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AERODYNAMIC UNSEADY FORCES OF THE ROTOR BLADES IN THE TURBINE STAGE WITH STEAM EXTRACTION

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The aerodynamic unsteady forces of the rotor blades in the turbine stage 13UC100 with steam extraction were calculated. In this stage the rotor blade failure was reported. The unsteady forces of the rotor blade were calculated for four steam extraction conditions. It was found that for the maximal steam extraction the 10th harmonic (400 Hz) of the axial force and moment was close to the first natural frequency of the bladed disc with one nodal diameter. So the steam extraction can become a cause of the blade failure.

1. INTRODUCTION

The aerodynamic unsteady forces of the rotor blades in the turbine stage 13UC100 with steam extraction were calculated. In this stage the blade failure was reported.

The numerical calculation of the 3D transonic flow of an ideal gas through turbomachinery blade rows moving relatively one to another 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 (Gnesin and Rzadkowski 2002).

In order to find the pressure distribution of steam parameters behind the rotor blades calculation were done for steady flow through the stage and steam extraction channels using the program SPARC.

Next, the unsteady forces of the rotor blade were calculated for four steam extraction conditions. It was found that for the maximal steam extraction the 10th harmonic (400 Hz) of the axial force and moment was close to the first natural frequency of the bladed disc with one nodal diameter. So the steam extraction can become the cause of the blade failure.

2. NUMERICAL SIMULATION OF THE FLOW IN STEAM EXTRACTION AREA

The pressure distribution of steam parameters behind the rotor blades were calculated for the steady flow through the stage and steam extraction channels (see Fig. 1) using the program SPARC. The rotor and stator blades were not taken into consideration. The steady calculations were done for four extractions regimes presented in Tab. 1. Regimes one to three are nominal.

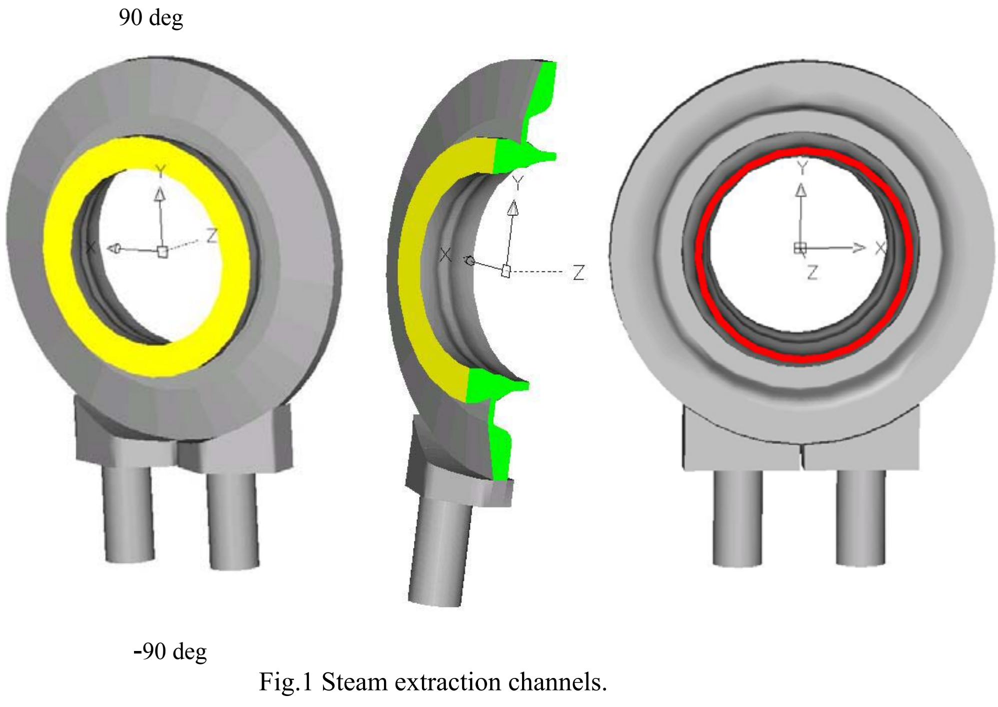

Fig.1 Steam extraction channels.

TABLE 1 Flow parameters in the steam extractions regimes

Flow parameters	Regime 1	Regime 2	Regime 3	Regime 4
p_0 [Pa]	328100	314100	300440	282400
T_0 [K]	411.15	409.67	408.27	406.38
Tu [%]	0.3	0.3	0.3	0.3
Eddy viscosity ratio [-]	1	1	1	1
p_{wylot} [Pa]	248500	239400	228000	215300
p_{upust} [Pa]	317300	299700	280950	252950
m [kg/s]	80.76	80.77	80.68	80.42
m_{upust} [kg/s]	5.42	8.42	11.51	15.38

p_0	- stagnation pressure;
T_0	- stagnation temperature;
Tu	- turbulence level at inlet;
Eddy viscosity ratio	- at inlet;
p_{wylot}	- outlet static pressure;
p_{upust}	- static pressure at the extraction pipe outlet;
m	- mass flow rate at inlet;
m_{upust}	- extraction mass flow rate.

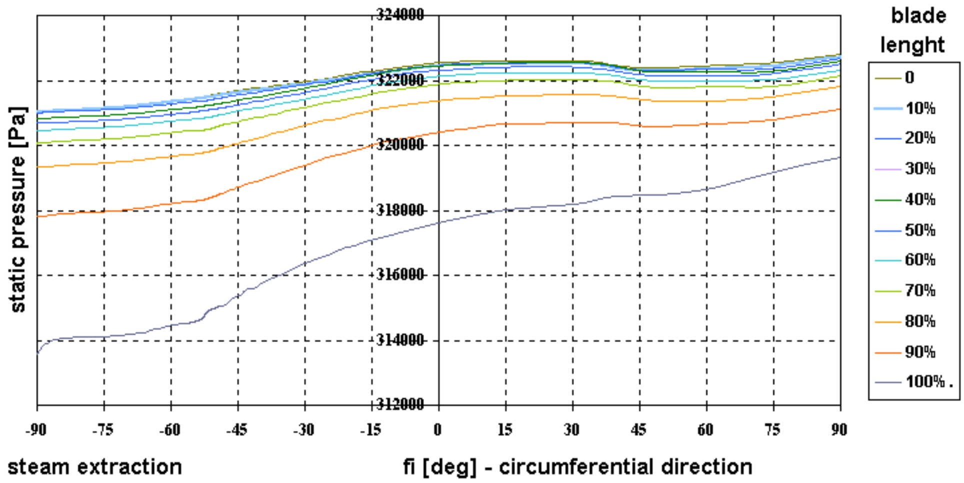

Fig. 2. Pressure distribution behind the rotor blades for regime 4, (m_{up}=15.38 kg/s).

Fig. 2 presents the pressure distribution behind the rotor blades for 4th regime (see Tab. 1). The upper curve shows the circumferential pressure distribution at the radius of blade root (0%). The lowest curve concerns the inlet to the extraction slot (100%). Circumferential angle –90 deg concerns lower part (between extraction tubes, Fig. 1) and 90 deg concerns upper apex of the turbine (see Fig. 1).

2. UNSTEADY FORCES OF ROTER BLADES IN STEAM EXTRACTION AREA

The numerical calculations of the 3D transonic flow of an ideal gas through turbomachinery blade rows moving relatively one to another have been carried out.

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 (Gnesin 2001).The numerical calculations presented below were carried out for the stage of the turbine with rotor blades of length 0.1341 m. The number of stator blades is equal to 46, the number of rotor blades is equal to 126.

The unsteady axial, radial, circumferential forces of the rotor blade were calculated for four steam extraction conditions (m_{up}= 5.39, 8.41, 11.51, 15.38 [kg/s] see Tab. 1). Figs 3 a, b present the unsteady axial forces in the top cross-section for m_{up}= 5.39 [kg/s]. The high frequency excitation (2300 Hz) is equal to 10% of the steady force A_o=32.04 [N].

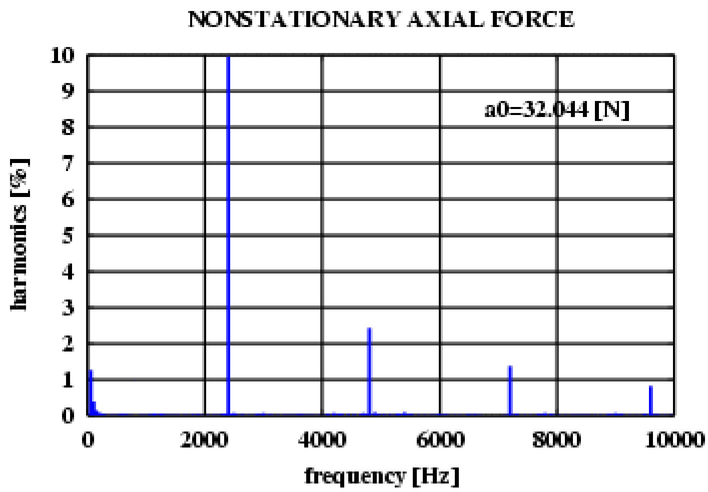

Fig. 3a Harmonics of unsteady axial force: Regime 1, top cross-section , m_{up}= 5.39 [kg/s].

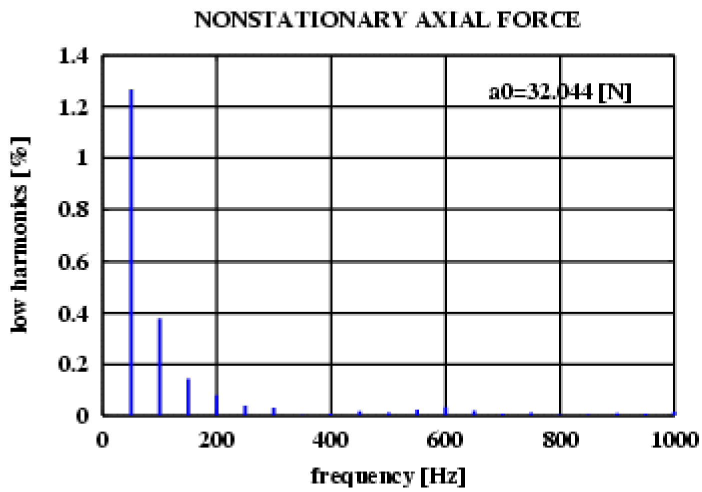

Fig. 3b Low frequency harmonics of unsteady axial force: Regime 1, top cross-section , m_{up}= 5.39 [kg/s].

The low frequency excitation caused by non-uniform pressure distribution is 1,28% for 50 Hz (see Fig. 3b) and 0.39% for 100 Hz.

Figs 4 a, b present the unsteady axial forces in the top cross-section for m_{up}= 15.32 [kg/s]. The high frequency excitation (2300 Hz) is equal to 10.4% (4.38 [N]) of the steady force 42.05 [N]. The low frequency excitation caused by non-uniform pressure distribution is 3.2% (1.33 N) for 50 [Hz] (see Fig. 4b) and 1.1% for 100 [Hz].

In the case of the steam extraction mass flow m_{up}= 15.32 [kg/s] additional unsteady harmonics appeared for 300 Hz, 350, 400, 450, 500, 750, 800, 850 Hz at the 100% and 90% of the blade length. For example the low frequency axial unsteady forces (see Fig. 4b) calculated at the 100% of the blade length are: 0.8% of the steady part for 300 Hz, 1.7% for 350 Hz, 1.9% for 400 Hz, 1.4% for 450 Hz, 0.7% for 500 Hz, 0.3% for 600 Hz, 0.7% for 750 Hz, 0.9% for 800 Hz, 0.7% for 850 Hz.

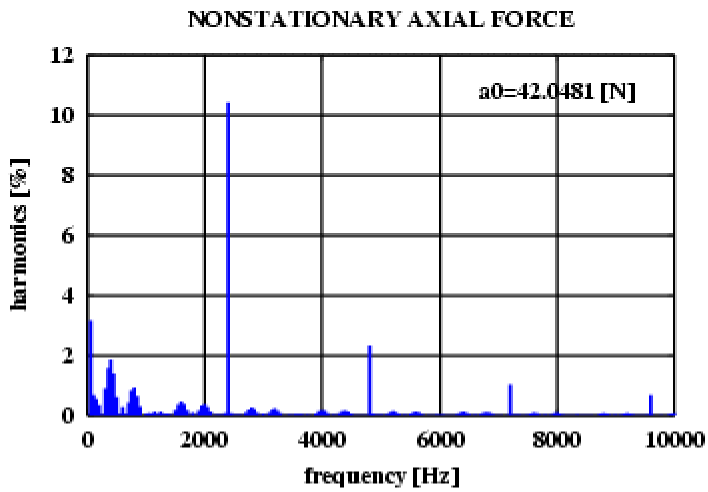

Fig. 4a Harmonics of unsteady axial force: Regime 4, top cross-section, m_{up}= 15.32 [kg/s].

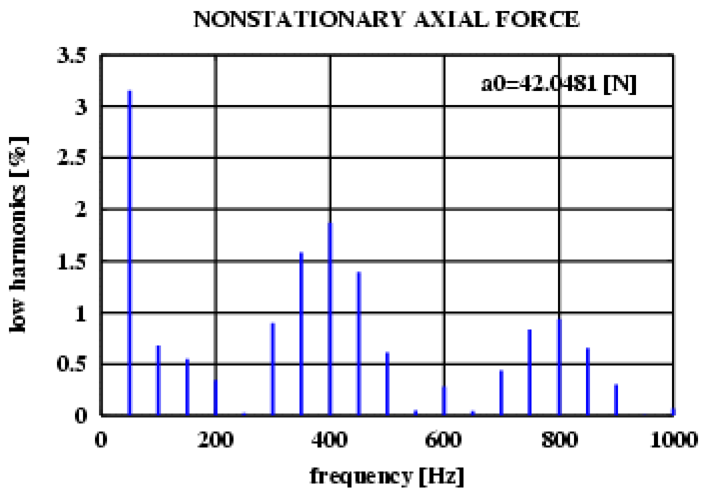

Fig. 4b Low frequency harmonics of unsteady axial force: Regime 4, top cross-section, m_{up}= 15.32 [kg/s].

Tab.2, 3, 4 presents the unsteady axial, radial, circumferential force harmonics of the rotor for four steam extraction conditions (m_{up}= 5.39, 8.41, 11.51, 15.38 [kg/s] see Tab. 1).

- When the steam extraction mass flow increases the steady components of the unsteady forces also increase;
- The largest change of the unsteady forces is observed at the 100% and 90% of the blade length;
- The largest harmonics appeared for high frequency excitation (46 * 50) 2300 Hz, where 46 is the number of stator blades. For axial forces it is equal to 10% -10.41% of the steady component, for moment it is equal to 12-13% of the steady moment and for the circumferential forces it is equal to 82-62% of the steady part;
- Non symmetrical distribution of the pressure behind the rotor blades cause by the steam extraction creates the low frequencies components of the unsteady forces for harmonics 50-900 Hz;
- In the case of m_{up}= 5.39, 8.41, 11.51[kg/s] the low frequency excitation appeared at 50, 100, 150 Hz. The largest one is equal to 4% of the steady part (see Tab 2, 3, 4);
- For m_{up}= 15.38 [kg/s] the largest low frequency excitations are for 50 Hz and are equal to 3.4 –4% of the steady part. The additional harmonics appeared at frequencies 300 -850 Hz.

TABLE 2. Harmonics of the unsteady torsion moments for mass flow steam extraction (m_{up}= 5.39, 8.41, 11.51, 15.38 [kg/s]) at the top blade cross- section

m_{up} [kg/s]	Steady part [Nm]	50 [Hz] [%]	100 [Hz] [%]	150 [Hz] [%]	200 [Hz] [%]	400 [Hz] [%]	600 [Hz] [%]	800 [Hz] [%]	2300 [Hz] [%]	4600 [Hz] [%]
5.39	0.3399	1.2	0.5	0.2			0.2		12	0.5
8.41	0.3704	2.4	0.7	0.3					12.9	0.7
11.51	0.4028	3.6	1.1	0.4					12.9	1.1
15.32	0.4427	3.4	0.6	0.5	0.4	2,01	0.4	1.1	13	0.6

The natural frequencies of the blade and bladed discs were calculated. The natural frequencies of the bladed discs are shown in the interference diagram (see Fig. 5). The natural frequency of the bladed disc with one nodal diameter is close to 400[Hz], so in the case of the steam mass flow extraction m_{up}= 15.38 [kg/s] the blades can work at resonance.

TABLE 3 Harmonics of the unsteady circumferential forces for mass flow steam extraction (m_{up}= 5.39, 8.41, 11.51, 15.38 [kg/s]) at the top blade cross- section

m_{up} [kg/s]	Steady part [N]	50 [Hz] [%] (N)	100 [Hz] [%] (N)	150 [Hz] [%] (N)	200 [Hz] [%] (N)	400 [Hz] [%] (N)	600 [Hz] [%] (N)	800 [Hz] [%] (N)	2300 [Hz] [%] (N)	4600 [Hz] [%] (N)
5.39	11.83	2.35 (0.28)	0.3	0.2			0.5		82 (9.7)	2 (0.36)
8.41	14.018	3,5 (0.47)	1.1 (0.15)	0.3					74 (10.41)	2.8 (0.402)
11.51	16.34	4.64 (0.76)	1.45 (0.24)	0.5					68 (11.16)	2.49 (0.407)
15.32	19.23	4.06 (0.78)	0.9	0.8	0.5	2,07 (0.4)	0.3	1.04 (0.2)	62.56 (12.03	1.98 (0.38)

TABLE. 4 Harmonics of the unsteady axial forces for mass flow steam extraction (m_{up}= 5.39, 8.41, 11.51, 15.38 [kg/s]) at the top blade cross- section

m_{up} [kg/s]	Steady part [N]	50 [Hz] [%] (N)	100 [Hz] [%] (N)	150 [Hz] [%] (N)	200 [Hz] [%] (N)	400 [Hz] [%] (N)	600 [Hz] [%] (N)	800 [Hz] [%] (N)	2300 [Hz] [%] (N)	4600 [Hz] [%] (N)
5.39	32.04	1.28 (0.41)	0.4	0.15			0.5		10 (3.2)	2.5 (0.8)
8.41	35.05	2.3 (0.8)	0.69 (0.24)	0.3					10.4 (3.64)	2.45 (0.86)
11.51	38.19	3.43 (1.31	1.04 (0.398)	0.39					10.4 (3.99)	2.38 (0.91)
15.32	42.05	3.16 (1.33)	0.5	0.6	0.5	1,87 (0.79)	0.3	0.94 (0.39)	10.41 (4.38)	2.33 (0.98)

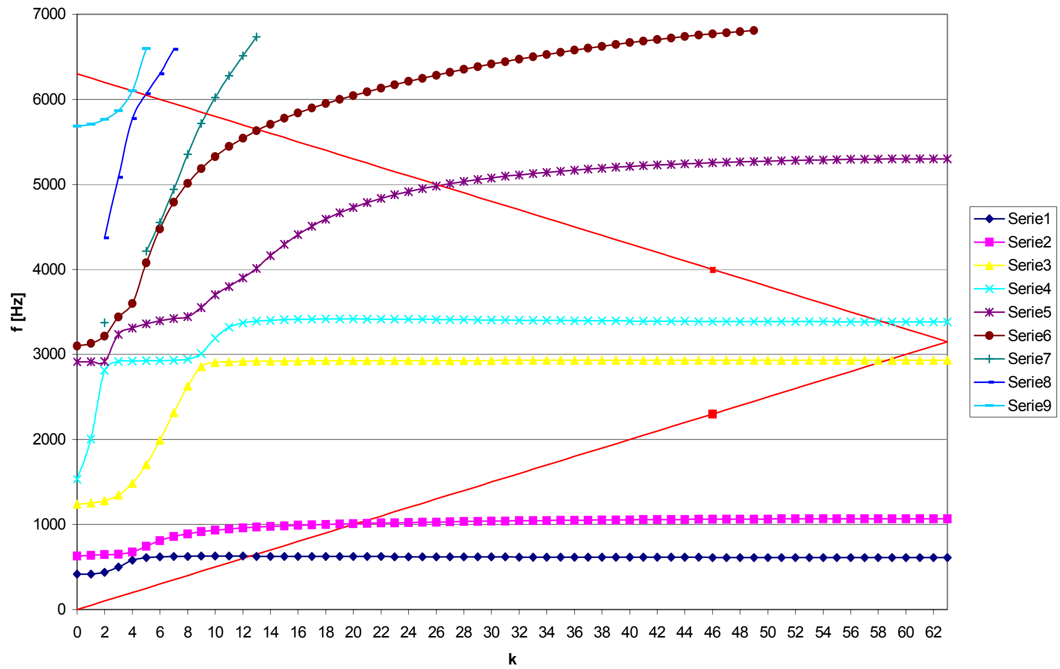

Fig 5. Interference diagram of the bladed disc

6. CONCLUSIONS

The unsteady forces of the rotor blade were calculated for four steam extraction conditions. It was found that for the maximal steam extraction m_{up}= 15.38 [kg/s] the 10th harmonic (400 Hz) of the axial force and moment was close to the first natural frequency of the bladed disc with one nodal diameter. So the steam extraction can be the cause of the blade failure.

The disadvantage of this calculation is that the pressure distribution in the extraction plane was calculated without the stator and rotor blades. The unsteady forces were calculated for one stator-rotor blade channel for boundary conditions calculated from the steady simulations and following approximations in circumferential direction. In future the unsteady calculations of the stage with stator and rotor blades and steam extraction channels must be done.

REFERENCES

[1] Gnesin V.: Program Package, 3D Flow, Theory Manual, Opr. wew. IMP PAN, Nr. arch. 1652/01, Gdańsk 2001

[2] Gnesin V., Rządkowski R.: A Coupled Fluid-Structure Analysis for 3D Inviscid Flutter of IV Standard Configuration, *Journal of Sound and Vibration*, 251(2), 315-327, 2002

[3] Rządkowski R., Gnesin V.: 3D Unsteady Forces of the Transonic Flow Through a Turbine Stage with Vibrating Blades**,** *ASME Paper* GT-2002-300311, 2002.

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NIESTACJONARNE SIŁY DZIAŁAJACE NA ŁOPATKI WIRNIKOWE W OBSZARZE UPUSTU

Niestacjonarne siły działające na łopatki wirnikowe 16 stopnia turbiny parowej 13UC100 z upustem zostały obliczone. Siły te obliczono dla czterech różnych masowych wydatków upustu. Stwierdzono, że dla najwyższej wartości masowego wydatku pojawiają się harmoniczne o częstotliwości 400 Hz, które są bliskie częstotliwości ułopatkowanej tarczy z jedną średnicą węzłową. Stwierdzono, że dla pewnych masowych wydatków upustu łopatki mogą pracować w rezonansie.