Comments to vibrations and pressure oscillations induced by the rotor stator interaction in a hydraulic turbine

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Abstract

Both high frequency pressure oscillations and high frequency vibrations can arise in a water turbine due to the rotor stator interaction. Vibrations of several hydraulic turbine components such as rotor, runner, guide vanes, head cover etc. can be excited. In this paper comments focused on lateral rotor vibrations, spiral case pressure oscillations as well as high head pump turbine runner vibrations are presented. The comments are based on the theory as well as on the praxis.

Some examples from pumped storage power plants equipped with pump turbines are also presented.

Keywords

Rotor stator interaction, vibrations, pressure oscillations.

1. Introduction

The rotor stator interaction (RSI) generates an unsteady pressure field in a hydraulic machine. Using some simplifications this field can be described such as one having diametrical nodes. The number of the diametrical nodes is given by a simple criterion derived by several authors many years ago [1], [3], [5], [6].

$$mz_R - nz_S = k \qquad (1)$$

Where:

z_R ......number of rotor blades

z_S ......number of stator vanes

$m = 1,2,3$ arbitrary integer

$n = 1,2,3$ arbitrary integer (usually $n = 1$)

k .... number of diametrical nodes

Frequency in the stationary system: $$f_S = mz_R f_O \qquad (2)$$

Frequency in the rotating system: $$f_R = nz_S f_O \qquad (3)$$

Where:

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1.1 Stationary system

From the point of pressure oscillations as well as rotor vibrations the stationary system is decisive. Dynamic component (excitation) of the pressure along the runner diameter is given by the following formula

$$p_{(\varphi)} = A_b \cos(\omega t - k\varphi) \qquad (4)$$

where

A_b...... amplitude of the excitation

ω....... circural frequency in the stationary system

t........ time

$2k$....... number of pressure nodes at given diameter

φ....... angle in the stationary system

For k = 0 the pressure field is of the pulsating nature. For k ≠ 0 the pressure field is of the rotating nature. In the case k > 0 (k< 0) the field rotates in the same (opposite) sin as the runner.

The Eq. (4) can be expressed in the following form

$$p_{(\varphi)} = A_b \cos k\varphi \cdot \cos \omega t + A_b \sin k\varphi \cdot \sin \omega t \qquad (5)$$

It means that a rotating excitation can be taken as two nonrotating (pulsating) components – the cos-component as well as sin-component.

1.2 Rotating system

From the point of runner vibrations the rotating system is decisive. Under the same frequency several pressure fields having different diametrical nodes should be taken into consideration. To keep the problem simple enough only two excitation fields are taken into consideration. The dynamic component of the runner periphery pressure is given by the following formula

$$p_{(\Phi)} = A_b \cos(\Omega t - k_1 \Phi) + A_b \cos(\Omega t - k_2 \Phi) \qquad (6)$$

Where:

Ω …….. circural frequency in the rotating system

Φ …….. angle in the rotating system

The numbers of diametrical nodes are taken as follows

$$m_1 z_R - n z_S = k_1$$
$$m_2 z_R - n z_S = k_2$$
$$m_2 = m_1 + 1$$

The Eq. (6) can be expressed in the following form:

$$p_{(\Phi)} = A_b[(\cos k_1\Phi + \cos k_2\Phi)\cos \Omega t + (\sin k_1\Phi + \sin k_2\Phi)\sin \Omega t] \qquad (7)$$

Similarly to Eq. (5) the rotating excitation can be taken as two pulsating components – the cos-component as well as sin-component.

2. Rotor lateral vibrations (k = ± 1)

For k = ± 1 both the radial force and the bending moment acting on the runner force lateral rotor vibrations. The corresponding excitation rotates forward (k = +1) or backward (k = -1). It is evident that a rotor with a rated speed lying below the first critical speed passes, due to hydraulic RSI excitation, through several resonances during start into the turbine operations.

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2.1 Praxis

Combinations z_S/z_R leading to k = ±1 by m = 1 and n = 1, such as 18/17 by hydraulic turbines or 8/7 by storage pumps, are well known as “bad” ones from the past and are not discussed here. However, in the case of pump turbines combinations leading to k = ±1 by m > 1 and n = 1 are frequently used. For example

$z_S/z_R = 20/7$; m = 3; n = 1; k = +1; $f_S = 21\ f_0$
$z_S/z_R = 19/9$; m = 2; n = 1; k = -1; $f_S = 18\ f_0$

How can this feature affect the rotor vibration of the unit?

2.2 Steady state operation

Usually no problems with high frequency rotor vibrations can be observed at rated speed. However, the excitation frequency is high. For example by a nominal speed 428,6 rpm and z_S/z_R = 20/7 the excitation frequency is 150 Hz. Due to the high frequency some problems with guide bearing vibrations measured at vibration velocity in mm/s (see ISO standard 10816-5), could appear. Fig. 1 shows a resonance peak of a 300 MW pump turbine rotor. The resonance peak was determined by varying the speed at no load operation. The peak is located in the vicinity of the rated speed. The high frequency vibration dominates in this unit above all on the lower generator bearing [10]. At full load turbine operation the amplitude at rated speed reaches 3 mm/s – see [8].

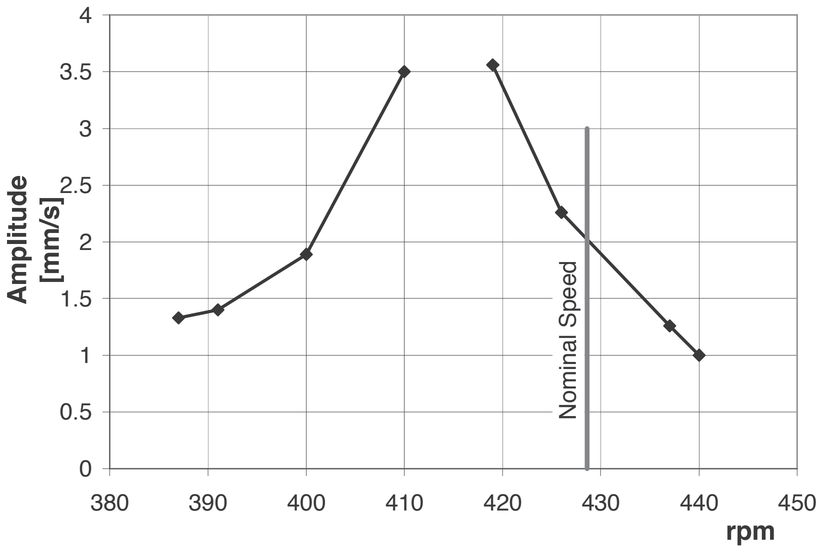

Fig. 1. *Amplitude of high frequency vibration of the motor-generator lower guide bearing at no load operation. The frequency corresponds to 21 times the rotating frequency. The pump turbine has the guide vane to rotor blade combination 20/7.*

It is interesting that small loops are clearly expressed [8] in the shaft orbit. The number of these small loops is 20 – see Fig.2. This type of the shaft movement at the bearing does not cause any problem.

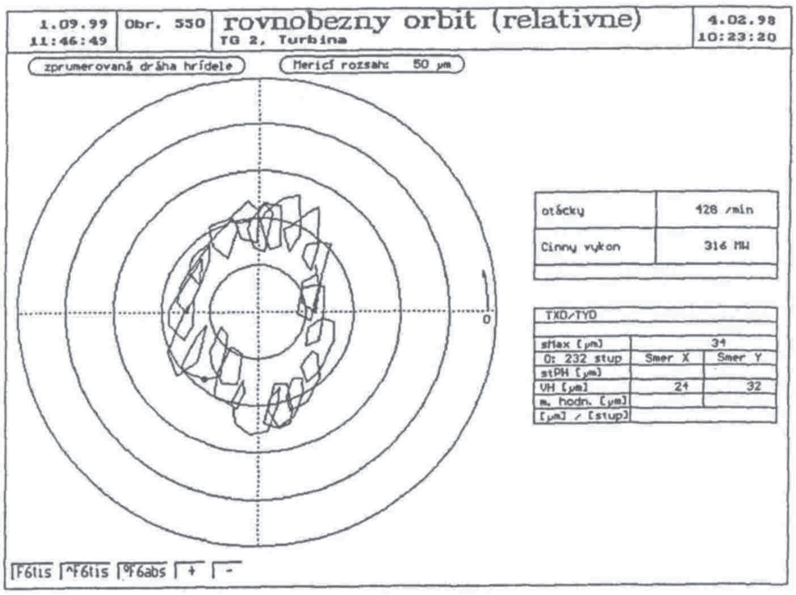

Fig. 2. *Shaft orbit at the 300 MW pump turbine guide bearing [8].*

2.3 Start into the turbine operation

As mentioned above, during running up of a pump turbine into the turbine operation the rotor is passing trough several resonance regions. Remarkable increase of rotor vibration amplitudes is usually expressed at the first resonance peak only. The rotor accelerates from the still stand and passes the first resonance region relatively slowly. At the beginning the running up of the turbine is therefore rough enough. Then the rotor speed increases more rapidly and passing trough the other resonance regions is relatively smooth. Some examples

Pump turbine 20/7 H = 120 m, D = 6 m, P = 180 MW, n = 166,7 rpm

The rotor is passing the first resonance frequency 9,0 Hz at 15% nominal speed.

Pump turbine 20/7 H = 500 m, D = 4,57 m, P = 300 MW, n = 428,6 rpm

The rotor is passing the first resonance frequency 11,5 Hz at 7,6% nominal speed.

3. High frequency pressure oscillations (k = ± 1, ± 2)

It is well known that the high frequency pressure oscillations are present in both the vaneless space and the spiral case. The situation in the penstock is usually not exactly known because it is not easy to measure any pressure oscillation along the whole penstock. In spite of this lack of measured data there are criteria, procedures as well as software enabling to predict or compute the pressure oscillation in the hydraulic system of the turbine.
However, the main problem is how to model the whole hydraulic system.

Let us assume a one-dimensional continuous system consisting of a spiral case and a simple penstock. The excitation is propagating from the vaneless space into the spiral case.
There are two limit cases how to model the hydraulic system.

3.1 Standard dynamic system

Due to the superposition of traveling and reflecting waves standing waves arise in the system. The system behaves like a standard vibrating one having negligible low damping. There are natural frequencies (eigen frequencies) and normal (eigen) modes. There are fixed nodes of the pressure as well as of the discharge. The phase lag between the pressure and discharge is 90°.
The standing waves can be excited by a standing excitation. In our case it means the sin or cos component of the pressure field rotating in the vaneless space - see Eq. (5). Under resonance condition the amplitude of pressure oscillation can reach dangerous values.

3.2 Semi-infinite system

The system behaves like a semi-infinite one. It means a system where reflections are not returned from the end. (In the praxis it can be a long system. Due to damping in the system waves do not return back. In the theory the so called no reflecting end of the system can be applied.) There are no natural frequencies, no normal modes, no resonance peaks. Only traveling waves are propagating trough this system with the wave speed (celerity). There is not any phase lag between the pressure and the discharge.
The traveling waves are induced due to rotating pressure field – see Eq. (4).
It is obvious that the amplitude of the traveling waves reaches its maximum if the length of the traveling wave λ corresponds to the length λ_b of the rotating pressure field at a given diameter D. These lengths λ and λ_b are definite by the following formulas:

$$\lambda = \frac{a}{f} \qquad \lambda_b = \frac{\pi D}{|k|} \tag{8}$$

Where:

a …….. wave speed (celerity)

The condition $\lambda = \lambda_b$ is satisfied if

$$\frac{a}{f} = \frac{\pi D}{|k|} \tag{9}$$

Using Eq. (2) as well as Eq. (1) we can get

$$\frac{a}{m z_R f_o} = \frac{\pi D}{|m z_R - n z_S|} \tag{10}$$

This equation can be modified into the following form

$$\frac{z_R}{z_S}\left(1 \pm \frac{\pi D f_o}{a}\right) = \frac{n}{m} \tag{11}$$

It is well known Nechleba's criterion (specifying disadvantages combinations) derived by quit another procedure [2]. Other authors have derived similar criterions [4], [5]. Based on the procedure presented the criterions seem to be valid above all at the semi-infinite systems.

3.3 Praxis

Based on hydraulic model measurement results [9] it can be assumed that a standard hydraulic model stand circuit behaves like a system in which the standing waves predominate. There are resonance peaks – see Fig. 3 – and a fixed pressure node in the spiral case. The amplitude measured at the tongue reaches surprising high value.

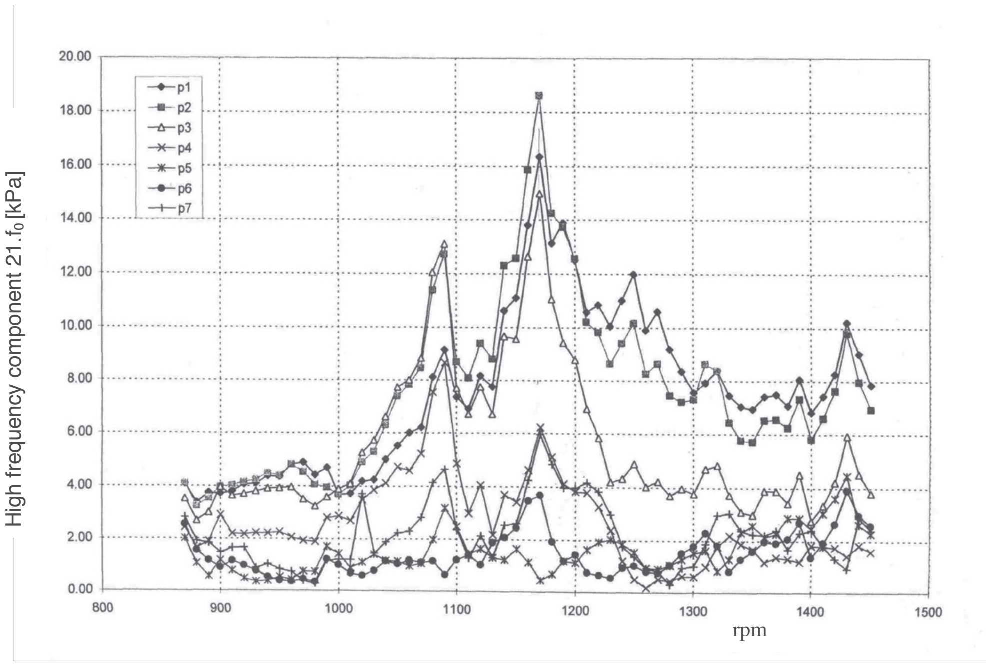

Fig. 3 *Amplitude of high frequency pressure oscillations measured along the spiral case of a hydraulic model [9]. Transducer p1 is located at the tongue, p7 at the inlet. The pump turbine has 20 guide vanes and 7 runner blades. Turbine mode of operation.*

At the prototype corresponding to the model in question only pressure oscillations at the spiral case inlet has been measured. The amplitude of the high frequency pressure oscillations was very low, comparable with the value measured on the model (directly without any stepping!). How to explain it? Is a node of the pressure oscillation at the prototype spiral case inlet? Are there traveling waves on the prototype instead of the standing ones on the model? Measurement mistakes? Without a comprehensive pressure oscillation measurement this question cannot be answered.

To the authors there is known only one comprehensive field measurement of high frequency pressure oscillations from which the type of waves can be defined. It was published by Ohura at al. 1990 [7]. The high head pump turbine (20/6, k = -2, nominal speed 514 rpm) was operated at no load operation from 350 rpm up to 560 rpm. The pressure oscillations were measured at 12 points located on the spiral case circumference. The results show clearly that up to 365 rpm standing waves predominate in the spiral case. However, in the region 415 to 560 rpm traveling waves predominate in the spiral case. It means that the hydraulic system of the prototype behaves in the nominal speed region like a semi-infinite one.

The main problem of high frequency pressure oscillations seems to be the determination of the damping in the whole hydraulic system.

4. High head pump turbine runner vibrations

Runners of high head pump turbines are sensitive to vibrations and fatigue cracks can develop above all in the crown [6]. It is possible to compute the runner response on the RSI excitation. Usually such computation must be several times repeated by dimensioning the runner to find an appropriate solution. Some parameters such as damping as well as added mass are not exactly known. It makes the computation results a bit questionable. Therefore some theoretical considerations before any computation seem to be advantageous.

A pump turbine runner consists of two disks (crown and band) connected by blades.
It is well known that the normal modes of the runner can be grouped in the following way:

- Single-disk-like modes. The runner behaves like a single disk having diametrical nodes. The crown and the band are vibrating in-phase.
- Double-disks-like modes. The crown and the band are not vibrating in-phase.

An example of two different normal modes (computation results) is presented in Fig. 4.

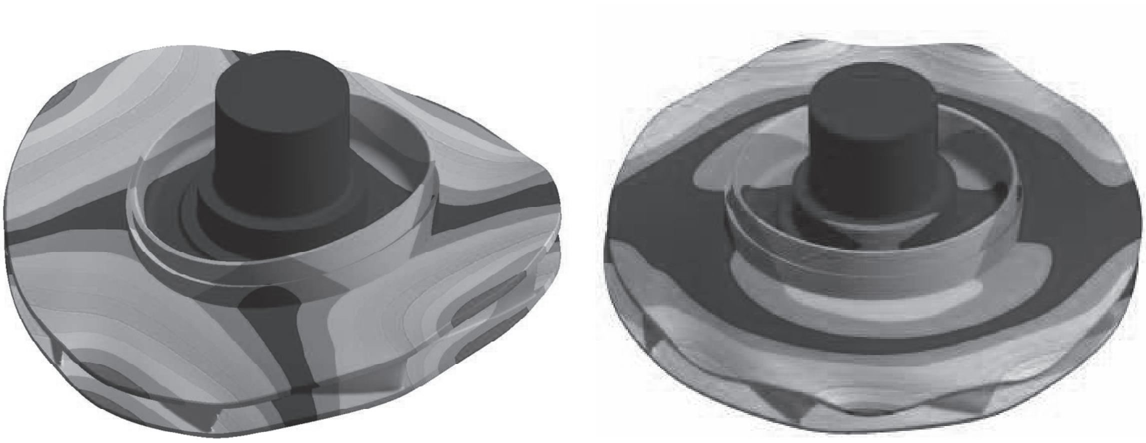

Fig. 4 *An example of a single-disk-like as well as double-disk-like normal modes*

The excitation frequency given by Eq. (3) is relatively high. There are many natural frequencies of the runner in the frequency region in question. If the vibrations are forced by an exciter (shaker) corresponding number of very sharp resonance peaks can be excited. However, model [6], [11] as well as field stress measurements show clearly, that the runner response on the excitation due to RSI has only single broad resonance peak when rotating speed increases continuously from zero. It resembles a response of a system having one degree of freedom. How to explain it?

The forced vibrations can be dangerous (under low damping) if the following conditions are satisfied:

- The excitation frequency is equal to a natural frequency (resonance).
- The excitation is affined to the corresponding normal mode.

(Theoretically if the excitation is orthogonal to the corresponding normal mode no vibrations can be excited in spite of the resonance.)

The unsteady pressure field due to RSI can excite above all the double-disk-like modes. (Contrary to Tanaka's procedure [6] the runner vibration is in our case not excited on blades but on the crown as well as on the band.)

The problem is of 3D nature, however, let as assume only the circumferential direction along the runner diameter. The displacement of the crown as well as of the band at a certain normal mode can be compared with the excitation given by Eq. (7). The double normal modes are excited by both the cos-component and the sin-one. If the double natural frequencies differ considerably in the reality only vibrations excited by one component predominate.

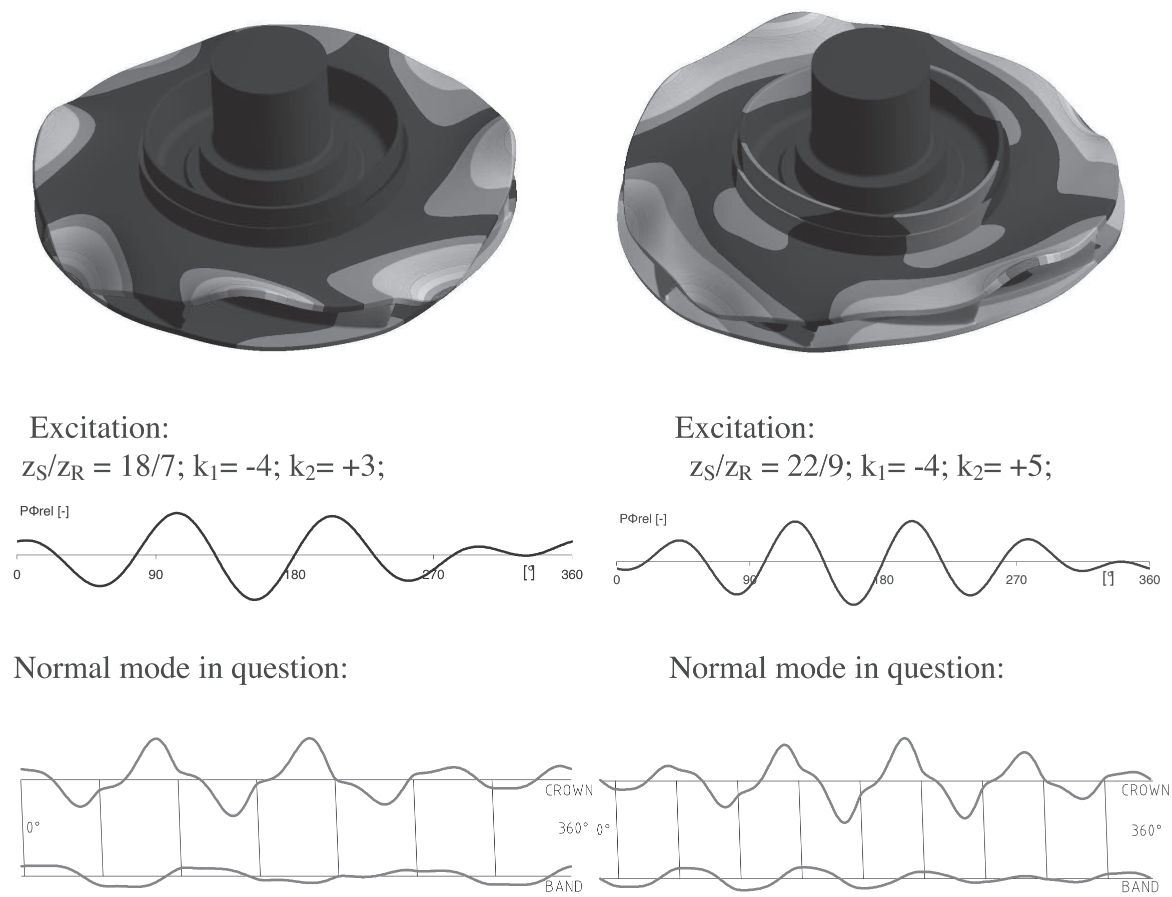

Fig. 5 *The displacement of the crown as well as of the band at given normal modes compared with the excitation given by Eq. 7 . The left runner has 7 blades, the right one has 9 blades.*

Two examples of similarity between natural modes and the excitation are presented in Fig. 5. Such situations are typical for guide vane to runner blade combination which satisfies the following condition:

$$|k_1 + k_2| = 1,2,3 \qquad k_1; k_2 \neq 0$$

Similarly with [6], it can be shown that the combinations:

$$k_1 + k_2 = 0$$

are “bad ones”. In this case the number of runner blades is always even.
In general, by dimensioning the runner a possible resonance must be always avoided from the rated speed region. If the excitation is similar (or affined) to the normal mode in question – see Fig. 5 - the resonance peak is broad and dangerous. Therefore it is difficult at certain combinations to shift the high dynamic stress region considerably enough from the rated speed region by dimensioning the runner. Changing number of guide vanes seems to be most effective.

It causes a change of:

- the excitation frequency,
- the shape of excitation on the periphery of the runner

Using this procedure the vibration problem can be solved.

5. Conclusions

Every pump turbine tends due to RSI to certain types of high frequency vibrations as well as pressure oscillations. The combination of guide vane number to runner blade one is decisive.
At a combination leading to $k = \pm 1$ the RSI could cause relatively high guide bearing vibrations measured at vibration velocity in mm/s.
The high frequency pressure oscillation problem seems not to be exactly solved up to day. The high head pump turbine runner dynamic stress caused by RSI can be kept in an acceptable limit above all by an appropriate combination of guide vane number to runner blade one.

5. References

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