

Penstock resonance due to RSI pressure pulsations on Francis turbines: a case study in a revamped plant

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Introduction

This paper aims to report a case study of rotor-stator interaction (RSI) pressure pulsations issue that occurred in a revamped plant, focusing on how it appeared, what effects it had and how the mitigations acted. The revamping project considered the replacement of the units and of the penstock within an Italian hydropower plant older than 50 years due to modernization.

The new turbines and penstock showed a strong noise in the powerplant and on the penstock as an RSI effect during commissioning phase, therefore it was necessary to analyse the topic as reported.

The following are the operating data of the plant:

- | | |
|---------------------|-----------------------|
| • Head | 100 m |
| • Maximum discharge | 2,8 m ³ /s |
| • Nominal power | 2,5 MW |
| • Turbines type | Francis |
| • Penstock type | Aerial |
| • Penstock diameter | 1,2 m |
| • Penstock material | Steel. |

1. Acoustic and vibration phenomena experienced

The strong acoustic phenomena experienced during unit commissioning obliged to perform a campaign of vibration measurements on the unit and on the penstock.

The measurements have been conducted at several values of load (33%, 67%; 100% rated power) and locations (turbine guide bearing; penstock pos. A and B, see next reference sketch).

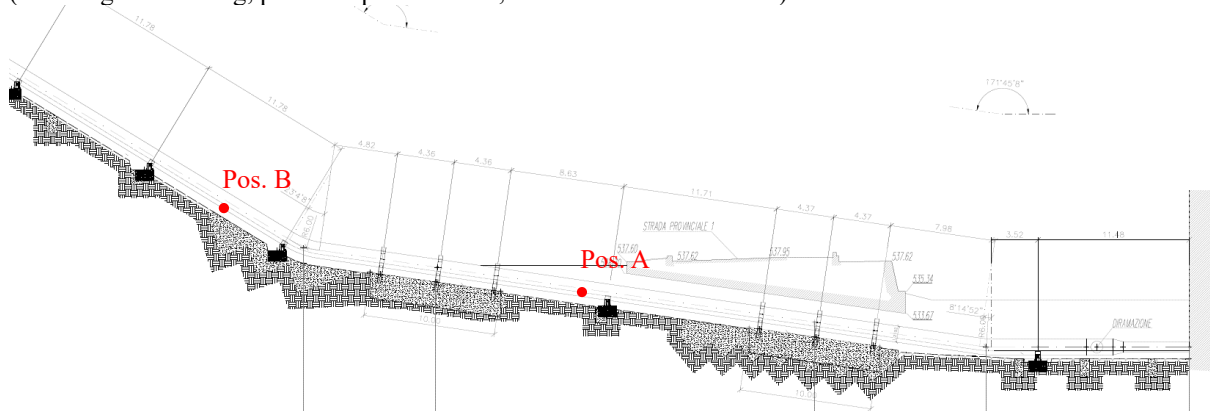


Fig.1. Penstock positions for vibration measurements

The results of the first measurement are shown on the next tables.

			Turbine guide bearing		Penstock positions			
			V_{rms}	V_{rms}	V_{rms}	V_{rms}	V_{rms}	V_{rms}
Unit	Distr.	press.	-	-	Pos.A	Pos.A	Pos.B	Pos.B
tested	opening	turbine in.	rad.P0	rad.P90	CH1	CH2	CH1	CH2
(-)	(%)	(m)	(mm/s)	(mm/s)	(mm/s)	(mm/s)	(mm/s)	(mm/s)
1	30	9,71	1,39	1,39	2,29	2,01	1,30	1,00
1	57	9,65	1,72	1,55	2,36	2,02	0,82	0,81
1	75	9,55	1,74	1,27	2,69	2,17	1,30	0,85

Table 1 – First vibration measurements on the unit and on the penstock during commissioning, at different loads.

Unit	Distr.	W_{gt}	L_{eq}	L_{min}	L_{max}
tested	opening	-	POS.A	POS.A	POS.A
(-)	(%)	-	(dB)	(dB)	(dB)
1	75	A	83,50	82,20	85,2

Table 2 - First noise measurement on the penstock during commissioning.

The vibration amplitudes were higher than expected, especially on the turbine guide bearing. The noise measurements showed at full load a L_{eq} equivalent to 83.5 dB(A), i.e. almost 20 dB(A) higher than the diurnal limit, according to current Italian applicable environmental laws. The vibration and noise analysis through the spectra gave further hints for the first interpretation of the phenomena.

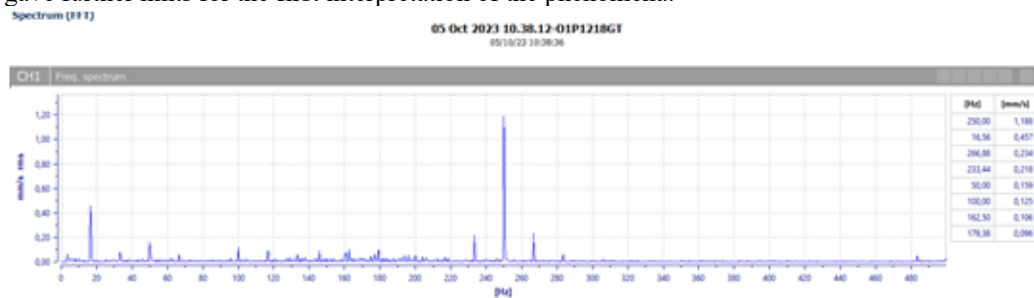


Fig.2. Turb. guide bearing vibration at 100% nom. power – CH1=sensor parallel to spiral case inlet.

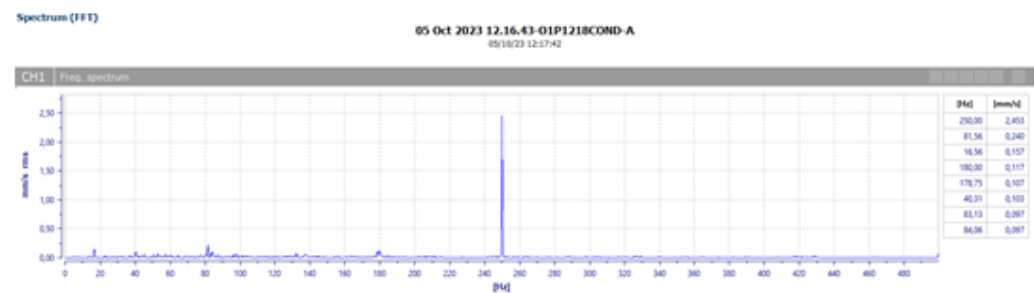


Fig.3. Penstock vibration pos.A at 100% nom. power – CH1=sensor horizontal oriented.

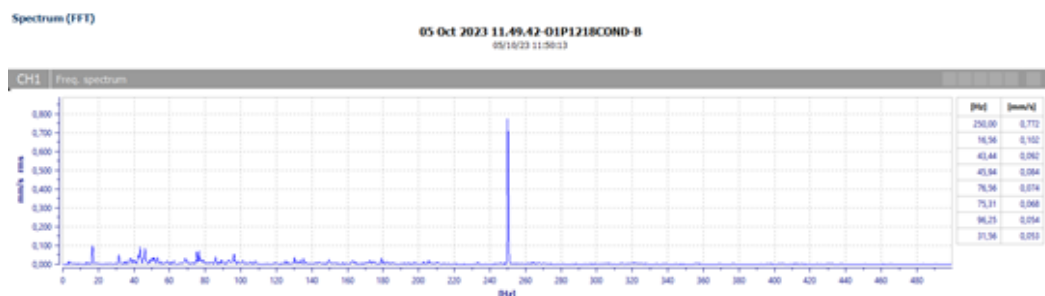


Fig.4. Penstock vibration pos.B at 100% nom. power – CH1=sensor horizontal oriented.

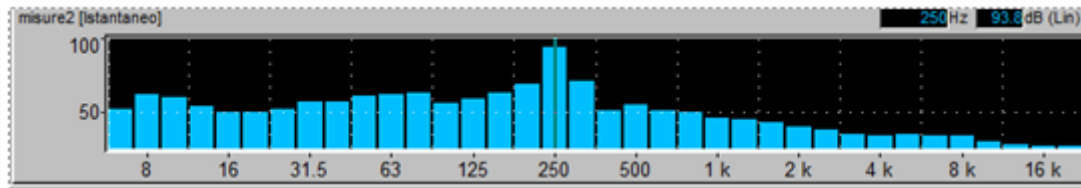


Fig.5. Noise spectra at pos.A with unit at 100% nom. Power.

The 3 vibration spectra and the noise spectrum gave the same indication: the dominant vibration was at 250 Hz, that corresponds basically to the runner blade passing frequency, but also to one of the main frequencies of the Rotor-Stator Interaction (RSI), see para 2.1. From acoustic point of view, the situation found was worse because the emitted sound appeared with a strong tonal effect¹, which obliged to increase the measured $L_{eq}(A)$ by 3 dB(A). A bump test has been performed on the penstock to check its natural vibration modes.

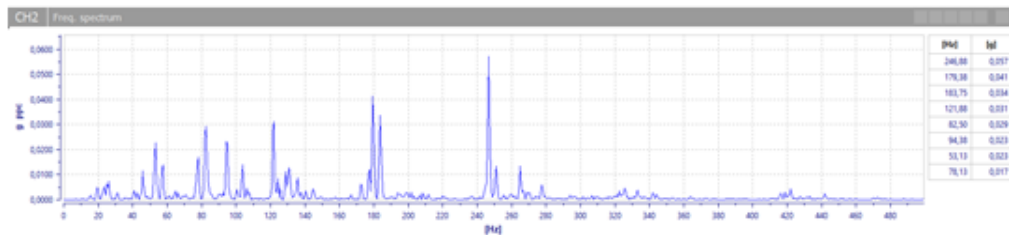


Fig.6. Bump test on penstock at position A results.

The result, showed above, confirms the possibility to excite the penstock through a forcing frequency of 250 Hz. The penstock resonance can, therefore, be the main cause of the noise with strong tonal effect.

In the meantime, it was crucial to check whether the vibration amplitudes found on the penstock could be dangerous from structural point of view. As first verification the Steel Penstock Manual n°79 of American Society of Civil Engineers (ref. 5) was used to interpretate the vibration amplitudes: this manual recommends that vibration associated incremental stress does not exceed 20% of the design stress or the radial deformation does not exceed penstock internal diameter/1.000. A numerical check has been done and both conditions have been positively matched with significant margins therefore, it was possible to define the penstock safe from structural point of view.

2. Identification of the forcing cause

2.1. Brief summary on RSI pressure pulsation and penstock resonance

The 2 phenomena are well known since decades, all their details can be found in the technical literature and below is reported a brief summary.

In hydraulic machines the interaction between *runner blades* and *guide vanes* may produce pressure variations on the fluid. In case of Francis turbines, a pressure pulsation is generated whenever one of the runner blades matches a single guide vane. Due to the rotation of the runner this single pressure pulsation is transformed into a periodic wave which depends on the rotational speed of the machine and on the number of the blades. This phenomenon occurs for each guide vane resulting in a generation of pressure pulsations (one for each guide vane), each starting from the respective guide vane. Considering the wave propagation in the fluid, two main directions can be identified: a) upstream to the spiral inlet, b) downstream to the end of the spiral.

Considering the time and the location of all matches (each blade crossing each vane), it's possible to study the synchronization of all pressure pulsations when they arrive at the inlet and when they arrive at the end of the spiral, if they are synchronized the pressure pulsation gets amplified or vice versa gets dampened.

The riskiest runner blade and wicket gate are those which fulfil the following relationship between Z_r runner blade number and Z_d wicket gate number:

- For turbines: $Z_r = Z_d - 1$

¹ A tonal effect occurs when most of power is concentrated on a specific frequency. The Italian laws require to add 3 dB(A) at measurements in case of strong tonal effect to account for the more harmful effect on the human ear.

- For pumps: $Z_r = Z_d + 1$

It corresponds to the combination most favourable to the occurrence of the phase resonance phenomenon between the elementary waves (phase synchronization). If waves depart temporally and angularly with a delta of time similar to a sequence of the propagation speed and mode along the spiral, the pulses get tuned to a single frequency equivalent to the product of the rotational speed and the number of runner blades.

$$f_{tuned} = \frac{rpm}{60} \cdot Z_r$$

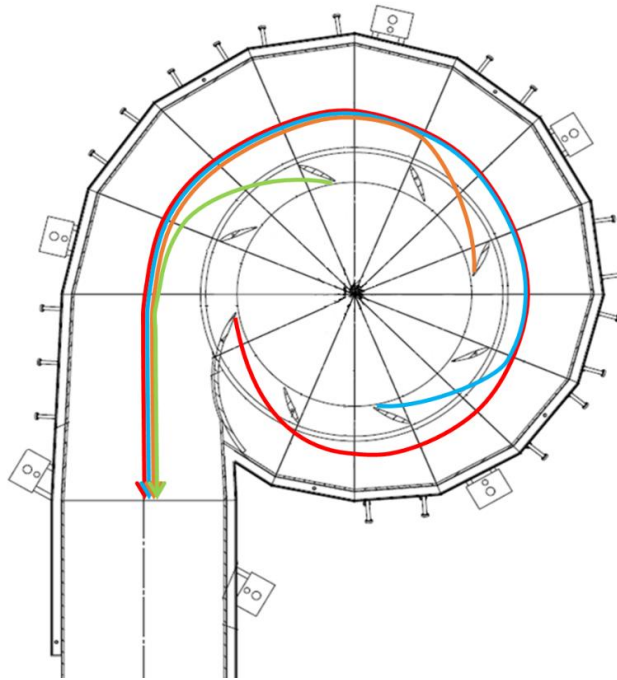


Figure 1 - Conceptual illustration of waves (coloured) propagation towards the spiral inlet of a Francis turbine

Steel penstocks with external laying are elastic structures with little damping; if a forcer excites its own natural vibrating frequencies, they can easily resonate emitting noise. A penstock can have several natural vibration modes because they can vibrate even 3-dimensionally for its circumferential mode flexing, especially for large D/t ratio (D as diameter and t as thickness).

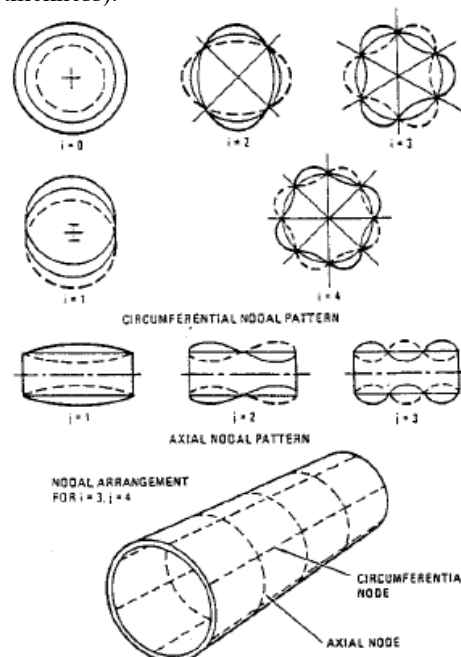


Figure 2 - Axial and circumferential nodal patterns for supported cylinder without axial constraint (ref. 5)

For that reason, the penstock can resonate at frequencies with high power and high sensitive to human ear.

2.2. Possible cause and numerical analysis for confirmation

In the case study, the measurements (see para 1) showed a clear monotone impulse at frequency 250 Hz that keep the same even changing the flow rate.

Following are the specific data of the turbine for the further numerical analysis, whose aim was to confirm the origin of the phenomena and to find a solution, which should mitigate the acoustic issue.

- rotational speed 1.000 rpm
- number of runner blades 15
- number of guide vanes 16
-

In order to get a final confirmation of this conclusion, a numerical calculation of the RSI pressure pulsation synchronization factor (i.e synchronization of pulsation phases) at spiral inlet and at spiral end has been made, using a dedicated program, which requires the definition of the spiral case geometry and of the relevant wave speed distribution. Considering that RSI pulsations are perfectly in phase each other when the synchronization factor is equal to 1.00, the following results have been found for the exciting frequency of 250 Hz:

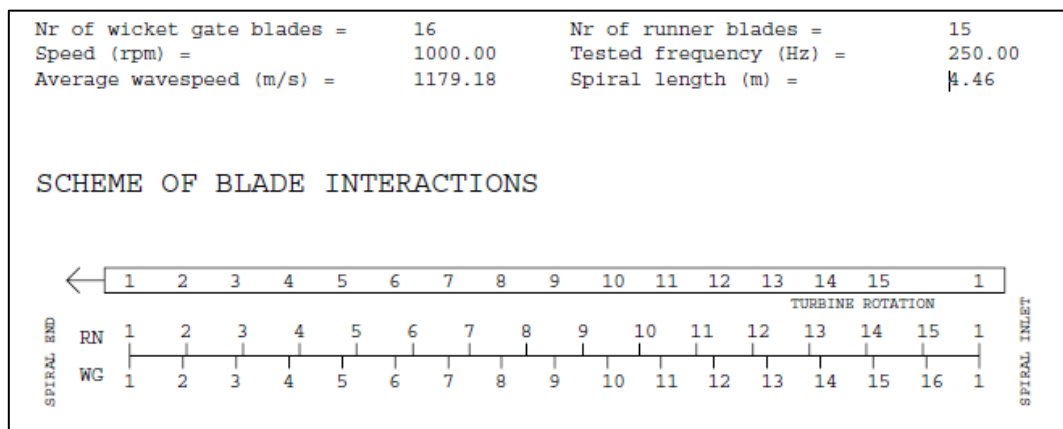


Figure 3 - Scheme of the RSI synchronization factor computation for the exciting frequency of 250 Hz

Synchronization factors (250 Hz)	
Turbine rotation, to spiral inlet direction	0,975000
Turbine rotation, to spiral end direction	0,112997

Table 3 - Results of RSI synchronization factors with $Z_d=16$ and $Z_r=15$

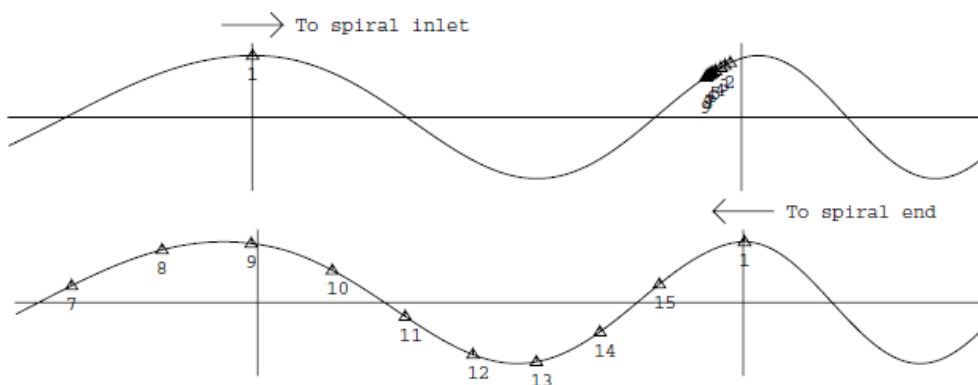


Figure 4 - Representation of impulses on spiral base wave with $Z_d=16$ and $Z_r=15$

The results, which showed an extremely high pressure pulsation synchronization factor (0.975) towards spiral case inlet confirmed that the RSI pressure pulsation were the cause of the penstock resonance excitation.

3. Mitigations & results

Since the noise measurements indicated an excess of almost 20 dB(A) with respect of the current applicable Italian environmental laws, the plant has not been able to operate at full power and mitigations were necessary in order to restore the entire operation.

3.1. Compressed air injection in spiral case

The first experienced mitigation has been the use of pressurized air at spiral case inlet with the aim to damp the elementary waves. A bladder full of pressurized air in the spiral case was not considered feasible because of the small room available.

After several trials, an air flow of 0.5% of the nominal water flow has given the best results, without any negative effects on turbine performances, avoiding gates and blades erosion. The pressurized air has been injected in 4 holes located at spiral case inlet.

The positive effect of the compressed air on the penstock emitted noise at pos. A can be seen in the next figure, it shows the Leq(A) time history during a failure of the air compressor with the unit at 15% of power.

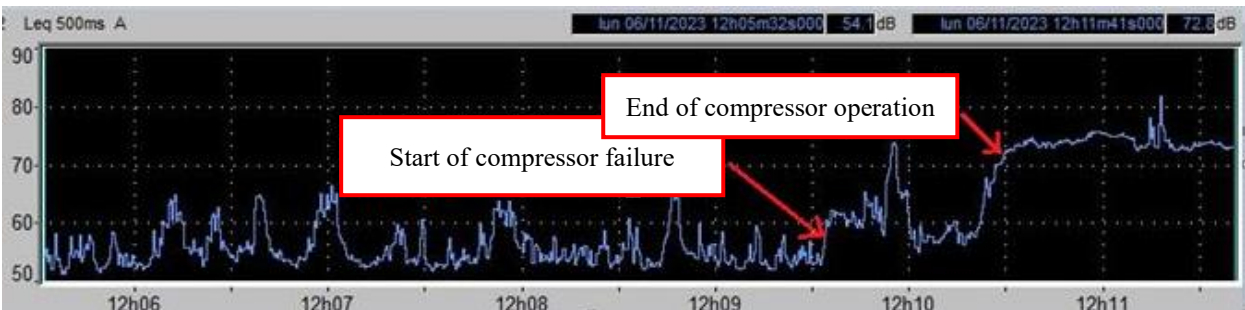


Fig.8. Leq(A) time history at pos.A with unit at 15% of power during air compressor failure.

The compressed air injection resulted in a reduction of about 20 dB(A) at diurnal operation at low load, but it was not enough to respect the laws limit when considering full load diurnal operation, and also night operation, when the limit of sound emission is 10 dB lower.

3.2. Runner replacement

Due to the significant noise effect, it was considered necessary, as main mitigation effect, the replacement of the existing runner with a 13-blades new one.

The same calculation program for RSI pressure pulsation synchronization factor has been used to evaluate the solution with 13 runner blades, founding the following results, in this case for the exciting frequency of 216.67 Hz:

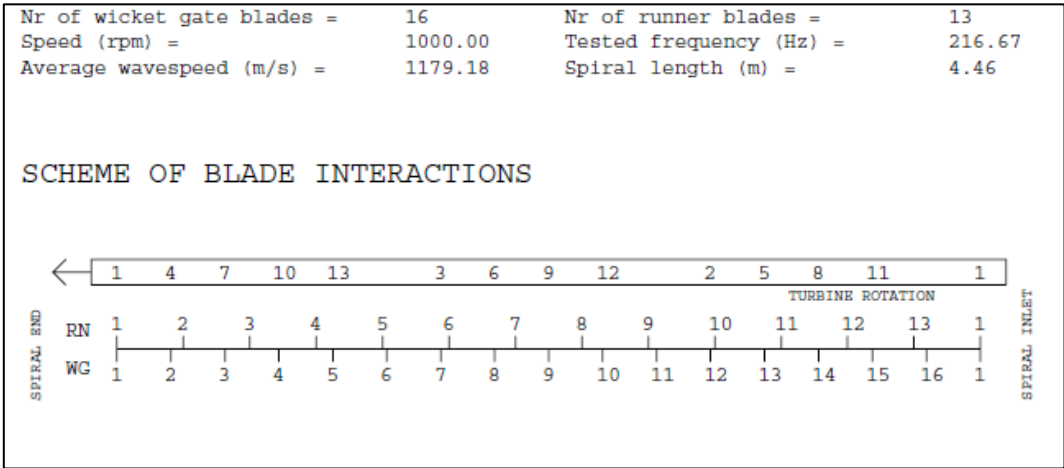


Figure 5 - Scheme of the RSI synchronization factor computation for the exciting frequency of 216,67 Hz.

Synchronization factors (216,67 Hz)	
Turbine rotation, to spiral inlet direction	0,086613
Turbine rotation, to spiral end direction	0,241449

Table 4 - Results of RSI synchronization factors with $Z_d=16$ and $Z_r=13$

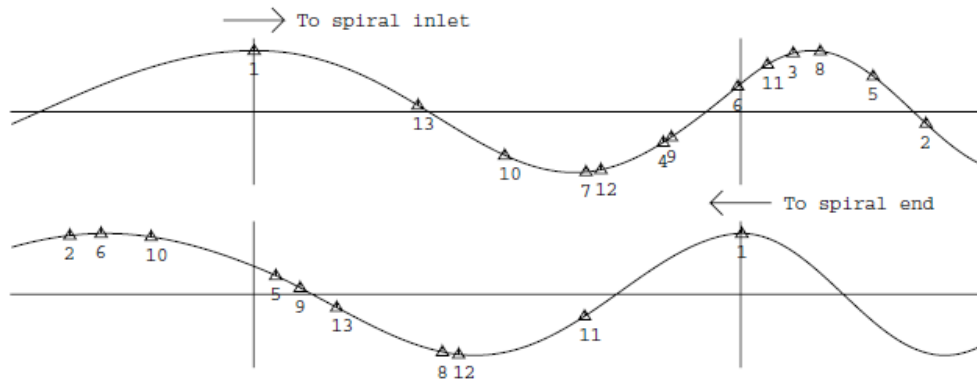


Figure 6 - Representation of impulses on spiral base wave with $Z_d=16$ and $Z_r=13$

The results of the calculations confirmed that a runner with 13 blades creates a RSI pressure pulsations synchronization factor far lower than the one created by a runner with 15 blades, mitigating significantly the penstock vibration and noise.

The results of the site-tests with the new 13-blades runner, in terms of turbine spiral case, guide bearing and penstock vibration, are summarized in the next figures.

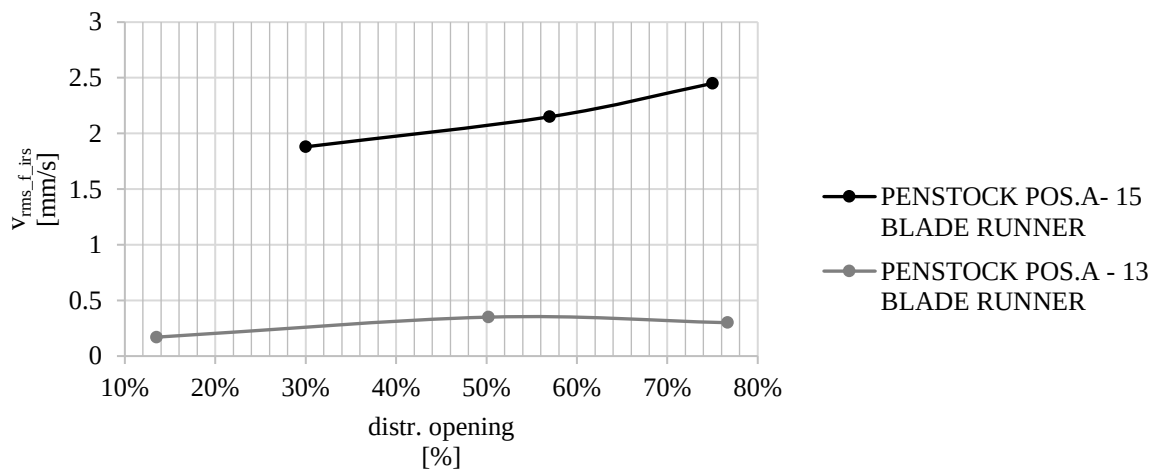


Figure 7 - Penstock vibration on dominant RSI frequencies as function of power and number of runner blades.

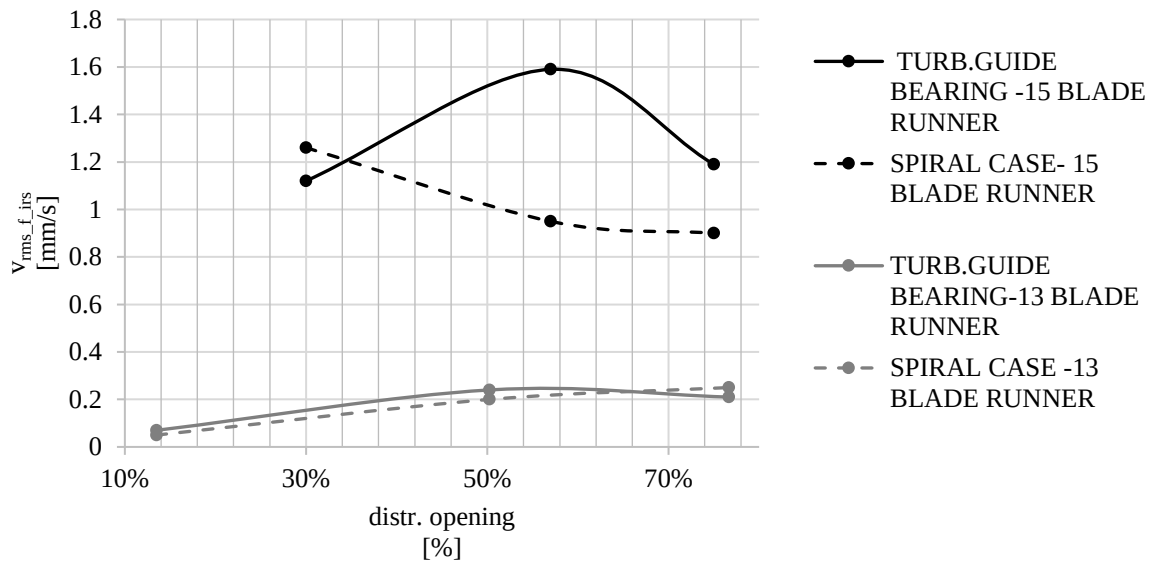


Figure 8 - Turbine case and guide bearing vibration on dominant RSI frequencies as function of power and number of runner blades.

The positive effect of the new runner with 13 blades is showed also on the penstock emitted noise, the equivalent sound pressure level $Leq(A)$ is reduced of more than 20 dB(A) at full load with respect of the 15-blade runner.

4. Conclusion

The case study has demonstrated that, for Francis turbines with specific speed $n_q < 45$, it's crucial analysing RSI and phase resonance effects to reduce the probability of occurrence of penstock vibration and/or noise emission, in particular when the selection of the wicket gate and runner blade numbers are potentially risky.

The RSI and phase resonance study allows to find the safer combination of gates and blades so its choice can reduce strong effects that can, even, impact up to the penstock emitted noise.

Should RSI strong effects still occur, the spiral case air injection has shown that it can have a not negligible effect on the vibration intensity and, consequently, on the penstock emitted noise.

In any case, if the penstock is made of steel, is installed aerial and passes close to inhabited areas, a soundproof cover is suggested to respect the stricter environmental conditions.

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