the extraction chamber. Not coming into details of the patent [1], the advantage of such a solution results from:

- eliminating the swirl zone in the flow since the ring removes the so-called aerodynamic curtain resulting from transonic steam flow within the tip clearance. Simultaneously, the steam flow capacity of the stage behind extraction is significantly increased;
- eliminating the mixing of the tip leakage steam and the main stream steam, which results from directing the leakage steam directly to the extraction;
- thermal reloading of the first regenerative exchanger resulting from utilizing the high energy of leakage stream in the extraction chamber. It is noticed that mass flow rate of the leakage is usually equivalent to that of the extraction steam;
- eliminating liquid phase from the flow, since the ring operates as a separator of secondary water droplets existing in that part of steam turbine.

The new solution was checked in experimental investigations carried out on 200MW turbines [2, 3]. They were particularly important since in the old LMZ design:

- quite large radial clearances over the rotor blades existed;
- large divergence of blade channel was used with simultaneous retaining parallel to the turbine axis clearances causing generation of large dead zones;
- Bauman's stages with crosswise ledge were applied in the steam path; the stages were poorly adjusted to flow in respect of kinematics;
- apparent thermal underloading of first heat exchanger was found;
- erosion of last stage rotor blades was found, in the area considerably below tips.

The thermal measurements instrumentation is presented in Figure 2. Thanks to the plate probes inserted into the flow path, pressure, temperature, and velocity distributions have been evaluated with large accuracy.

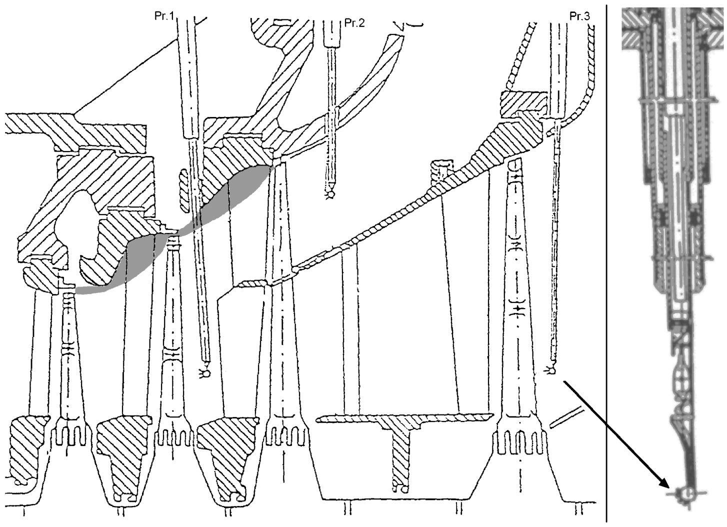

Figure 2. Thermal measurements equipment in the LP section of 200MW turbine with Bauman's stage; zones of influence of tip leakage flow are shadowed in grey

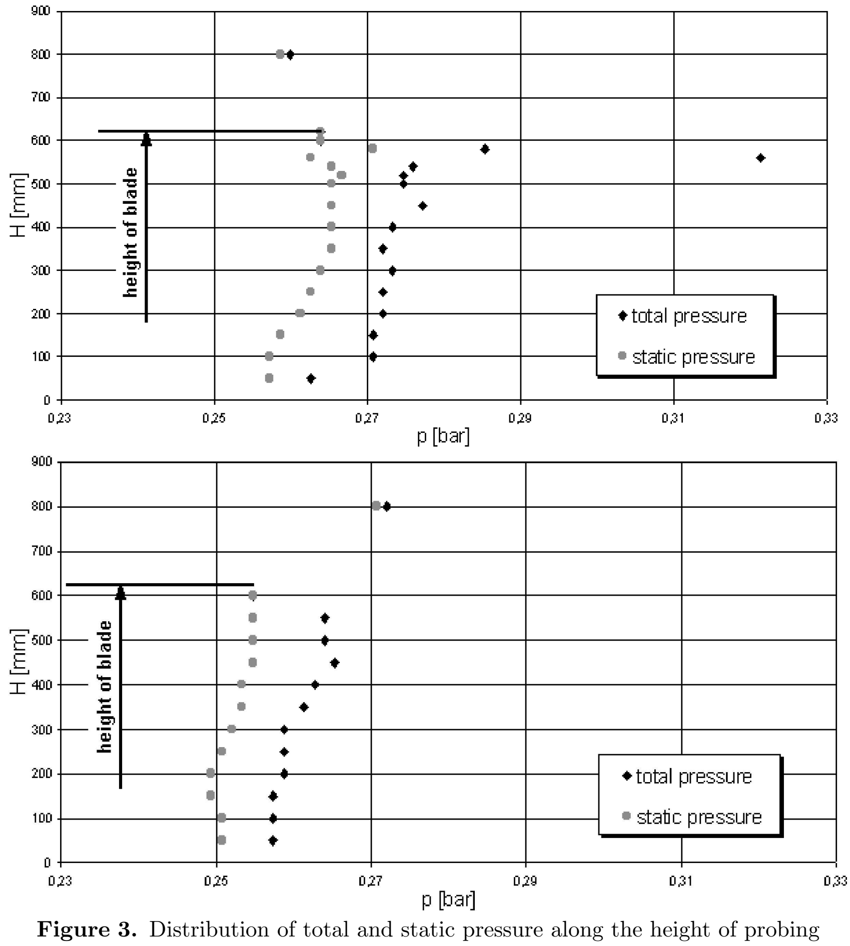

Figure 3. Distribution of total and static pressure along the height of probing before (upper diagram) and after (lower diagram) modernisation

Some measurement results taken before and after modernisation are shown in Figure 3 [2]. The effects of modernisation are clearly visible. Benefits from the patent were evaluated for 300–800kW depending on operating conditions. Over the period 1991–1999 the new solution has been applied on 28 turbines. No operational problems have been found [3] (see Figure 4).

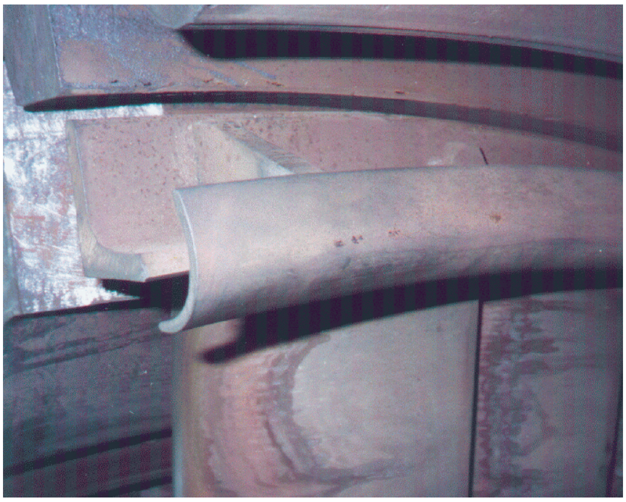

Figure 4. Ring inside the turbine flow path after six years of operation

3. Proposal of patent application in LP part of the turbines with ND37 exhaust

Basing on the experience from the patent application in old 200MW turbines, the authors proposed to install the ring in the modernised blade system with new exhaust. In this case, experimental studies performed with plate probes at Turow Power Plant [4] were used.

Schematic diagram of the measurement system is shown in Figure 5 and proposal of ring introduction in Figure 6.

4. Computational Fluid Dynamic analysis

To estimate benefits and check the new structure numerical computation technique CFD was employed [5]. The present subsection deals with analysis of fluid-flow and thermal calculations. Due to the fact that the flow in blade system is better organised, and the ring removed the effect of mixing utilizing the tip leakage energy in first heat exchanger, actual benefits were acquired.

Simulations were performed for the diffuser with extraction point between the third and fourth stage. These computations were done using standard code FLUENT [6] for axisymmetric model. The computational domain was discretised with hybrid grids, *i.e.* structured mesh near the walls and unstructured one for main flow zone. Number of cells, ranged between 50 000 for initial mesh and 300 000 for adapted one. The numerical simulations were carried out for compressible and viscous fluid (two-equation RNG k-ε turbulence model). It was assumed that the thermodynamic state

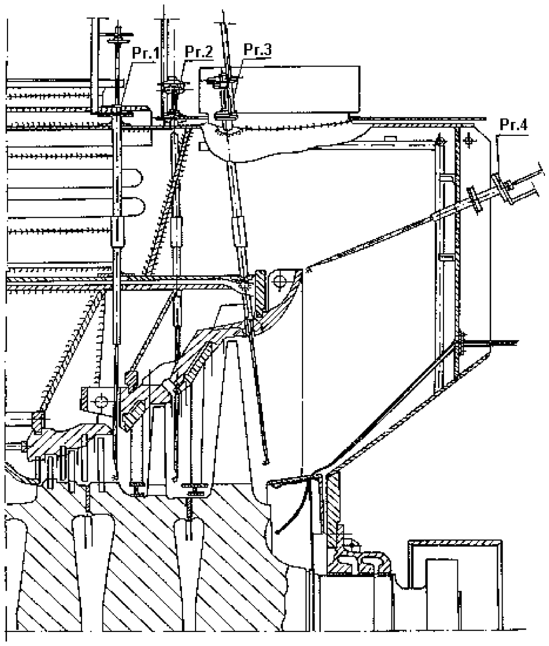

Figure 5. Measurement system in the new LP part of 200MW turbine

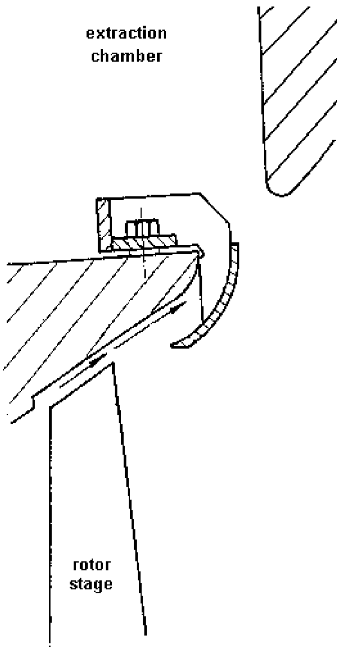

Figure 6. Proposal of ring installation under patent in the modernized exhaust

of steam is described as for the perfect gas assuming appropriate values for κ and R. Fluid-flow data include taken from the power station measurements distributions of total temperature and pressure, inlet angles and mass flow rates, supplemented by outlet static pressure [4]. Figure 7 shows examples of measured and calculated pressure distribution along the channel length for the case without a ring.

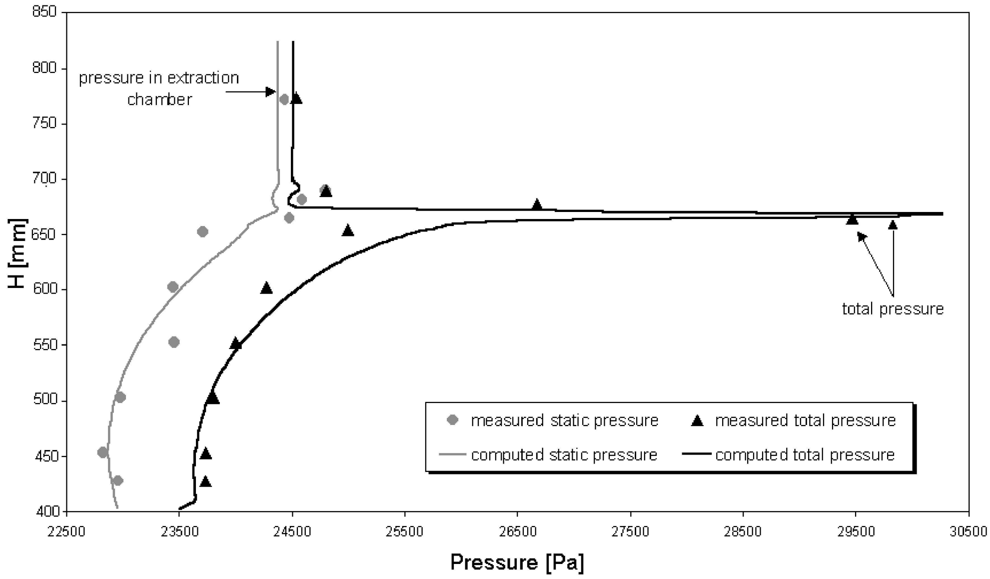

Figure 7. Comparison of computed and measured pressure profile along the probing distance for the case without the ring

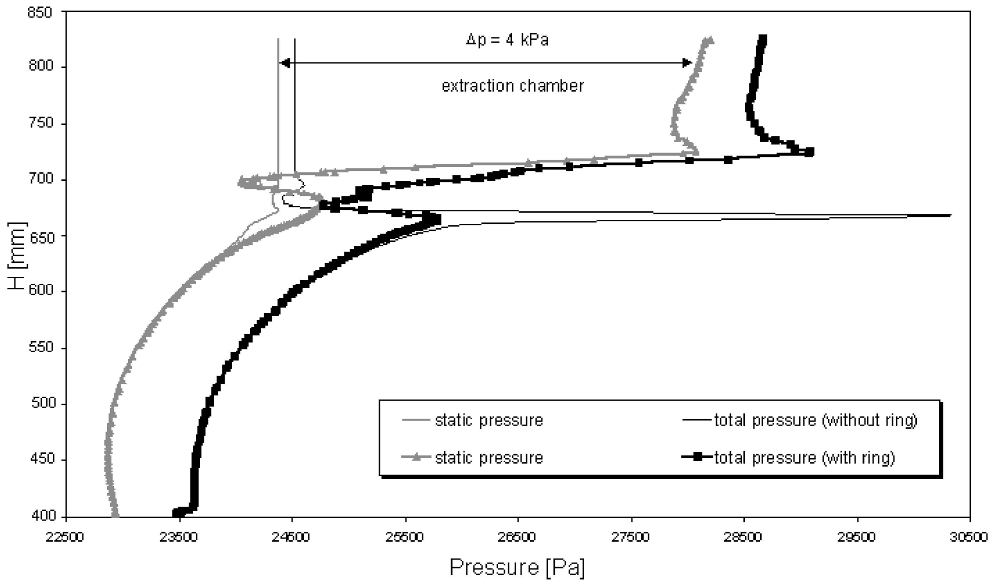

Figure 8. Comparison of computed pressure profile along the probing distance for the case with and without the ring

Comparison of results of calculated pressure distribution along blade height for the variants before and after modernisation at rated conditions are presented in Figure 8. Distribution of Mach number and velocity vectors are presented in Figures 9 and 10. Advantageous influence of the ring application is clearly seen since the ring removed the area of reverse flow before the extraction chamber. At the same time inflow of steam to the last stage stator blades was improved.

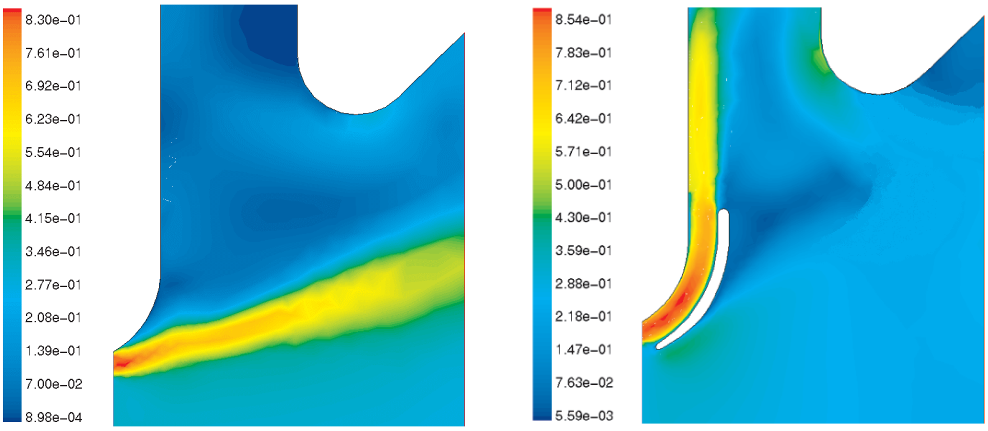

Figure 9. Distribution of Mach number before and after modernisation

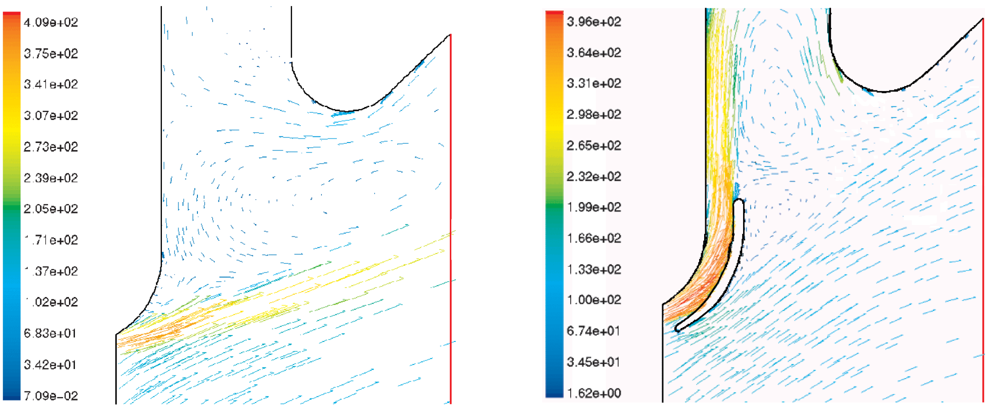

Figure 10. Velocity vectors before and after modernisation

From the pressure distribution within the extraction chamber before and after modernisation (see Figure 8), it is seen that increase of the heat exchanger pressure on the level of 4kPa is possible. It is noteworthy that such an increase was obtained after carrying out simplified optimisation of ring shape and position behind the third stage.

Knowing detailed heat cycle diagram, and having results of measurements taken on the modernised 200MW units [7], it was estimated from the balance calculations how the pressure difference in the heat exchanger influences the power output. To do it a computer code DIAGAR [8] was employed. More important results are collected in Table 1. They refer to two characteristic power outputs 200 and 140MW at condenser pressure 5.5kPa. Additional profit of power 161–223kW is obtained from better operation of heat exchanger. It is equivalent to heat rate decrease by 10kJ/kWh.

It should be pointed out that the mounted ring also improves the flow condition at the inlet of the next stage. It was confirmed by the calculations carried out with the aid of simplified code based on solutions of balance equations using streamline curvature method [9], where viscosity is described by empirical formulae. These

Table 1. Estimated power and heat rate gains for different load of turbine

Power output [MW]	Condenser pressure [kPa]	Power output gain [kW]	Heat rate gain [kJ/kWh]
200	5.5	223	9.7
140	5.5	161	10.7

calculations shown that the benefits of the power gain can be doubled, 300−400kW have been obtained [5].

5. Computational Solid Dynamic analysis

The modernisation was not performed taking into account efficiency aspects only, but included also strength considerations. The main aim of the strength analyses was to check mechanical strength of the structure, and ensure its safe operation in various operating conditions. The analysis was carried out with the help of commercial software ABAQUS [10].

From the forces acting on the ring, thermal load and mean steam forces are the most important factors. In numerical simulations the deadweight and impinging droplets are neglected.

The strength calculations include check of:

- welds (rib-angle, rib-clamping-ring),
- rib stresses,
- clamping ring stresses,
- screws (tension, shear),
- structure natural frequency.

5.1. Thermal model of casing and ring

Due to high susceptibility of the structure to temperature changes, heat transfer calculations in casing and ring were performed. At rated operating conditions the steam flowing through the stage has temperature about 70°C, while the casing temperature is about 115°C. Control system of the turbine protects against exceeding the casing temperature over 150°C. Nevertheless, a more critical is thermal state during start-up of the turbine from a warm state. After turbine shutdown the casing temperature is getting more uniform, and consequently its temperature is increasing to 210°C in the region of 3^{rd} stage, while the steam during windage has temperature 155°C. These are extremely severe operating conditions occurring during some start-ups. Based on the measurement results [4] it is known that during start-ups from a warm state steam temperature drops to 40°C, while the casing has at the same time about 150°C. Because the thermal stresses in the ring are not determined by its temperature, but by temperature difference between the cold ring and hot casing, it was necessary to perform heat transfer calculations in the casing and ring. High heat capacity of the casing and short start-up time at run-up (7 minutes for the extreme thermal states mentioned above) required to perform unsteady computations, while the heat transfer conditions in the thin-walled ring allowed to limit calculations to steady-state only. Numerical model assumed for the casing was two-dimensional, while for the ring – three-dimensional with shell elements. For thermal simulation in transient states measurement results taken from [4] were applied as initial and boundary conditions. The casing metal and steam temperatures measured during start-up were treated as thermal loading. On external surfaces heat transfer was taken into account. Values of the heat transfer coefficient α were adjusted on the basis of fluid-flow calculations and analytical formulae given in [11, 12].

The results of temperatures variation during start-up are shown in Figure 11. The smooth TSR curve is a calculated carrier temperature in the area of clamping screw, and accounts for temperature averaging over the hole depth (Figure 12 shows temperature distribution in casing). This temperature is treated as thermal loading for the ring. The loading has dual character:

- temperature TSR is a thermal boundary condition for the heat transfer within the ring. The second boundary condition is heat convection on all exposed surfaces of the ring;
- casing is undergoing thermal deformation proportionally to heating. One of the possible deformations is circumferential elongation. On account of rotational symmetry the circumferential deformation of the carrier can be converted into radial deformation, and the latter into carrier radius increase. Finally, the radius increase may be calculated from the following formula:

$$\Delta R = R \cdot [1 + \beta \cdot (\mathrm{TSR} - \mathrm{TM})],$$

where R is the carrier radius, β stands for the thermal expansion coefficient, TSR – the carrier temperature, TM denotes temperature of ring assembly in the casing. It was assumed that TM = 30°C.

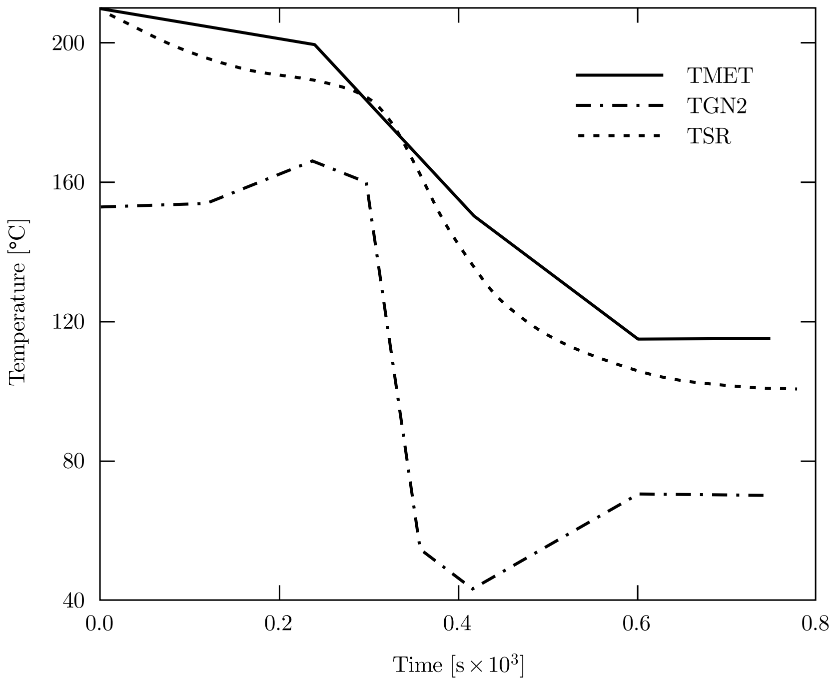

Figure 11. Temperature variation during warm start-up;
TMET – casing temperature, TGN2 – measured steam temperature,
TSR – calculated casing temperature in the region of clamping screws

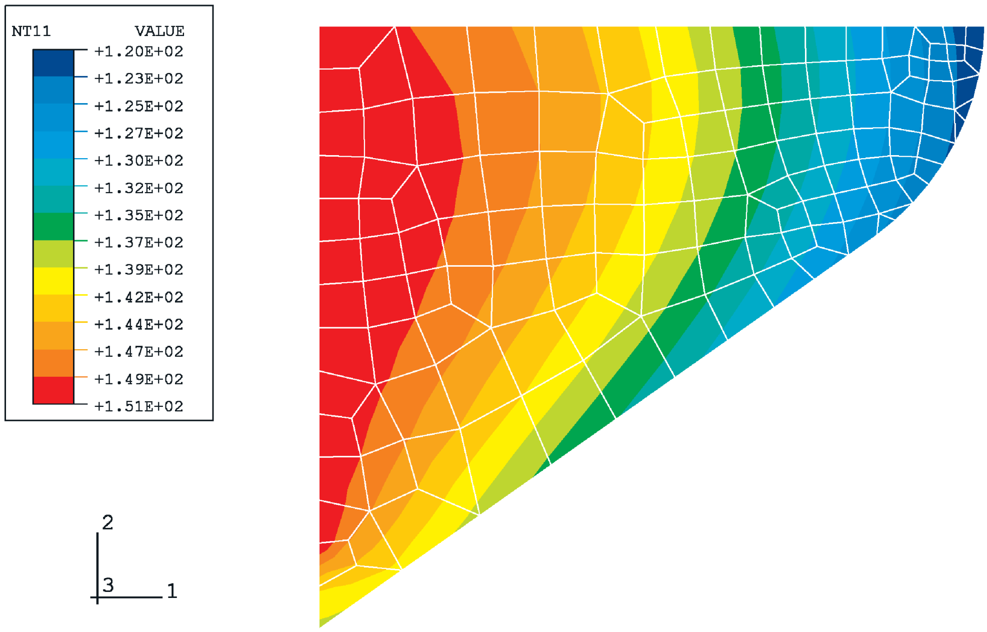

Figure 12. Calculated temperature distribution in casing at steam temperature 43°C (see Figure 11)

This means that the thermal calculations of the casing in transient states and the ring in stationary conditions are "manually" coupled by applying to the ring the temperature TSR and radially displacing supports by ΔR. As it was mentioned, the largest thermal stresses are generated at big casing-ring temperature difference, and not at high but uniform casing and ring temperature. That is why as a final loading the temperature difference at 360^{th} second from the beginning of simulation was selected from the history of heat transfer investigated. For this instant, the rotational speed is 3000 rpm, steam temperature – 54°C, casing temperature – 167°C, screws temperature – 163°C, and the ring has slightly higher temperature than steam (from 55 to 60°C), except for the regions close to the holes for screws. Thus, the ring is subjected to temperature difference about 105°C, and to loading from static and dynamic pressure at full flow rate.

6. Results of calculations

Spatial model of the structure, due to axial symmetry, includes $\frac{1}{2}$ of the ring only. The ring, ribs and angle were modelled by means of shell elements with reduced integration scheme and parabolic shape function. Along the shell thickness 5 integration points were introduced. Boundary conditions for the structure include supports in the region of screws entering the turbine casing, and conditions resulting from symmetry of the semi-ring. The screws joining the angle and the casing (blade carrier) were not modelled, but were treated as support points with locked translation degrees of freedom in axial and radial directions.

Numerical computations of forces in the structure were performed for the following loadings:

- mean steam force from stationary flow acting on the clamping ring; this force is uniformly distributed along the ring circumference and assumes a value: radial component – 1.3kN, axial component – 2.7kN;
- temperature difference between the carrier (43°C) and the ring (146°C) during start-up plus mean steam force;
- unsteady steam force from non-stationary flow fluctuations; excitation frequency – 50Hz, excitation amplitude – 30% PU, material damping – 0.4% critical.

Results of stress calculations are shown in Figure 13. As it follows from the analyses, the loading from steady-state flow only generates stresses in the shells (sheets) up to 25MPa, while the temperature difference during start-ups (together with steady-state flow) – up to 112MPa. For comparison, when thermal expansion of the ring is restricted (rigid attachment to the casing), the reduced stresses in the ring are much higher: clamping ring – 366MPa, ribs – 1037MPa, angle – 550MPa, and stiffening bolts – 750MPa.

This very low level of dynamic stresses in the structure results from sufficient separation margins. First ten natural frequencies are in the range 115–322Hz, and that is why the amplitude of dynamic stresses generated in sheets by unsteady flow is lower than 9MPa. Low stresses occur in clamping screws (up to 10MPa). Increased stresses occur in the extreme welds joining the rib and angle, and on account of this it is recommended to inspect quality of the welds. In the remaining welds the stress level does not exceed 40MPa. Forms of free vibration are mostly torsional, and do not coincide with pressure distribution along the ring circumference. Because the dynamic stresses have very low level, detailed fatigue analysis was omitted while checking the load capacity of the structure.

7. Conclusions

- Application of the ring directing the leakage stream to the extraction chamber in steam turbine stages with unshrouded blades results in actual benefits. The benefits result from eliminating swirls, mixing and water droplets, and from the possibility of heat exchanger reloading.
- The effects were found on twenty eight 200MW turbines with old design LP parts, where the power output increase was estimated on 400–800kW. The rings have operated properly, and over the 9-year operation no failures have been found. Additionally, erosion of the last stage was reduced.
- It turns out that this cheap solution can also be applied in modernized blade systems of 200MW turbines with another type of exhausts. As the calculations revealed, in this case primary benefits result from improved operation of heat exchanger being able to utilize the relatively high energy of the leakage. They were estimated on at least 200kW. Taking into consideration better efficiency of the last stage (4^{th} stage) the benefits should reach even 400kW.
- The results of steam flow within the extraction obtained has revealed that the shape and position of the ring significantly influence the effects of patent application. Optimisation 3D calculations are started, and the stage geometry

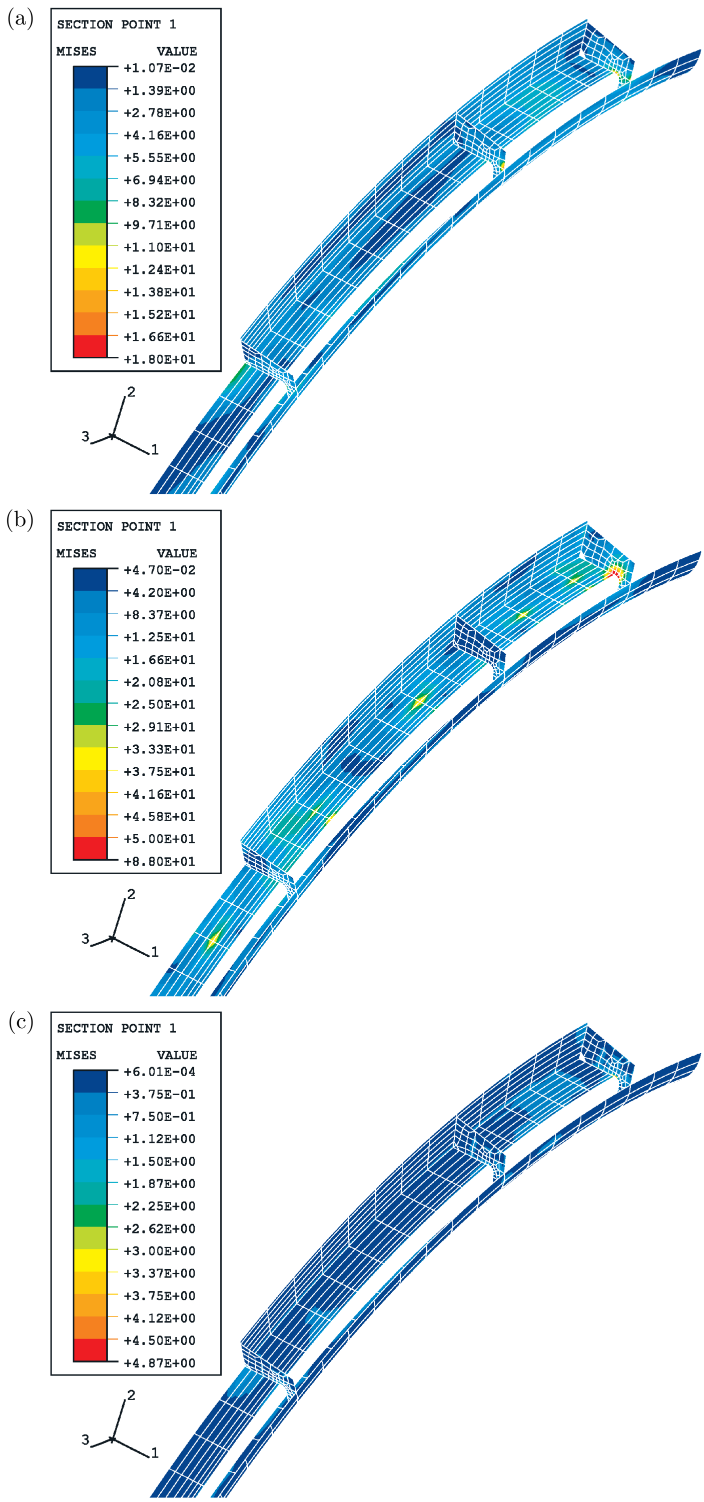

Figure 13. Huber-Misses reduced stresses [MPa] resulting from: (a) steady-state steam flow, (b) temperature difference during start-up, (c) unsteady steam load

takes into account blade rims in order to prove additional benefits resulting from more efficient flow in the last stage.

- The ring structure proposed completely satisfies strength and dynamic requirements at rated operating conditions and during start-ups. The load capacity margin is estimated at 43% of the ring extreme load.
- Considering in the calculations the thermal loads at turboset start-ups and shutdowns showed a considerable role of this loading in the structure effort.

It is provided that the new solution will be applied on one modernized 200 MW turboset at Turow Power Plant where simultaneously fluid-flow and thermal measurements will be made.

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