SHAFT RADIAL OSCILLATIONS INDUCED BY PHENOMENON IN RUNNER LABYRINTHS OCCURRING AT FREQUENCIES ABOVE UNIT SPEED OF ROTATION

mag. Damir DOLENC*

Litostroj Power d.o.o., Slovenia

Denis UDOVIČ

Litostroj Power d.o.o., Slovenia

France KRMELJ

Litostroj Power – associate, Slovenia

ABSTRACT

In course of commissioning of newly installed Francis unit excessive shaft relative vibrations in unit load operation were detected. During first loading of the unit in 2010, measured shaft relative vibrations showed to be above 200µm Smax, which was well above acceptable limits. Several attempts were conducted to reduce shaft relative vibrations including balancing and air admission, but initially with no success.

Additional measurements and analyses showed that shaft vibration frequency did not match the frequency of unit rotation and furthermore, »shaft oscillations« did not rotate in the direction of unit rotation.

During measurements taking place characteristic frequency previously determined on the shaft oscillations were found to be present in most other measured signals like water pressures in spiral case, draft tube, turbine cover and labyrinths as well as on bearing vibrations. At first, it made it difficult to allocate the source of shaft oscillation phenomena.

It was discovered that the motor of the shaft radial oscillation was runner labyrinth construction which acted as de-balancing effect on the shaft.

After detailed investigation a new set of lower labyrinths was installed on the runner and the result was immediate. All shaft oscillation frequencies previously measured on several signals disappeared.

This paper presents the physical background of the phenomena, measurements and investigation efforts that finally led to the successful solving of the problem.

KEYWORDS

Francis Turbine, Runner Labyrinths, Shaft Oscillation, Pressure Pulsations

* *Corresponding author:* Research Department, Litostroj Power d.o.o., Litostrojska 50, 1000 Ljubljana, Slovenia, phone: +386 1 5824 299, email: damir.dolenc@litostrojpower.eu

1. INTRODUCTION

There has been a new hydro power plant with 3 units built in 2010. Two of the units are vertical Francis units with the maximum power of 22 MW each. The third one is a horizontal 1 MW Francis unit. High levels of vibrations have been discovered during the commissioning of the large vertical units in 2010.

Vertical Francis units		Rated	Maximal
Rated speed of rotation	n [min^{-1}]	187,5	360
Turbine power	Pt [MW]	21,25	22,04
Turbine discharge	Q [m^3/s]	60	64
Net head	Hn [m]	38,18	40
Runner diameter	D_g [m]	2,933	/

Tab. 1 – Turbine technical data

Soon after the first wet tests of the two vertical units started, exceeded vibrations and shaft runout were discovered. Both units have shown the same behavior.

2. DISCOVERING THE PHENOMENON

Units were started and put into mechanical run for the first time in August 2010. The generator has been balanced in mechanical and excited mechanical run, so the units were ready for testing under load. Partial (controlled) runaway speed was tested at the speed of 145% for more than one minute without higher unexpected vibrations and runout. During the partial runaway the dominant frequency of runout and vibrations was equal to the frequency of rotation.

When 5 MW, which represents about 25% load, was reached, exceeded vibrations were discovered. Power was increased up to 7 MW but as there was no change in unit behavior the unit was stopped (see Fig. 1). The same problem emerged also on the second unit. Shaft runout and vibration values were significantly higher than it was expected. At that time it was not known what the source of the problem was.

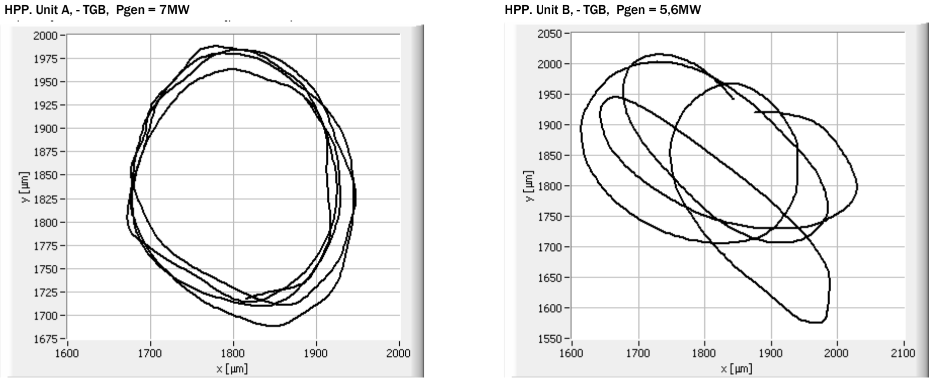


Fig. 1: Shaft runout at turbine guide bearing measured at 7 MW for unit A and at 5,6 MW for unit B

Measurements have shown exceeded shaft runout and increased vibrations on both turbine and generator guide bearings. Both units had the same phenomenon as shown on the Fig. 2.

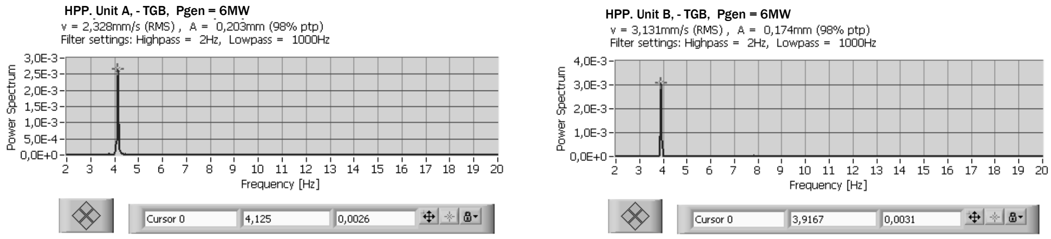


Fig. 2: Radial vibrations measured at the turbine guide bearing (unit A and unit B)

The vibration velocity RMS value measured on the turbine guide bearing was between 2 and 4 mm/s which could not be interpreted as excessive. This was due to the low vibration frequency at approximately 4 Hz. On the other side the displacement of the housing has been high, reaching between 0,20 and 0,30 mm of radial displacement which is way too high as according to standard ISO 10816/5 the limit for group C is 0,08 mm of peak- to-peak displacement.

Further testing under load was stopped. Both generator and turbine guide bearing gap adjustment showed to be according to the design documentation. Turbine guide bearing was readjusted to the lower admissible limit at 0,16 mm to each bearing pad. After restarting, the same phenomenon was discovered, but now with slightly lower vibration and runout values. As units were already double-checked for correct mechanical erection, it was decided to continue tests at higher loads. During the emergency shut down and load rejections the vibration phenomenon strangely disappeared.

Exceeded vibrations have been discovered from the power of 5 MW till the maximum load equally on both units.

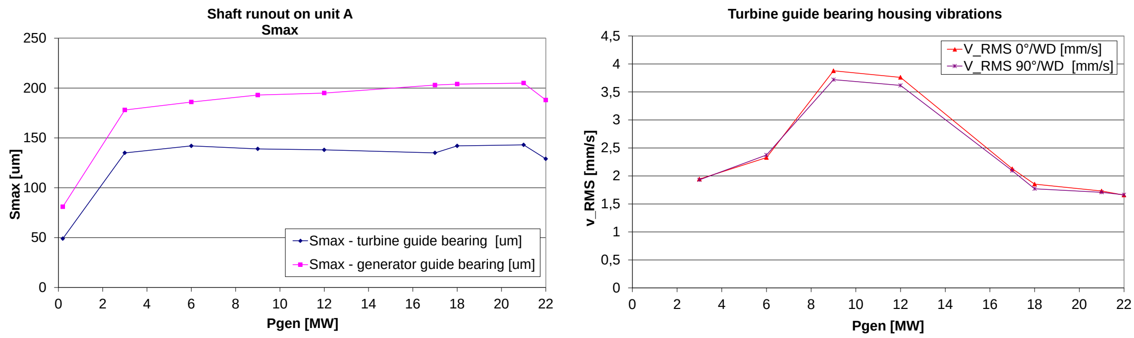


Fig. 3: Shaft runout and vibrations on the turbine and generator guide bearing

3. INVESTIGATION AND MEASUREMENTS

In mechanical and excited mechanical run the phenomenon of exceeded vibrations was not present. The frequency analysis of the shaft runout has shown the dominant frequency equal to the frequency of rotation (187,5 rpm = 3,125 Hz).

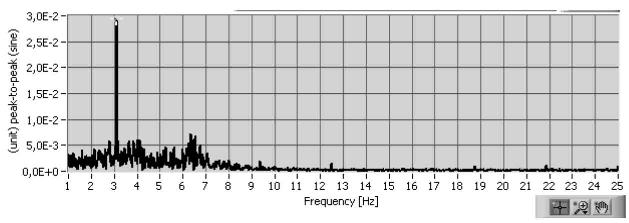


Fig. 4: Frequency analysis of the shaft runout signal in unit mechanical run

Further it was discovered that the dominant frequency of vibrations and shaft runout of approximately 4 Hz was present in all signals for loads above 5 MW and it slightly changed with the load between 3,9 Hz up to 4,4 Hz. It was not a constant multiple of the frequency of rotation. Fig. 5 presents frequency analysis of the measured signals at the load of 15 MW on unit A. Except from the signal of the pressure in the draft tube, all show the dominant frequency of 4,2 Hz.

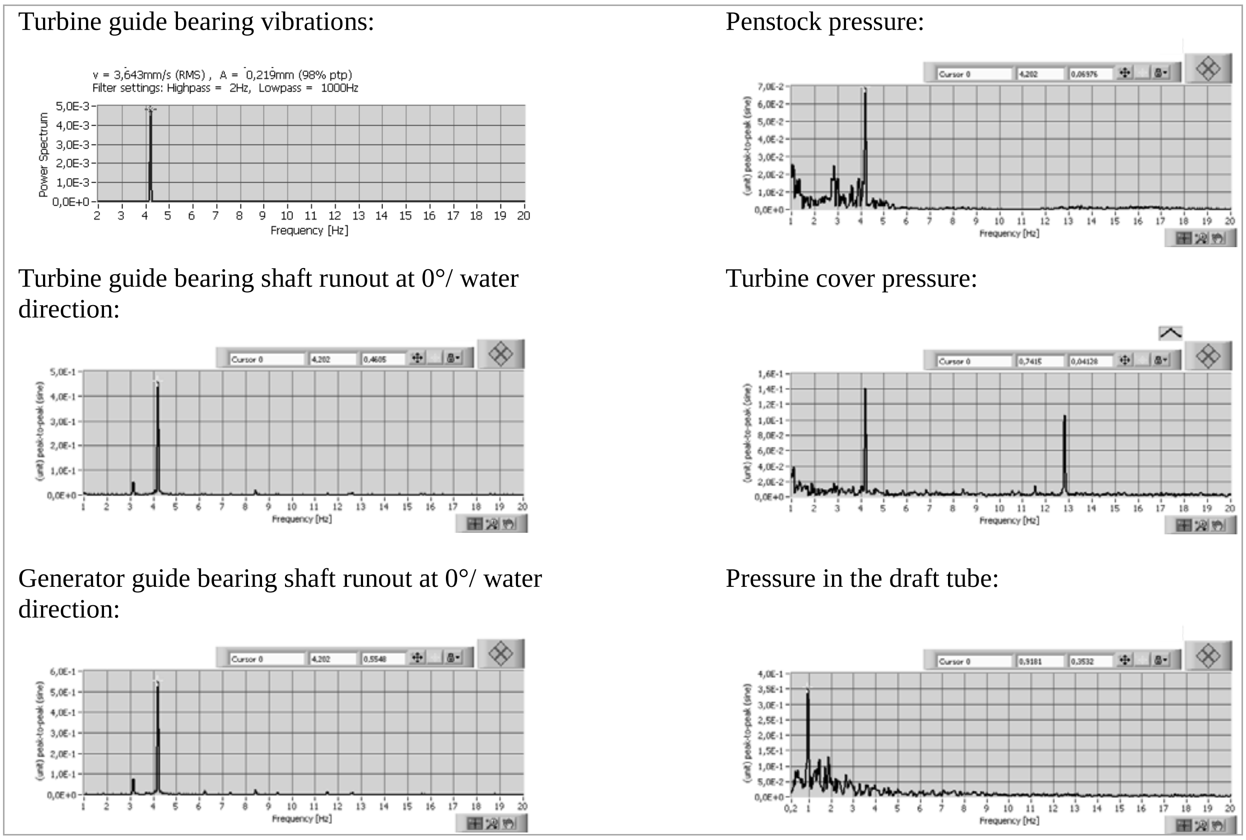


Fig. 5: Frequency specter in measured signals

Few attempts were conduct to see the effect on the vibrations:

First influence of the water pressure below turbine cover was tested. By manually closing the drain adjustment valve, the pressure in the runner-turbine cover chamber was slowly increased toward the upper allowed limit of 2 bar and oscillatory phenomenon showed to be decreasing. The improvement achieved was connected with the increase of downward axial force on the runner. Nevertheless, frequency at about 4 Hz could still be seen in all the signals and the vibration as well as runout values were still too high.

Units were than tested with different air admission valve adjustment. Vibration values slightly changed with different air admission adjustments but it did not affect the dominant oscillatory frequency and vibration and runout values remained too high.

Fig. 7 presents the raw data measured at the shaft runout sensors. Shaft runout was measured on both bearings in water direction and on 90° to water direction – see Fig. 6. It can be seen from the signals that the phase between sensors at turbine and generator guide bearings measured in the same vertical plane was approximately 130°.

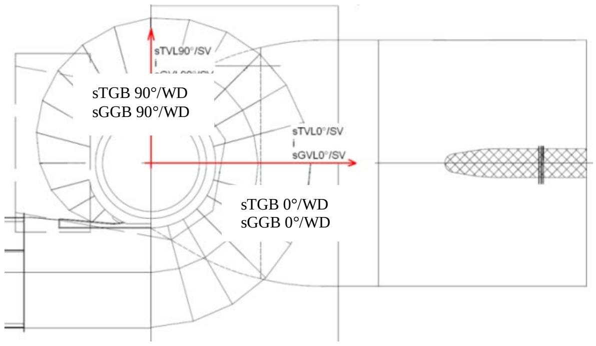


Fig. 6: Coordinate system of the shaft runout

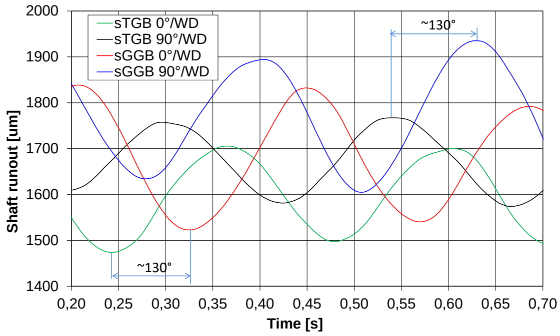


Fig. 7: Shaft runout raw signal data measured at 17MW (75% load)

Another interesting fact was that the shaft runout on both bearing was rotating in the opposite direction to the direction of rotation. Fig. 8 presents measured shaft runout during the time of one shaft revolution (t=0,32 seconds).

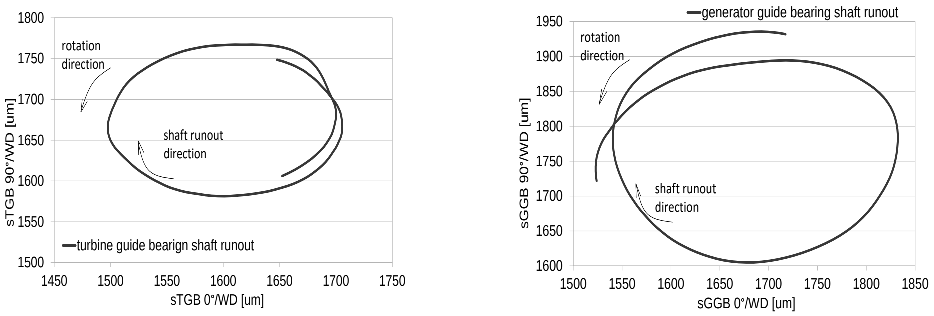


Fig. 8: Shaft runout orbit measured on the turbine and generator guide bearing

This phenomenon was caused by the increase of friction in the radial bearings. The schematic diagram of acting forces is presented on Fig. 9. The radial force presses the shaft against the bearing thus increasing friction forces between the shaft and the bearing. Due to the resultant force the shaft starts to roll like a wheel in the bearing.

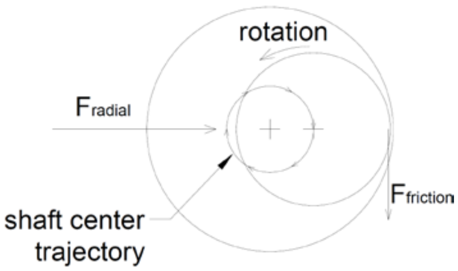


Fig. 9: Schematic diagram of forces in the bearing

The source of the phenomenon was discovered after the measurement of pressure pulsations in the runner labyrinths (see Fig. 10).

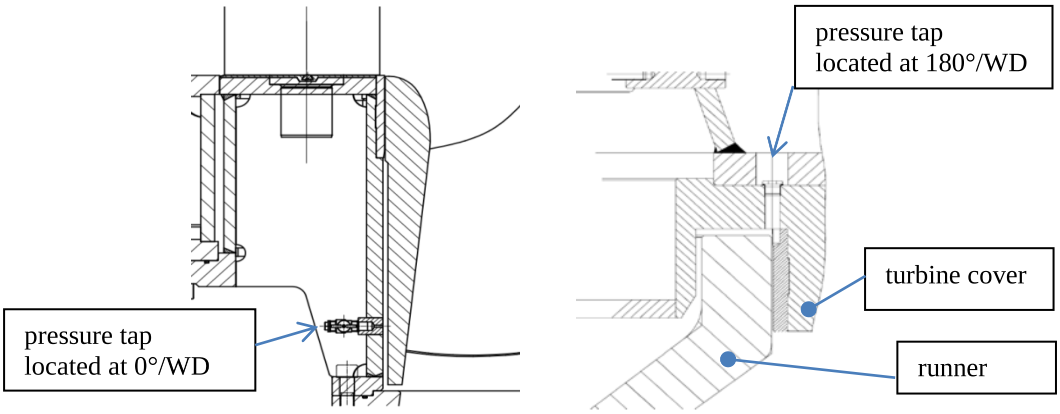


Fig. 10: Location of pressure measurements in the runner labyrinths

It was discovered that pressure pulsations have the same dominant frequency and are in phase with the shaft displacement when measured in the same direction (see Fig. 11.). That proved that when the shaft moves from the center, the force that is generated tries to increase the displacement. Shaft displacement in this setting is limited by the bearing wall, and the resultant acting force makes the shaft start rotating in the opposite direction to the direction of rotation.

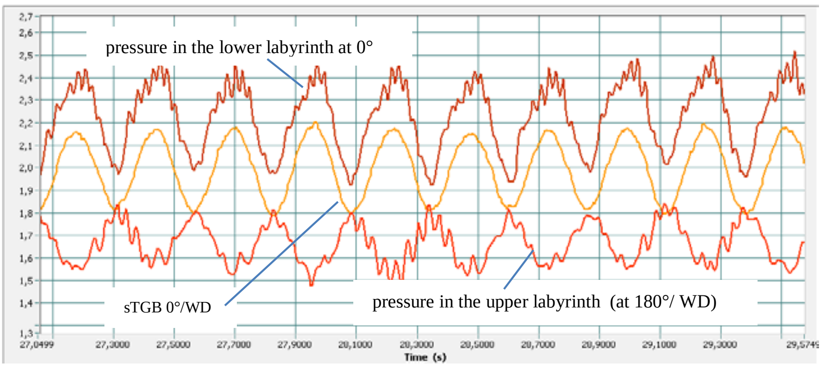


Fig. 11: Pressure measurement in the runner labyrinths

Fig. 12 shows that if the runner moves toward direction B the clearance in labyrinth copies the movement and increases the clearance on the side A. Consequently the pressure on side A will increase while the pressure on the side B will decrease. The resultant force from the pressure acting in the labyrinths has decentering effect by pushing the runner away from the ideal center of rotation. As seen from Fig. 13 the runner crown area is high and already small pressure pulsations cause large force pulsations.

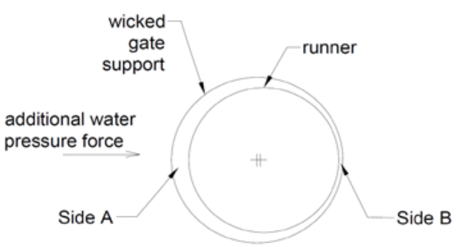


Fig. 12: Runner labyrinths view from the above

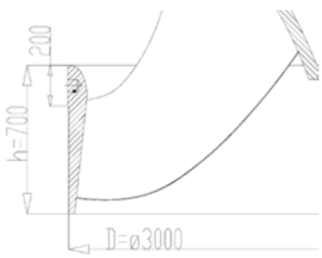
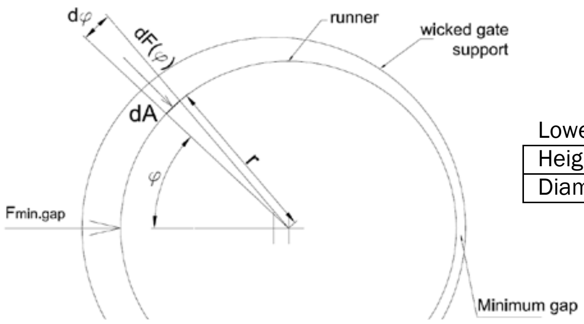


Fig. 13: Detail of the runner crown

4. CALCULATION OF RESULTANT FORCE IN LABYRINTHS

According to the measured pressures in the runner labyrinths a radial force of water acting on the runner can be calculated. Measured pressure pulsation was approximately 0,45 bar in all operating steady state conditions.



Lower labyrinth dimensions:

Height	h=~0,7 m
Diameter	D= ~3 m

Fig. 14: Schematic of forces in the lower runner labyrinth

The differential of the radial force acting on the runner crown can be calculated as:

$$dF(\varphi) = p(\varphi)\cdot h\cdot r\cdot d\varphi = p_0\cdot\cos\varphi\cdot h\cdot r\cdot d\varphi \qquad (1)$$

The above calculated force has the direcion towards the center of the runner. The differential force acting on the runner crown in the direction of the smallest labyrinth gap equals:

$$dF_{\min.gap} = dF(\varphi)\cdot\cos\varphi = p_0\cdot\cos^2\varphi\cdot h\cdot r\cdot d\varphi \qquad (2)$$

With the integration of this force it is obtained the total force of the water acting in the direction of the smallest gap:

$$F_{\min.gap} = \int_{-\frac{\pi}{2}}^{\frac{\pi}{2}} dF_{\min.gap} = p_0\cdot h\cdot r\int_{-\frac{\pi}{2}}^{\frac{\pi}{2}}\cos^2\varphi\cdot d\varphi = p_0\cdot h\cdot r\cdot\left(\frac{\varphi}{2}+\frac{\sin 2\varphi}{4}\right)_{-\frac{\pi}{2}}^{\frac{\pi}{2}}$$

$$F_{\min.gap} = p_0\cdot h\cdot r\cdot\frac{\pi}{2} = 45000\frac{N}{m^2}\cdot 0{,}7m\cdot 1{,}5m\cdot\frac{\pi}{2} = 74220N \qquad (3)$$

From the above calculation it can be seen that a force of 74kN tons was acting in the runner labyrinth in the radial direction.

5. SOLUTION

To solve the problem a new labyrinth was added to the bottom part of the runner crown (see Fig. 15). It is important that the lower labyrinth has a smaller gap than the upper one. In addition water was supplied directly from the penstock. Tests have shown that the lower labyrinth with the smaller gap than the upper one efficiently solved the problem even if the water admission from the penstock was closed.

To evaluate operation with new labyrinth added, figure to Fig. 12 can be drawn. Because the lower labyrinth has a smaller gap, majority of pressure drop across the labyrinth setup is achieved on the lower labyrinth. Local pressure above the lower labyrinth remains constant even when the shaft radial displacement is present. Now, with the pressure being constant, the whole runner crown acts like a bearing.

If the runner moves eccentric to the wicked gate support (like shown on Fig. 12) the pressure on side A will be lower and the pressure on side B higher. In this improved situation water pressure in the labyrinth is trying to move the runner to the center.

After the modification implemented, the frequency of approximately 4 Hz totally disappeared.

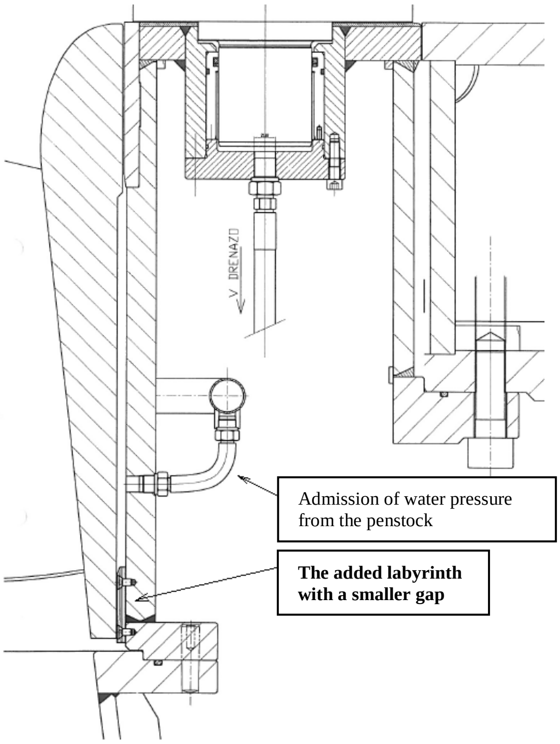


Fig. 15: The added lower labyrinth

6. CONCLUSION

The source of the exceeded vibrations was the absence of lower runner labyrinth. Due to the high surface of the runner crown, the pressure pulsations that were present locally caused high forces with de-balancing effect. This was modified by the addition of the second labyrinth on the bottom of the runner crown, and most importantly, installed was a labyrinth with smaller gap as the upper one.

After the implemented modification on lower labyrinth, vibrations and shaft runout stabilized within the limits of group A/B according to ISO 7919 and ISO 10816. It was concluded that the unit could operate safely and it was ready for long term operation.

7. NOMENCLATURE

RMS		*Root-mean-square*
TGB		*Turbine Guide Bearing*
GGB		*Generator Guide Bearing*
Smax		*evaluated shaft displacement (runout) according to ISO 7919:5*
WD		*direction of water flow*
F	[N]	*force*
h	[mm]	*height*
r	[mm]	*radius*
f	[Hz]	*frequency*
φ	[°]	*angle*
p_o	[Pa]	*pressure*
v	[mm/s]	*vibration velocity*