15th International Research/Expert Conference
"Trends in the Development of Machinery and Associated Technology"
TMT 2011, Prague, Czech Republic, 12-18 September 2011

CHARACTERISTICS OF NON-STATIONARY THERMAL STRESSES IN THE LOW-PRESSURE PART OF THE ROTOR

Zvonimir Guzović, Branimir Matijašević, Tihomir Mihalić
Faculty of Mechanical Engineering and Naval Architecture
Ivana Lučića 5, 10000 Zagreb
Croatia

ABSTRACT
In the paper "Modelling of Non-stationary Thermal Stresses in Steam Turbine Rotors" [1] presented at TMT 2004, the algorithm and the results of non-stationary thermal stresses modelling in steam turbine rotor by means of the users software package are shown. Non-stationary thermal stresses are stipulated by pressure change on turbine exit. The results of non-stationary thermal stress calculations (i.e. of the numerical modelling) show on several characteristic regions of the rotor thermal stressed state: a) the rotor central bore; b) the low-pressure rotor; c) the disc of the last turbine stage, and d) the rear-end labyrinth gland. As follow-up to the paper "Characteristics of Non-Stationary Thermal Stresses in Steam Turbine Rotors" [2] presented at TMT 2006, these characteristic regions are additionally analysed. As in the low-pressure part of the rotor the high gradients of thermal and mechanical quantities (temperature, heat flux, deformation, stresses) are determined, so this region of steam turbine rotor is analysed in detail. In this paper the results of this analysis are presented.
Key words: steam turbine, low pressure rotor, non-stationary thermal stresses, numerical modelling

1. INTRODUCTION

One-dimensional analysis along the central bore of the rotor shows that the thermal and mechanical quantities have significantly changed depending on the exit pressure (back pressure) only after the 9th turbine stage, and that changes are most pronounced in the low-pressure turbine part. Therefore, this paper presents the results of one-dimensional analysis in the area of low pressure rotor (from the 12th to 21st stage) in 10 sections of the rotor, from the central bore to its outer edge. These sections are shown schematically in Figure 1 and marked with numerals from *I* to *X*.
Here, the results only in three most characteristic cross-sections are given, which are hereinafter labeled with letters a), b) and c):

- section a): the area below the 12th stage (corresponds to section *I*);
- section b): the area below the 17th stage (corresponds to section *VI*);
- section c): the area below the 21st stage (corresponds to section *X*).

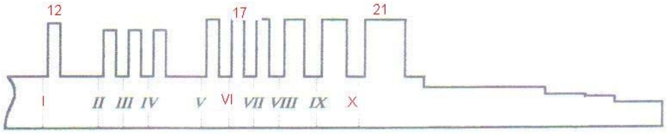

Figure 1. Cross-section of 30 MW power steam turbine

The calculations of non-stationary temperature fields and the respective thermal stresses have been carried out for five characteristic working regimes, for on-design value of the exit pressure 0.03 MPa and for four off-design values: 0.055; 0.085; 0.115; 0.14 MPa. One-dimensional graphical representations are given parametrically depending on the turbine exit pressure and relate to the radial distance from the central axis of rotor (i.e. from axis of rotation) - the abscissa of all graphical presentations is the radial distance from the rotation axis of the rotor.

2. ANALYSIS OF THERMAL-STRESSED STATE IN THE LOW-PRESSURE PART OF THE ROTOR

Temperature distributions over sections depending on the back pressure are shown in Figure 2 [3]. In all sections the temperature increases with pressure. On each of these sections the temperature increases from the periphery of the rotor towards the central bore, as a result of better heat removal in the interstage seals than in the interior of the rotor closer to the bore [4]. Downstream (in the direction of steam flow) through a given cross-sections the temperature values decrease, since by expansion of the steam through turbine flow part its temperature decreases.

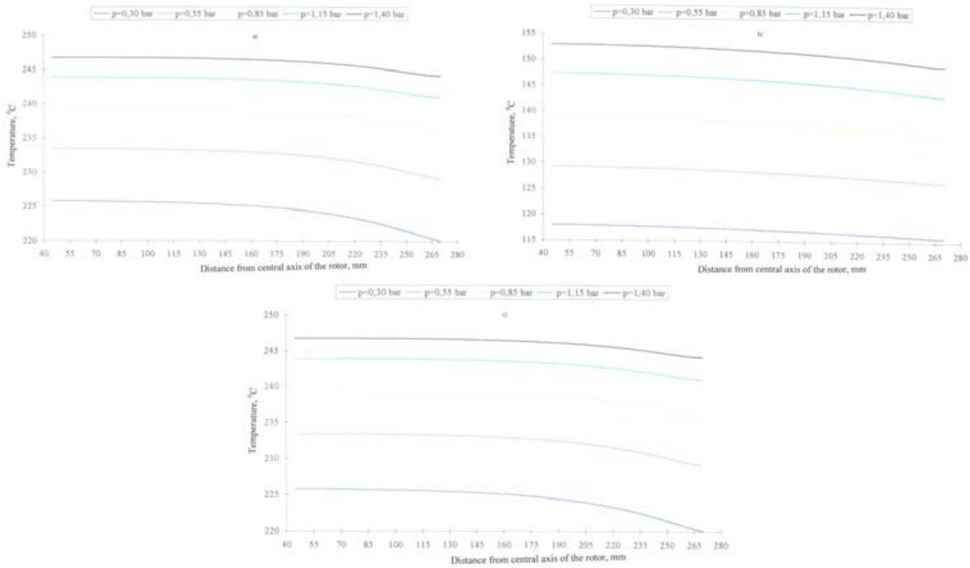

Figure 2. Temperature distributions over sections a), b) and c)

Thermal stresses in sections a), b) and c), conditioned by established temperature field and the geometry of the rotor are shown in Figures 3 to 7 [3].

Characteristics of axial stresses given in Figure 3 are:

- the stress values grow from section a) to section c) due to the increase of thermal deformations on those sections;
- there is a change in character of stresses in the direction from the bore of the rotor to its outer radius - in the wider area of the bore of the rotor axial stress has a tensile character, and closer to the periphery compressive character. This is a consequence of the increased thermal deformations in the axial direction in the areas of the periphery of the rotor relative to the same at its bore. Thereby, the inner parts of the rotor closer to the bore resist axial deformation of the outer part of the rotor which in the peripheral part causes compressive stresses. By reducing the relative deformation of the rotor in the axial direction from section a) to c), the downstream cross-sections of the rotor increasingly resist deformation of the rotor cross-section upstream from them which causes in them increasing values of axial compressive stress in the periphery of the rotor. The tensile stress at the bore of the rotor increases identically;
- in sections a) and c) the value of the axial stress decreases with increasing back pressure, while it increases in section b), which again confirms the complexity of the mutual influence of stress and temperature distribution. This effect of direction change of stresses in the vicinity of section b) on all the other stresses (radial, tangential, shear and equivalent stress) will be observed.

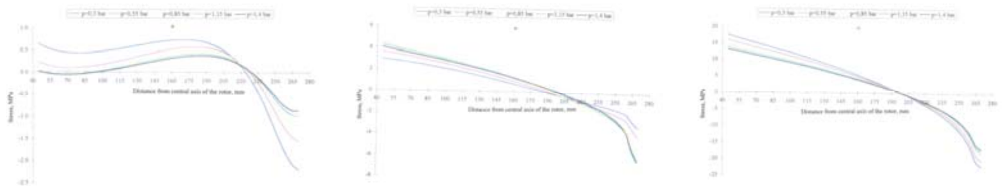

Figure 3. Distributions of the axial stresses over sections a), b) and c)

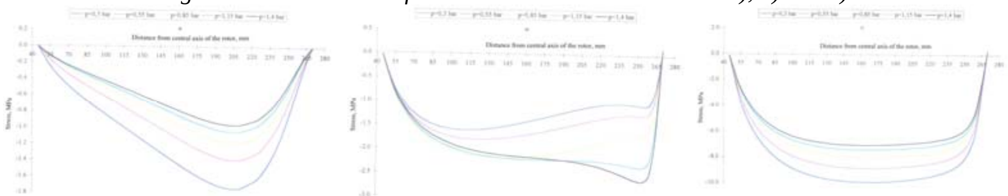

Figure 4. Distributions of the radial stresses over sections a), b) and c)

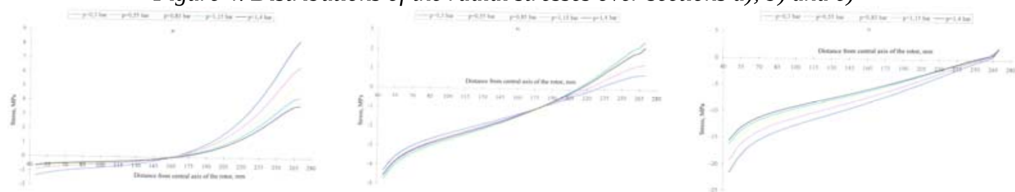

Figure 5. Distributions of the tangential stresses over sections a), b) and c)

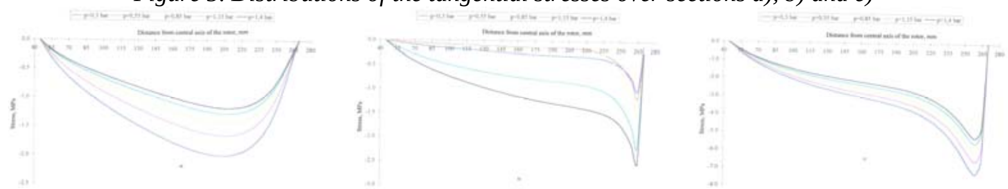

Figure 6. Distributions of the shear stresses over sections a), b) and c)

Characteristics of the radial stresses given in Figure 4 are:

- stresses have compressive character because the area of rotor bore is hoter than its periphery, and the thermal deformations are higher in bore region than on the periphery. Thus peripheral layers of the shaft compressive strain the inner layers.

Characteristics of the tangential stresses given in Figure 5 are:

- stresses have the compressive character in the bore region of the rotor, and the tensile character in the periphery;
- the place of the sign change of stresses has moved radially outward as it goes downstream from section a) to section c). The compressive stress in the interior of the rotor is due to its deformed shape, where due to the stronger tangential deformations in the region of smaller radius the higher compressive tangential stress is established.

Characteristics of the shear stresses given in Figure 6 are:

- stress intensity increases downstream along the rotor (from section a) to section c)), and reaches its maximum at the periphery of the rotor where the deformations are simultaneously higher;
- along all the sections these stresses are negative, as result of higher thermal deformations upstream of section rather than downstream, which caused the deformation of the rotor shape, which is by convention on the sign of shear stress negative.

Characteristics of the equivalent von Mises stress given in Figure 7 are:

- in section a) in the areas of the bore of the rotor and its periphery, the tangential and axial stresses dominate, while the radial and shear stresses have their maximum in the middle of the rotor shaft. The calculation of these values gives the maximum equivalent stress in the periphery of the rotor, while the same are lower in the vicinity of the bore of the rotor;

- in section b) the maximum value of stresses are the result of axial and tangential stresses at the rotor bore and on its periphery, and the radial and shear stresses close to the periphery of the rotor shaft. The calculation of this stress distribution shows an increase of the equivalent stresses in the bore region of the rotor, while the stresses on the periphery remain high;
- the distributions of axial, radial, tangential and shear stresses that are established in section c), in the calculation of the equivalent von Mises stresses, lead to approximately same distribution as it is the case in section b), but there is an increase in equivalent stress at the bore of the rotor.

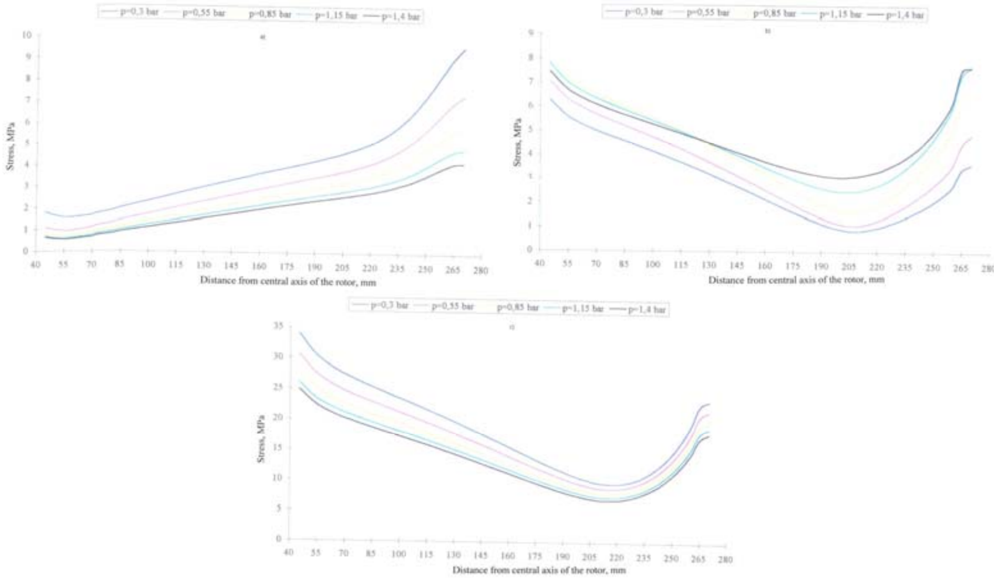

Figure 7. Distributions of the equivalent von Mises stresses

3. CONCLUSION

Regardless of the nature of the distribution of equivalent von Mises stress in the individual sections, it can be generally concluded that the value of these stresses in the area of low pressure rotor increases downstream of the turbine from section a) to section c). Thus, the maximum of about 9.7 MPa at the periphery of section a) is amended to approximately 34 MPa in the bore of section c).

Generally, on the basis of the obtained thermal and mechanical values it is possible to conclude that the thermal-stressed state of the low pressure part of the steam turbine rotor is acceptable: maximal equivalent von Mises stress is below 50 MPa. These values of equivalent von Mises stress in the low pressure part of the rotor are specific for the majority of steam turbines due to the nature of the thermal state and conditions which exist in this region [5].

4. REFERENCES

[1] Guzović Z., Matijašević B., Sviderek S.: Modelling of Non-Stationary Thermal Stresses in Steam Turbine Rotor, TMT 2004, Faculty of Mechanical Engineering Zenica, Neum, Bosnia and Herzegovina, 2004.,

[2] Guzović Z., Matijašević B., Mihalić T.: Characteristics of Non-Stationary Thermal Stresses in Steam Turbine Rotor, TMT 2006, Faculty of Mechanical Engineering Zenica, Barcelona-Lloret de Mar, Spain, 2006.,

[3] Novko I.: Modelling of Transient Thermal and Mechanical Phenomena in Rotor of Steam and Gas Turbines, (in Croatian with abstract in English), M.Sc. Thesis, Faculty of Mechanical Engineering and Naval Architecture, Zagreb, 2004.,

[4] Guzović Z.: Applicability of Existing Correlations for Heat Transfer Determination at Steam and Gas Turbines (in Croatian with abstract on English), Ph.D. Thesis, Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Zagreb, 1998.,

[5] Guzović Z.: Design and Strength of Steam and Gas Turbine Parts (in Croatian), Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Zagreb, 1994.