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- Gas and liquid flows with heat transport, particularly two-phase flows,
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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.

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TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY

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ROMUALD RZĄDKOWSKI*[a,b], VITALY GNESIN[c] and MAREK SOLIŃSKI[a]

High and low frequency excitations of the last stage rotor blades and shaft of the steam turbine 13K215

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Abstract

A three-dimensional numerical analysis for aerodynamic unsteady forces of the last stage steam turbine 13K215 rotor blades and total unsteady forces acting on the shaft from considered stage, have been presented. The numerical calculations were performed for different pressure distribution behind the rotor blades in circumferential direction (p_2 =6000, 7500, 9000, 7500 Pa). The analysis of the 3D transonic flow of an ideal gas through turbomachinery blade rows moving relatively one to another without taking into account the blades oscillations is presented. An ideal gas flow through the mutually moving stator and rotor blades with periodicity on the whole annulus is described by the unsteady Euler conservation equations, which are integrated using the explicit monotonous finite-volume difference scheme of Godunov-Kolgan and moving hybrid H-H grid. The algorithm proposed allows to calculate turbine stages with an arbitrary pitch ratio of stator and rotor blades. The unsteady forces acting on the rotor blades in axial, tangential and radial directions were found. Due to non-uniform pressure distribution the low frequency excitation appeared. Values of the low frequency excitations are greater than the high frequency excitation caused by the stator blades. Next, the total unsteady forces acting on the shaft from all rotor blades in considered stage were calculated. Values of unsteady forces are very small in comparison to the single blade. *The low frequency excitation disappeared. The* high frequency excitation is depended on the number of rotor blades.

Keywords: Unsteady forces; Steam turbine; Blades; Shaft

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1 Introduction

Unsteady phenomena in turbines generally are divided into three large groups: phenomena caused by circumferential nonuniformity of mean flow [5-8,12-13,15], phenomena caused by rotor blades vibration in the flow [1,4,10,14-16] and coupled fluid-flow interaction in the stage incorporating the blade vibration [2,3,9].

In the first and third group the unsteady rotor blade forces were calculated assuming the uniform pressure distribution behind the rotor blades.

In this paper the unsteady forces acting on the rotating and non-vibrating rotor bladed were calculated for the non-uniform pressure distribution behind the rotor blades in circumferential direction. This type of excitation was simulated for 94 rotating blades with 54 nozzles. It was assumed that the pressure behind the rotor blades is changing in the circumferential direction (measured by the angle around the axis of rotation of the turbine). For circumferential angle ($\alpha_o \in (0, 90^o)$ $p_2 = 6000$ Pa, $\alpha_o \in (90^o, 180^o)$ $p_2 = 7500$ Pa, $\alpha_o \in (180^o, 270^o)$ $p_2 = 9000$ Pa, $\alpha_o \in (270^o, 360^o)$ $p_2 = 7500$ Pa. The unsteady forces acting on the rotor blade in axial, tangential and radial directions were found. Next, the total unsteady forces acting on the shaft from all rotor blades in considered stage were calculated.

2 Aerodynamic model

A 3D transonic flow has been considered of ideal gas through a space multipassage blade row [6,7]. In a general case the flow is assumed to be aperiodic function from blade to blade (in pitchwise direction), so the calculated domain includes all blades of the whole assembly.

The flow equations will be written for a three dimensional Cartesian coordinate system which is fixed to a rotating blade row.

The spatial solution domain is discretized using linear hexahedral elements. The equations of motion are integrated on a H-H – type grid with use of explicit monotonous second-order accuracy Godunov–Kolgan finite different scheme [6,11].

We assume that the unsteady fluctuations in the flow are due to rotor blade rotation, and the flows far upstream and far downstream from the blade row are at most small perturbations of uniform freestreams. Therefore the boundary conditions formulation is based on one-dimensional theory of characteristics, where the number of physical boundary conditions depends on the number of characteristics entering the computational domain.

In general case, when axial velocity is subsonic, at the inlet boundary initial values for total pressure, total temperature and flow angles are used in terms of the rotating frame of reference, while at the outlet boundary only static pressure has to be imposed. Non-reflecting boundary conditions can be used, i.e., incoming waves (three at inlet, one at the outlet) have to be suppressed, which is accomplished by setting their time derivative to zero. On the blade surface, zero flux is applied across the solid surface (the grid moves with the blade) [7,11].

3 Numerical results

The numerical calculations have been carried out for the last stage of the steam turbine 13K215. The important properties of the blade disc are: the disk inner radius: $r_o = 0.27$ m, the bladed-disk junction radius: $R = 0.667$ m, the blade length: $L = 0.765$ m. All geometrical parameters of the blade are presented in [15].

In real turbine the interblade phase angle is dependant on the number of stator blades. The computational results presented below were carried out for the last stage of steam turbine with stator to rotor blade number ratio of 56:96 (7:12).

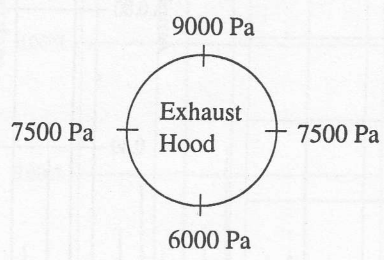


Figure 1. The pressure distribution behind the rotor wheel.

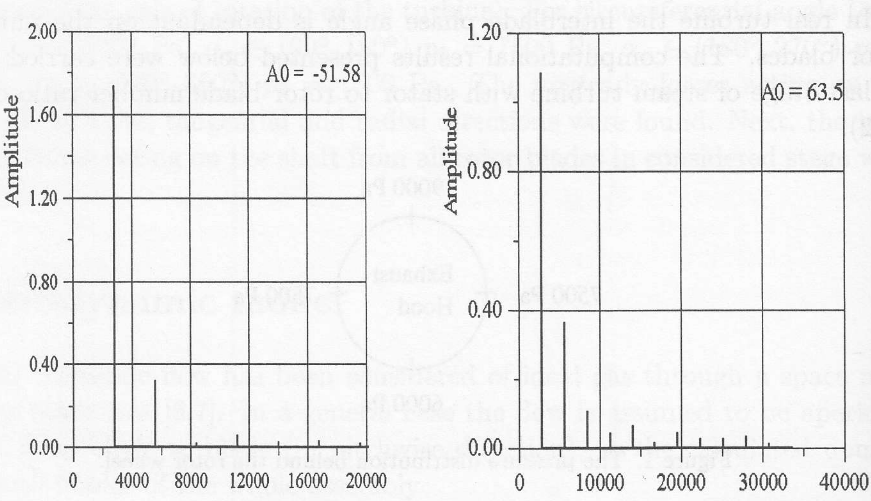
The low frequency excitation arise mainly from low frequency harmonic distortions of the flow field. Harmonic flow distortions may arise from the inlet flow guide passage spacing, e.g., from guide vane pitching and or gauging errors during manufacture or assembly of the guide rows. These excitations also arise from construction features such as diaphragm joints, and from radial or angular eccentricity of the moving row and nonsymmetrical pressure distribution behind the exhaust hood in circumferential direction. This type of excitation was simulated for 94 rotating blades with 56 nozzles. It was assumed that pressure behind the rotor blades is changing in the circumferential direction (measured by the angle around the axis of rotation of the turbine). For circumferential angle (see Fig. 1) $\alpha_o \in (0, 90°)$ $p_2 = 9000$ Pa, $\alpha_o \in (90°, 180°)$ $p_2 = 7500$ Pa, $\alpha_o \in (180°, 270°)$ $p_2 = 6000$ Pa, $\alpha_o \in (270°, 360°)$ $p_2 = 7500$ Pa.

The unsteady forces acting on the rotor blades in axial, tangential and radial directions are in the form:

$$\mathbf{P}_i(x) = \mathbf{P}_{oi}(x) + \sum_{j=1}^{k} \mathbf{P}_j(x) \sin(j\omega t + \phi_j(x) + j(i-1)\varphi)$$

where x is the co-ordinate of the cross-section of the blade, $\mathbf{P}_{oi}(x)$ is the steady part of the unsteady force, j is the harmonic of the force, ω is the rotation frequency, $\phi_j(x)$ is the phase angle of the j-th harmonic, $\varphi = 2\pi z_s/z_r$, z_s is the number of stator blades, z_r is the number of rotor blades.

As the first step the unsteady forces of the stage were calculated for the pressure behind the rotor blades equal to 6000 Pa, 7500 Pa and 9000 Pa respectively.



(a) (b)

Figure 2. Harmonics of the unsteady forces for a pressure behind the rotor blades equal to 6000 Pa (a) circumferential force; (b) axial force.

Figures 2 present the harmonics of the unsteady cicumferential and axial forces for pressure behind the rotor blades equal to 6000 Pa. It is seen that high frequency excitation of the unsteady cicumferential force appeared for 2800 Hz and is equal to 3.1% of the steady part at the level of 51.58 N. The high frequency excitation of the unsteady axial force appeared for 2800 Hz (56·50) and is equal to 1.7% of the steady part 63.5 N. The unsteady radial force components are very small in comparison to the unsteady axial and circumferential forces.

The sum of all blades unsteady moments and unsteady axial forces acting on the shaft follows from equation:

$$\mathbf{P} = \sum_{i=1}^{z_r} \sum_{k=1}^{n} \mathbf{P}_i(x_{ik}) =$$

$$= \sum_{i=1}^{z_r} \left[\sum_{k=1}^{n} \mathbf{P}_{0i}(x_k) + \sum_{j=1}^{k} \sum_{k=1}^{n} \mathbf{P}_j(x_k) \sin(j\omega t + \phi_j(x_k) + j(i-1)\varphi) \right]$$

where z_r is the number of rotor blades, n is the number of cross-section of the blade and k is the number of harmonics.

Figure 3 presents the harmonics of the unsteady axial forces from all blades acting on the shaft for pressure behind the rotor blades equal to 6000 Pa. It is seen that all harmonics are very small in comparison to the unsteady forces acting on the single blade.

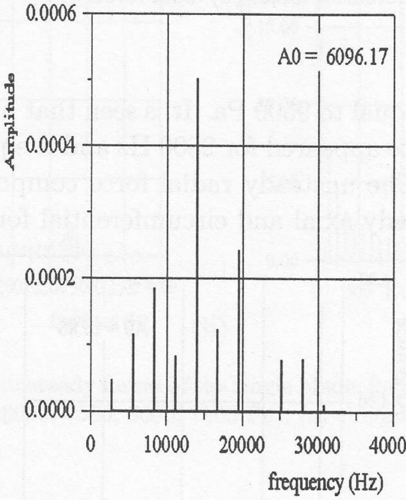


Figure 3. Harmonics of the unsteady axial forces acting on the shaft for a pressure behind the rotor blades equal to 6000 Pa.

Figures 4 present the harmonics of the unsteady cicumferential and axial forces for pressure behind the rotor blades equal to 7500 Pa. It is seen that high frequency excitation of the unsteady cicumferential force appeared for 2800 Hz and is equal to 5.2% of the steady part 24.06 N. The high frequency excitation of the unsteady axial force appeared for 2800 Hz (56·50) and is equal to 2.3% of the steady part at the level of 40.89 N. The unsteady radial force components are very small in comparison to the unsteady axial and circumferential forces.

Figure 5 presents the harmonics of the unsteady axial forcer for pressure

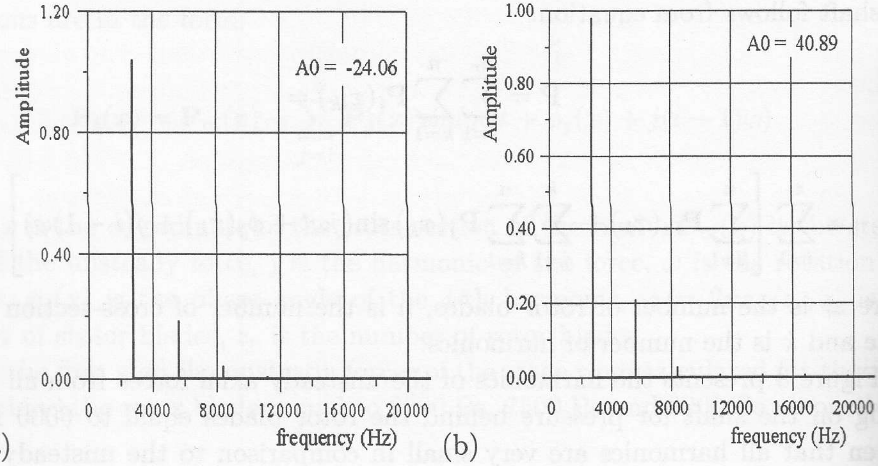


Figure 4. Harmonics of the unsteady forces for a pressure behind the rotor blades equal to 7500 Pa (a) circumferential force; (b) axial force.

behind the rotor blades equal to 9500 Pa. It is seen that high frequency excitation of the unsteady axial force appeared for 2800 Hz and is equal to 36% of the steady part equal to 15.88 N. The unsteady radial force components are very small in comparison to the unsteady axial and circumferential forces.

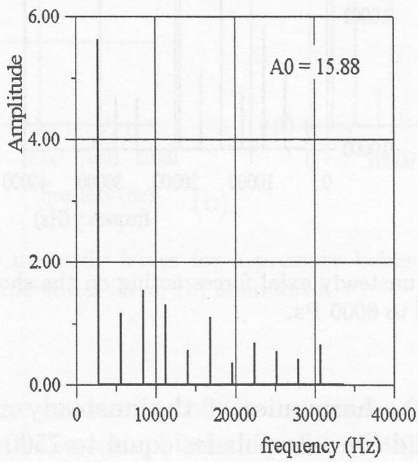


Figure 5. Harmonics of the unsteady axial force for a pressure behind the rotor blades equal to 9000 Pa.

Next, the unsteady forces for all blades were approximated for circumferential angle (see Fig. 1) $\alpha_o \in (0, 90^o)$ $p_2 = 6000$ Pa, $\alpha_o \in (90^o, 180^o)$ $p_2 = 7500$ Pa,

$\alpha_o \in (180^o, 270^o)$ $p_2 = 9000$ Pa, $\alpha_o \in (270^o, 360^o)$ $p_2 = 7500$ Pa.

Figures 6 present the harmonics of the unsteady circumferential and axial forces acting on a blade for non-uniform pressure distribution. It is seen that low frequency excitation of the unsteady cicumferential force appeared for 50 Hz and is equal to 93.1% of the steady part 24.7 N an so on. The high frequency excitation appeared for 2800 Hz and is equal to 10% of the steady part.

The low frequency excitation of the unsteady axial force appeared for 50 Hz and is equal to 54.6% of the steady part equal to 40.29 N and so on. The high frequency excitation appeared for 2800 Hz and is equal to 4.96% of the steady part.

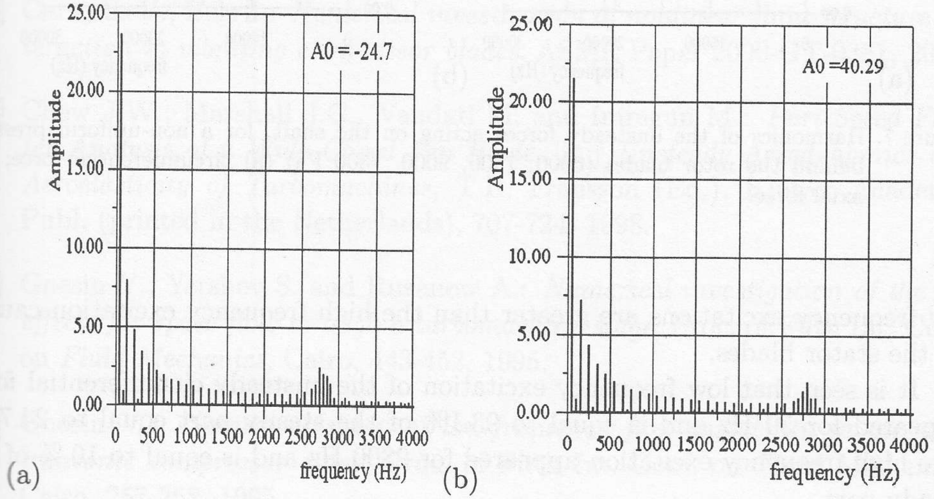


Figure 6. Harmonics of the unsteady forces of the single blade, for a non-uniform pressure behind the rotor blades (6000, 7500, 9000, 7500 Pa), (a) circumferential force; (b) axial force.

Figure 7 presents the harmonics of the unsteady cicumferential and axial forces acting on a shaft for non-uniform pressure distribution. It is seen that low frequency excitation disappeared. The high frequency excitation appeared for 4800 Hz (96 rotor blades multiplied by 50 Hz) and is equal to 0.6.% of the steady part for axial and circumferential forces. The values of the unsteady forces acting on the shaft are very small in comparison to unsteady forces of the blade.

4 Conclusions

The unsteady forces acting on the rotor blades in axial, tangential and radial directions of the last stage of 13K215 turbine were found. Due to non-uniform pressure distribution the low frequency excitation appeared. The values of the

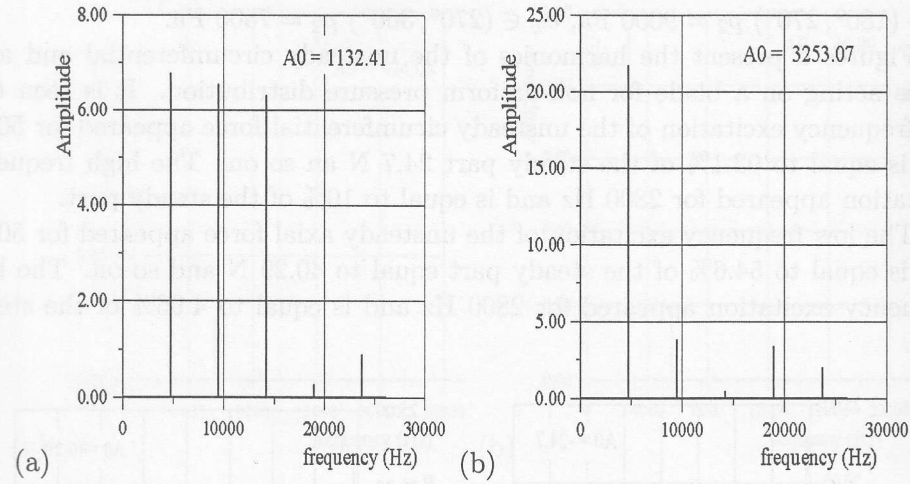


Figure 7. Harmonics of the unsteady forces acting on the shaft, for a non-uniform pressure behind the rotor blades (6000, 7500, 9000, 7500 Pa) (a) circumferential force; (b) axial force.

low frequency excitations are greater than the high frequency excitation caused by the stator blades.

It is seen that low frequency excitation of the unsteady cicumferential force appeared for 50 Hz and is equal to 93.1% of the steady part equal to 24.7 N. The high frequency excitation appeared for 2800 Hz and is equal to 10.% of the steady part.

The low frequency excitation of the unsteady axial force appeared for 50 Hz and is equal to 54.6% of the steady part at the level of 40.29 N. The high frequency excitation appeared for 2800 Hz and is equal to 4.96% of the steady part.

Next, total unsteady forces acting on the shaft from all rotor blades in the considered stage were calculated. The low frequency excitation disappeared. The high frequency excitation are dependant on the number of rotor blades. The high frequency excitation appeared for 4800 Hz (96 rotor blades multiplied by 50 Hz) and is equal to 0.6.% of the steady part for axial and circumferential forces. Values of unsteady forces acting on the shaft are very small in comparison to unsteady forces of the blade.

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