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OSCILLATORY ANALYSIS OF PISTON PUMPS

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Abstract: *The present paper deals with an oscillatory analysis on pumps with axial pistons. A vibrations analysis strategy is described. An oscillatory analysis on a pump with wear bearings was undertaken and the diagrams of the displacement amplitude spectra. After oscillatory analysis on a pump with worn pistons.*

Keywords: oscillatory process, vibration, piston pumps, diagnosis.

1. INTRODUCTION

The key issue for the development of a faults and wear diagnose system, is the difficulty of choosing, from the set of data that results from such an analysis, those that allow direct assessment of the technical condition of pump components. Regarding this, there are a number of technical possibilities to exploit such information to be correlated with defects causing changes in the pump's body wave. [4,5]

One possibility is that the pump is monitored, permanently aiming the evolution of body vibration or noise. Their main change is to be interpreted as being caused by abnormal growth of states (for example wear). [4,5]

Experimental investigations have confirmed that the body vibrations are more intense as the energy source that produces them is stronger.

The increasing complexity of products, as well as the large number of components, which generally, are technically designed to have the same lifetime, are arguments justifying the possibility of increased number of accidental falls of the facility in which the product is part.

Basically hydraulic parts have a durability limit imposed by the moment of exceeding permissible states of wear. The wearing of different product components, leads to the situation where system performance requirements are not provided anymore and, therefore it cannot be used. This wear, which does not involve a fundamental change in functional parameters, occurs gradually over time and leads, from a certain limit, to irreversible damage to the facility.[1,2,3]

2. VIBRATION ANALYSIS STRATEGY

Pump vibration behavior analysis is an increasingly common method due to high informational content that it provides. Limits on the possible use of this method are determined by the high costs of systems required to collect measurement values.[4]

The performed oscillatory analysis is structured in three main steps:

- Observation - which aims temporal monitoring of the wear indicators (anomalies);
- Status control - that evaluates the machine' state by analyzing wear indicators (anomalies);
- Wear diagnosis (anomalies) - component that locates and describes defects by analyzing observation indicators.

Figure 1 shows the general strategy of body wave analysis in order to determine the defects:

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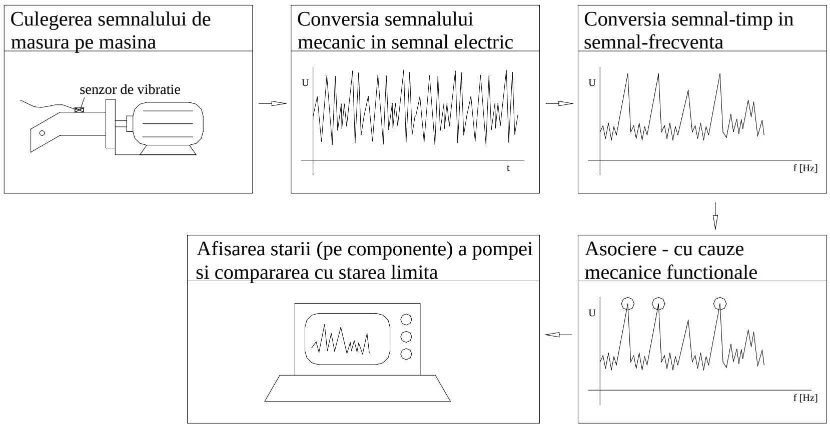

Fig. 1 Wave analysis model strategy for establishing the causes of defects

With the help of a contact sensor on the pump are collected and recorded oscillations in the housing and then turned into temporal electrical signals. These signals contain information that can be recovered by using a frequency analyzer.
The next important step is to determine the evolution of the amplitude peaks in the frequency spectrum, in order to pair them with the mechanical causes which led to these peaks.[3,4]
The machine status results from a qualitative interpretation of measurements by comparison with the frequency spectrum determined on a new pump without abnormal wear and without functional abnormalities.
In this paper, vibration reading on a pump was done by using either a piezoelectric acceleration sensor or an acoustic emission one. Further signal processing involves using a power amplifier and a computer with which the signal's frequency analysis was made.
The three components of analysis that need to be made are shown in schematic representation in figure 2.

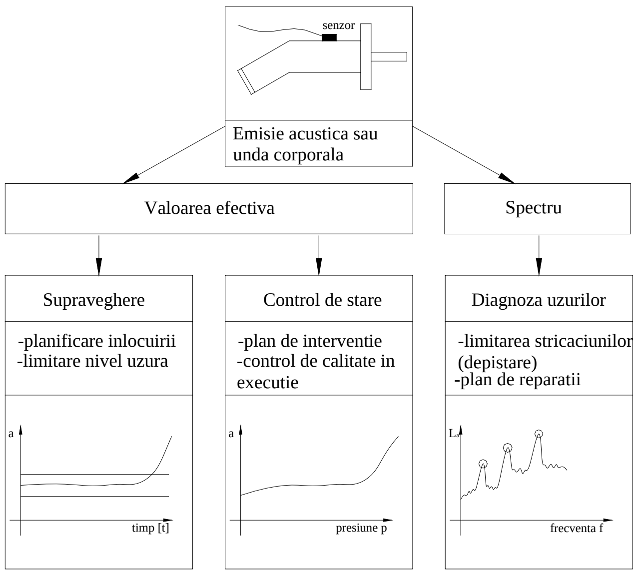

Fig. 2 Representation of the body wave analysis for pump

Pump operation control should be carried out in relation to a reference value, which if changed outside preset limits shall be interpreted as a necessity of intervention to prevent expansion of the anomaly that caused the change of this value.
Root - mean - square (RMS) is the cheapest and most commonly used mathematical method for assessing the average level of a oscillating signal.

This is calculated by the formula:

$$U_{RMS} = \sqrt{\frac{\int_0^t u^2(t)dt}{t}}. \quad (1)$$

As interpretation, it reflects the level of constant signal that produces the same effect as the original signal under the same conditions.
When analyzing U_{RMS} reported to a functional parameter (id pressure) judgments can be made on the pump' state at a certain time.
Such analyzing systems can be introduced for pumps' quality control to the manufacturer. This prevents measurement of working parameters, raising curves and other operations that require a large amount of work and instrumentation.
The "oscillation" behavior of the pump related to the pressure load, which then compared with the shape obtained on a reference pump, allows feedback on the state tested pump.
Using this method for checking the quality of the pump, test costs decrease. The method is very good, being able to provide product warranty.
If body wave analysis is limited to the above, it leads to an appreciation of "good / bad". If the verdict is "bad" further diagnosis is required to determine the exact fault for which the verdict was given.
Additional diagnosis aims at exploiting the information included in frequential representation of the u(t) function or AE that describes the wavy process.
Once highlighted in the frequency spectrum of u(t) function an abnormal operation, diagnose system must allow, based on previously established methods, the exact determination of abnormality's characteristic from the pump's mechanical structure.
Based on this information a decision is taken on when to intervene for repairs (replacement of comonents or compensate for wear or irregularity of form).
The diagnose method must therefore allow to show information about the further evolution of the fault, namely those concerning the measures that need to be taken to their slowdown.
The difficulty of using this method is that from all the information it provides, the essential information must be discerned in a simple manner.
In respect to this, the most efficient solution is to provide a database that contains the frequency spectra for progressive developments of the anomaly.
To create the "database" a program was developed, outlining the frequency spectra for defects or wear of pump that can occur during operation.
The condition for a practical use of this method is related to the existence of a database as complete as possible, containing the manifestation of every possible abnormalities.

3. OSCILLATORY ANALYSIS ON A PUMP WITH WEAR BEARINGS

For this experiment a train bearing (36 rolls) with tread wear to the allowable limit was used, with an estimated 3000 hours of operation.
The displacement amplitude spectra for the pump with bearing wear are shown in Figures 4 and 5.
In defect rolling bearing case, irregular vibration levels occur with the following frequencies of impact:

- For defects (wear) on the outer surface of the paths

$$f_1 = \frac{n}{2} \cdot f_r \left[1 - \frac{d_B}{d_m} \cdot \cos\beta\right] \quad (2)$$

- For defects (wear) on the inner surface of the paths

$$f_2 = \frac{n}{2} \cdot f_r \left[1 + \frac{d_B}{d_m} \cdot \cos\beta\right] \quad (3)$$

- For defects (wear) of balls or rollers

$$f_3 = \frac{d_m}{d_B} \cdot f_r \left(1 - \frac{d_B}{d_m}\right)^2 \cdot \cos^2 \beta \qquad (4)$$

the following notations were used:

n - number of balls or rolers

f_r - relative motion corresponding frequency between the outer and inner rings,

β - contact angle,

d_B - diameterof one ball or roller,

d_m - medium diameter of the bearing.

For the case of the bearings in the pump frequencies f_1, f_2 and f_3 are calculated with the numerical values: $n = 36$, $f_r = 25\,Hz$, $\beta = 180^\circ$, $d_B = 16\,mm$, $d_m = 70\,mm$.

For these values of working parameters we obtain the following numerical values for f_1, f_2 and f_3:

$$f_1 = 18 \cdot 25 \cdot 0{,}77 = 346{,}7\ Hz$$

$$f_2 = 18 \cdot 25 \cdot 1{,}23 = 553{,}5\ Hz$$

$$f_3 = 4{,}38 \cdot 25 \cdot 0{,}77^2 = 64{,}92\ Hz$$

Also amplitudes of the fundamental frequency increase in value when compared with the new bearings.

In large frequencies up to 6000 Hz, there is an explosive growth of a field frequency in 3000 ÷ 4000 Hz range, with maximum amplitude around the frequency value of 3400 Hz.

If we analyze the RMS level in the two situations (new pump and pump with bearings worn - Figure 3) there is an increase of this indicator by about 15 dB. Average slope variation of the RMS level with the pressure parameter is about the same.

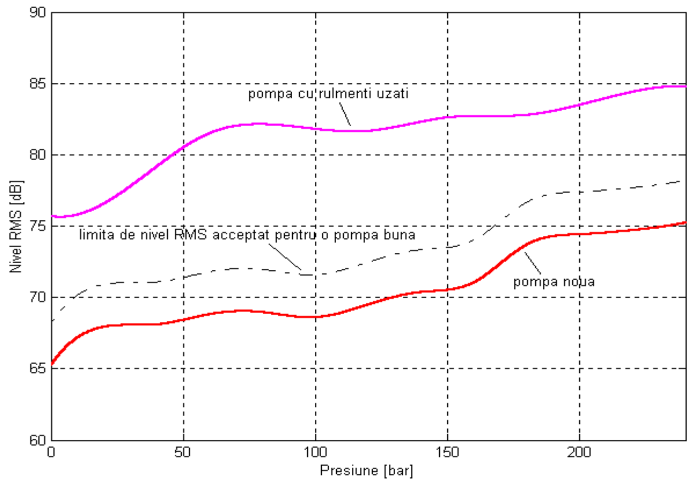

Fig. 3. RMS level for new pump and pump with worn bearings

Although experimental research has not allowed for reasons of time, continuous monitoring of the change in the RMS (Root Mean Square) during pump operation (3000 hours) there is a proportionality RMS level in dB.

Therefore, that increased RMS level is directly proportional to the degree of wear of the bearings.

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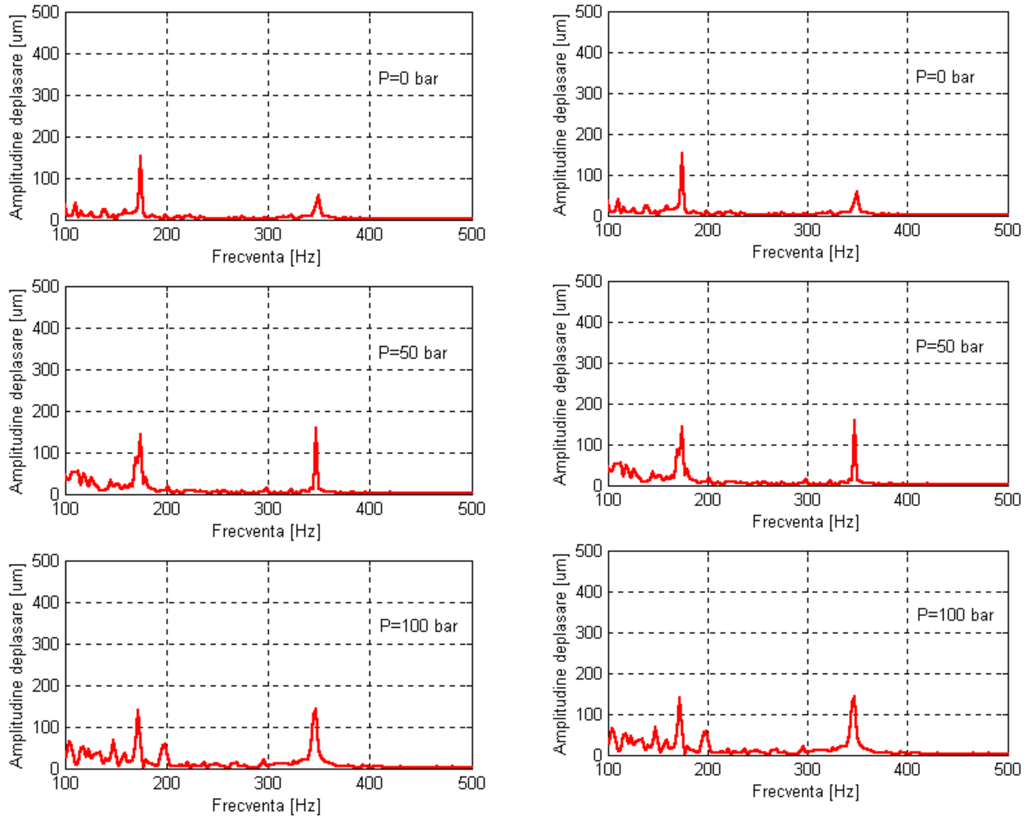
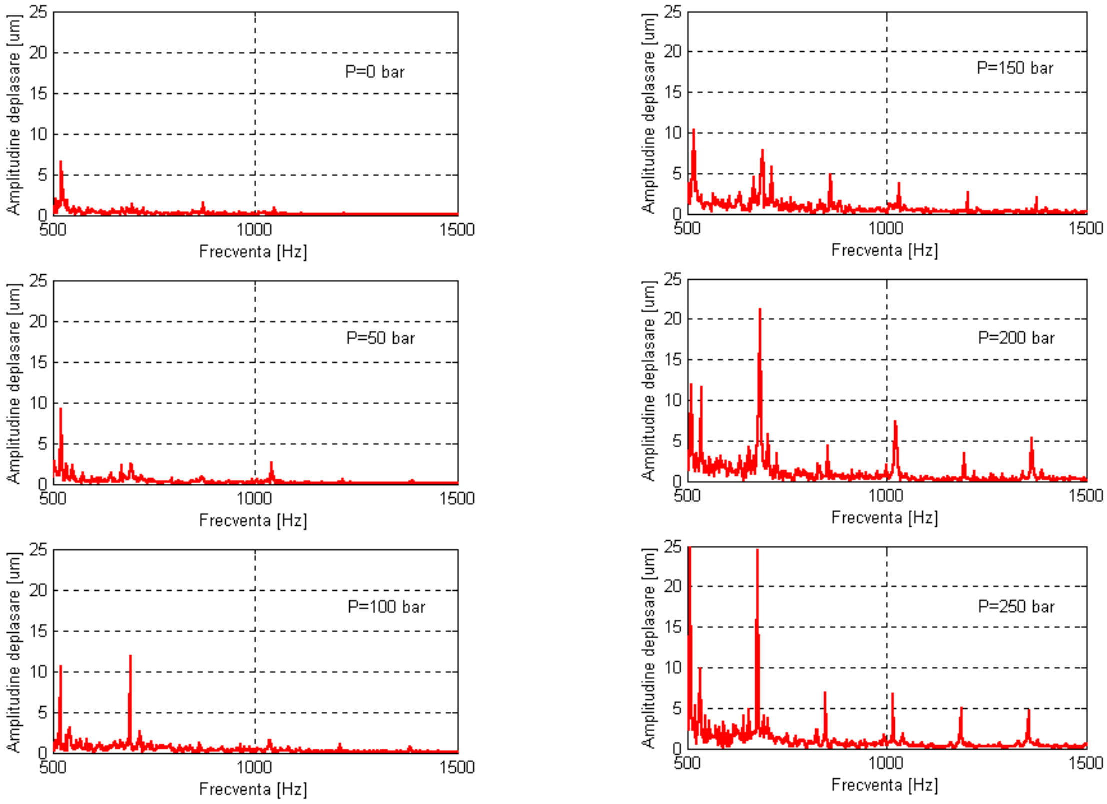
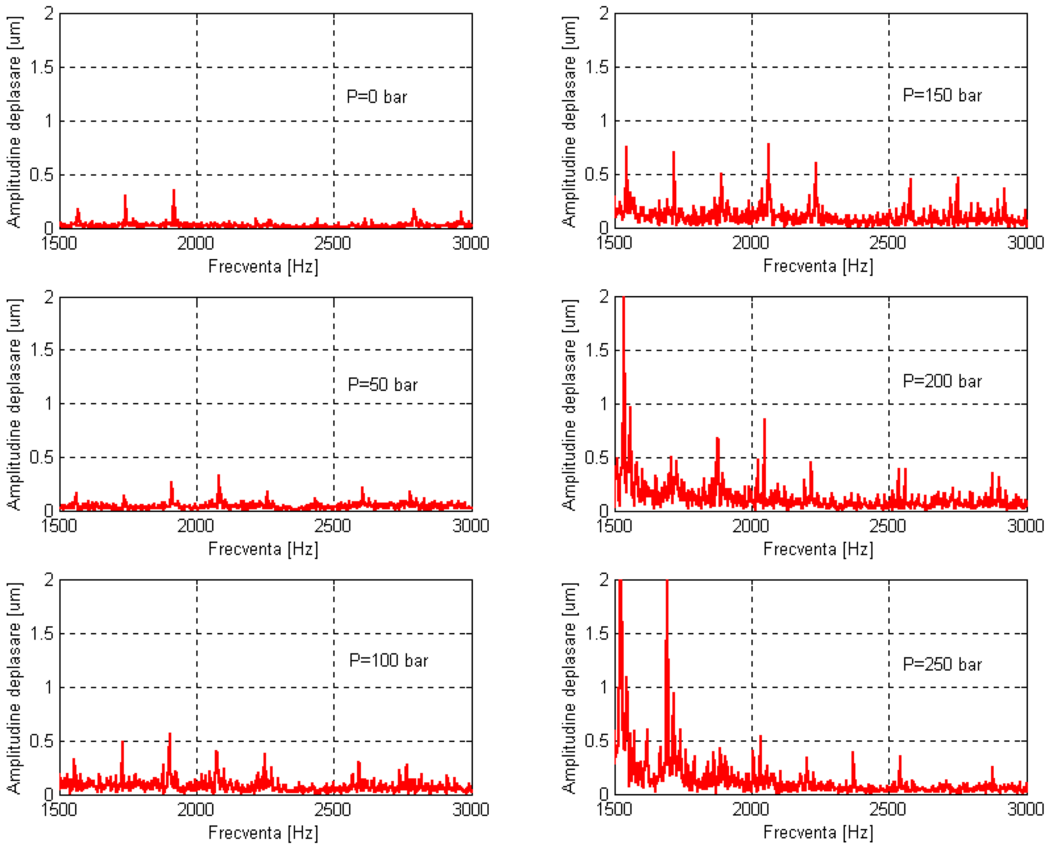


Worn bearing in 100-500 Hz range

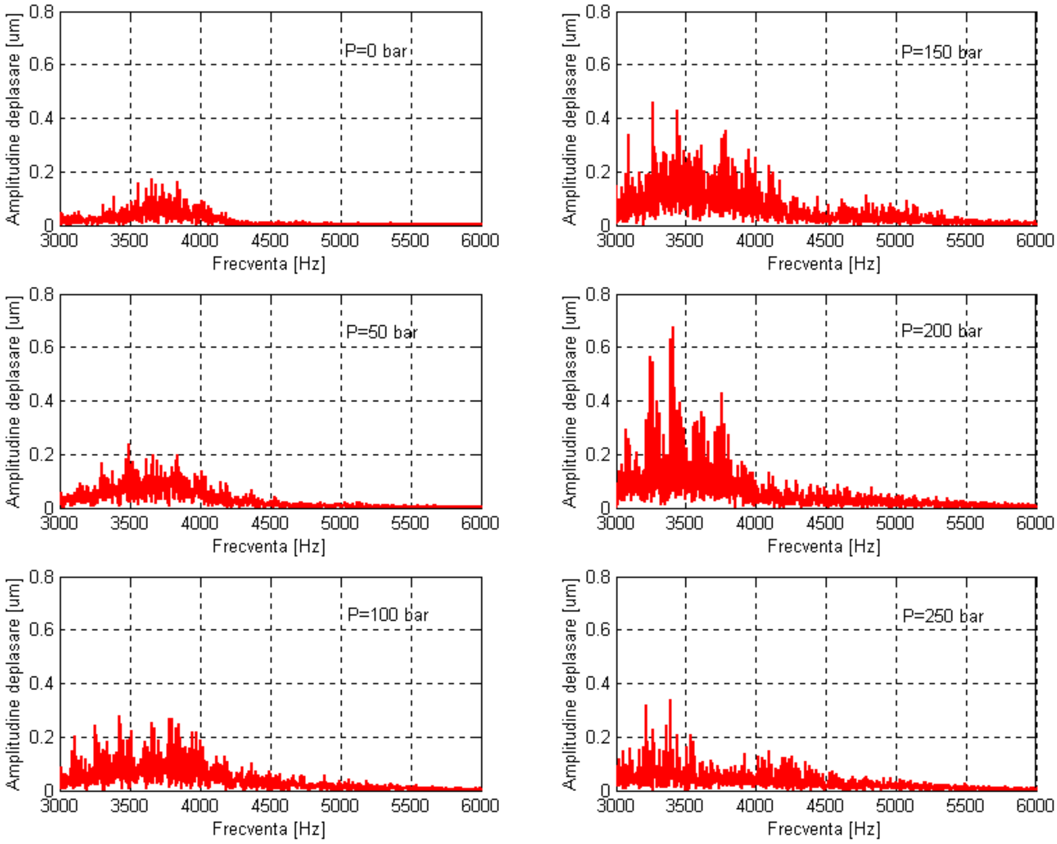


Worn bearing in 500-1500 Hz range

Fig. 4 Displacement amplitude spectra for pump with bearings worn in the range 0 ÷ 1500 Hz



Worn bearing in 1500-3000 Hz range



Worn bearing in 3000-6000 Hz range

Fig. 5 Displacement amplitude spectra for pump with bearings worn in the range 1500 ÷ 6000 Hz

4. OSCILLATORY ANALYSIS ON A PUMP WITH WORN PISTONS

For this test, the pump was equipped with a set of worn pistons with the clearance against the bore cylinders about 0.1 mm. Wear spectra caused in this case are shown in Figure 7.
The following features are highlighted:

- - Subharmonics of the fundamental frequency of 175 Hz appear in the low frequency area;
- - At pressures above 100 bar displacement amplitude is doubled in value;
- - Flow pulsation frequency (525 Hz) is dominant;
- - The frequency field enhance around 750 Hz;
- - In the high frequencies an explosive growth is observed in the frequency field 3500 ÷ 5000 Hz with maximum amplitude peaks around 4000 Hz. These fields are very intense at high pressures (100 ÷ 200 bar) their appearance being due to leakage losses.

If we analyze the RMS level of the worn piston pump with the new pump (Figure 6) related to pressure, an increase of about 20 dB is observed. the pressure variation curve has a steep slope for small values (up to 50 bar) and then is kept at the same value as that of the new pumps.

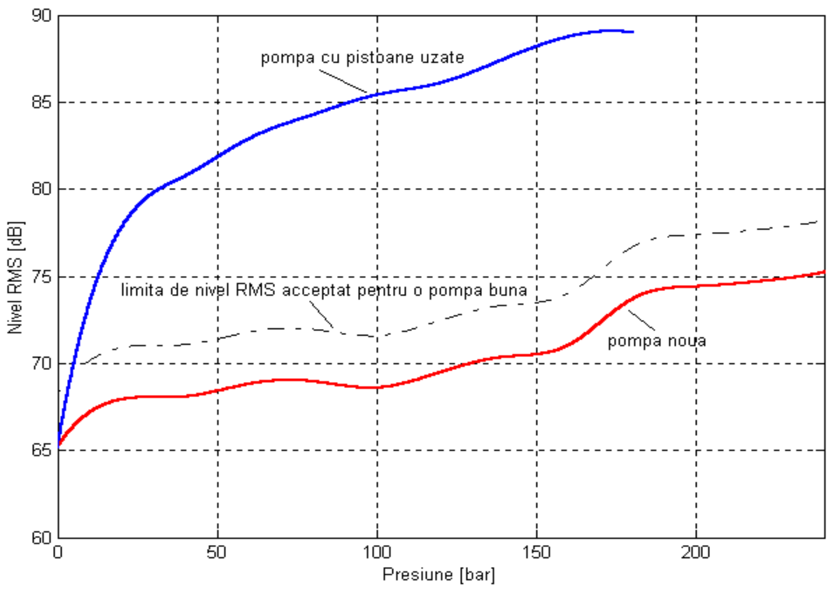
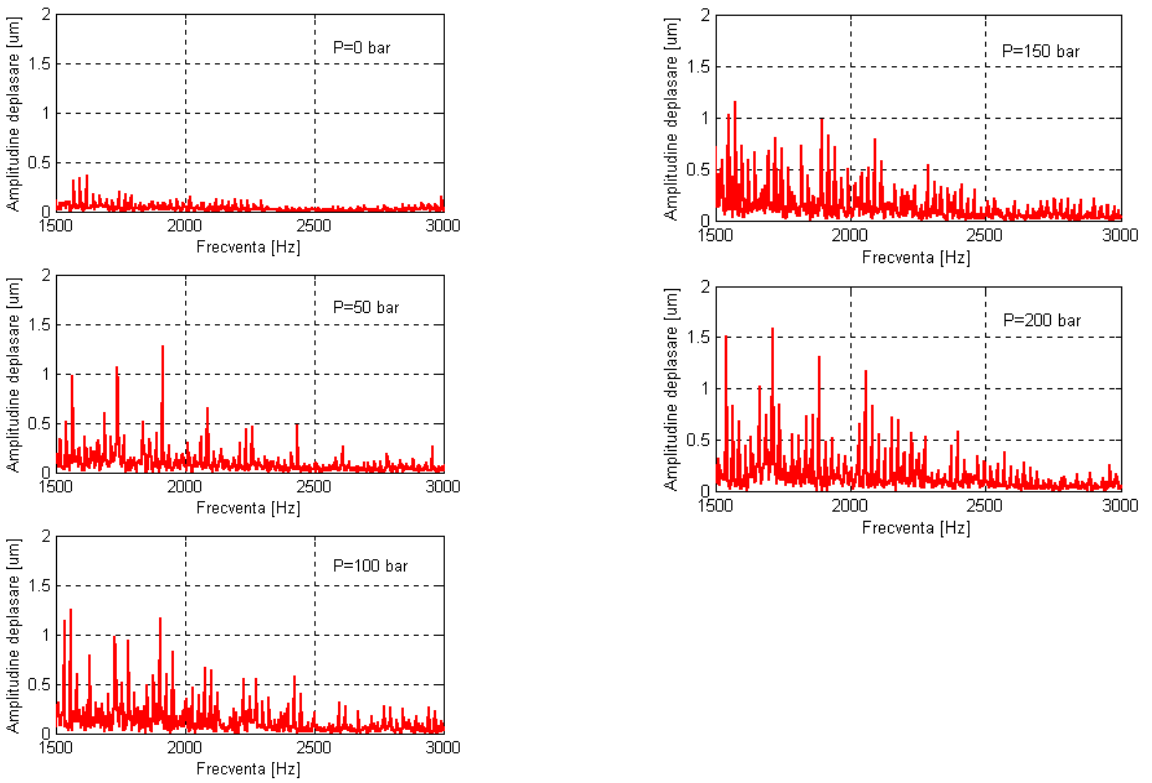
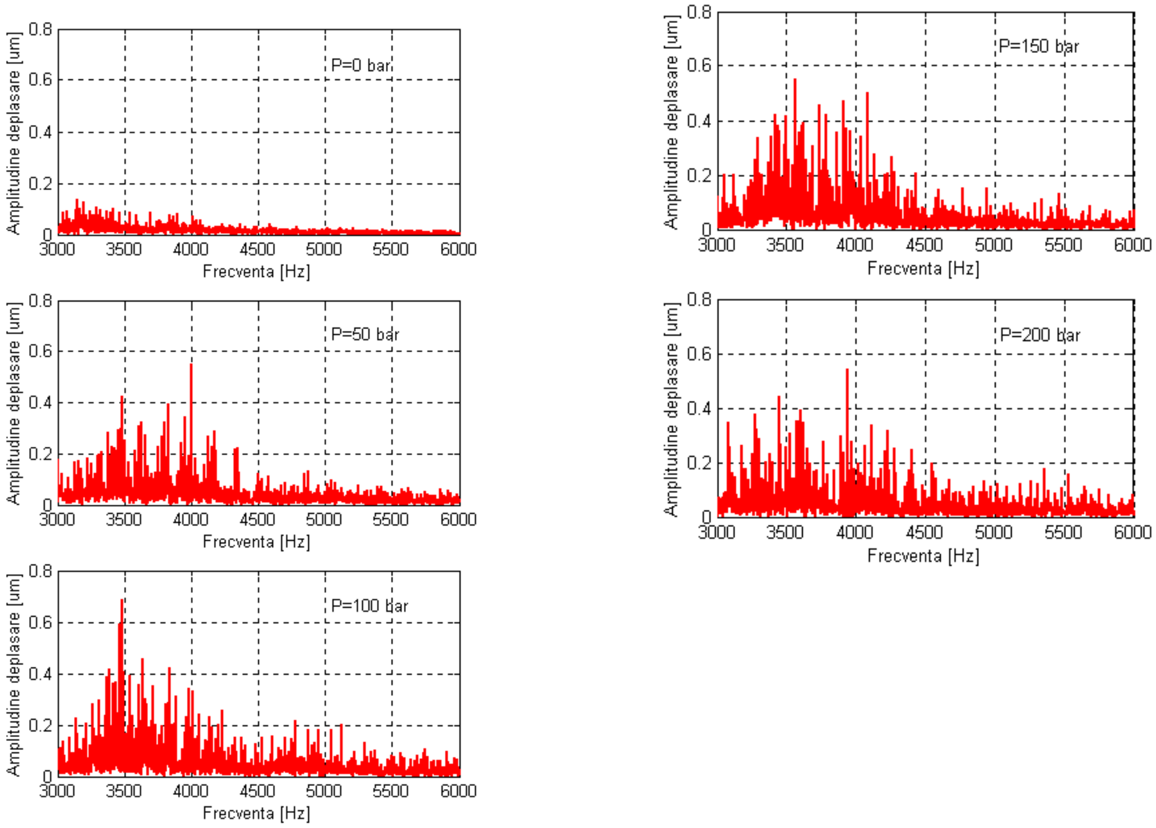


Fig. 6. RMS level for pump with worn pistons and new pump



Worn pistons 1500-3000 Hz range



Worn pistons 3000-6000 Hz range

Fig. 5 Displacement amplitude spectra for pump with pistonss worn in the range 1500 ÷ 6000 Hz

5. CONCLUSIONS

Based on information and associations commented on the two types of wear presented, data is obtained by interpreting the frequency spectrum for all types of wear or abnormalities that may occur during pump operation; this information is stored in a database.

The database allows a diagnose of wear and anomalies on pumps.

Interpretation of frequency spectra and the decision regarding the evolution of the state indicators is performed by computer.

The obtained results by diagnosis confirm the fact that such a process may be practical.

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