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RESONANCE RELATED PROBLEM, DIAGNOSTICS AND SOLUTION PROPOSAL ON VERTICAL FRANCIS HYDROGENERATOR

ABSTRACT

The problem of increased vibrations was noticed when two new Francis turbine driven hydro-units (5 MW, 750 rpm) were commissioned.

Both unit's behavior vibro-dynamical response was identical with a passage through critical speed at exactly 600 rpm (10 Hz). Vibrational velocities measured on the upper generator bearing exceeded 20 mm/s (RMS value) during the critical speed operation and were at acceptable limits 2-3 mm/s during normal operation. However, even during normal operation, the vibration levels intermittently increased and decreased without obvious cause.

Several unsuccessful measurement campaigns, with different approaches, were tried before the measurements presented in this article were conducted.

The full identification was conducted in a couple of steps with simultaneous multi-channel configuration which was proven to be of great importance to successfully diagnose the problem. The main conclusion was that there were significant irregularities in spiral and draft tube foundations, which allowed the unit to vibrate at 10 Hz resonant frequency.

The solution was proposed with to improve the unit foundations. It involved high-pressure concrete injection and addition of dozens of anchoring elements on spiral and diffusor in order to reinforce the bond.

After these actions – the resonant frequency was increased into the ~16 Hz range and both units are able to operate adequately in all available operating regimes and achievable rotational speeds. The critical speed is between the nominal and highest achievable rotational speed following 5 MW load rejection and is also below the theoretical overspeed.

Key words: resonance, resonant response, diagnostics, overhaul, repair works, multi-channel instrument, intermittently high vibrations

1. INTRODUCTION

The problem of excessive vibrations on two newly commissioned units was present from first operation. The units were both 5 MW, nominal operating speed 750 rpm equipped with Francis turbines and with rolling elements bearings.

It was observed that the vibrations increase during run-up and run-down in a way that greatly resembles passage through critical speed – at the ~600 rpm rotational speed (10 Hz). It was also observed that, when the unit was operating at load, vibrations increased intermittently (and abruptly), without visible cause and also decreased after a certain time period. When left unaddressed, large vibration levels, such as the one measured on these units, can lead to fatigue cracks, putting the units at risk of a catastrophic failure.

Multiple unsuccessful independent measurement campaigns were organized in order to diagnose and recognize this problem but none resulted in permanently achieving acceptable vibration levels.

At the end, vibration diagnostic measurements were performed in a couple of different measurement configurations with the main purpose in recognizing the problem's root cause and suggesting overhaul procedures that would lead to its solution.

2. MEASUREMENT CONFIGURATION

The measurement configuration used during the first measurements is given in the following paragraph

Fig. 1~~Fig. 1~~ shows the photo of the unit with the measurement directions marked. The directions X and Y are radial and the direction Z is axial. The appropriate bearing planes are also marked – upper bearing is axial-radial and the lower bearing is radial. Both are rolling element bearings.

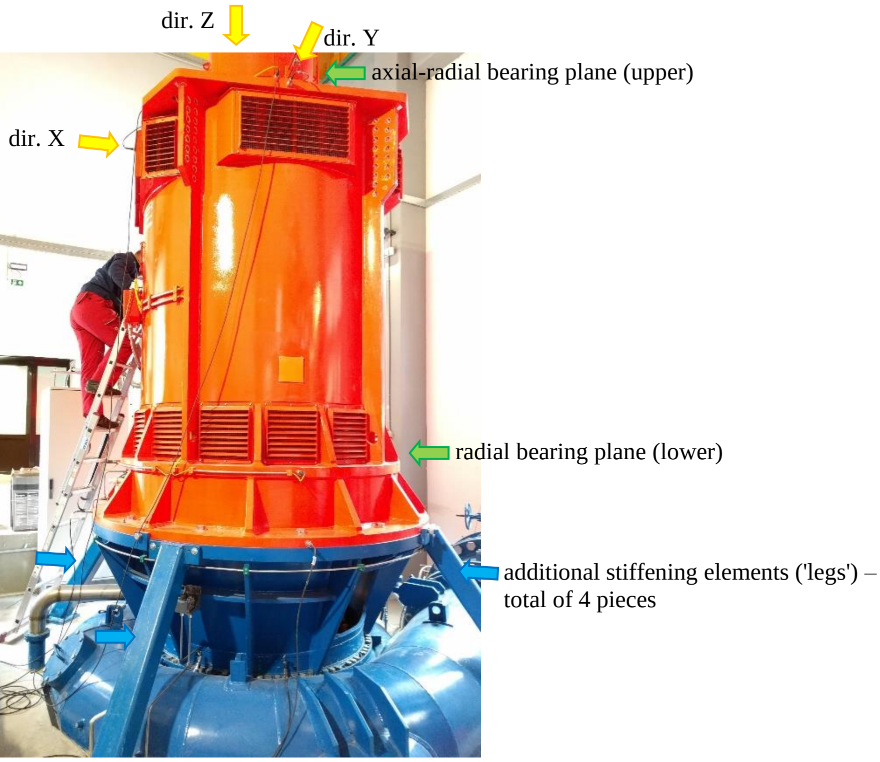


Fig. 1 Photo of the unit with typical measurement directions and planes marked. Measurement directions are X, Y and Z.

The photograph also shows additional stiffening elements ('legs') whose bottom part is anchored to the concrete foundation and the upper part is next to the generator pedestal flange. As needed, these 'legs' can be attached to the flange or separated from it – in both radial and axial directions.

Three types of sensors were used:

- optical sensor to measure the rotational speed (which also serves as a diagnostic trigger)
- inductive proximity probes Balluff BAW M12 with 3.33 V/mm sensitivity
- accelerometers CTC AC 135 with 500 mV/g sensitivity


Axial and radial measurements on a flywheel


Radial displacement measurement in the turbine cover area


Radial generator flange displacement measurement relative to the turbine

Fig. 2 Photos of measurement sensors on some measurement positions.

The analog signals from the sensors are conditioned and digitized using the 18-channel measurement device CoDiS-PMU[1]. The digitized signals are then analyzed in a supporting software CoDiS-PDS[2]. During the measurements the signals were digitized with 2048 samples / second and, the following vibration descriptors were calculated continuously:

- overall vibrational descriptors (for displacement sensors those were Smax, according to ISO 7919-5 and Sp-p – peak-to-peak values and for accelerometers VRMS = vibration velocity effective value)
- amplitudes (peak values – for displacement sensors) and amplitudes RMS (for accelerometers) and phases of the first three rotational frequency harmonics
- REST value which is an indicator of the vibrations which exist outside the first three rotational frequency harmonics
- statical (DC) shaft position relative to the sensor mounting position (only for displacement sensors)
- rotational speed (frequency)

3. MEASUREMENT RESULTS

In this part measurement results for unit 2 are given. Unit 1 has a similar behavior and, since the same overhaul procedures were proposed and conducted, these results are omitted.

First measurement campaign on unit 2

During the first measurement campaign, the unit was started and stopped 6 times. At the end of the first and second start, a load rejection from 3 MW was performed, followed by free run-down starting from, approximately, 900 rpm.

During the first run, the unit was synchronized and loaded with 1, 2 and 3 MW, respectively, after which the load rejection was performed.

[1] CoDiS-PMU = Computerized Diagnostic System – Portable Monitoring Unit
[2] CoDiS-PDS = Computerized Diagnostic System – Portable Diagnostic System

The second run was warm-up and lasts from 16:20 h to ~18:00 h. From 16:20 h to 17:30 h the unit load was increased each 10-15 minutes for 1 MW. From 17:30 h to ~18:00 h the unit was operating on 5 MW load. At 18 h the load was reduced to 3 MW after which the unit was shut down.

The most relevant vibrational descriptors (RMS = Root Mean Square value of vibrational velocity) for accelerometers on bearing housings (X and Y direction) are, for axial-radial and radial bearing planes, are shown on Fig. 3~~Fig. 3~~.

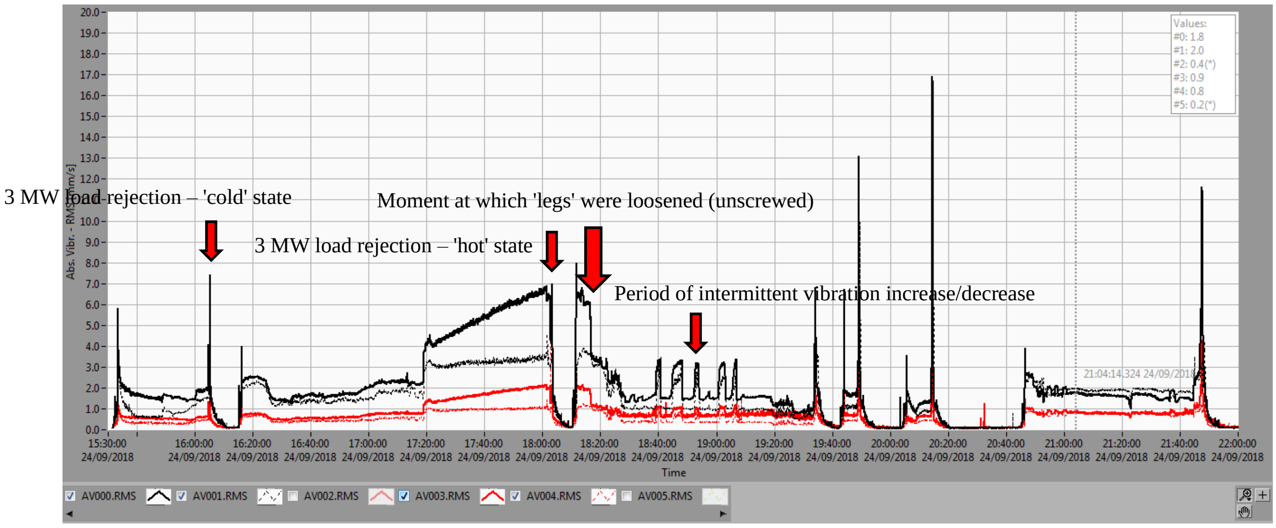


Fig. 3 RMS value of vibrational velocity in different measurement planes. Axial-radial plane (black), radial plane (red). Directions: X = full line, Y = dashed line.

An abrupt increase and decrease in vibrations on run-up and run-down is visible from the diagram. This resembles critical speed passage in turbine generating units. Additionally, around 17:19 h there is an abrupt increase in vibrations, mostly in axial-radial bearing X direction although no regime changes occurred at that moment. So, the question is why this increase occurred? Also visible is the vibration increase with temperature (the increase in vibrations from 17:19 h up until 3 MW load rejection in warm state.

An example of the absolute bearing housing vibration in the axial-radial plane is show on Fig. 4 below

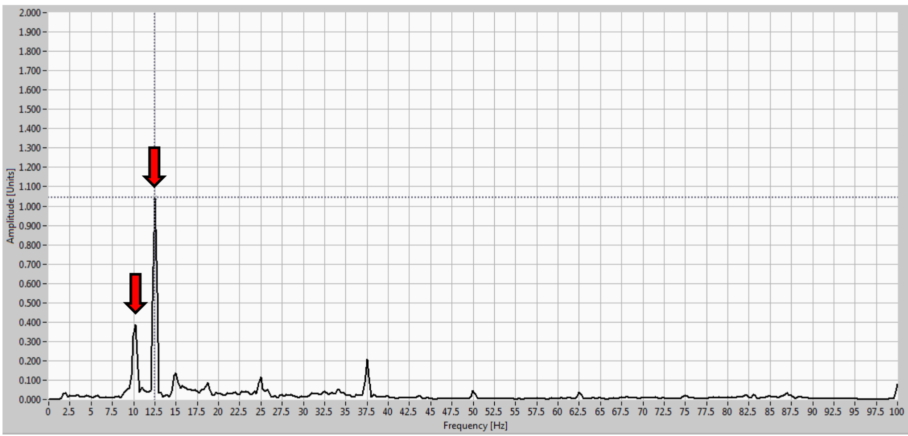


Fig. 4 Spectrum (amplitude RMS) of absolute bearing housing vibrations in axial-radial plane in direction X. The highest lines on 12.5 Hz and 10 Hz are marked. Spectral composition up to 100 Hz.

Two main lines are visible: the one on 12.5 Hz, which corresponds to the 1x harmonic of rotational frequency (750 rpm = 12.5 Hz) and is related to the rotor rotation and the line at 10 Hz whose existence is related to the vibrational response on system's natural frequency (this was additionally confirmed with a 'bump' test). Additionally, harmonics of rotational frequency are also visible (lines on 25 Hz and 37. 5 Hz) but are smaller in amplitude.

The increase in vibrational response with temperature can be explained through natural frequency increase as a result of additional stiffness increase related to temperature. During the measurements, the additional stiffening elements ('legs') were attached to the flange with appropriate screws and the warm-up increased the natural frequency towards the nominal rotational frequency. The vibrational responses with rotational speed increase (as a result of 3 MW load rejections) for the period around 16 h ('cold' state) and around 18 h ('hot' state) were compared. For this reason, Fig. 5~~Fig. 5~~ is given on which it can clearly be seen what the system natural frequencies are for both situations (~625 rpm for 'cold' and ~730 rpm for 'hot' state).

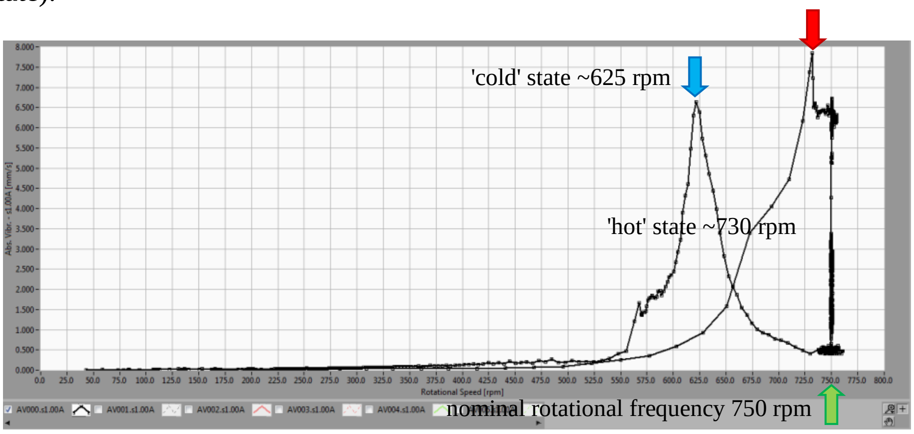


Fig. 5 Natural frequencies in 'cold' (blue arrow) and 'hot' (red arrow) during the 3 MW load rejection tests which resulted in rotational speed increase followed with a run-down.

For this reason, it was decided to loosen (unscrew) the additional stiffening elements ('legs') and this was done at 18:17 h (the moment is marked on Fig. 3~~Fig. 3~~). It can be notice that the vibrations were immediately reduced and the largest change is in direction X of the axial-radial bearing. In the next period it can be seen that there exist intermittent increases and decreases in vibration levels (marked on Fig. 3~~Fig. 3~~) but the levels were not too high.

After that, unit rotor was balanced to further decrease the vibration levels and, below 1.3 mm/s. The problem of increased vibrations during unit run-up and run-down remained but these vibrations did not last too long so it was decided to leave the unit in this state and follow its vibrational response.

In the following period, the permanently installed SCADA system which included the main vibrational descriptors (RMS values) showed increased vibrations still outside of acceptable limits for the end-user (below 2.5 mm/s). Therefore, the problem was not solved and additional measurement configurations was proposed to solve the issue.

Second measurement campaign on Unit 2

In order to further diagnose the problem, the focus was given on the identification of the generator pedestal stiffness and its mounting. Therefore, the sensors and measurement configuration were chosen help identify the foundations construction stability. The measurements included:

- rotational speed (as in previous measurements)

- absolute vibrations in the axial-radial bearing plane in X and Y directions (as in previous measurements)
- axial vibrations (direction Z) of the pedestal flange (absolute vibrations) next to the spiral case on 4 positions mutually under 90° starting from direction X
- relative vibrations of generator flange in two radial directions mutually at 90° measured relative to the foundation (the sensors were placed on additional stiffening elements, which are free from the generator flange)
- relative vibrations of the generator flange in axial direction on 4 positions relative to the foundation (sensors were placed on all 4 additional stiffening elements)

As an example of sensor placement, Fig. 6~~Fig. 6~~ is given.

Relative flange vibrations in radial and axial directions | Sensor placement on a unit | Axial flange vibrations next to the spiral

Fig. 6 Photos of the sensors on measurement positions relevant in evaluation of pedestal stiffness and foundations stability.

The first important results were related to the axial flange position and its displacement when the water entered the spiral case. It is a statical (DC) position, that is, a distance from the sensor's tip of 4 sensor placed on additional stiffening elements. It should be noted (as it is visible from Fig. 6~~Fig. 6~~) that the stiffening elements are separated from the flange and, since they are firmly connected to the foundations (screwed into it) and do not vibrate when the unit is running (which was proven with measurements) they should be treated as a foundation. Therefore, flange displacement relative to foundation can be measured.

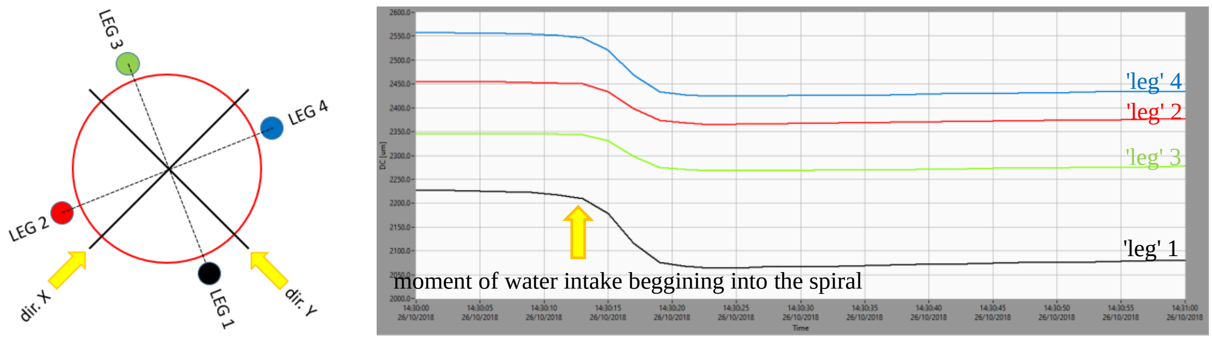


Fig. 7 Sensor configuration in axial direction (Z). LEFT: Top view with radial sensor measurements X i Y and stiffening elements . RIGHT: Shaft static centerline position (DC) in axial direction (Flange to sensor)

Fig. 7~~Fig. 7~~ trends clearly show how the spiral changes its volume (increases) due to water hydrostatic pressure (~15 bar) and, since the generator pedestal is mounted on it, as a result it also lifts. This results in reduction of distance between pedestal and sensor on additional stiffening elements. It is also visible that this increase is asymmetrical – that is it is not the same for all stiffening elements. For example, on 'leg' 1 it is the largest and is ~160 µm while on 'leg' 3 it is the smallest and is ~80 µm.

Additionally, large vibrations in axial (direction Z) direction on the flange between the generator pedestal and spiral case were noticed when the unit was running. These are unusually large values where one would expect the vibrations to be negligible. Also, it was noticed that the vibrations on opposite sides are out-of-phase.

Fig. 8~~Fig. 8~~ shows the visualization of the vibration deflection shape according to vibration vectors obtained from the sensors on the flange, next to the spiral case, with the expected vibrational vector on the measurement position near the position on top of the generator. It should be additionally pointed out that these are 10 Hz vibrations and not vibrations on nominal rotational frequency of 12.5 Hz.

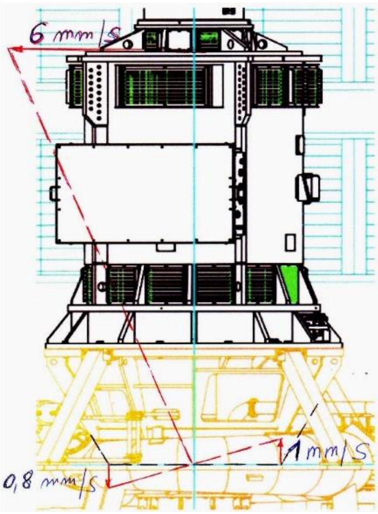


Fig. 8 Visualization of the unit deflection shape according to the measurements. Vibrations on 10 Hz (natural frequency).

As an additional proof that the vibrations are predominantly on 10 Hz, a spectra of absolute vibrations (larger from the two directions) is given on Fig. 9~~Fig. 9~~ at the moment of largest vibrations during measurements.

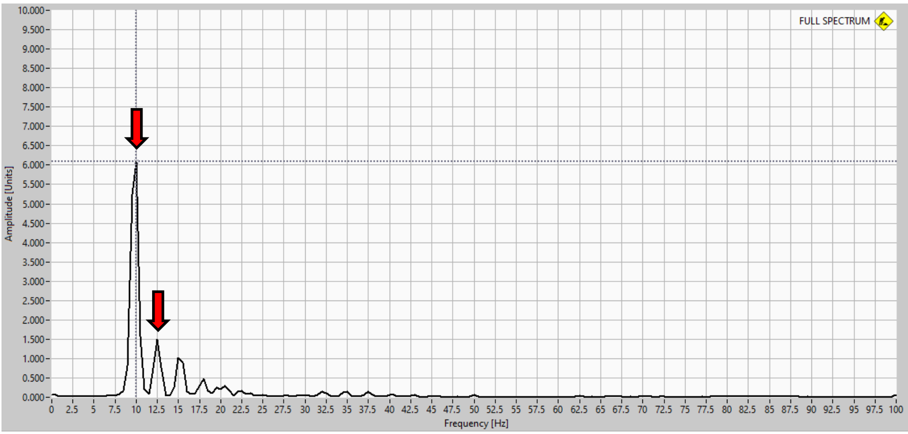


Fig. 9 Spectra (amplitude RMS) of absolute generator bearing housing vibrations in axial-radial plane in direction Y. Two largest lines on 10 Hz and 12.5 Hz are marked. Spectra is shown up to 100 Hz.

It can be clearly seen that the dominant vibrational response is at natural frequency of 10 Hz so the problem needs to be solved in such a way as to increase this frequency above nominal operating frequency of 12.5 Hz. This will be done by actions in the construction foundation region.

Another important question is why the vibrational response is largest at 10 Hz (which corresponds to the lowest construction natural frequency) if there is no external force exciting the system at this frequency? The answer to this question is related to the foundation state below the generator pedestal flange. The contact of the steel part and concrete is weak and poorly defined. It is also continuously changing due to water axial force. Therefore, the steel part of the foundation structure is 'bumping' against the concrete which results in the construction response on natural frequency. When the 'bumping' stops there is no natural frequency vibrations and/or the vibrations on 10 Hz are damped. This is exactly the reason why the vibrations on 10 Hz are not present the entire time but occur intermittently.

From Fig. 8~~Fig. 8~~ and the geometry one can see that ~6 mm/s is expected for a perfectly rigid pedestal. Since ~7 mm/s was measured, which is a very similar number, it was concluded that the generator pedestal's stiffness is adequate and that for the existing vibrational state the structure stiffness distribution below the spiral flange is responsible. The generator is founded on this flange through the bearing pedestal. Therefore, the overhaul procedures should be defined in changing the properties of this structure. It is a part of the spiral case with concrete filament from the spiral case to the draft tube. This part of the structure is not available for visual inspection but it exactly here that the repair works must be done in order to increase stiffness, that is, structure resistance to axial vibrations.

Therefore, since the statical (when the water entered the spiral) and dynamical (large flange axial vibrations) unit instabilities were confirmed – the overhaul procedure should only take into account the unit foundations. In other words, the problem solution lies in increasing the stiffness below generator pedestal.

For this reason – two overhaul suggestions were proposed to the power plant personnel and the repair works crew:

PROPOSITION 1 – concrete injection under pressure:

The basis of this proposition is injection of concrete under pressure into the area between the spiral case, draft tube and lower part of the spiral flange. It is evident that there are voids where concrete should be present and this reduces the unit's foundation stiffness. Additionally, this proposition included insertion of additional steel rods (reinforcement) which will increase stiffness to the structure.

PROPOSITION 2 – new pedestal:

The basis of this proposition is fabrication and installation of new generator pedestal which would be mounted directly on the foundation without any connection to the spiral. This suggestion is more complicated to carry through but it eliminates the influence of spiral deformations and the part of the concrete which is below the spiral on the statical and dynamical unit stability.

Since it's possible to apply proposition 2 even if proposition 1 does not provide satisfactory results and proposition 1 is easier to carry through, it was decided to go with proposition 1 and check what is the influence on unit vibrational response.

During the repair works, a relatively large zone was found in which the concrete and steel construction contact is poor. This is marked on Fig. 10~~Fig. 10~~ (left part of the figure).

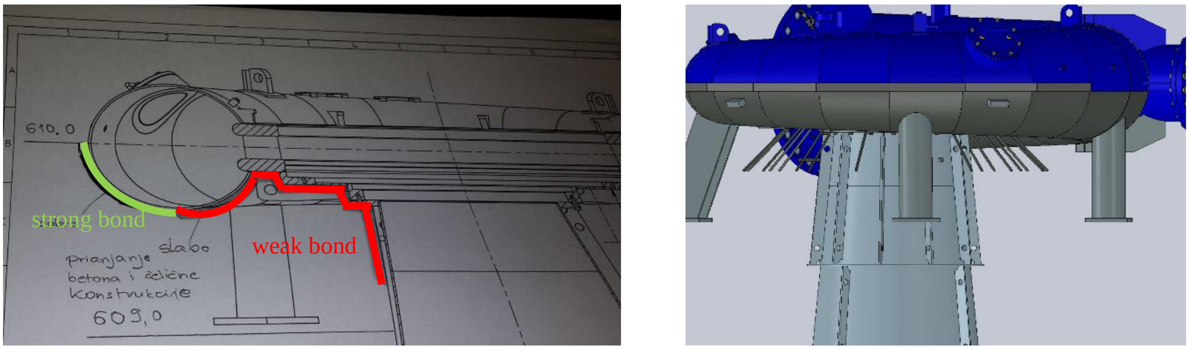


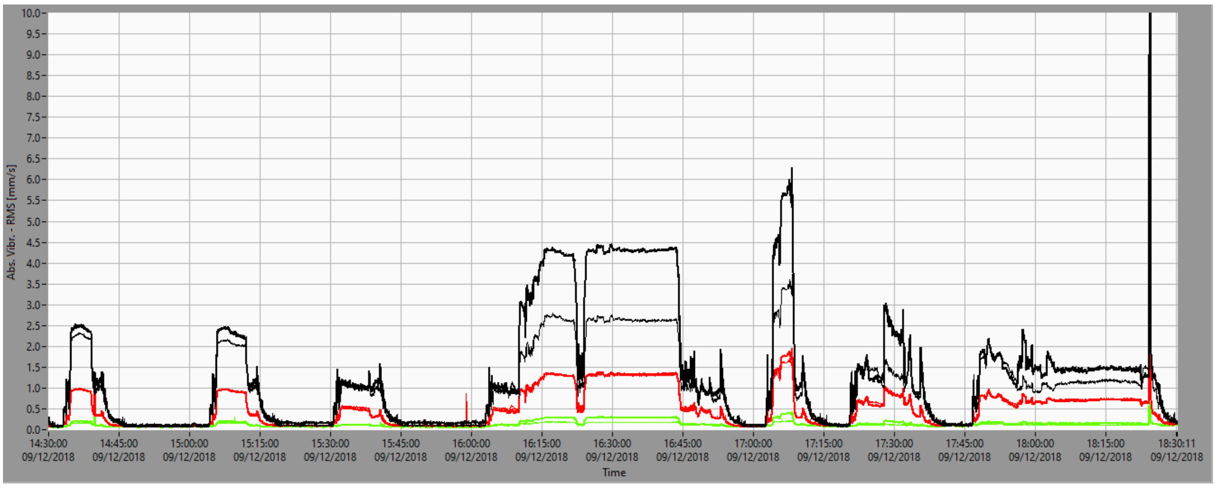
Fig. 10 LEFT: Overview of as-found state during repair works. RIGHT: Additional steel rods (concrete reinforcement) added during repairs.

On the right side of the same figure, proposed steel rod reinforcements added during repair works are shown.

Measurements after the overhaul

After the overhaul, vibration measurements were performed. Since the goal of the overhaul was to move the natural frequency of the unit above the nominal rotational frequency, additional stiffening elements ('legs') were also taken into account and screwed onto the flange.

On Fig. 11~~Fig. 11~~ results of the RMS values of vibrational velocity (upper diagram) and 1x harmonic of rotational frequency of vibrational velocity (lower diagram) are shown. Since the RMS value is almost identical to the 1x harmonic, it is clear that the 1x harmonic is dominant. This was confirmed by spectral analysis and it was additionally established that the response at 10 Hz is no longer present. This is a proof that the natural frequency is above the nominal rotational frequency.



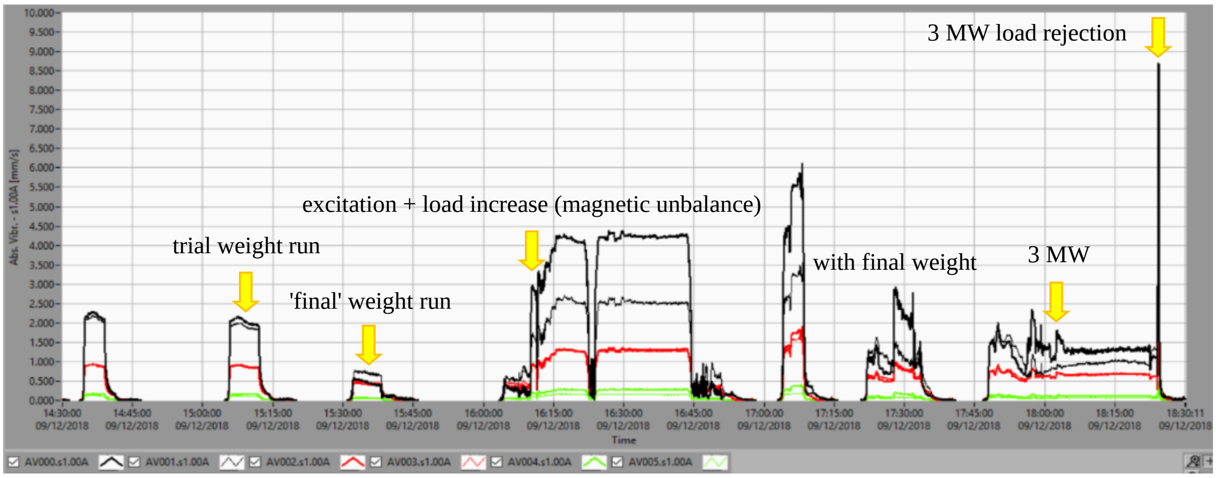


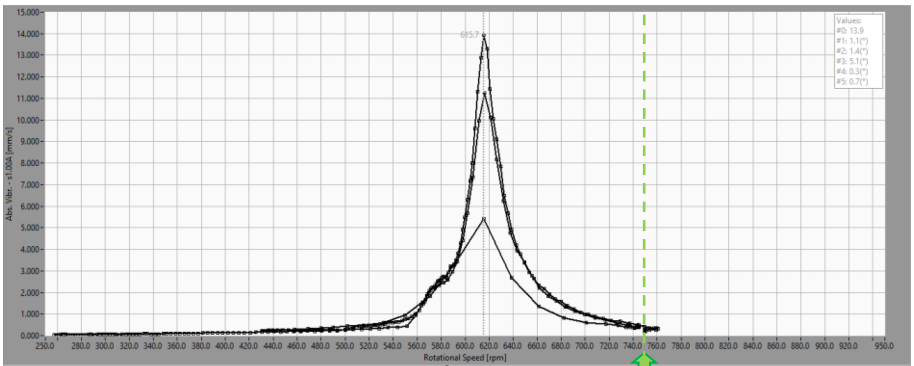
Fig. 11 UPPER DIAGRAM: Trends of vibrational velocity RMS values [mm/s] LOWER DIAGRAM: Trends of 1x harmonic of vibrational velocity (amplitude RMS) for: axial-radial bearing plane (black, X and Y directions), radial bearing plane (red, X and Y directions), flange connection of generator pedestal to the spiral (green, Z direction on two positions).

At first, the unit was balanced in mechanical rotation. By comparison of the vibration phase relations in the axial-radial and radial planes it was established that the vibrations are in phase. Therefore, it was sufficient to conduct the balancing in one plane only. From Fig. 11~~Fig. 11~~ it can be seen how the balancing procedure was conducted during the first three unit runs (run-up and run-down). The first rotation was without balancing weight, the second with the trial weight in one plane and the third with the 'final' weight for mechanical rotation. It can be seen that the vibrations during the third run are now very low and within limits acceptable to the end-user (below 2.5 mm/s).

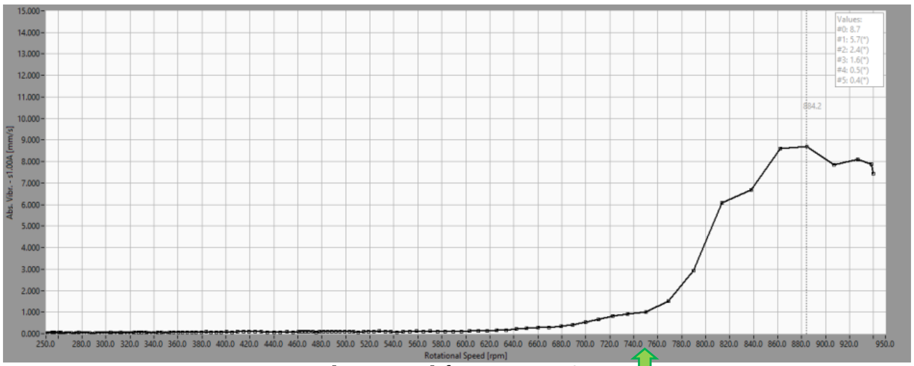
After that, the unit was started and excitation applied, followed by synchronization and load increase. The abrupt increase in the vibrations is visible which occurs with excitation as a consequence of the rotor magnetic unbalance(. Therefore, additional balancing was necessary (in second plane) in order to obtain vibrationally acceptable state in all of the operating regimes. In other words, vibrations needed to be redistributed to be acceptable in all operating regimes.

The criteria of acceptable vibrations for long-term problem free operation was defined by the end-user to not exceeding ~2.5 mm/s in normal operation.

From Fig. 11~~Fig. 11~~ it is visible how this was achieved during the 7th run and then, the decision was made to perform 3 MW load rejection in order to make possible the comparison of unit's vibrational behavior prior to and after the overhaul. The main question was at what value the unit's natural frequency was increased. For this reason. Fig. 12~~Fig. 12~~ is given showing the natural frequency before and after the overhaul.



nominal rotational speed 750 rpm / nominal frequency: 12.5 Hz



nominal rotational frequency 750 rpm

Fig. 12 UPPER DIAGRAM: 28.09.2018. – before the overhaul. Natural frequency ~10 Hz. LOWER DIAGRAM: 09.12.2018. load rejection from 3 MW – after the overhaul. Natural frequency ~14.75 Hz

4. CONCLUSION

Vibration related issues on 5 MW units was (for both units) a consequence of the inadequate spiral case foundation and the fact that the generator pedestal was bolted to the spiral case. This caused resonance on every run-up and run-down. Additionally, due to poor spiral to foundation contact, natural frequency at 10 Hz was intermittently excited and the unit was vibrating on this frequency during normal operation.

During the overhaul, concrete was injected under pressure below the spiral. Additionally, steel rods reinforcements were added and additional stiffening elements used from previous overhaul attempts.

As a result, the natural frequency was increased from ~10 Hz (600 rpm) to ~15 Hz region (900 rpm).

If the unit rejects load when operating above 2.5 MW, resonance effect will be present. But, during the runaway speed the unit will pass through resonance and the response at runaway speed will be acceptable not causing problems for the unit.

This state is perfectly acceptable and both units are operating without problems from the overhaul (end of 2018.).

[1]