HIGH-FREQUENCY PULSATIONS MODELING

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This paper focuses on mathematical modeling of high frequency pulsations in pump turbines, which are the source of high-cycle continuous stress of the spiral casing cover, wicket gates and runner. The problem is solved by transfer matrix method. F-A char. and MS Excel programs have been used for data processing. Final results are presented by graphs of frequency-amplitude characteristic depending on running speed of motor, and plots of the pressure mode shape in selected phase shifts. Modeled variants are those of wheels with 7; 9 blades and 7 blades and 7 splitter-blades. The number of distributing blades is 20.

Keywords: *turbine, pump, high-frequency pulsations*

1. Introduction

Paper presents results of pressure pulsations measurement for 3 designs of the runner. Design with 7 and 9 blades and with 7 long blades and 7 splitter blades. Comparison of measured and computed eigen mode shapes both for pump and turbine modes of operation are presented.

2. Comparison of measured and computed pressure pulsations

Mathematical model for high-frequency pulsations computation is composed of pipes and nodes. Scheme of the model is in Figs. 1 and 2.

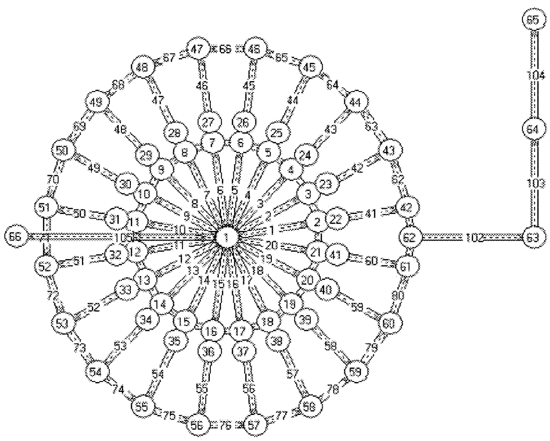

Fig.1: Model of a turbine with a feeder and a draft tube

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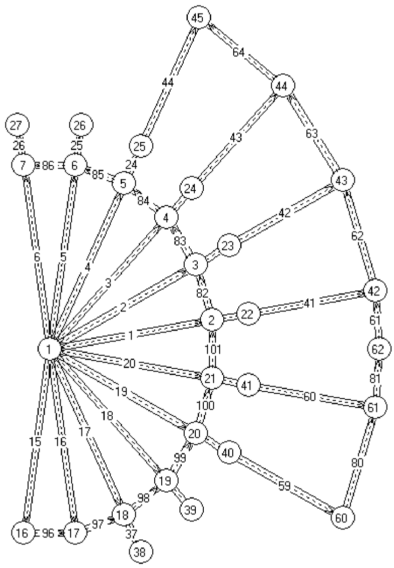

Fig.2: Detailed turbine model of Fig. 1

Description mathematical model:

Node:

1	simulates an area under the impeller,
2–21	simulates a fluid entry into the impeller during the turbine flow,
22–41	simulate a fluid exit from distributor (VLS),
42–62	simulate outlets from the spiral into the stator blades section,
65	simulates an edge condition of the upper reservoir, a constant pressure is entered there,
66	simulates an edge condition of the lower reservoir, a constant pressure is entered there.

There is a flow edge condition entered in other nodes, so the continuity relation must be applicable.

Tube:

1–20	simulate the impeller,
21–40	simulate pressure jump on the perimeter of the vaneless space (VLS),
41–60	simulate the spiral; each has a bigger area corresponding with the area of the spiral in the particular area,
61	simulates spiral entrance,
81	simulates spiral nose.

Following spiral pipes in the direction of turbine flow (clockwise) have diminishing cross-section.

41–61	simulate the space between the blades of the distributor (VLS),
82–101	simulate the space between the splitter blades,
102–104	simulate the feeder,
105	simulate sucking part of the turbine.

Table 1 shows parameters of individual pipes of the model. The first column shows the pipe index, the second one shows the area of the pipe, the third shows the length, the fourth shows the speed of the sound, the fifth shows the linearized resistance on the entrance, the sixth shows the linearized resistance when leaving the pipe and the seventh shows the second viscosity. The second viscosity shows frequency dependent energy dissipation. This should be considered together with the second viscosity.

The values were obtained from the drawing documentation of the model and from measuring the model PVE Dlouhé Stráně.

Tube	Surface	l	v	R [1]	R [2]	ζ
	m^2	m	m/s	$Pa\,s/m^3$	$Pa\,s/m^3$	$m^2\,s$
1–20	0.001	0.52	972	4.6796078	0	4.22
41–60	0.0015	0.2	991	0	9.97E–10	6.66
82–101	0.0014079	0.07	880	0	0	5.13
61	0.00008	0.0622	1033	6.513E–07	0	25.00
62	0.00229	0.126	1033	0	0	25.00
63	0.00385	0.129	1033	0	0	29.20
64	0.00554	0.131	1066	0	0	29.50
65	0.00754	0.133	1066	0	0	28.40
66	0.00882	0.134	1066	0	0	29.80
67	0.00985	0.135	1066	0	0	29.90
68	0.0113	0.136	1066	0	0	30.00
69	0.0133	0.138	1111	0	0	30.00
70	0.015	0.139	1111	0	0	29.90
71	0.0167	0.14	1111	0	0	29.60
72	0.0191	0.141	1111	0	0	29.60
73	0.0206	0.143	1111	0	0	28.60
74	0.0227	0.144	694	0	0	25.10
75	0.0249	0.145	694	0	0	28.10
76	0.0266	0.146	694	0	0	27.60
77	0.029	0.147	694	0	0	29.90
78	0.0308	0.148	694	0	0	13.10
79	0.0327	0.149	773	0	0	14.10
80	0.0346	0.15	773	0	0	15.10
81	0.0346	0.0748	773	0	0	26.20
102	0.0707	70	1500	0	0	10.00
103	0.0962	10	500	0	0	15.00
104	0.332	35	500	0	0	15.00
105	0.332	35	1400	0	0	10.00

Tab.1: Arithmetic means

3. Evaluation

The pressure pulsation values are described in a dimensionless parameter defined as the head pulsation divided by turbine drop.

In this equation:

$$\frac{\Delta H}{H} = \frac{\dfrac{\sum\limits_{i=1}^{n} p_{\mathrm{A}i}}{n\,g\,\varrho}}{H}\,100\,\%$$

VLS vaneless space (8–16), CASE spiral (1–7), n [1] number of pressure points, $p_{\mathrm{A}i}$ [Pa] pressure amplitude with the index i, H [m] head, g [m.s^2] gravity acceleration.

Pressure pulsation values $p_{\mathrm{A}i}$ were calculated after DFT of the time pressure record.

		T CASE	T VS	P CASE	P VS
7 blades	speed fr.	0.18	0.3	0.35	0.5
	blade fr.	0.1	3.9	0.13	1.4
	3× speed fr.	0.6	1.3	0.35	0.6
9 blades	speed fr.	0.1	0.1	0.8	0.85
	blade fr.	0.1	5	0.15	1.4
	3× speed fr.	0.14	2.1	0.04	0.5
7 blades and 7 semi blades	speed fr.	0.35	0.4	0.2	0.25
	blade fr.	0.09	4	0.07	2.5
	3× speed fr.	0.15	1.65	0.15	0.3

Tab.2: Evaluating the graphs of the pulsation in the percentage of H

4. Calculating the cycle for each version of the wheels in the turbine and pump mode

Lines show the calculated pressure pulsation. Small circles show the measured pressure. Drive using the pressure jump is assumed. The versions of 7 blades, 7 blades and 7 splitter blades are in the shape of rotating excenter Fig. 3, rotating together with the rotating impeler. 9 blade version looks like a rotating biscuit Fig. 4, rotating against the direction of the rotating impeler.

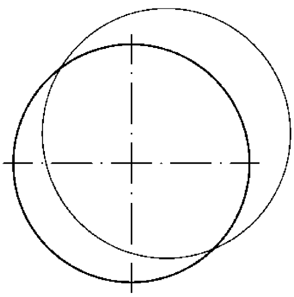

Fig.3: Rotating excente

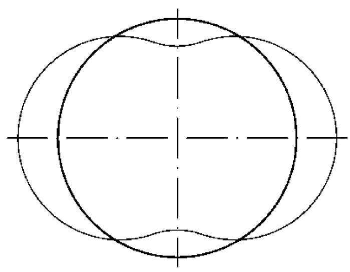

Fig.4: Rotating biscuit

5. Conclusion

Tab. 2 summarizes average values for spiral casing and vaneless space for turbine and pump mode of operation.

The original 7-bladed wheel brings about high-frequency pulsations on 21 fold of rotation frequency in fixed coordinate system. Regarding the generation in rotating coordinate system, on 20 fold of rotation frequency. Pressure pulsations in MLP are on 0.9 % of the head and in spiral on 0.5 % of the head.

The 9 bladed wheel brings about high-frequency pulsations on 18 fold of rotation frequency in fixed coordinate system. Regarding the generation in rotating coordinate system, on 20 fold of rotation frequency. Pressure pulsations in (VLS) are on 1.2 % of the head and in spiral on 0.06 % of the head.

The 7 bladed and 7 splitter-bladed wheel brings about high-frequency pulsations on 21 fold of rotation frequency in fixed coordinate system. Regarding the generation in rotating coordinate system, on 20 fold of rotation frequency. Pressure pulsations in MLP are on 0.9 % of the head and in spiral on 0.15 % of the head.

Absolute oscillation shapes for individual wheel versions

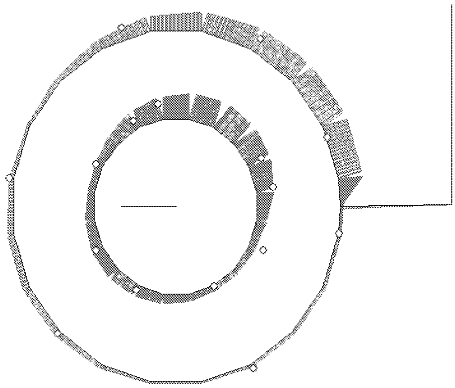

Fig.5: Turbine 7 blades at frequency 490.10 Hz

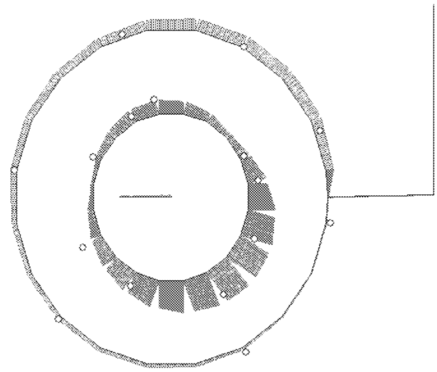

Fig.6: Pump 7 blades at frequency 493.29 Hz

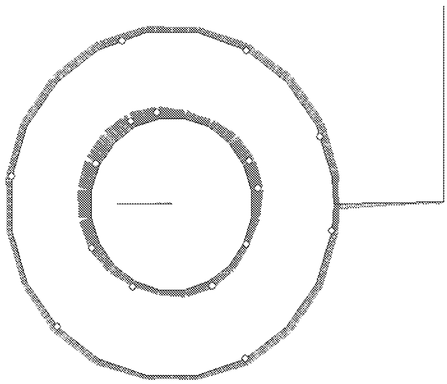

Fig.7: Turbine 9 blades at frequency 420.23 Hz

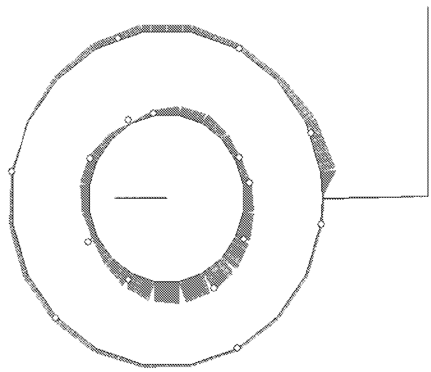

Fig.8: Pump 9 blades at frequency 424.30 Hz

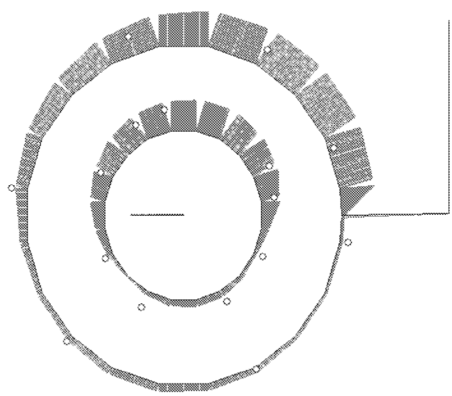

Fig.9: Turbine 7 blades and 7 splitter blades at frequency 386.86 Hz

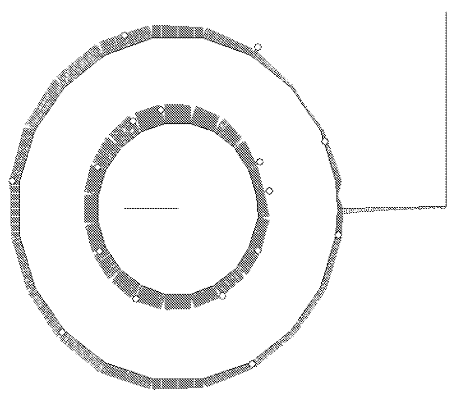

Fig.10: Pump 7 blades and 7 splitter blades at frequency 375.15 Hz

Pressure oscillation in individual phases of 7 blades and 7 splitter blades – turbine

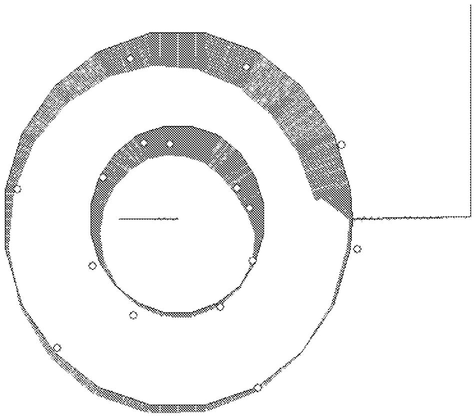

Fig.11: Pressure for $\phi = 0°$

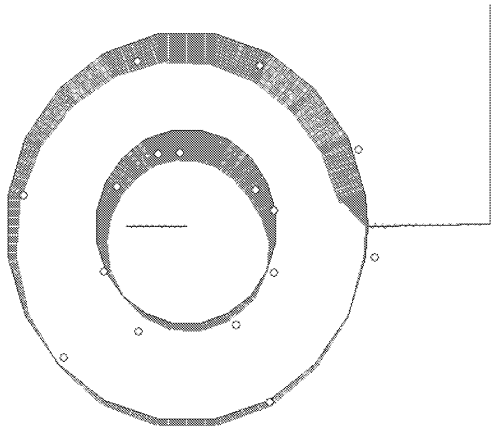

Fig.12: Pressure for $\phi = 60°$

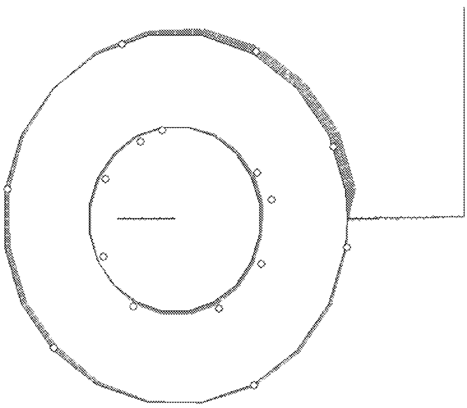

Fig.13: Pressure for $\phi = 120°$

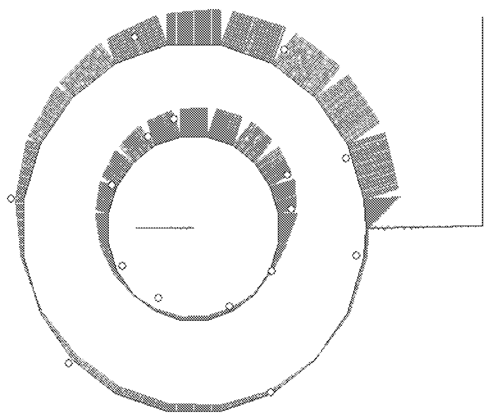

Fig.14: Pressure for $\phi = 180°$

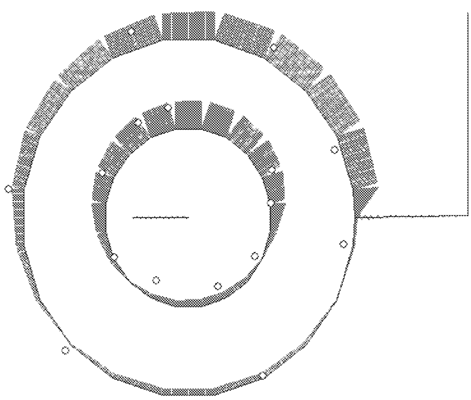

Fig.15: Pressure for $\phi = 240°$

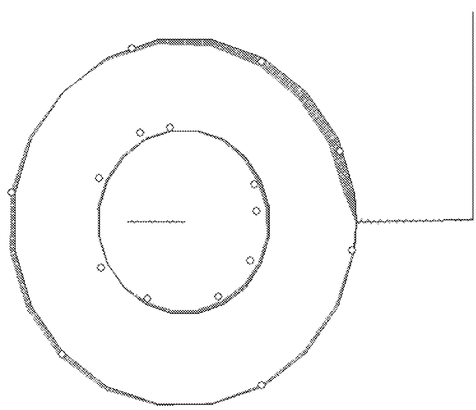

Fig.16: Pressure for $\phi = 300°$

Pressure oscillation in individual phases of 7 blades and 7 splitter blades – pump

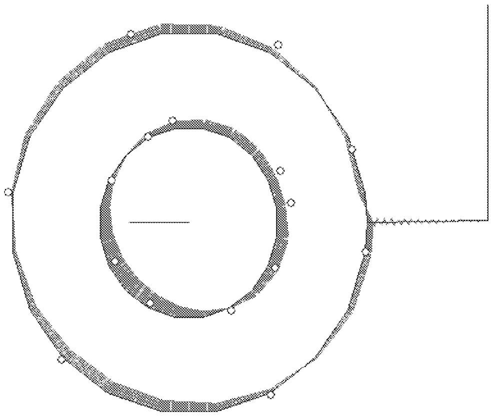

Fig.17: Pressure for $\phi = 0°$

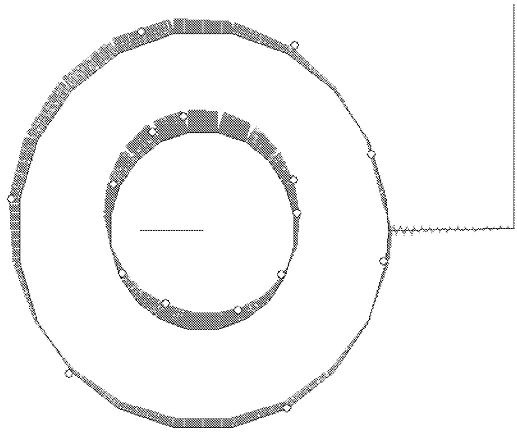

Fig.18: Pressure for $\phi = 60°$

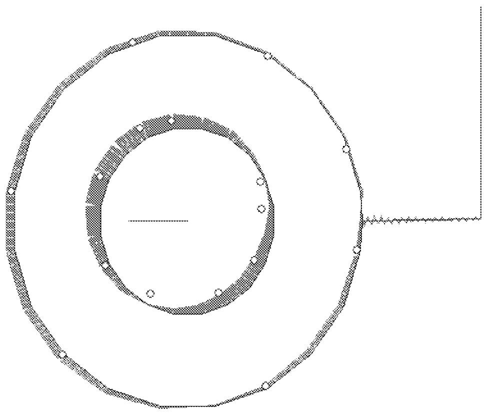

Fig.19: Pressure for $\phi = 120°$

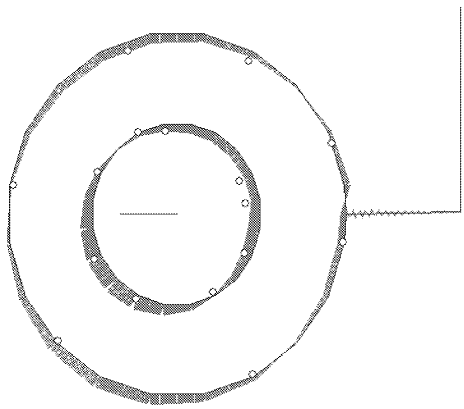

Fig.20: Pressure for $\phi = 180°$

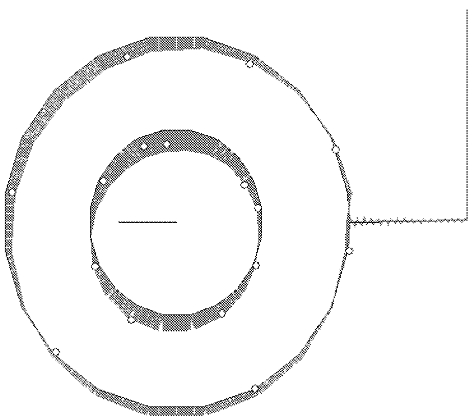

Fig.21: Pressure for $\phi = 240°$

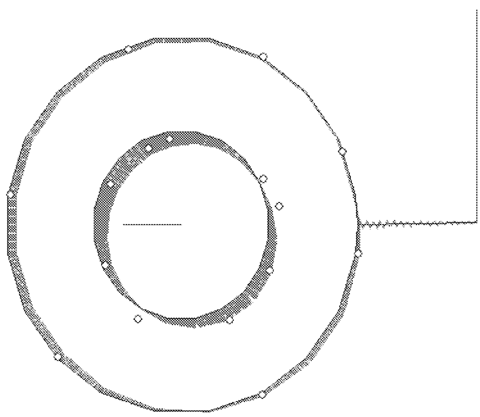

Fig.22: Pressure for $\phi = 300°$

The best design from this point of view is 7 runner blades, then 7 runner blades with 7 splitter blades and finally 9 runner blades. The best design in case that pulsations in spiral casing are the most problematic is 7 runner blades with 7 splitter blades, then 9 runner blades and finally 7 runner blades. Design with 7 runner and 7 splitter blades has 1.7 times lower pressure amplitude than design with 7 runner blades only.

The Chapter 4 shows the comparison of calculated and measured values of the pulsations. It is shown in the form of the oscillation shape. The average correlation coefficient is 0.75 .

The above shows that we are able to predict the frequency of high frequency pulsations with regard to the revolutions. We can also distinguish a stationary wave from the running wave with regard to the forced shape of the oscillation. We also can describe the shape of the pressure oscillation and calculate the amplitude of pressure pulsation within 35 % accuracy.

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