THE INFLUENCE OF STEAM EXTRACTION PARAMETERS ON UNSTEADY ROTOR FORCES

ROMUALD RZĄDKOWSKI[1] AND VITALY GNESIN[2]

[1] *Institute of Fluid-Flow Machinery,*
Polish Academy of Sciences,
Fiszera 14, 80-952 Gdansk, Poland
z3@imp.gda.pl

[2] *Department of Aerohydromechanics,*
Institute for Problems in Machinery,
Ukrainian National Academy of Sciences,
2/10 Pozharsky st., 310046 Kharkov, Ukraine

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Abstract: Aerodynamic unsteady forces of rotor blades in a turbine stage with steam extraction were calculated and blade failure was reported. A numerical calculation of the 3D transonic flow of an ideal gas through turbomachinery blade rows moving one in relation to another is presented. An ideal gas flow through mutually moving stator and rotor blades with periodicity on the whole annulus is described with unsteady Euler conservation equations, integrated using the explicit monotonous finite-volume difference scheme of Godunov-Kolgan and a moving hybrid H-H grid. In order to find the pressure distribution of steam parameters behind the rotor blades, calculations were performed for a steady flow through the stage and the steam extraction channels using the SPARC program and the unsteady forces of the rotor blade were calculated for four steam extraction conditions. For the maximal steam extraction the 10^{th} harmonic of the axial force and moment was found to be close to the first natural frequency of the bladed disc with one nodal diameter. Thus, steam extraction can be the cause of the blade's failure.

Keywords: steam extraction, blade, unsteady forces

1. Introduction

An asymmetry of the pressure behind the rotor blades appears in the first stage, stages with steam extraction and the last steam turbine stage, producing low frequency excitation components of the unsteady forces acting on the rotor blades. These components can be dangerous from the point of view of blade stress when they coincide with natural frequencies of the blade or the bladed disc.

In the present paper, the seventh stage of a turbine with steam extraction was considered (see Figure 1, extraction channels of the 17^{th} stage marked in gray). It was in this stage that the failure of the blade was reported.

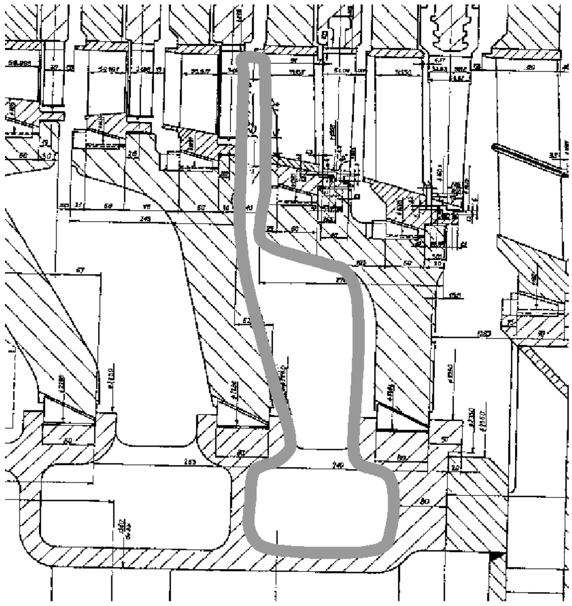

Figure 1. The 17th stage of the 13UC100 turbine

The unsteady forces acting on the rotor blades were calculated using the 3D transonic flow model of an ideal gas through turbomachinery blade rows moving one in relation to another. The unsteady Euler conservation equations were integrated using the explicit monotonous finite-volume difference scheme of Godunov-Kolgan and a moving hybrid H-H grid [1, 2].

The pressure distribution of steam parameters behind the rotor blades were found for a steady flow through the stage and the steam extraction channels using the SPARC program (*cf.* [3]). The geometry of individual stator and rotor blades was neglected.

The unsteady forces of the rotor blade were then calculated for four steam extraction conditions, assuming asymmetrical pressure distribution behind the rotor blades. For the maximal steam extraction, the 10^{th} harmonic of the axial force and moment was found to be close to the first natural frequency of the bladed disc with one nodal diameter. Thus, steam extraction can be the cause of the blade's failure.

2. Numerical simulation of the flow in the stage with steam extraction

The distribution of pressure behind the rotor blades in the stage with steam extraction is asymmetrical and depends on the extraction mass flow rate. Calculations of unsteady forces acting on the rotor blade should include this non-uniformity of pressure. In principle, the calculation should be performed by introducing the full geometries of the stage with the stator and rotor blades and of the extraction channel into the numerical program. However, such a numerical model would be huge and the calculation of unsteady forces would take a whole month.

In this paper, the calculation of unsteady forces has been divided into two steps. In the first step, the pressure distribution of steam parameters in the stage

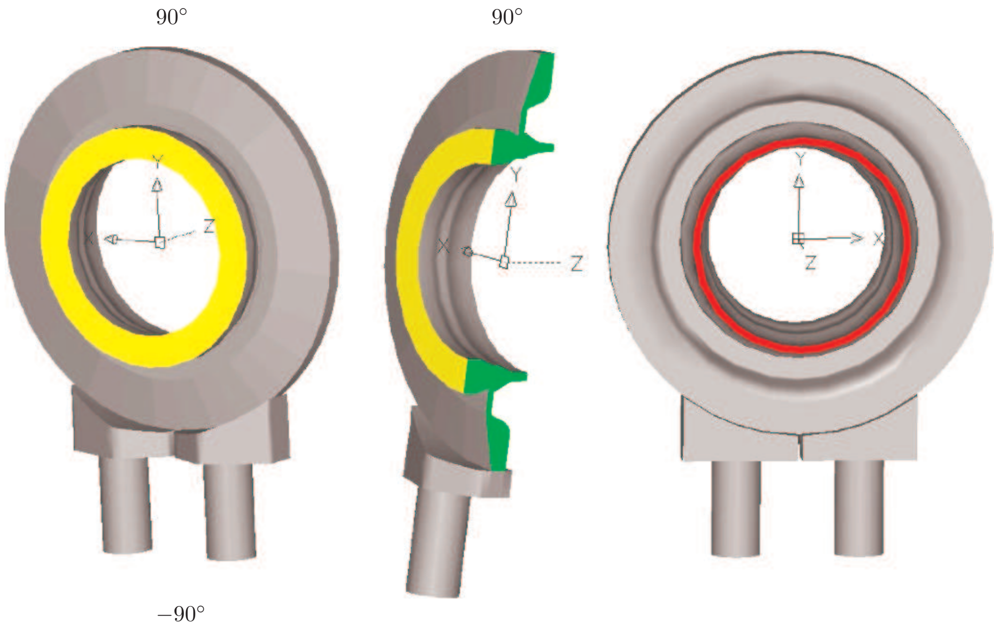

Figure 2. Steam extraction channels

Table 1. Flow parameters in steam extraction 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
$P_{\rm outlet}$ [Pa]	248500	239400	228000	215300
$P_{\rm extr}$ [Pa]	317300	299700	280950	252950
m [kg/s]	80.76	80.77	80.68	80.42
$m_{\rm extr}$ [kg/s]	5.42	8.42	11.51	15.38

p_0 – stagnation pressure;
T_0 – stagnation temperature;
Tu – turbulence level at the inlet;
eddy viscosity ratio – at the inlet;
$P_{\rm outlet}$ – static pressure at the outlet;
$P_{\rm extr}$ – static pressure at the extraction pipe's outlet;
m – the mass flow rate at the inlet;
$m_{\rm extr}$ – the extraction mass flow rate.

was calculated. The model of the stage with extraction channels was simplified (see Figure 2), the rotor and stator blades were neglected.

In the next step, the calculated pressure behind the rotor blade was used as the boundary condition in the stage exit to calculate the unsteady forces acting on the rotor blade.

The steady pressure distribution in the stage for a different extraction mass flow rate (see Table 1) was calculated using the SPARC program [3]. The four extraction

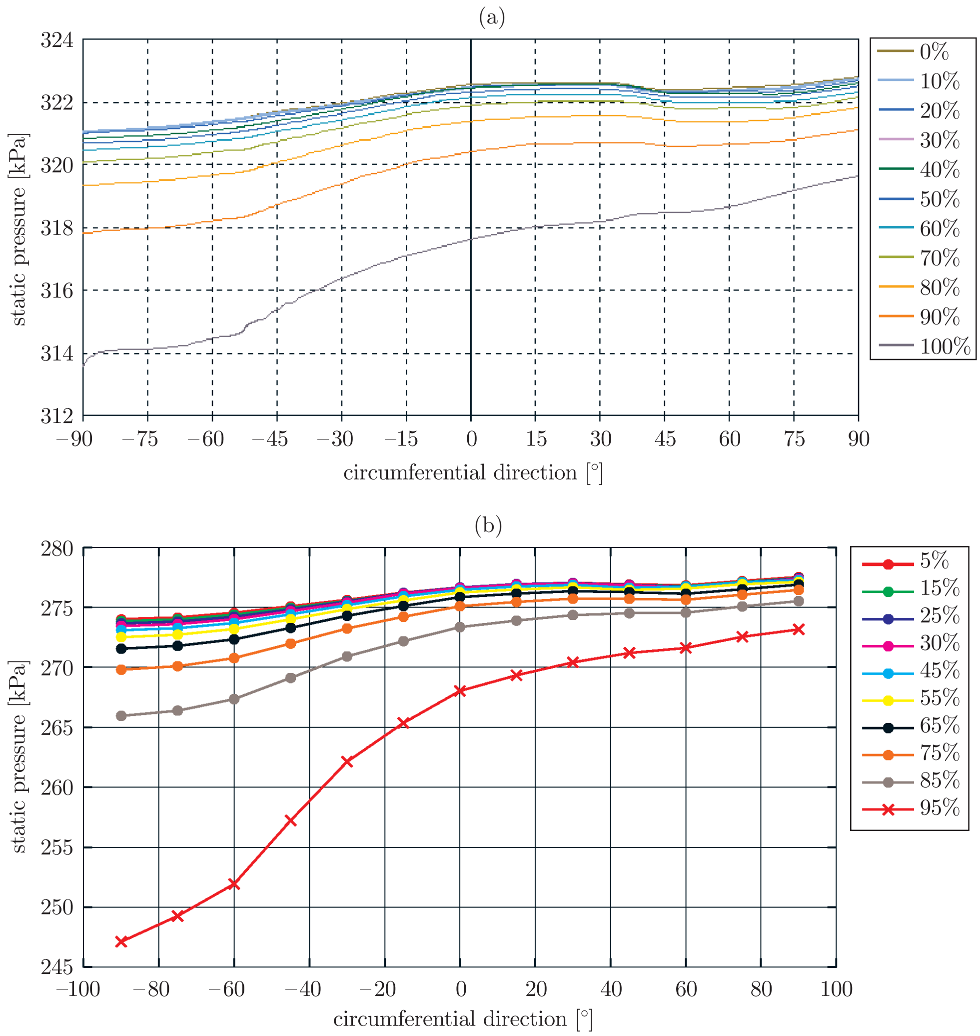

Figure 3. Pressure distribution behind the rotor blades: (a) for regime 1 ($m_{\rm extr} = 5.42$kg/s); (b) for regime 4 ($m_{\rm extr} = 15.38$kg/s)

regimes were obtained from thermodynamical calculations presented in Table 1, the first regime being nominal.

The pressure distribution behind the rotor blades is shown for regime 1 ($m_{\rm extr} = 5.42$kg/s, see Table 1) in Figure 3a and for regime 4 ($m_{\rm extr} = 15.38$kg/s) in Figure 3b. The upper curve shows the circumferential pressure distribution at the blade's root (0%), the lowest curve – that at its top (100%). The circumferential angle of $-90°$ concerns the lower part of the extraction channel (between extraction tubes, Figure 2), while that of 90° concerns the upper part of the channel (see Figure 2). The boundary conditions for unsteady blade forces' calculations are flow parameters p_0, T_0 at the stage inlet (see Table 1), incidence angle in the stator blade equal to zero, and the pressure behind the rotor blade for the various circumferential angles.

3. Unsteady forces of rotor blades in the stage with steam extraction

Numerical calculations of a 3D transonic flow of an ideal gas through turbomachinery blade rows moving one in relation to another have been carried out.

An ideal gas flow through mutually moving stator and rotor blades with periodicity on the whole annulus is described by the unsteady Euler conservation equations, integrated using the explicit monotonous finite-volume difference scheme of Godunov-Kolgan and a moving hybrid H-H grid [1]. The numerical calculations presented below were carried out for the stage of the turbine with cylindrical blade of length 0.1341 m. The number of stator blades was 46, that of rotor blades was 126.

A cross-sections of the stage and the stator and rotor blades are presented in in Figure 4.

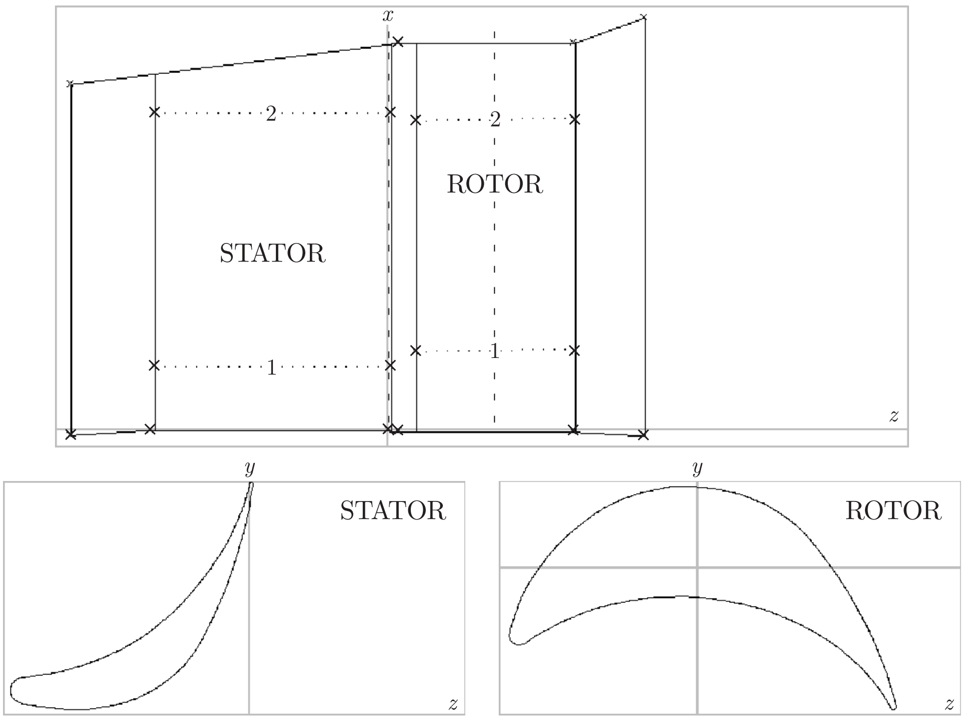

Figure 4. Cross-sections of the stage (top) and the stator and rotor blades (bottom)

The unsteady axial, radial and circumferential forces of the rotor blade were calculated for four steam extraction conditions ($m_{\mathrm{extr}} = 5.42$, 8.41, 11.51, 15.38 kg/s, see Table 1). The unsteady axial forces in the rotor blade's top cross-section for $m_{\mathrm{extr}} = 5.42$ kg/s are presented in Figure 5. The high frequency excitation (2300 Hz) was equal to 10% of the steady force, $A_0 = 32.04$ N.

The low frequency excitation due to non-uniform pressure distribution was 1.28% for 50 Hz and 0.39% for 100 Hz (see Figure 5b).

The unsteady axial forces acting in the top cross-section at $m_{\mathrm{extr}} = 15.38$ kg/s are presented in Figure 6. The high frequency excitation (2300 Hz) was equal to

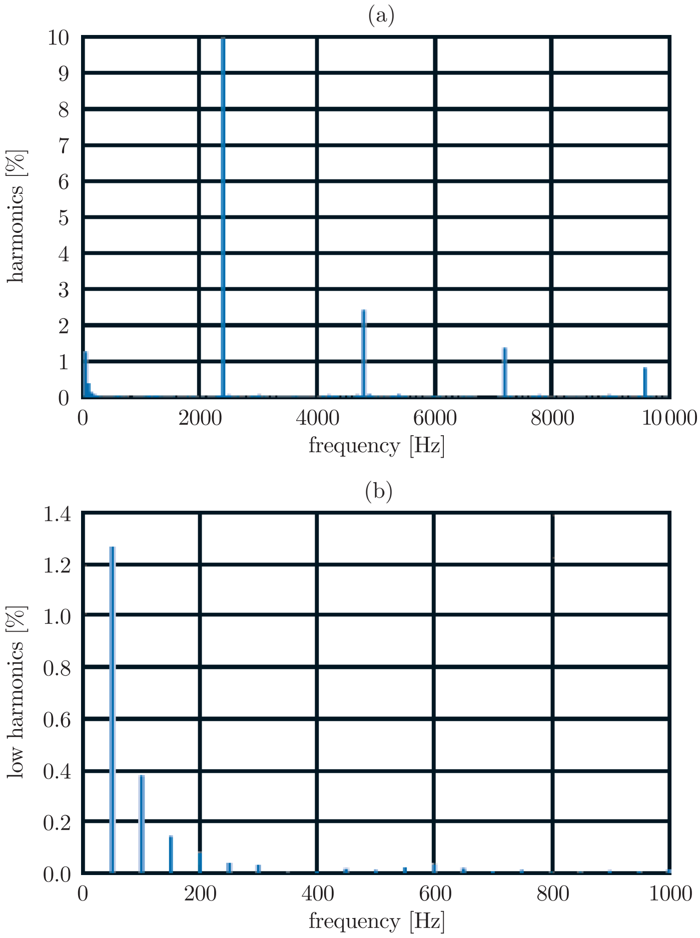

Figure 5. Harmonics (a) and low harmonics (b) of the unsteady axial force (regime 1, top cross-section, $m_{\mathrm{extr}} = 5.42\,\mathrm{kg/s}$, $A_0 = 32.044\,\mathrm{N}$)

10.4% (4.38N) of the steady force of 42.05N. The low frequency excitation caused by non-uniform pressure distribution was 3.2% (1.33N) for 50Hz and 0.7% for 100Hz (see Figure 6b).

Additional unsteady harmonics appeared in the steam extraction mass flow of $m_{\mathrm{extr}} = 15.38\,\mathrm{kg/s}$ for 300, 350, 400, 450, 500, 750, 800, 850Hz at 100% and 90% of the blade's length. For example, the low frequency axial unsteady forces (see Figure 6b) calculated at 100% of the blade's length were 0.8% of the steady part for 300Hz, 1.7% for 350Hz, 1.9% for 400Hz, 1.4% for 450Hz, 0.7% for 500Hz, 0.3% for 600Hz, 0.7% for 750Hz, 0.9% for 800Hz and 0.7% for 850Hz.

The unsteady axial, radial and circumferential force harmonics of the rotor are presented in Tables 2–4 for four steam extraction conditions ($m_{\mathrm{extr}} = 5.42$, 8.41, 11.51, 15.38kg/s, see Table 1).

- When the steam extraction mass flow increases, the steady components of the unsteady forces increase too.

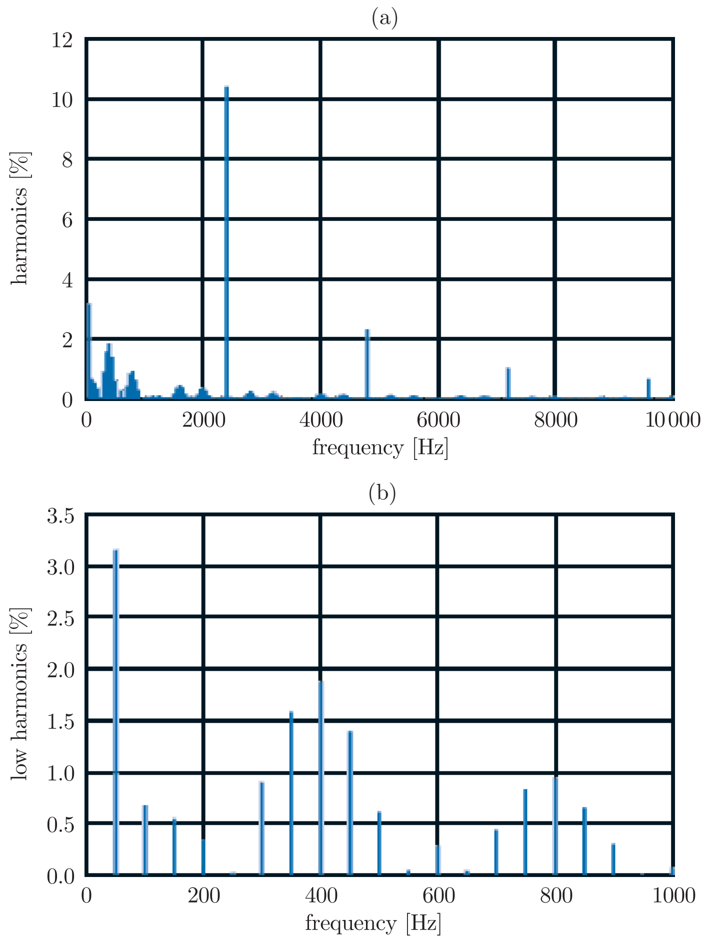

Figure 6. Harmonics (a) and low harmonics (b) of the unsteady axial force (regime 4, top cross-section, $m_{\mathrm{extr}} = 15.38\,\mathrm{kg/s}$, $A_0 = 42.0481\,\mathrm{N}$)

- The greatest change in the unsteady forces is observed at 100% and 90% of the blade's length near its top.
- The largest harmonics appear 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 equals 12–13% of the steady moment; for the circumferential forces, it is 82–62% of the steady part.
- The asymmetrical distribution of pressure behind the rotor blades creates the low frequencies components of the unsteady forces for the 50–900 Hz harmonics.
- In the case of $m_{\mathrm{extr}} = 5.42$, 8.41, 11.51 kg/s, low frequency excitation appears at 50, 100 and 150 Hz. The largest one equals 4% of the steady part (see Tables 2–4).
- For $m_{\mathrm{extr}} = 15.38\,\mathrm{kg/s}$, low frequency excitations are the strongest for 50 Hz and equal 3.4–4% of the steady part. Additional harmonics appear at frequencies of 300–850 Hz.

The natural frequencies of the blade and bladed discs were calculated. Those of the bladed discs are shown in Figure 7 in dependence of the number of nodal diameters. The modes of bladed disc are classified by analogy with axisymmetric modes, which are mainly characterized by nodal lines lying along the structure's diameters and having constant angular spacing. There are zero ($k=0$), one ($k=1$), two ($k=2$) or more ($k>2$) nodal diameter bending or torsional modes. Series 1 is associated with the first natural frequency of a single cantilever blade, Series 2 is associated with the second natural frequency of a single cantilever blade and so on. The red rectangle shows the excitation of nozzles.

The natural frequency of the bladed disc with one nodal diameter is close to 400Hz, so the blades can work at resonance in the case of the steam mass flow extraction of $m_{\mathrm{extr}}=15.38$kg/s.

Table 2. Harmonics of the unsteady torsion moments for mass flow steam extraction ($m_{\mathrm{extr}}=5.42$, 8.41, 11.51, 15.38kg/s) at the top blade cross-section

m_{extr} [kg/s]	Steady part [N]	50Hz [%]	100Hz [%]	150Hz [%]	200Hz [%]	400Hz [%]	600Hz [%]	800Hz [%]	2300Hz [%]	4600Hz [%]
5.42	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
15.38	0.4427	3.4	0.6	0.5	0.4	2.01	0.4	1.1	13	0.6

Table 3. Harmonics of the unsteady circumferential forces for mass flow steam extraction ($m_{\mathrm{extr}}=5.42$, 8.41, 11.51, 15.38kg/s) at the top blade cross-section

m_{extr} [kg/s]	Steady part [N]	50Hz [%]	100Hz [%]	150Hz [%]	200Hz [%]	400Hz [%]	600Hz [%]	800Hz [%]	2300Hz [%]	4600Hz [%]
5.42	11.83	2.35	0.3	0.2			0.5		82	2
8.41	14.018	3.5	1.1	0.3					74	2.8
11.51	16.34	4.64	1.45	0.5					68	2.49
15.38	19.23	4.06	0.9	0.8	0.5	2.07	0.3	1.04	62.56	1.98

Table 4. Harmonics of the unsteady axial forces for mass flow steam extraction ($m_{\mathrm{extr}}=5.42$, 8.41, 11.51, 15.38kg/s) at the top blade cross-section

m_{extr} [kg/s]	Steady part [N]	50Hz [%]	100Hz [%]	150Hz [%]	200Hz [%]	400Hz [%]	600Hz [%]	800Hz [%]	2300Hz [%]	4600Hz [%]
5.42	32.04	1.28	0.4	0.15			0.5		10	2.5
8.41	35.05	2.3	0.69	0.3					10.4	2.45
11.51	38.19	3.43	1.04	0.39					10.4	2.38
15.38	42.05	3.16	0.7	0.5	0.3	1.87	0.3	0.94	10.41	2.33

4. Conclusions

The unsteady forces of a rotor blade were calculated for four steam extraction conditions. For the maximal steam extraction mass flow of $m_{\mathrm{extr}}=15.38$kg/s, the 10$^{\mathrm{th}}$

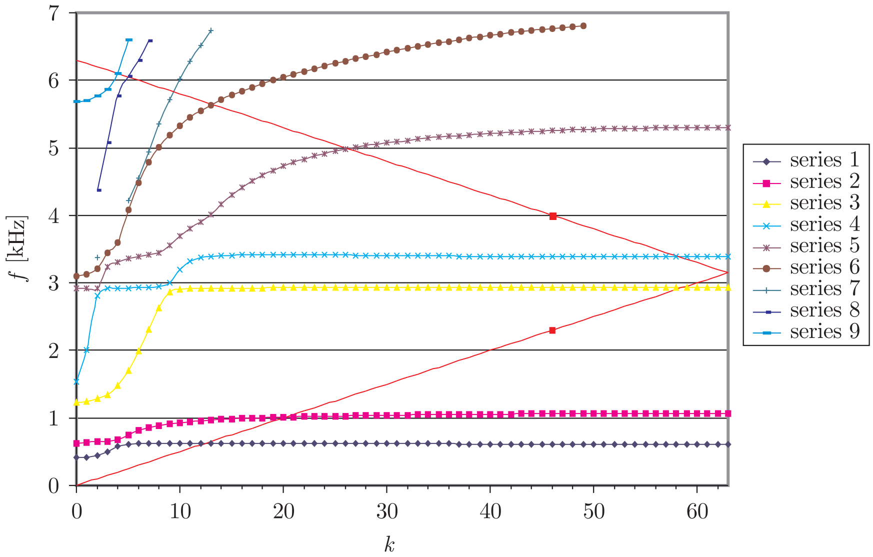

Figure 7. Interference diagram of the bladed disc

harmonic (400 Hz) of the axial force and moment was found to be close to the first natural frequency of the bladed disc with one nodal diameter. Thus, steam extraction can be a cause of the blade's failure.

The disadvantage of this calculation is that the pressure distribution in the extraction area 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 the circumferential direction. Unsteady calculations of the stage with stator and rotor blades and steam extraction channels remain to be performed.

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