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TRANSIENT BEHAVIOR ANALYSIS. STUDY CASE: PUMPING STATION GÂLCEAG

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In the spring of year 2006, both hydro units of Gâlceag pumping station were damaged, the diffuser vanes being broken or deformed. This kind of accidents takes place as a result of the impeller-diffuser interaction which occurs in unsteady operating regimes.

The transient flow in multistage centrifugal pumps resulted from the impeller-diffuser interaction was studied numerical and experimental by many research specialists.

These observations were made using analogical measurement devices (manometers) and diagrams obtained from the pressure transducers.

The impeller-diffuser interaction effect makes the flow through the pump to be a very complex phenomenon. One of its aspects refers to the pressure variations which occur in the pump inlet and outlet.

Important variations of pressure values in upstream and downstream pipe were observed during in site measurements and in all pump operation stages. Those pressure variations were observed in all pump operation stages.

Keywords: impeller, diffuser, transient, interaction, pressure variations.

1. Introduction

Between the moving impeller (with z blades) and the steady post-rotoric diffuser (with z_0 blades) there is an important interaction. This interaction consists in circumferential pressure fluctuations in impeller, developing secondary flows, characterized by non uniformities, stream detachment and others. The magnitude of this phenomenon is bigger during the increase or decreasing of the pumps' rotational speed (at the pumps' starting and stopping).

There are many papers written in the last decade about this interaction, changing the concepts of components design.

Theoretical and experimental studies [1] shown that the rotor – stator interaction is basically due to pressure fluctuations inside the vaned diffuser. The

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fundamental harmonic is the most important component of pressure fluctuation. It is shown that the effect of pressure fluctuation is more significant than the transient component of the pressure variation.

In similar experiments and numeric simulation [2] the head, the discharge and the radial force evolution during the pumps transient operation regimes were analyzed. Also, the influence of the radial clearance between the impeller and the diffuser on the pressure and radial force variation was studied. The results showed that the radial force decreases with the increase of the clearance.

Considering these papers and all the others written in the last 15 years about the impeller–diffuser interaction, the conclusion is that the accident at P.S. Gâlceag was due to the impeller–diffuser interaction during the transient operating regimes of the pumps.

2. Experimental work

The experiments were performed in a centrifugal pump which has already been described in [3].

During the experimental investigation the pump's discharge Q, head H and the consumed electric power, P were measured in order to determine the pump operating parameters, for different operating regimes of the pumping station, and the characteristic curves $H = H(Q)$ and $\eta = \eta(Q)$ were obtained.

2.1. Discharge measurement (Q)

For discharge measurement was used a portable flowmeter CONTROLOTRON 1010P. It was connected to a KUSB – 3100 data acquisition module (see figure 1).

Fig. 1. Flowmeter and data acquisition module.

2.2. Head measurement (*H*)

The pump's head was obtained using the following equation:

$$H = \frac{p_2 - p_1}{\rho \cdot g} + \frac{v_2^2 - v_1^2}{2 \cdot g} + h \qquad (1)$$

where p_1, v_1 are the water pressure and velocity at the pump's inlet; p_2, v_2 are the water pressure and velocity at the pump's outlet, and h represents the level difference between the pump's inlet and outlet.

The pressure was measured using two transducers, one set at the pump's inlet and the second one set at outlet.

The transducers were connected to the same data acquisition module as the flowmeter and the entire data acquisition chain was lead by means of a program realized in LabVIEW 8.0.

The sampling frequency was 333 Hz for each of the three analog input channels (discharge, inlet pressure and outlet pressure), for a sampling period of 3 minutes and 10 minutes.

The acquired pressure values were compared with the pressure values read by means of two pressure manometers.

2.3. Consumed electric power (*P*)

The electric power consumed by the pump's motor was read at the control panel display (a very modern multimeter with a good precision class and the possibility to record data).

3. Data processing

3.1. Steady operating regimes

The first step in data processing analyze was to compare the acquired data with the data provided from the constructor. The pump characteristic curves ($H = H(Q)$ and $\eta(Q)$) obtained from the acquired data, and from the constructor data with a certainty interval of 3 % are presented in figures 2 and 3.

It can be seen that there is a domain of discharge values where the pump characteristic curves obtained from measurements are close to the constructor characteristics.

For discharge value lower than 2.5 m^3/s the constructor head characteristic is inside the certainty interval and for discharge value between 1.75 and 3 m^3/s the pump efficiency respects the guaranty.

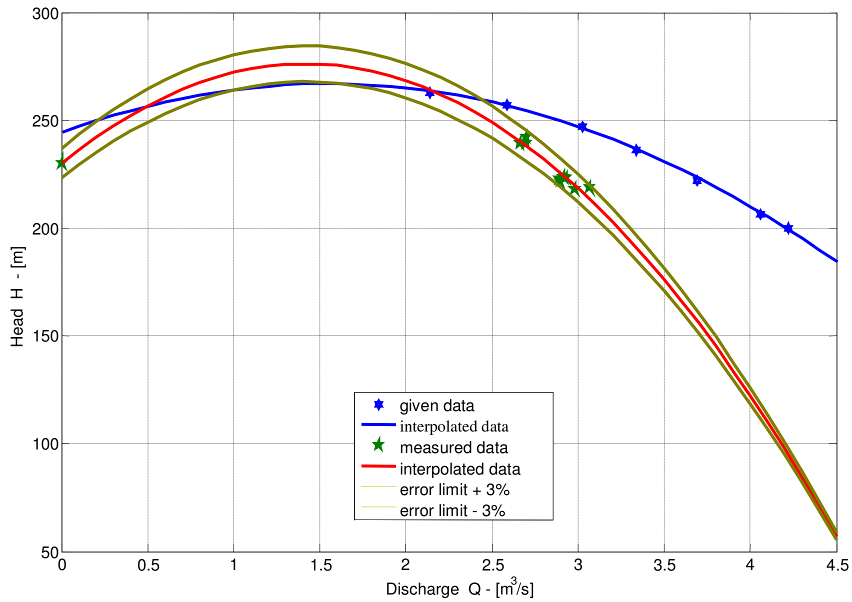

Fig. 2. Head characteristic curve $H = H(Q)$ provided by the constructor and resulted from measurements and the uncertainty domain.

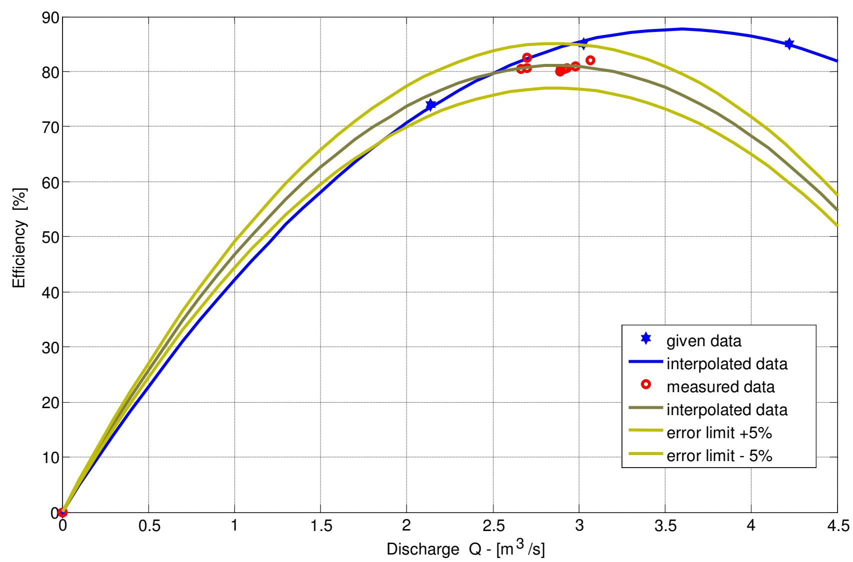

Fig. 3. Efficiency characteristic curve $\eta = \eta(Q)$ provided by the constructor and the resulted from measurements and the uncertainty domain.

These differences between the two pump's characteristics (provided by the constructor and the characteristics obtained from the measurements) are mainly due to the impeller – diffuser interaction.

The effect of impeller – diffuser interaction leads to a great complexity of the flow through the pump. One of the aspects of the flow refers to pressure fluctuations which appear at the inlet as well as at the outlet of the pump.

3.2. Frequency domain analyse

In order to identify the possible operating disorders, as well as their causes, the signal for each measurement was processed.

The pressure variation in time in the outlet pipe is shown in figure 4. In the range of (25.7 ... 27) bar important outlet pressure fluctuations can be observed. The signal has the shape of a continuous oscillation around a mean value, its amplitude being of 0.6 bar.

A representative sample of data was used and the continuous component of the signal was extracted in order to compute the FFT. This leaded to indications over vibrations and distortions sources.

The sampling period being 0.003 seconds, resulted a "*cutting frequency*" of 166.5 Hz (the Nyquist frequency) and a resolution of 0.645 Hz. The results of spectral analyze are synthetically shown in figure 5.

The results show that in the frequency domain of 0…163.5 Hz the spectrum is quasi-symmetric around the value of 100 Hz. The main pressure fluctuations are at 24.914 Hz, 43.12 Hz, 86.722 Hz, 99.258 Hz and 115.371Hz.

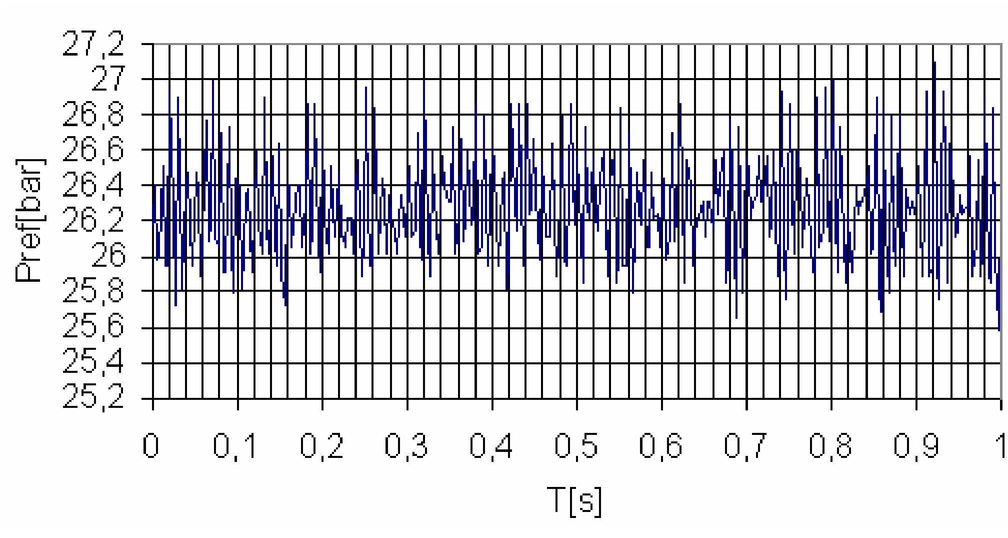

Fig. 4. The time evolution of the outlet pipe pressure in steady operating regime.

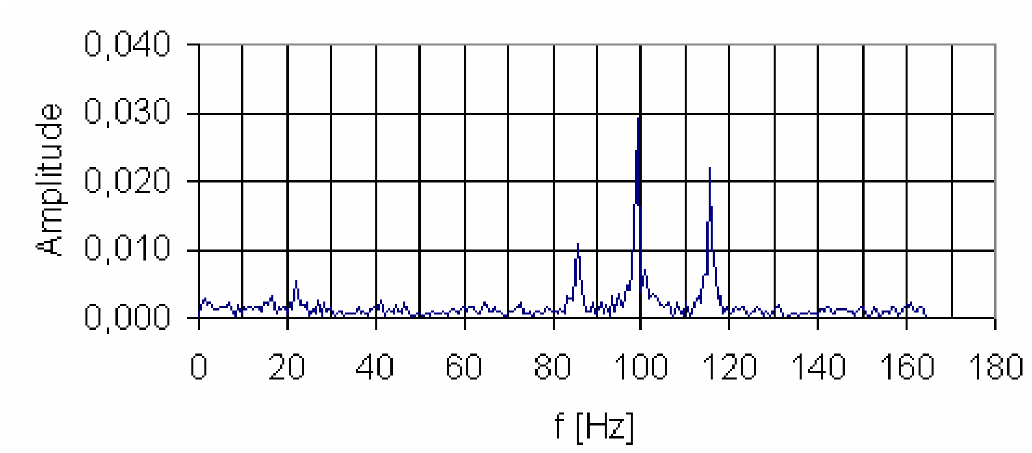

Fig. 5. Pressure spectrum in outlet pipe.

The frequency at 1000 rot/min motor rotational speed is:

$$f_m = \frac{1000}{60} = 16.666 \text{ Hz}, \tag{2}$$

and the fundamental frequency is almost 100 Hz which corresponds to the above frequency multiplied by the impeller's number of blades (6 blades):

$$f = \frac{N}{60} \cdot n = \frac{1000}{60} \cdot 6 = 100 \text{ Hz}. \tag{3}$$

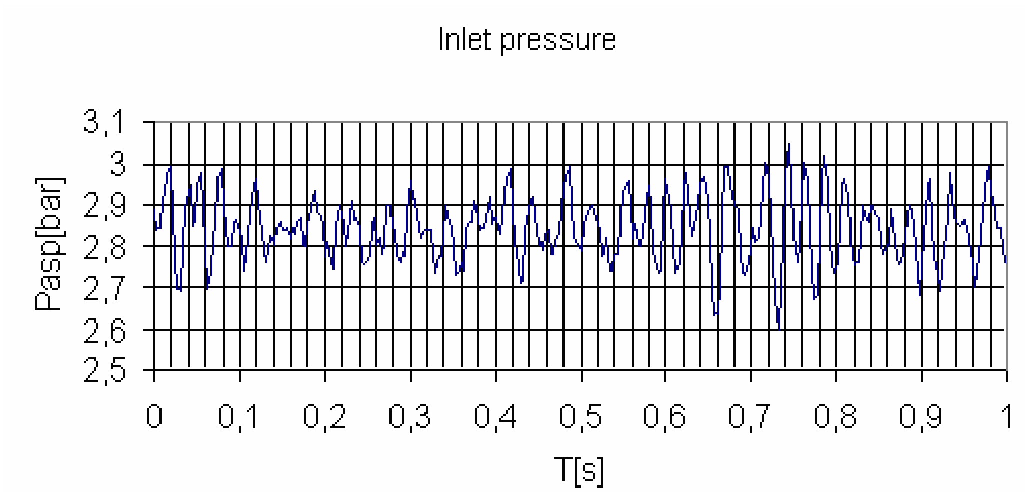

Fig. 6. The time evolution of the inlet pipe pressure in steady operating regime.

The frequencies of 86.722 Hz and 115.371 Hz are situated quasi-symmetric related to the fundamental frequency, at a "distance" of almost 16.66 Hz and they represent the 5th and the 7th multiples of this frequency.

The time evolution of the pressure in the inlet pipe presents a smaller fluctuation frequency with the amplitude of 0.2 bar (see figure 6).

The shape of the pressure variation on the inlet pipe is similar with the one of outlet pipe (the signal shape is a continuous oscillation around a mean value).

The pressure spectrum in the inlet pipe (figure 7) has higher amplitude at 26.426 Hz, 29.648 Hz, 43.184 Hz and 48.984 Hz in the range 0…163.5 Hz.

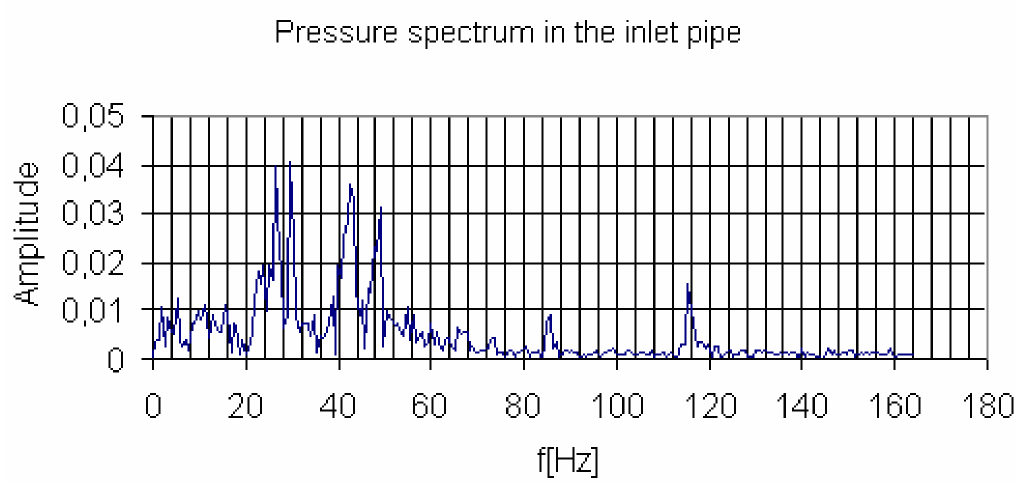

Fig. 7. Pressure spectrum in the inlet pipe in pump's steady operating regime.

These frequencies are generated by the pump's inlet hydraulic circuit and they cannot be correlated with the pump's specific frequencies. It can be considered that these frequencies (24.914 Hz and 43.12 Hz), which are present also in the outlet pressure spectrum, derive from the inlet hydraulic circuit and are sent through the pump in the outlet hydraulic circuit.

Also, there are higher amplitudes in the signal at 86 Hz and 116 Hz, the inlet pressure being influenced by the impeller's movement. It can be seen that these frequencies are present in the outlet pressure spectrum as well having higher amplitudes than in the inlet hydraulic circuit.

4. Conclusions

During the measurements there were observed important pressure fluctuations inside the inlet and outlet pump's pipes, which are present in all pump's operating stages. In the present paper a steady operating regime was presented.

The pressure oscillations are generated by the pump's inlet hydraulic circuit and they can not be correlated with the pump's specific frequencies.

It can be seen that in the frequency domain analysis, the main amplitude appears at the blades passing frequency of $f = \frac{N}{60} \cdot n = \frac{1000}{60} \cdot 6 = 100$ Hz, the other amplitudes observed representing harmonics of this fundamental frequency.

So, it can be considered that the main effect of the increase of the radial clearance between the vaned diffuser and the pump's impeller was decreasing the pressure fluctuation inside the pump's hydraulic circuit and reducing the vibrations' level, which will be the subject for a further study.

REFERENCES

[1]. *W. Qin and H. Tsukamoto*, "Theoretical study of pressure fluctuations downstream of a diffuser pump impeller – part 1: Fundamental analysis on rotor-stator interaction", in Journal of Fluid Engineering, **vol. 119**/647, September 1997.

[2]. *M. Zhang, H. Wang and H. Tsukamoto*, "Transient hydrodynamic forces on a diffuser pump impeller caused by rotor-stator interaction at fast start-up", in IARH Symposium on Hydraulic Machinery and Systems, September 9 – 12, 2002, Lausanne.

[3]. *E. C. Isbă□oiu et al.*, "Stația de pompare Gâlceag. Studiul privind funcționarea în tandem a stației de pompare Gâlceag cu hidrocentrala Gâlceag şi studierea posibilității de montare a unui organ suplimentar de închidere pe refularea pompelor", Contract 65/2005, Beneficiar S.C. HIDROELECTRICA S.A.

[4]. *V. Câmpian*, "Măsurători şi probe efectuate pe pompa nr. 1 de la stația de pompare Gâlceag înainte de punerea în funcțiune", Raport Tehnic nr. U-03-400-55.