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EXPERIMENTAL ANALYSIS OF VIBRATIONS AT A PELTON TURBINE

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The paper presents results of an experimental study on the vibration levels of a small Pelton turbine operating both with balanced and unbalanced runner. To unbalance the runner, a concentrated mass was added on a bucket, close to the runner periphery. Vibrations were measured in axial, transverse and vertical directions at different nozzle openings, i.e. at different discharges, and at different runner speeds and turbine loads. The results obtained suggest that the vertical vibrations at large nozzle openings and runner speeds are mostly caused by the flow and less by the turbine load. The axial vibrations are larger for the balanced runner. Maximum values of displacement RMS at multiples of the rotational frequency indicate a shaft misalignment and bearings issues. The influence of the added mass at small discharges and runner speeds is marginal.
Key words: vibration, experimental analysis, mass unbalance, hydraulic turbine, Pelton.

1. INTRODUCTION

The monitoring of vibrations generated by hydraulic equipment that is used in hydro power plants (HPP) is an actual issue strongly connected with the operating status of the hydropower system in power generation. The vibrations of rotating and non-rotating parts of the hydraulic equipment of an HPP have mechanical, hydraulic, and electrical causes [1]. The complexity of the flow inside the runner of a hydraulic reaction turbine, the hydraulic imbalance on the buckets of a hydraulic impulse turbine, the mechanical imbalance of the rotating parts of the turbine, the electrical imbalance of the electrical generator assembly, the misalignment between the turbine shaft and the shaft of the electrical generator, cavitation phenomena, abrasive erosion are only a few examples of sources of vibration [1]. For this reason, the vibration measurements, as part of the vibration condition monitoring method, is a powerful tool in predictive maintenance [2, 3].

Even operating regimes that deviate from the Best Efficiency Point (BEP), which is usually the rated operating point, can cause vibrations in a hydraulic turbine [4] and the mechanical system must respond to extreme off-design operating conditions, when the whole dynamic behavior of the turbine (runner, shafts line, and electrical generator) is changed [5].

Research papers in hydraulic machinery show that the vibrations of a hydraulic turbine can be divided into vibrations generated by mechanical and technological factors and vibrations generated by added masses, that are usually caused by changes in the discharge of the hydraulic turbine at different operating conditions.

Studies on the vibration behavior of a body with an added mass that is working in water or air started to be carried out at the beginning of the twentieth century on hydro and aerospace vehicles with cylindrical walls [6]. With the improvement in the design methods of hydraulic turbines, new issues regarding the dynamic behavior arose. The influence of an added mass that is uniformly distributed around the runner of a hydraulic turbine was studied by the team of “Center of Industrial Diagnostics and Fluid Dynamics” at Technical University of Catalonia, Barcelona, Spain, which was led by Professor Eduard Eguisquiza. In their papers results are presented, which were obtained by carrying out theoretical, experimental and numerical studies on the behavior of the runner of a vertical Francis turbine, operating with different added masses of air or water [5, 7-10]. The experimental results showed that the natural frequencies of the runner filled with water are reduced when compared to those obtained for the same runner filled only with air [5, 10] and the mode-shapes obtained with air are similar to those obtained with water. The term "added mass effect" was introduced.

In this paper, the natural frequencies of a horizontal Pelton turbine are measured at different operating regimes. Two cases were considered: *(i)* the balanced Pelton runner (denoted further as PRCE) with uniformly distributed mass and *(ii)* the unbalanced Pelton runner (denoted further as PRNE) with a concentrated mass of 0.275 kg added close to its periphery in order to simulate the behavior of the turbine when the mass of the runner is no more uniformly distributed. In situ, the latter case can occur when the runner losses some of its mass, usually due to abrasive erosion caused when the runner buckets are attacked by the high velocity water jets that can contain small silt particles.

In the following, the experimental set-up and the measuring procedure are described. Obtained results are then presented and discussed. Based on the results, conclusions are subsequently drawn.

2. EXPERIMENTAL SET-UP

The experimental installation is presented in Figure 1. The Pelton turbine (1) is supplied with water from a pump station through an upstream pipe (2). The turbine runner has 14 buckets casted in one piece with the runner wheel. A shut-off valve (3) mounted on the upstream pipe allows to adjust both the head and the discharge of the turbine. The discharge is measured with and orifice meter (4) installed in the upstream pipe. The pressure taps of the orifice meter are connected to a differential pressure manometer with mercury (5). The upstream pipe ends with the turbine nozzle, whose spear, mounted on a threaded rod, can be manually adjusted by means of the hand wheel (6). The static pressure at turbine entry, i.e. at nozzle entry, is measured with a pressure gauge (7). After exchanging energy with the turbine runner, the water falls freely into a tail race (8) located below the turbine that leads the water back into the suction reservoir of the pump station. The turbine runner is mounted on the turbine shaft that sits on two ball bearings installed inside the turbine casing. A jaw coupling (9) connects the turbine shaft to the shaft of a hydrodynamic brake (10) that allows to adjust different loads of the turbine by changing the degree to which the brake is filled with water. The brake casing can rotate to some extent, being linked to a scale (11) for measuring the braking moment, M, which equals the torque at the turbine shaft. The turbine speed, n, corresponding to a turbine load was measured with a digital tachometer Testo 470 (12) with a measuring range of 1...9,999 rpm and an accuracy of 0.02% of the reading. The vibrations were measured with three Brüel & Kjaer DeltaTron accelerometers (13) of type 4507 B 001 with built-in preamplifier. They were mounted on the turbine casing, close to the nozzle, along three perpendicular directions: axial (parallel with the turbine shaft), transverse (parallel with the nozzle and perpendicular to the shaft) and vertical. The accelerometers have the reference sensitivity of 20 m/s^2 RMS at 159.2 Hz, the measuring range ±7000 m/s^2 peak, and the frequency ranges of 0.1 Hz to 6 kHz for amplitude and 0.5 Hz to 5 kHz for phase. Via a NI USB-9233 portable bus-powered USB carrier (14), the accelerometers were connected to a USB port of a laptop (15) for data acquisition.

Tests were carried out at different nozzle openings, a_0, expressed as revolutions of the hand wheel (6). For the balanced runner, the openings were of 0.25, 0.5, 0.75, 1, 2, 3, and 4 revolutions. For the unbalanced runner, tests were carried out only at small openings of 0.25, 0.5, and 0.75 revolutions, to avoid damaging the turbine due to the additional forces caused by the added mass. At each nozzle opening, the differential height at the differential pressure manometer, Δh, and the static pressure at turbine entry, p_1, were measured. As long as the nozzle opening remains unchanged, these values do not change with turbine load and, consequently, the discharge and turbine head do not change either with turbine load. This behavior is normal since the Pelton turbine is a pure impulse turbine, whose runner operates at constant pressure. With the help of the hydrodynamic brake, four loads were adjusted at each nozzle opening, going from overspeed, when the brake was empty, to the lowest speed that was possible to attain when the brake was fully filled with water.

For each load, the turbine speed, n, the shaft torque, M, and the vibrations were measured.

Based on the differential height Δh, the turbine discharge was obtained. Figure 2 shows the discharge depending on nozzle opening. As expected, the discharge increases with the nozzle opening, yet the increase become gentler as a_0 increases. This is typical for Pelton turbines.

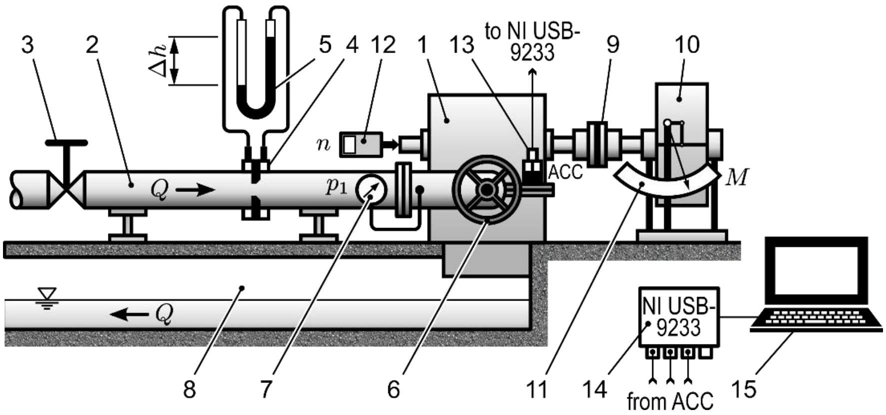


Figure 1 - Sketch of the experimental set-up.

The static pressure at turbine entry, p_1, was used to calculate the turbine head. Finally, with the turbine speed, *n*, and the shaft torque, *M*, the turbine efficiency was obtained both for PRCE and for PRNE. Figure 3 shows the efficiency curves depending on turbine speed and nozzle opening. An interesting finding is that the runner seems to have higher efficiencies when it is unbalanced. However, that could be most probably the result of the fact that the nozzle opening cannot be adjusted very accurately.

The measured vibrations were processed with the dbTrait software. All accelerations were filtered with an RMS filter and their Fast Fourier Transforms (FFTs) were subsequently calculated. The FFTs of the accelerations were then used to obtain FFTs of velocities and displacements.

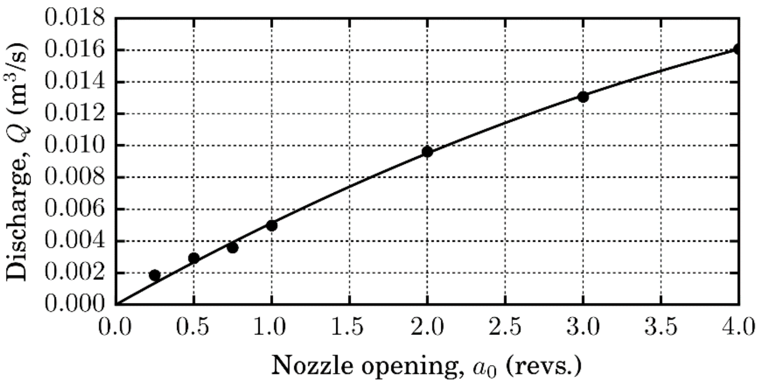


Figure 2 - Turbine discharge depending on nozzle opening.

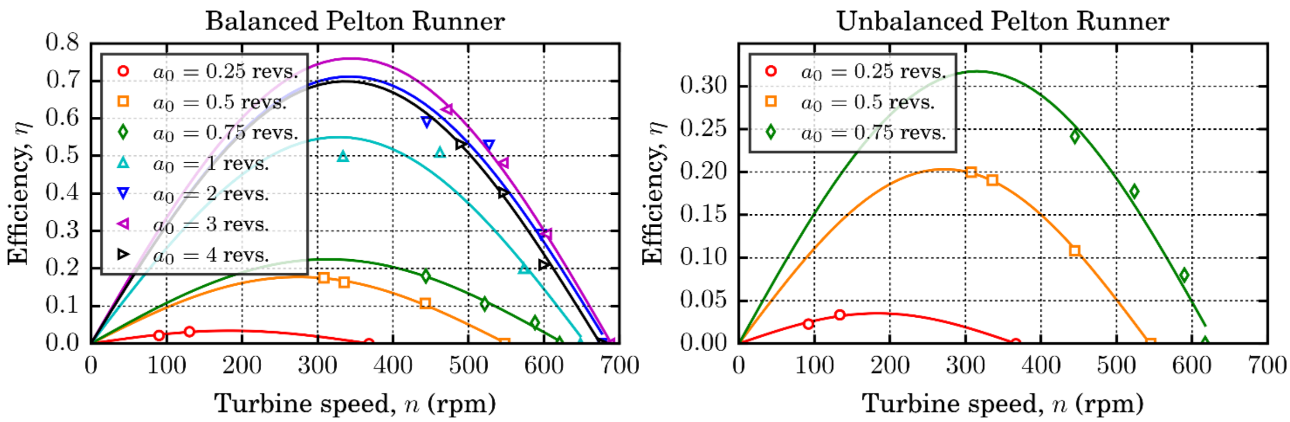


Figure 3 - Efficiency curves of the Pelton runner operating both balanced and unbalanced.

3. RESULTS

The vibration analysis reveals that, overall, the vertical vibration level is significantly lower than that of axial and transverse vibrations, regardless of whether FFT charts are being analyzed in velocities or displacements and independent of nozzle opening and runner speed (i.e. independent of turbine discharge and load). However, at large nozzle openings (Figure 4a) the differences decrease as opposed to the case of small openings (Figure 4b). The global RMS level of axial vibrations is quite close to that of transverse vibrations (both as displacements and velocities) and independent of nozzle opening and runner speed. It can be noticed that the level of global RMS of vibrations in axial direction is slightly superior to those of transverse vibrations. The differences between them are lower when the discharge is higher, i.e. at larger nozzle openings (Figure 5).

a)

b)

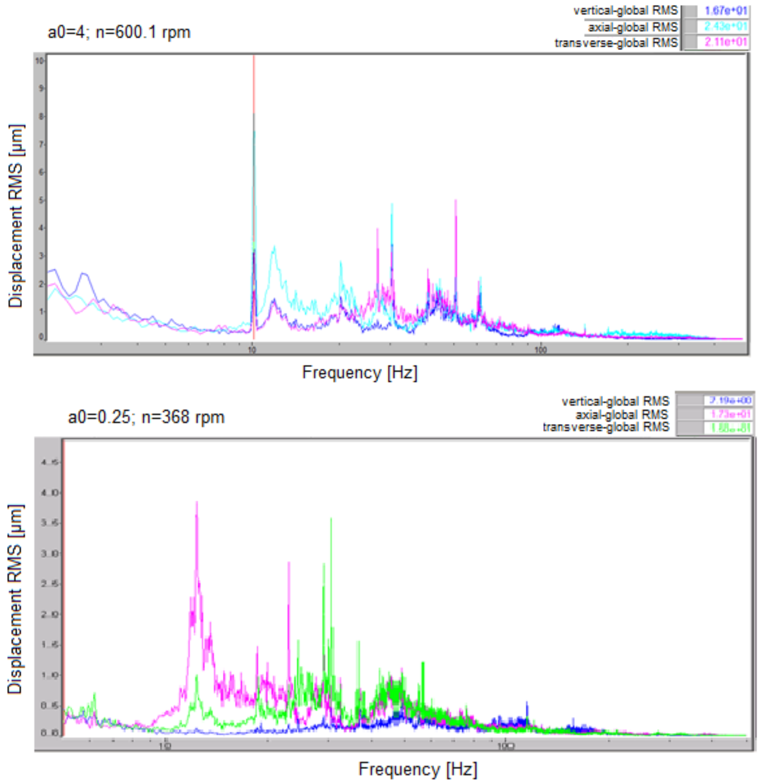


Figure 4 - Comparison of vibration levels in the three directions.

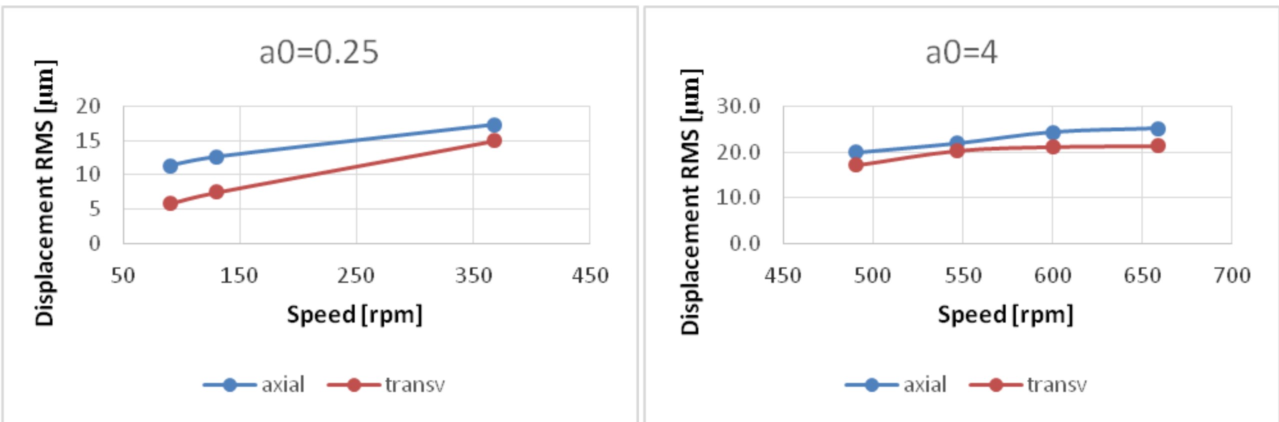


Figure 5 - The level of global RMS of vibrations.

In the case of the PRNE it was found that the global vibrations recorded in axial and transverse direction show, from the point of view of displacements RMS, the following characteristics (Figure 6):

- the level of transverse vibrations is higher than that of axial vibrations;
- vibration levels increase with increasing nozzle opening, i.e. with increasing discharge;
- at overspeed (i.e. no turbine load) the vibration levels are superior to those recorded when the load increases and the speed decreases.

When comparing the results obtained for PRNE with those obtained for PRCE, the following remarks can be made:

- in case of PRNE, the global RMS level of axial vibrations is reduced by almost 50%, while in transverse direction it is increased by about 20%, proportionally to the increase of nozzle opening and to the load decrease (Figure 7);

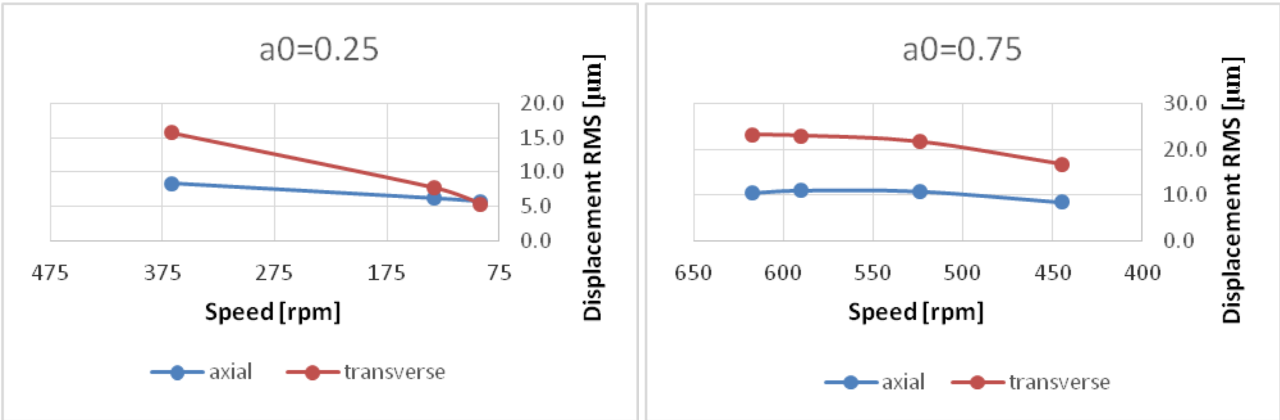


Figure 6 - Levels of global RMS of vibrations for the balanced Pelton runner.

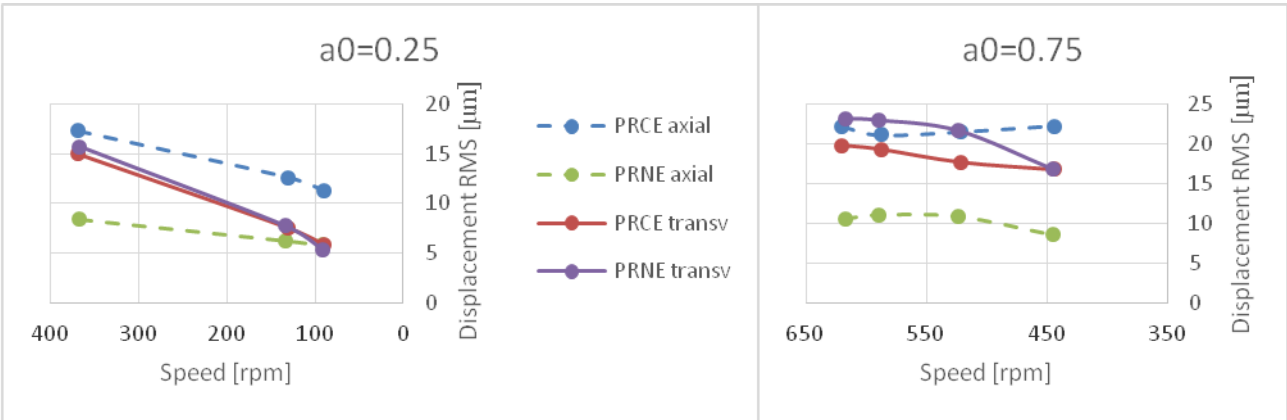
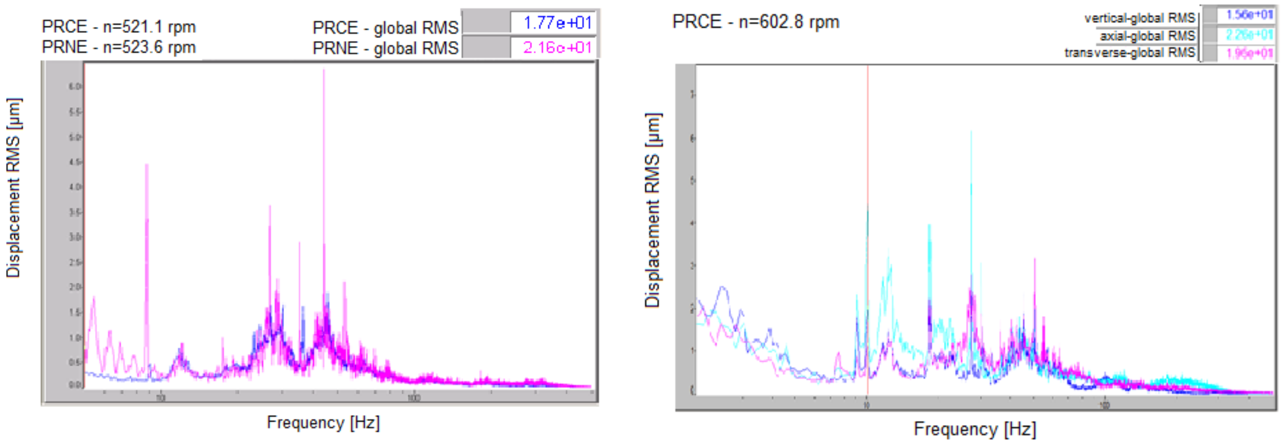


Figure 7 - Comparison between global RMS levels of vibrations.

- in case of PRNE, high (sometimes the highest) RMS amplitudes are observed at rotational frequency, while in case of PRCE the amplitudes are small or even very small (Figure 8a);
- in case of PRCE, maximum values of displacement RMS are observed at upper harmonics of 2X, 3X, 4X, and even 5X the rotational frequency at all nozzle openings (Figure 8b);
- the maxima corresponding to rotation frequencies and their harmonics become more significant with the increase in discharge and runner speed and the decrease in turbine load.



a) a_0 = 0.75 b) a_0 = 3

Figure 8 - Evolution of the displacements RMS at different nozzle openings, runner speeds and directions of vibration

4. CONCLUSIONS

The paper presents an experimental vibration analysis at a Pelton turbine for different operating regimes. Such analysis can be useful in vibration diagnosis as a method for monitoring vibrations of hydraulic equipment.

It can be generally said that the global vibrations studied have mechanical causes. Several conclusions can be drawn.

The vertical vibrations have levels comparable to those of axial and transverse vibrations at large nozzle openings and runner speeds. That means that the vibrations are mostly caused by the flow and less by the turbine load.

In case of PRCE, the axial vibrations have superior levels. Also, maximum values of displacement RMS (in both axial and transverse direction) are observed at upper harmonics of 2X, 3X, 4X, and even 5X the rotational frequency. This is a behavior specific to a coupling fault or incorrect adjustment of the shaft or to bearings issues.

In case of PRNE, the influence of the added mass on the transverse vibrations at small discharges and speeds is insignificant. As the discharge increases at low turbine loads, the level of transverse vibrations increases when compared to the case of PRCE. This is a normal behavior due to the mass imbalance.

The occurrence of maximum RMS amplitudes at rotational frequency maxima is a specific element due to the mass imbalance in both axial and transverse directions.

From the analysis of the obtained spectrograms, both in displacements and in velocities, it was observed that there are frequencies at which maximal RMS amplitudes appear, which depend neither on the discharge nor on the speed. These are of roughly 12.5 Hz, 28 Hz, and 45 Hz. They could most probably have hydraulic causes.

REFERENCES

1. Mohanta, R.K. Chelliah, T.R., Allamsetty, S., Akula, A., Ghosh, R., *Sources of vibration and their treatment in hydro power stations-A review*, Engineering Science and Technology, an International Journal, **20**, 637–648, 2017.
2. Carp-Ciocardia, C., Dragomirescu, A., Safta, C-A., Schiaua, M., Magheti, I., *Experimental identification of possible failures of Bánki turbine*, J. of Acta Technica Napocensis, Series: Applied Mathematics, Mechanics and Engineering, Vol. 60, Issue IV, pp. 545-550, Nov. 2017.
3. Mircea, V., Radu, T., Danut, D., *Critical analysis of vibration sources of hydro aggregates in operating regime*, J. Sustainable Energy **2**, 1–8, 2011.
4. Roman, R., Dunca, G., Bucur, D. M., Cervantes, M., Panaitescu, V. N., *Added properties effect in hydraulic turbines*, U.P.B. Sci. Bull., Series D, Vol. 78, Iss. 4, 225-236, 2016.

5. Rodriguez, C.G., Egusquiza, E. Escaler, X., Liang, Q.W., Avellan, F., Experimental investigation of added mass effects on a Francis turbine runner in still water, Journal of Fluids and Structures 22, 699–712, 2006.
6. Palmer, E. W., *The Influence of Added Mass on The Natural Vibrations and Impulse Response 'Of Long*, Thin- Cylindrical Shells, Depart. Of the Navy Naval Ship Research and Development Center, Technical Report 3395, October 1970.
7. Egusquiza, E., Valero, C., Liang, Q., Coussirat, M., Seidel, U., *Fluid Added Mass Effect in The Modal Response of A Pump-Turbine Impeller*, Proceedings of the ASME 2009 International Design Engineering Technical Conferences & Computers and Information in Engineering Conference, IDETC/CIE 2009, August 30 - September 2, 2009, San Diego, California, USA, DETC 2009, 86830.
8. Valentin, D., Presas, A., Egusquiza, E., Valero, C., *Influence of the added mass effect and boundary conditions on the dynamic response of submerged and confined structures*, 27th IAHR Symposium on Hydraulic Machinery and Systems (IAHR 2014), IOP Conf. Series: Earth and Environmental Science **22** (2014) 032042, doi:10.1088/1755-1315/22/3/032042.
9. Presas, A., David Valentin, D., Egusquiza, E., Valero, C., Seidel, U., *On the detection of natural frequencies and mode shapes of submerged rotating disk-like structures from the casing*, Mechanical Systems and Signal Processing 60-61, 547–570, 2015.
10. Presas, A., David Valentin, D., Egusquiza, E., Valero, C., Seidel, U., *Dynamic response of a rotating disk submerged and confined. Influence of the axial gap*, Journal of Fluids and Structures 62 332–349, 2016.