TECHNICAL UNIVERSITY OF CLUJ-NAPOCA

ACTA TECHNICA NAPOCENSIS

Series: Applied Mathematics, Mechanics, and Engineering
Vol. 60, Issue IV, November, 2017

EXPERIMENTAL IDENTIFICATION OF POSSIBLE FAILURES OF A BÁNKI TURBINE

Crăița CARP-CIOCÂRDIA, Andrei DRAGOMIRESCU, Carmen-Anca SAFTA, Mihai SCHIAUA, Ioan MAGHEȚI

Abstract: *The aim of the present paper is to investigate the vibrations that occur during the operation of a low power Bánki turbine. The turbine was tested at different operating regimes in a laboratory installation, where it can operate with or without a variable load provided by a hydrodynamic brake coupled to the turbine shaft. Vibrations were measured on the turbine casing, on one of the turbine bearings, and on the brake. Dominant frequencies pertaining to each subassembly were identified. Measurements were also carried out with an added mass that simulates a piece of debris stuck inside the runner, causing an unbalance. The results obtained suggest that the bearings are a source of high frequency vibrations while relatively low frequency vibrations are caused by the turbine load. Very low frequency vibrations were also identified, which might be transmitted from the installation through the upstream pipe. The results underline the importance of vibration diagnostics to identify possible failures of a Bánki turbine, as a predictive maintenance method specific to rotating machines.*

Key words: *small hydraulic turbine, vibration diagnostics, added mass, unbalance, bearing, dominant frequencies, predictive maintenance.*

1. INTRODUCTION

The method of vibration condition monitoring used in hydropower plants is a predictive maintenance tool which refers to online vibrations monitoring applied on different mechanical and electrical parts of the rotating machine [1]. Various mechanical, hydraulic, and electrical factors are sources of vibrations of a hydro generator. Wear or breakdown of a bearing, mechanical unbalance caused by an added or subtracted mass, cavitation and abrasive erosion, hydrodynamic interaction between guide vanes and turbine runner, shafts misalignment, eccentricity of the air gap in the electromagnetic field of the generator, or pressure fluctuations are possible sources of abnormal vibrations [1-7].

Even in small hydropower plants (SHPs), the vibration condition monitoring is important for the same reasons: low cost of maintenance process, high efficiency in operation, longer lifetime of the SHP equipment.

A special attention is payed currently to the usage of Bánki turbines in SHPs. These turbines are partial admission cross-flow water turbines used to harvest renewable energy. They usually produce low powers that span from a few kilowatts to a few hundred kilowatts. Due to their advantages – simple construction and maintenance, low price, flat efficiency curve, good behavior with partial loads, the possibility to operate unattended – these turbines are well suited to provide electricity in remote areas that are not connected to the grid.

In this paper a small Bánki turbine is investigated. The turbine is mounted in a laboratory installation were different operating regimes can be simulated. The vibrations of the turbine both at normal operation and when operating with unbalanced runner were measured and analyzed. In the following, the experimental installation and the turbine are described. Experimental results are then presented and discussed. Based on the results, several conclusions regarding the sources of vibrations and possible failures are formulated.

2. EXPERIMENTAL INSTALLATION AND TURBINE PERFORMANCE

The experimental installation is presented in Figure 1. The turbine is fed with water from a pump station through an upstream pipe foreseen with a shut-off valve and a Venturi tube for discharge measurement. After passing through the turbine runner, the water falls freely into a tail race. The turbine is coupled through a bush pin type flange coupling (i.e. an elastic coupling) to a hydrodynamic brake for simulating different loads and for measuring the torque (or shaft moment). The turbine and the flow inside it, obtained by numerical simulations by Dragomirescu and Schiaua [8], are depicted in Figure 2.

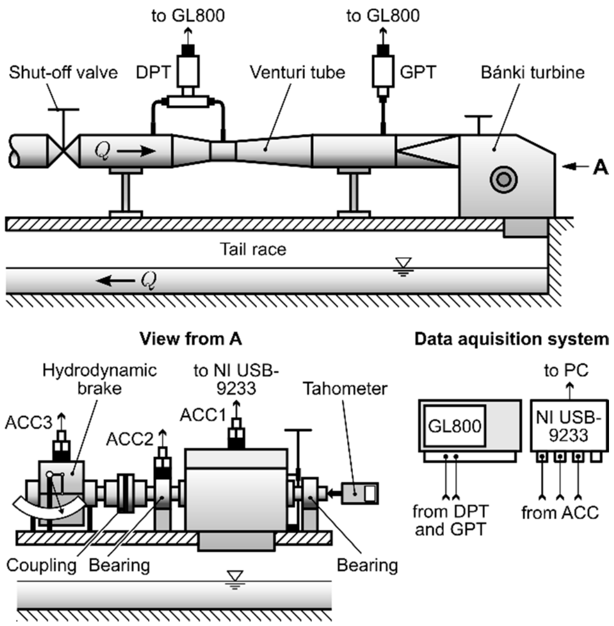

Fig. 1. Experimental installation

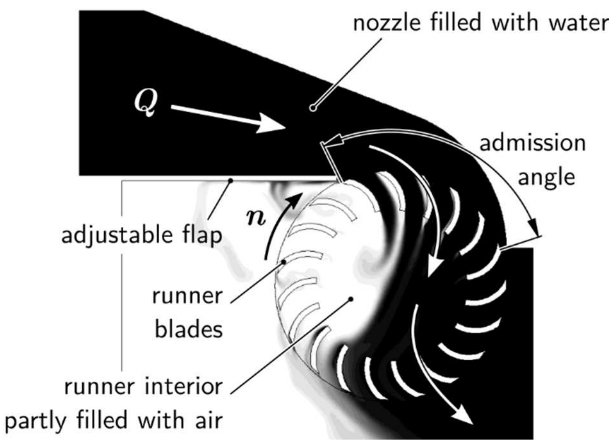

Fig. 2. Turbine cross-section and the flow inside it [8]

The water enters into the turbine through a nozzle which creates a water jet that attacks the runner on a limited angle of its periphery, called admission angle. An adjustable flap mounted inside the nozzle allows the variation of the discharge. The runner consists of blades mounted between two end discs at their rims.

After passing through the blade channels from runner outside to inside, the water crosses the bladeless inside of the runner, flows through the blade channels a second time, from runner inside to outside, and then exits the runner and falls into the tail race. Since the water passes through the blade channels twice and crosses the runner, the turbine is a two-stage cross-flow one. The runner operates at atmospheric pressure and therefore no risk of cavitation is present. Only a part of the runner is filled with water, the other part is filled with air. In the present study, the runner has 19 blades and its outer and inner diameters are of 250 mm and 165 mm, respectively. The runner breadth is of 300 mm. The runner shaft sits on two ball bearings.

The differential pressure at the Venturi tube and the static pressure at turbine inlet were measured with a differential pressure transducer (DPT) and a gauge pressure transducer (GPT) having measuring ranges of 0...100 mbar and 0...2.5 bar, respectively, and accuracies of 0.5% of the full scale (FS). Both transducers were connected to a GL800 data logger for data acquisition. The vibration levels were measured in transverse, axial, and vertical directions using three Brüel & Kjaer DeltaTron accelerometers (ACC) of type 4507 B 001 with built-in preamplifier, connected to a laptop via a NI USB-9233 portable bus-powered USB carrier. 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. The transverse direction was along the axis of the upstream pipe, perpendicular to the turbine axis, while the axial direction was along the turbine axis. The runner speed, n, was measured with a Testo 470 digital tachometer with a measuring range of 1...9,999 rpm and an accuracy of 0.02% of the reading (RD). The shaft moment, M, was read at the scale with which the hydrodynamic brake is foreseen.

Tests were carried out for three openings of the nozzle flap, a_0, i.e. at three different discharges, and at five runner speeds for each opening, both for normal operation of the turbine and for operation with unbalanced runner. For the normal operation, the flap openings were of 25%, 50%, and 75%. To unbalance the runner, a mass of 0.262 kg was added inside a blade channel at a radius of roughly 104 mm. This corresponds to the situation that can appear during turbine operation, when a piece of debris carried by the water flow remains stuck between two runner blades. With a blade channel partially obstructed by the added mass, the discharge and the runner speed decreased slightly for a given flap opening. As a result, it was necessary to increase the flap openings for the unbalanced runner to 35%, 60%, and 85% in order to obtain maximum runner speeds roughly equal to those attained at the openings used during normal operation. For all regimes of normal operation, vibrations were measured on all three directions at three positions: on top of the turbine casing (ACC1), on the bearing between turbine and brake (ACC2), and on top of the brake (ACC3). A number of 135 data sets were obtained. For all regimes of operation with unbalanced runner, vibrations were measured again in all three directions, but only on top of the turbine casing. A number of 45 additional data sets were obtained. Throughout all tests, accelerations were read at a sampling frequency of 2000 Hz, which was high enough to investigate the dynamic behavior of the turbine, since the maxim runner speed did not exceed 500 rpm, which means that the frequency of rotation, f_r, and the blade passing frequency, f_b, were not higher than 8.33 Hz and 158.3 Hz, respectively. Unfortunately, the GL800 data logger allows a maximum sampling frequency of only 10 Hz, which means that no correlations between vibrations and pressure pulsations at turbine inlet were possible.

3. RESULTS AND DISCUSSION

According to the specifications presented in the previous section, a total number of 180 data sets were measured. The signals obtained experimentally were first filtered using the dBTrait software. With a custom-made program written in Python, the Fast Fourier Transform (FFT) was subsequently applied to the filtered signals that were also integrated to obtain velocity amplitudes, which were found to offer a clearer image of the dynamic behavior of the turbine. Due to the limited space, only some relevant data are presented in this paper. However, the discussion and conclusions remain valid for the other data as well.

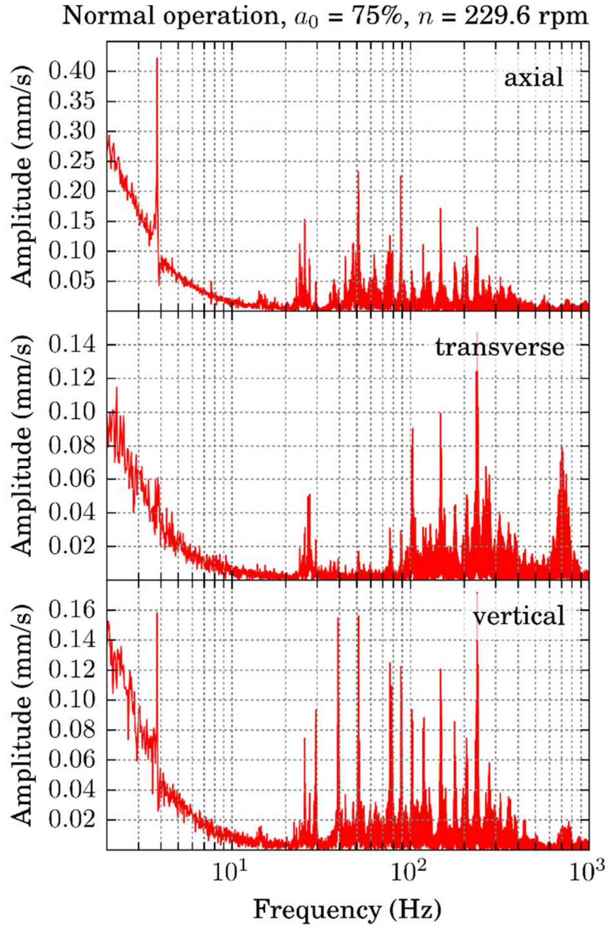

Fig. 3. FFTs of vibrations measured at a flap opening of 75% and a runner speed of roughly 229 rpm, on top of the turbine casing.

Figure 3 presents FFTs of vibrations measured along axial, transverse and vertical directions on top of the turbine casing during normal operation at a flap opening of 75% and a runner speed of roughly 229 rpm. In terms of amplitudes it can be observed that the vibrations in axial and vertical direction are dominant, while, in transverse direction the amplitudes increase significantly only in the range of high frequencies, above 100 Hz. An interesting aspect is the high amplitudes at frequencies below the rotational frequency, which will be discussed below.

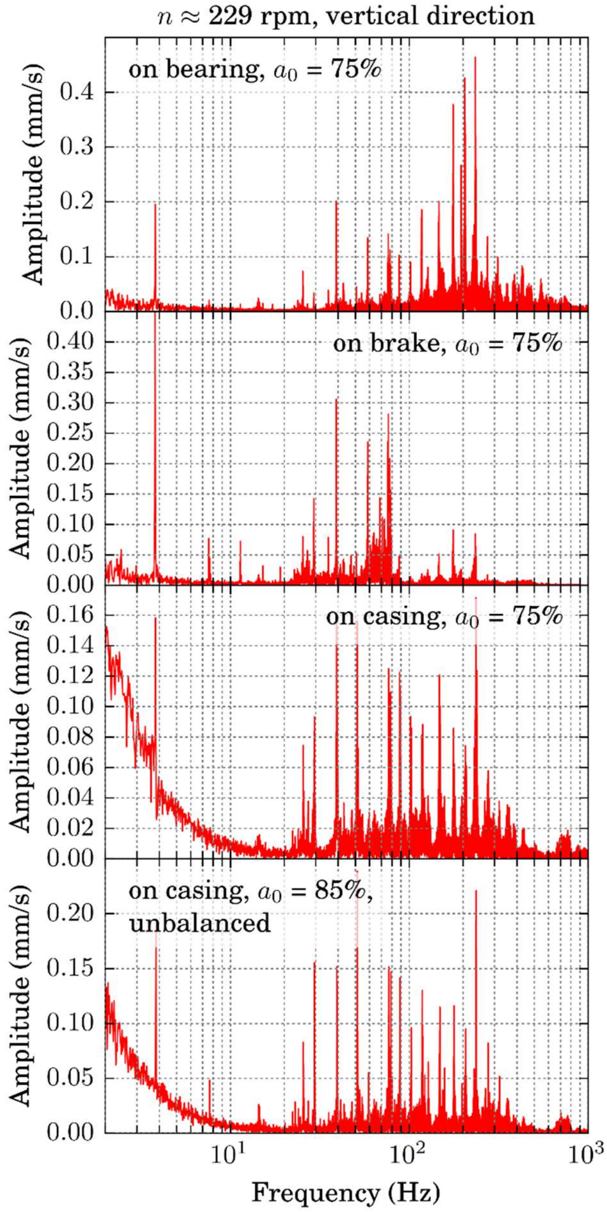

Fig. 4. Comparison between FFTs of vibrations measured at roughly 229 rpm in vertical direction.

Figure 4 shows a comparison between FFTs of vibrations measured at roughly 229 rpm in vertical direction at different positions both during normal operation and during operation with unbalanced runner. On the bearing, the dominant frequencies span over the entire frequency domain, although the highest amplitudes are attained at frequencies higher than 100 Hz. The vibrations measured on the brake are characterized by dominant frequencies that are mostly below 80 Hz. The results indicate that the high frequency and high amplitude vibrations are most probably caused by the ball bearings, while lower frequencies seem to be generated by the brake, i.e. the turbine load. Another source of low frequency vibrations could be the misalignment between the lines of the turbine and brake shafts. The results also suggest that high and low frequency vibrations are transmitted between brake and bearing, but with decreased amplitudes, being dampened most probably by the rubber bushings of the coupling. The measurements on the casing indicate that the vibrations generated both by the brake and by the bearings are transmitted, although dampened, to the casing.

Additionally, below the frequency of rotation of 3.82 Hz, very low frequencies of relatively high amplitudes, already observed in Figure 3, appear clearly on the turbine casing. This could indicate that very low frequency vibrations might be propagated to the turbine through the upstream pipe, being probably generated by the inherent pressure pulsations in the experimental installation. For the unbalanced runner, the frequency spectrum obtained on the casing does not change significantly, apart from a clear increase in vibrations amplitude mostly in the high frequency domain and from the clear appearance of the second harmonic of the frequency of rotation at 7.64 Hz.

When comparing different sets of data, it is customary to scale them down by dividing them by properly chosen reference values. In Figure 5, scaled FFTs of vibrations measured on top of the casing in vertical direction at different runner speeds during normal operation with the flap opened at 75% are presented.

At each runner speed, the scaling was accomplished by dividing all frequencies in the frequency domain by the rotational frequency. It can be seen that, when going from one runner speed to another, the scaled dominant frequencies do not show important changes, which indicates that the identification of the sources of vibrations and, thus, of the possible sources of failure does not depend significantly on the operating regime. When the first comments on Figure 4 are also considered, it can be concluded that the vibrations due to the brake have scaled dominant frequencies of up to 80 Hz divided by 3.827 Hz, i.e. to roughly 21. The pressure pulsations from the hydraulic installation have scaled frequencies mostly below 1. The bearings are a source of vibrations on the subdomain of large scaled frequencies.

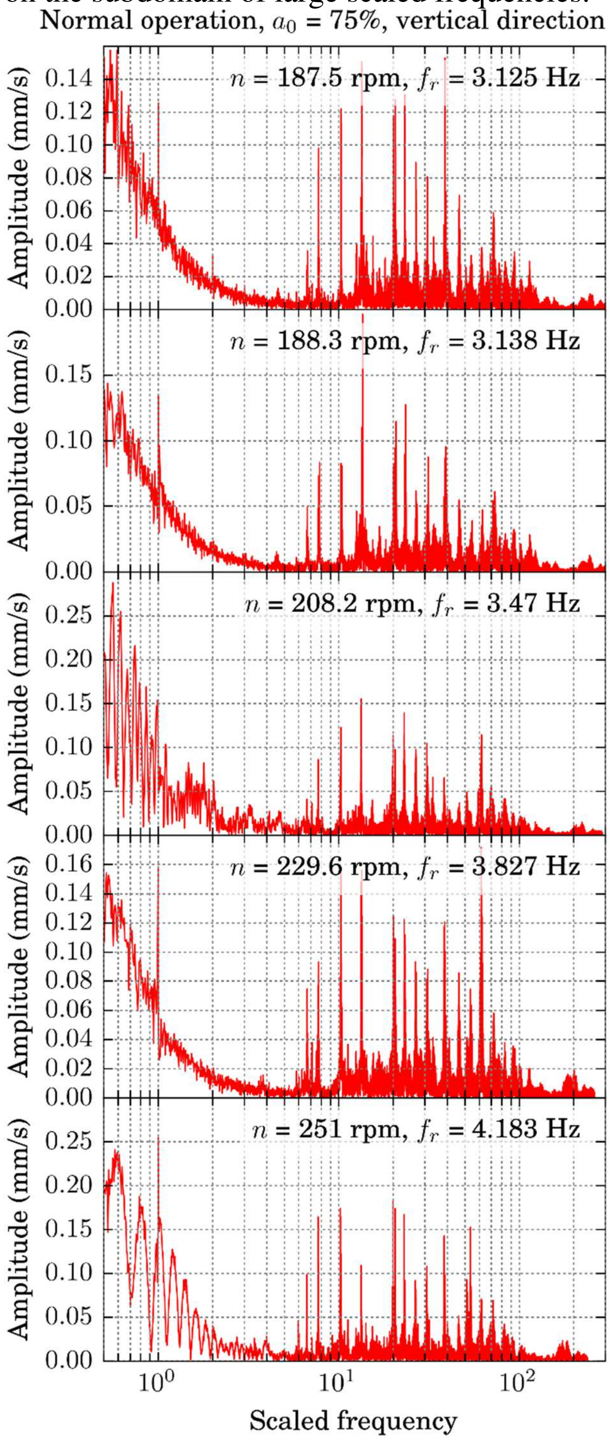

Fig. 5. Scaled FFTs of vibrations measured on top of the casing at different runner speeds.

4. CONCLUSIONS

The paper underlines the utility of vibration measurements as a predictive maintenance method for hydraulic turbines. In particular, the paper shows how to identify vibrations that occur due to different mechanical and hydraulic elements specific to a Bánki turbine which operates in a hydraulic installation.

Frequencies of rotation and their harmonics, possible vibrations due to pressure fluctuations in the installation, vibrations specific to the ball bearings that support the shaft of the Bánki turbine, and vibrations due to the hydrodynamic brake have been identified.

It was found that the vibrations in axial and vertical direction are dominant in terms of amplitude. The high frequency and high amplitude vibrations are likely to be caused by the bearings. The load of the turbine could be a source of lower frequency vibrations.

Vibrations of very low frequency, below the frequency of rotation were identified as well. Their cause could be the inherent pressure pulsations in the installation through which the turbine is supplied with water.

The mass added on the runner caused an increase in the amplitude of the vibrations mostly in the high frequency domain as well as the clear appearance of the second harmonic of the frequency of rotation, when the vibrations are measured on the turbine casing. These conclusions could be used to identify the situation in which a piece of debris remained stuck between two blades of the turbine runner.

To easily compare different operating regimes, the frequency domains were scaled by dividing the frequencies by the frequency of rotation corresponding to each operating regime. In the scaled frequency domain, the hydrodynamic brake has dominant frequencies of up to 21. The analysis shows also that the vibrations due to the bearings have larger amplitudes at higher scaled frequencies. The pressure pulsations from the hydraulic installation are likely to generate scaled frequencies mostly below 1.

When going from one operating regime to another, the scaled dominant frequencies do not show important changes. This suggests that, in the scaled frequency domain, the operating regime does not influence significantly the identification of the sources of vibrations and, thus, of the possible sources of failure.

The rest of the vibrations that occur due to the hydrodynamic flow can only be speculated. Additional experimental and numerical studies are required to properly identify the vibrations caused by the complicated flow through the turbine runner.

5. 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, pp. 637-648, 2017.

[2] Shi, F.X., Yang, J.H., Wang, X.H., *Analysis on Characteristic of Pressure Fluctuation in Hydraulic Turbine with Guide Vane*, International Journal of Fluid Machinery and Systems, 9 (3), pp. 237-244, 2016.

[3] Vereş M., Ţurcan R., Dumitrescu D., *Critical analysis of vibration sources of hydro aggregates in operating regime*, Journal of Sustainable Energy, 2, pp. 1-8, 2011.

[4] Hermod, B., *A Review on Work on Oscillatory Problems in Francis Turbines, New Trends in Technologies*: Devices, Computer, Communication and Industrial Systems, Er, M.J. (Ed.), ISBN: 978-953-307-212-8, 2010.

[5] Roman, R., Bucur, D., Aidanpaa, J.O., Cervantes, M.J., *Damping, stiffness and added mass in hydraulic turbines. Preliminary experiments and analysis*, ENERGIFORSK, ISBN 978-91-7673-106-2, 2015.

[6] Poll, H.G., Zanutto, J.C., Ponge-Ferreira, W., *Hydraulic power plant machine dynamic diagnosis*, Shock and Vibration, 13, pp. 409–427, 2006.

[7] Carlsson J., Mechanical vibrations in hydraulic machines. Suggested vibration limits based on analysis of the IEC TK4 vibration database, ISBN 978-91-7673-124-6, 2015.

[8] Dragomirescu, A., Schiaua, M., Experimental and numerical investigation of a Bánki turbine operating far away from design point, Energy Procedia 112, pp. 43-50, 2017.

Identificarea experimentală a posibilelor defecte la funcționarea unei turbine Banki

Rezumat: Scopul lucrării este de a studia vibrațiile care apar în funcționarea unei turbine hidraulice de putere mică, de tip Bánki. Turbina a fost încercată la diferite regimuri de funcționare într-o instalație de laborator, unde poate lucra cu sau fără încărcarea furnizată de o frână hidrodinamică cuplată la arborele turbinei. Vibrațiile au fost măsurate pe carcasa turbinei, pe unul din lagărele turbinei și pe frână. Au fost identificate frecvențele dominante corespunzătoare fiecărui subansamblu. Au fost realizate măsurători și cu o masă adăugată, care simulează un rest plutitor care a rămas blocat în interiorul rotorului, cauzând un dezechilibru. Rezultatele obținute sugerează că lagărele sunt o sursă de vibrații de înaltă frecvență, în timp ce vibrațiile de frecvență joasă sunt cauzate de sarcina turbinei. Au fost identificate și vibrații de foarte joasă frecvență, care ar putea fi transmise de la instalație, prin intermediul conductei amonte. Rezultatele subliniază importanța diagnozei pe baza vibrațiilor pentru identificarea posibilelor defecte ale unei turbine Bánki, ca metodă de mentenanță predictivă specifică mașinilor rotative.

Crăița CARP-CIOCÂRDIA, PhD. Eng., Assoc. Professor, University Politehnica of Bucharest, Department of Mechanics, craita.carp@upb.ro, +40214029250, Bucharest, Romania.

Andrei DRAGOMIRESCU, PhD. Eng., Lecturer, University Politehnica of Bucharest, Department of Hydraulics, Hydraulic Machinery, and Environmental Engineering, andrei.dragomirescu@upb.ro, +40214029710, Bucharest, Romania.

Carmen-Anca SAFTA, PhD. Eng., Professor, University Politehnica of Bucharest, Department of Hydraulics, Hydraulic Machinery, and Environmental Engineering, safta.carmenanca@gmail.com, +40214029119, Bucharest, Romania.

Mihai SCHIAUA, PhD. Eng., Researcher, University Politehnica of Bucharest, Department of Hydraulics, Hydraulic Machinery, and Environmental Engineering, mschiaua@yahoo.com, +40214029710, Bucharest, Romania.

Ioan MAGHEȚI, PhD. Fiz., Professor, University Politehnica of Bucharest, Department of Mechanics, ioan_magheti@yahoo.com, +40214029250, Bucharest, Romania.