

VIBRATIONS LEVEL ANALYSIS DURING THE OPERATION OF PUMPING STATION GÂLCEAG

Georgiana DUNCA¹, Eugen Constantin ISBĂŞOIU², Constantin CĂLINOIU³,
Diana Maria BUCUR⁴, Călin GHERGU⁵

Deoarece la începutul anului 2006 a intervenit ruperea palelor post-rotorice s-a considerat necesar să se verifice comportarea în funcționare a hidroaggregatelor de pompă din stația de pompă Gâlceag sub aspectul vibrațiilor.

În cadrul celei de-a treia campanii de măsurare, realizată în luna martie 2007 la stația de pompă Sebeș, echipa UPB a efectuat și măsurări de vibrații pe lângă determinările unor parametri hidraulici (debit, presiunea din amonte și presiunea din aval).

Scopul acestor măsurări a fost verificarea comportării în funcționare a hidroaggregatului de pompă nr. 2 care fusese scos din funcțiune pe durata celorlalte două campanii de măsurare și verificarea nivelului vibrațiilor celor două hidroaggregate.

During spring of 2006, both hydro units of the Gâlceag pumping station have been damaged, the diffuser vanes being broken or deformed. This is why it became important to analyze the operating behavior under the vibration level aspect for the pumping units.

In the third testing campaign realized in March 2007 at Gâlceag pumping station, there were performed vibration level measurements, in addition to the hydraulic parameters measurements (flow rate, upstream and downstream pressures).

The aim of this study is to analyze the vibration levels for the two pumping units of the Gâlceag station.

Keywords: vibration level, pumping unit, FFT, amplitude spectrum.

¹ Assistant, Dept. of Hydraulics and Hydraulic Machinery, University POLITEHNICA of Bucharest, Romania

² Professor, Dept. of Hydraulics and Hydraulic Machinery, University POLITEHNICA of Bucharest, Romania

³ Reader, Dept. of Hydraulics and Hydraulic Machinery, University POLITEHNICA of Bucharest, Romania

⁴ Assistant, Dept. of Hydraulics and Hydraulic Machinery, University POLITEHNICA of Bucharest, Romania

⁵ Assistant, Dept. of Hydraulics and Hydraulic Machinery, University POLITEHNICA of Bucharest, Romania

1. Introduction

Vibration is one of the most vexing problems with pumping machinery, and it is the cause of considerable altercation and litigation. Noise can become a significant, annoying problem. Excessive vibration from primary equipment can be transmitted directly to the building structure, which causes uncomfortable (and sometimes dangerous) structural vibration levels. Excessive vibration of equipment and piping can destroy portions of the equipment (such as drive shafts and seals), loosen or break pipe anchors, and even cause pipes to burst under certain conditions. Unlike the cavitation problems, the unsteady fluids flow in turbo machinery hasn't a long research history.

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 , the consumed electric power, P and the vibrations level were measured in order to determine the pump operating parameters, and to identify the possible operating disorders.

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:

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

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).

2.4. Vibrations level measurement

The vibrations level during the operation of the two pumping units was measured by means of the vibration transducer, type KD 20. The transducer was placed on: the bearing opposite to the engine radial orientated, the bearing next to the engine radial orientated, the left bearing axial orientated for pump 2, the bearing next to the engine, radial and axial orientated for pump 1.

3. Data processing

3.1. Steady operating regimes

The first step in data processing was analyzing the position of the obtained operating points on the pump operating characteristic. In the figures 2, 3, 4 there are presented the obtained results.

It is obvious that the operating points obtained respect the operating characteristic resulted after processing the data from the two previous measuring campaigns. (made for pump 1).

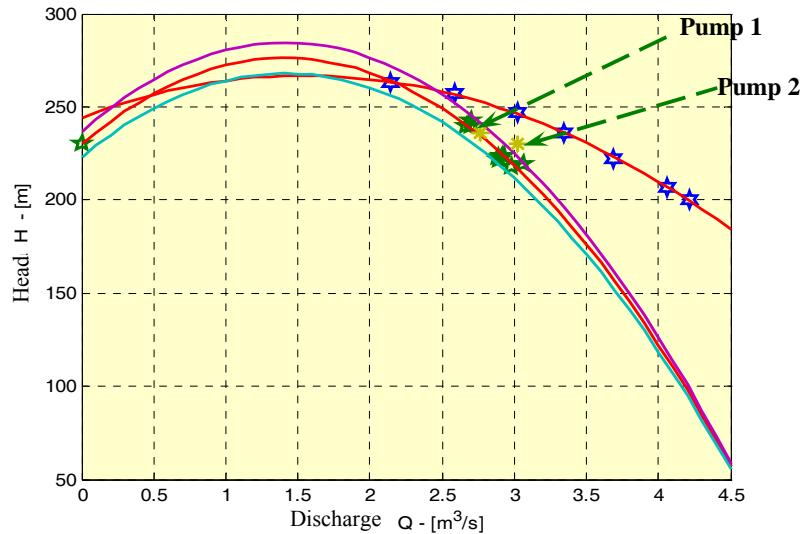


Fig. 2 – Head characteristic curve $H = H(Q)$ provided by the constructor and resulted from measurements and the uncertainty domain.

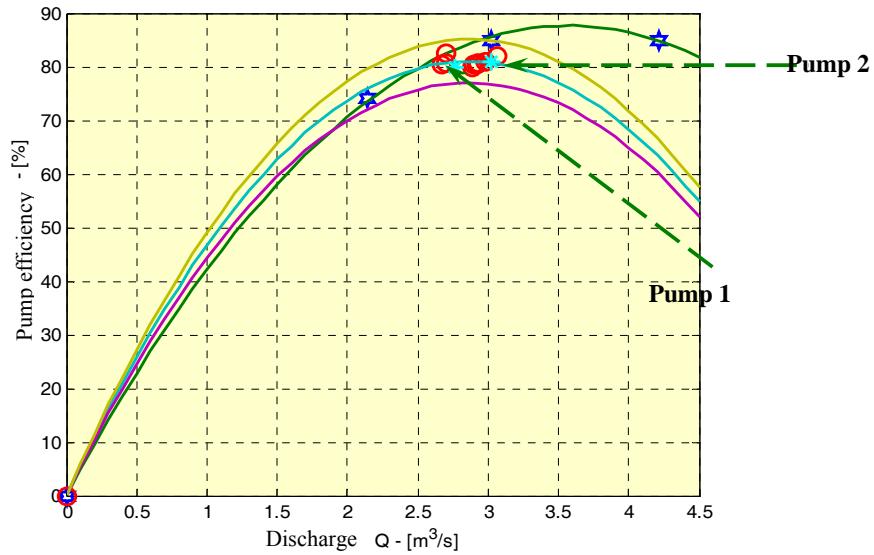


Fig. 3 – Efficiency characteristic curve $\eta = \eta(Q)$ provided by the constructor and the resulted from measurements and the uncertainty domain

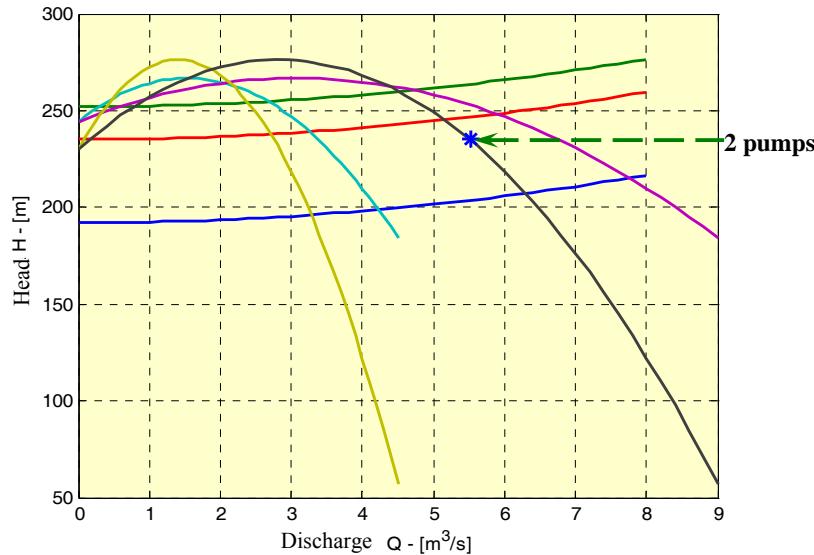


Fig. 4 – Determinarea punctelor de funcționare a stației pentru sarcinile geodezice maximă și minimă

3.2. Frequency domain analyse

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

These signals, acquired in volt had been transformed in acceleration units by means of the transducer calibration ecuation.

For each acquired vibration signal the frequency domain response was analyzed in order to identify the possible dangereus amplitudes.

The spectral analyze was made on $32768 (2^{16})$ test samples taken from the acquired signals. The Fast Fourier Transform (FFT) was used in order decompose the signal and to obtain the components amplitudes and frequencies.

In order to compare the results to the data obtained from the vibration analyze system PULSE, which are given in speed units, the amplitude spectrum obtained using FFT was integrated.

The maximum amplitudes spectrums in units of accelerations are presented in figures 5, 6, 7, 8 and 9.

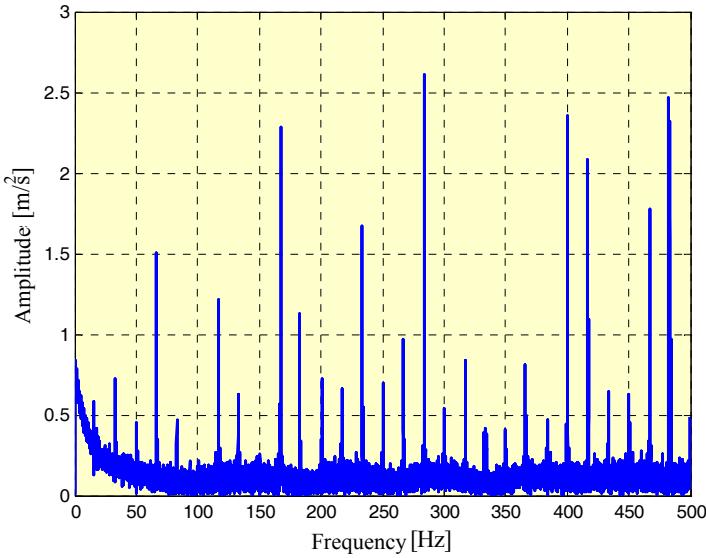


Fig.5 – The spectrum obtained when the vibration transducer is placed on the bearing opposite to the engine, radial orientated – pump 2

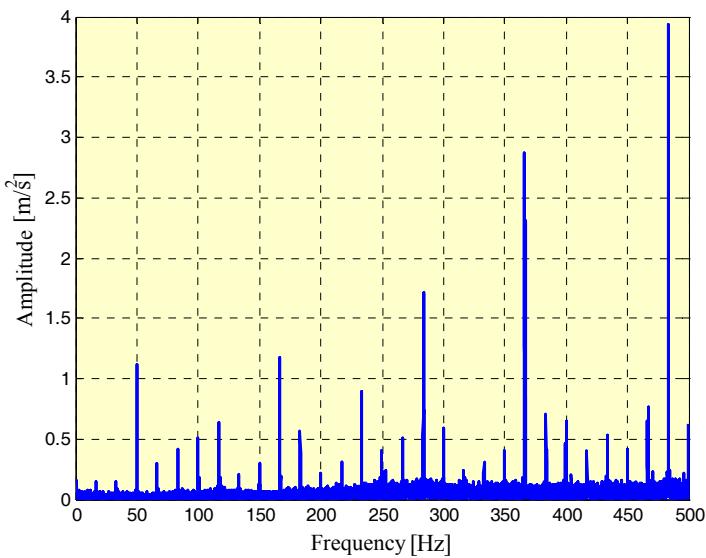


Fig.6 – The spectrum obtained when the vibration transducer is placed on the bearing next to the engine, radial orientated – pump 2

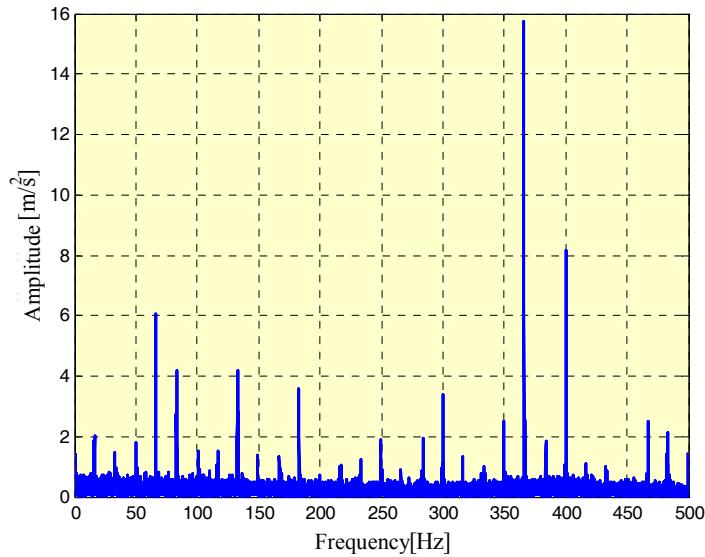


Fig.7 – The spectrum obtained when the vibration transducer is placed on the bearing next to the engine, axial orientated – pump 2

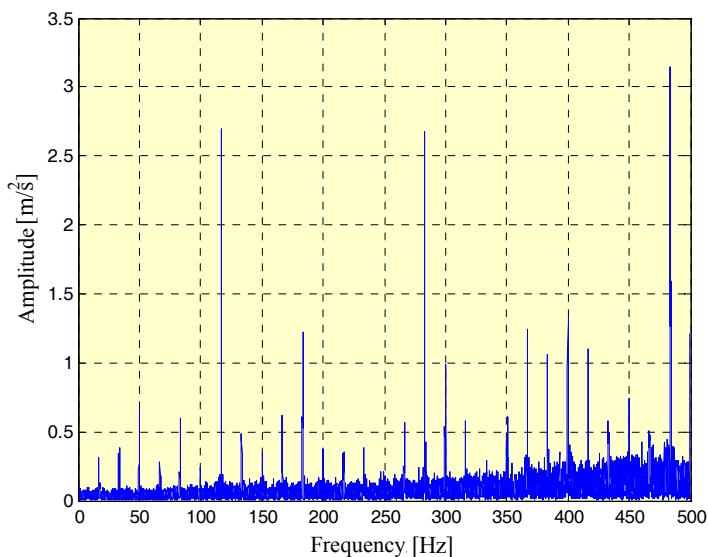


Fig.8 – The spectrum obtained when the vibration transducer is placed on the bearing next to the engine, radial orientated – pump 1

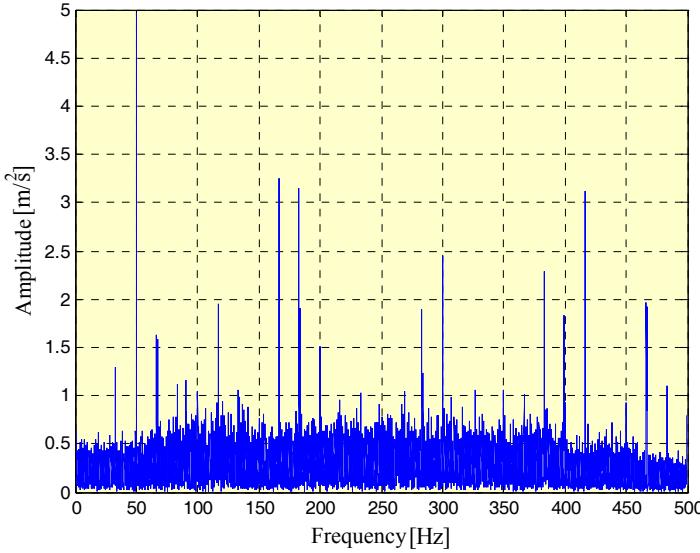


Fig.9 – The spectrum obtained when the vibration transducer is placed on the bearing next to the engine, axial orientated – pump 1

The frequency corresponding to the engine speed of 1000 rot/min is

$$f_m = \frac{1000}{60} = 16.666 \text{ Hz} \quad (2)$$

In every spectrum it can be seen that there are some major components at 16.666 Hz, 33.33 Hz, 50 Hz and their multiples. According to the specialty literature [1] (M. Băglăzan, „Turbine hidraulice și transmisii hidrodinamice”) these vibrations having frequencies of $1 \times f_m$ and superior harmonics represent mechanic vibrations of the impeller vanes and guide vanes, and possible hydrodynamic fluctuations due to the turbulence and Kármán vortices. These vibration signals different possible causes: lack of trimming and alignment, bent shafts etc.

In every spectrum it can be seen that the mean level is around 0.1 mm/s. In STAS 6910 – 74 there are established the following qualifiers for the maximum values of energetic units' vibrations:

- very well $<1.63 \text{ mm/s}$;
- well $1.63 - 2.96 \text{ mm/s}$;
- allowed $2.96 - 4.74 \text{ mm/s}$;
- allowed under surveillance $4.74 - 6.66 \text{ mm/s}$.

The component with the biggest amplitude was reached for the bearing next to the engine, axial orientated – pump 2 (6.8 mm/s) corresponding to a frequency of de 366.67 Hz.

4. Conclusions

A direct comparison between the results obtained from the vibrations analyze system (PULSE) and those resulted from the measurements is not possible. In the previous measurement campaign the vibrations were determined using a speed transducer and in the actual measurements campaign they were determined with an acceleration transducer, thus in order to obtain comparable values with the previous measurements that were made with the PULSE system, the signals where processed.

Because of the high amplitudes registered at relatively low frequencies it can be considered that, in the case of the pump 2 operation, there are mechanical vibrations of rotating parts (see fig. 5). Those high amplitudes are not found in the other cases of operation at pump 2 (fig. 6 and 7) and they can be neglected. In the measurements made at pump 1, similar values had not been found (fig. 8 and 9).

From the vibrations level point of view:

- it is considered that the bearing opposite to the engine at pump 2 has a very good running behaviour, the level of vibrations being 1.3 mm/s;
- the bearing next to the engine of pump 2 is considered to have a very good vibrations behavior, with a level of 0.16 mm/s on the radial direction and 1.3 mm/s on axial direction;
- the bearing next to the engine of pump1 is considered to have a very good operating behavior, the level of vibrations being 0.15 mm/s on radial direction and 0.91 mm/s on axial direction.

After comparing the analyzed signals obtained for the vibrations level with those from reports made before the reparation of the two pumping units of Gâlceag pumping station (based on the results provided by vibration analyze system PULSE), it is considered that now the units have a better behavior regarding vibrations, enlarging the space between the runner and stator vanes proving to be a good measure for improving that behavior.

It can be seen that the operating points for pump 1, respectively pump 2, operating isolated and in parallel, confirms the results from others measurements campaigns, the pumped flow rate being smaller that those given by the manufacturer, without supplementary energy consumption. The operating points are situated right on the curves (see fig. 2, 3, 4).

REFERENCES

- [1] *M. Bărglăzan*, „Turbine hidraulice și transmisii hidrodinamice”, Edit. Politehnica, Timișoara, 1999
- [2] *Christopher E. Brennen*, Hydrodynamics of Pumps, Oxford University Press, 1994
- [3] *E. C. Isbășoiu et al.*, „Stația de pompare Gâlceag. Studiul privind funcționarea în tandem a stației de pompare Gâlceag cu hidrocentrala Gâlceag și 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 și probe efectuate pe pompa nr. 1 de la stația de pompare Gâlceag înainte de punerea în funcțiune”, Raport Tehnic nr. U-03-400-55.
- [5] *M. Gafitanu, Sp. Crețu, B. Drăgan*, „Diagnosticarea vibroacustică a mașinilor și utilajelor”, Ed. Tehnică, București, 1989.
- [6] *Cyril M. Harris, Charles E. Crede*, „Șocuri și vibrații. Vol. II Analiza rezultatelor măsurărilor. Încercări. Metode de combatere a șocurilor și vibrațiilor”, Editura Tehnică, București, 1968.