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ANALYSIS OF THE INFLUENCE OF DESIGN AND OPERATING PARAMETERS ON PRESSURE PULSATIONS IN CENTRIFUGAL PUMPS.

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Abstract

The blade passing frequency (BPF) pressure pulsations in centrifugal pumps, caused by unsteady hydrodynamic interaction between the impeller flow and pump volute casing, consist of a "pseudo-sound" component and an acoustical component. The first component arises from the vortex incompressible motion of the liquid and does not propagate into the pump circuit. Generally this type of pulsation effectively exists near the impeller exit on a radial distance of about 10% of the impeller radius and in a local zone near the cutwater. The second component is acoustic waves generated by unsteady flow. This component interacts with boundary and impedance conditions at the pump casing exit and on the volute and diffuser walls.

There is a lack of publications on these interaction phenomena. Some qualitative experimental results published [1] show that the inlet and exit pipe impedances do not affect considerably pressure pulsations into the volute. The influence of local wall impedance (absorbent coating) has not been published before.

The present paper provides some computational results concerning the effects of pump geometry and boundary/impedance conditions on pressure pulsations.

1. GOVERNING EQUATIONS

The blade-passing frequency (BPF) pulsations are described by the following acoustic-vortex wave equation relative to enthalpy oscillations

$$\Lambda^2 \frac{\partial^2 h}{\partial \tau^2} - \nabla^2 h = s\,,$$

where:

$\Lambda = \dfrac{u_2 Z_1}{2\pi a}$ -- relative blade passing frequency,

$h = (H + H_{\mathbf{a}}) \approx \dfrac{p - p_0}{\rho u_2^2}$ --enthalpy (pressure) pulsation;

H -- "pseudo sound" pulsation,

$H_{\mathbf{a}}$ -- acoustic pulsation.

s – source function which depends on the velocity field of vortex incompressible flow

By using a local specific acoustic impedance Z (complex value), the boundary condition can be put in the form

$$\frac{\partial (h - H)}{\partial n} = -\frac{\Lambda}{Z} \frac{\partial (h - H)}{\partial \tau}$$

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2. HARMONY: A CODE FOR PRESSURE PULSATION COMPUTATION

The code Harmony [2] has been developed to serve as a user-friendly tool in the analysis of pressure pulsations within a centrifugal machine. It is a two-dimensional representation of the machine with the following features:

- Computation of absolute pressure pulsations and hydraulic forces acting on the casing.
- Computation of the unsteady pressure field in the working cavity.
- Optimization of the geometry (impeller, volute and diffuser) under a given operation mode.
- Analysis of centrifugal impeller flow.
- Prediction of the acoustic impedance effect on pressure pulsation within the working cavity.

In order to make computations more time-efficient, several simplifying hypotheses were applied :

- the flow is considered isentropic and subsonic
- the fluid is considered homogeneous and viscous diffusion is neglected
- the impeller flow is considered as axisymmetric.

The overall numerical procedure splits into 5 main steps which optimize the computation time and memory required:

⇒ Boundary condition for the vortex mode at the pump volute inlet.
⇒ Impedance boundary condition at the inlet and exit sections of the pump casing.
⇒ Incompressible flow (vortex mode) computation in a pump casing.
⇒ Source function computation (the right part of the main acoustic-vortex equation).
⇒ Solution of the main acoustic-vortex equation which is a non-uniform wave equation.

Definition of boundary condition for the vortex mode includes centrifugal impeller flow computation with 2D "discrete-vortex method" and subsequent determination of flow parameters at the impeller exit. Vortex mode and wave equations are solved on a 2-dimensional polar grid exponentially transformed with non-stationary finite-difference method. The vortex mode solver has the first time order upwind scheme for advection step and a fast direct procedure for solving the Poisson' equation. The wave equation is solved with the second time order direct procedure with the specified impedance boundary conditions for each harmonic of the blade passing frequency.

Efficient computational algorithms and original pre-processing and post-processing procedures provide an efficient environment for design optimization. A typical task can be computed within one day on a PC.

The applicability of computation is limited to

- Centrifugal pumps or ventilators with specific speed 40 < n_S< 150 under normal operation mode where $n_S = 193.3\omega\sqrt{Q}H^{-3/4}$.
- Normal operation mode guarantees the accuracy of computation 1.0--2.0 dB
- Delivery range 0.8--1.1 of the rated value.

Geometry may include arbitrary impeller blade profiles (additional short blades are also permissible) and arbitrary volute-diffuser geometry with one outlet pipe.

The interface for determination of impedance boundary conditions gives a possibility to take into account the connected hydraulic circuit. It includes:

- Direct impedance definition for each computed harmonic of the blade passing frequency.
- "Open-end" condition.
- "Infinite pipe" condition.
- Finite-length pipe with various end conditions
- Orifice

The new version of the code is under development which will include the bladed diffuser.

2.1 Interface options

Harmony provides a user friendly interface for the impedance data input including a possibility to add optional or user-defined devices or define the impedance directly for each harmonic.

On Figure 1 Harmony interface dialog shows these possibilities for an exit impedance condition. There is a list of standard pre-defined devices such as infinite pipe, pipe without flange, pipe with flange, output into tank, pipe with conical device, pipe with fair device, restrictor, pump (open-end-condition). Outlet Geometry Window contains graphic images of all of devices added to display the outlet chain.

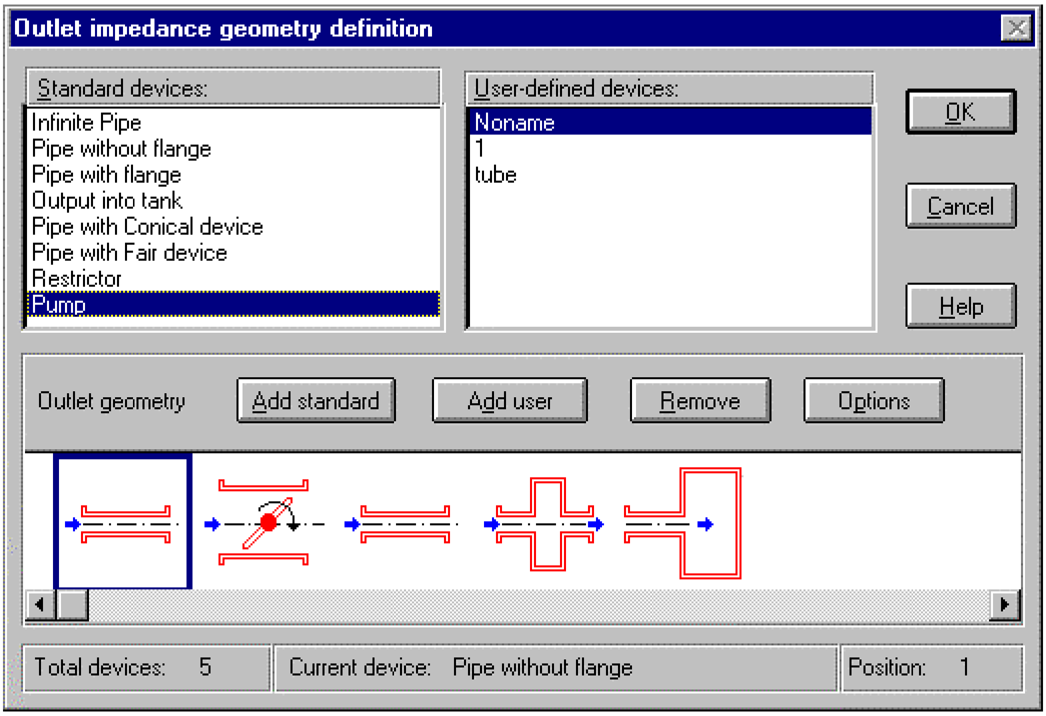

Figure 1 : Overview of the exit impedance input dialog.

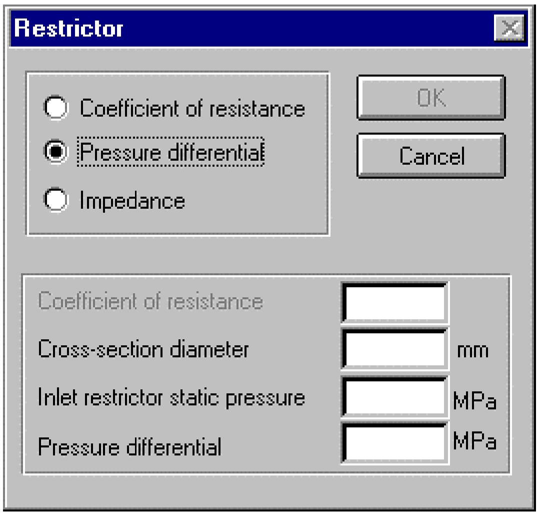

Figure 2 : Definition of a restrictor impedance.

Each standard device impedance, as shown for the restrictor on Figure 2, can be defined by geometry and other data input or with direct impedance input (see Figure 3). In the last case one should enter real and imaginary part of a complex specific impedance for each harmonic order.

There is a useful possibility to add a new device and specify the device's properties, such as the name of the device, a bitmap that HARMONY uses to represent the device, and some geometry and impedance data (see Figure 4).

Experience shows that the pressure pulsation in the pump cavity does not depend on the inlet pipe chain and the inlet impedance can be evaluated from the pump geometry only. Probably this conclusion can be corrected in case of an acoustic resonance in the inlet pipe.

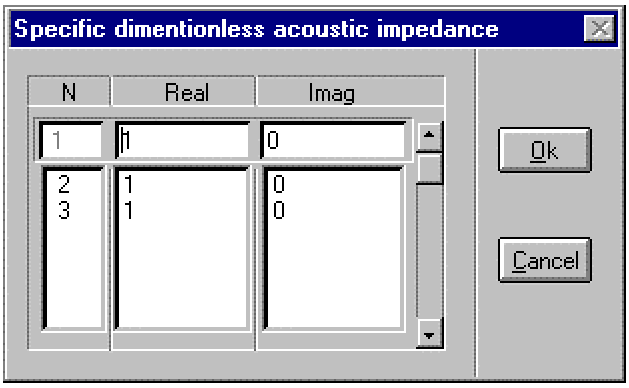

Figure 3 : Direct impedance input.

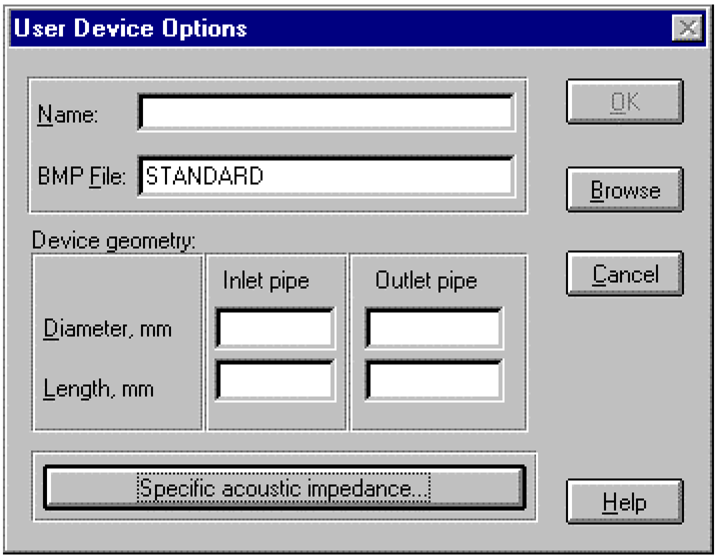

Figure 4 : User defined device.

There is another possibility that can be easily utilised in Harmony numerical procedure. This is a definition of local wall impedance to study effects of damping coating.

3. OPEN-END-CONDITION. COMPARISON WITH EXPERIMENTAL DATA

A centrifugal air pump model of CETIM [3] was used for Harmony validation. Experimental data and computation data were obtained for open-end-condition at the pump exit under following conditions. Number of measuring points in the volute >300. Number of finite-difference mesh nodes for the impeller channel span – 12. Number of overall mesh nodes >10,000. Time of computation on a PC 100MHz processor – 6 hours. Solution of wave equation for 7 harmonics of main BPF – 1.5 hours. RPM – 1400. Delivery – 0.0139 m^3/s.

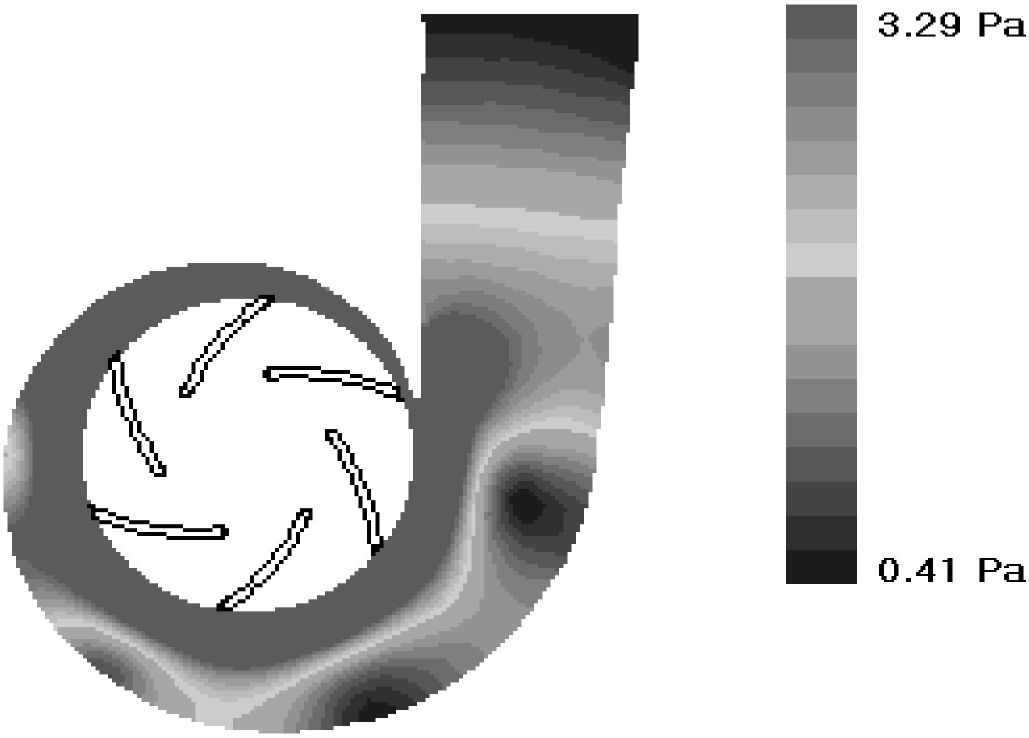

Figure 5 : Map of the first harmonic amplitude (Harmony computation).

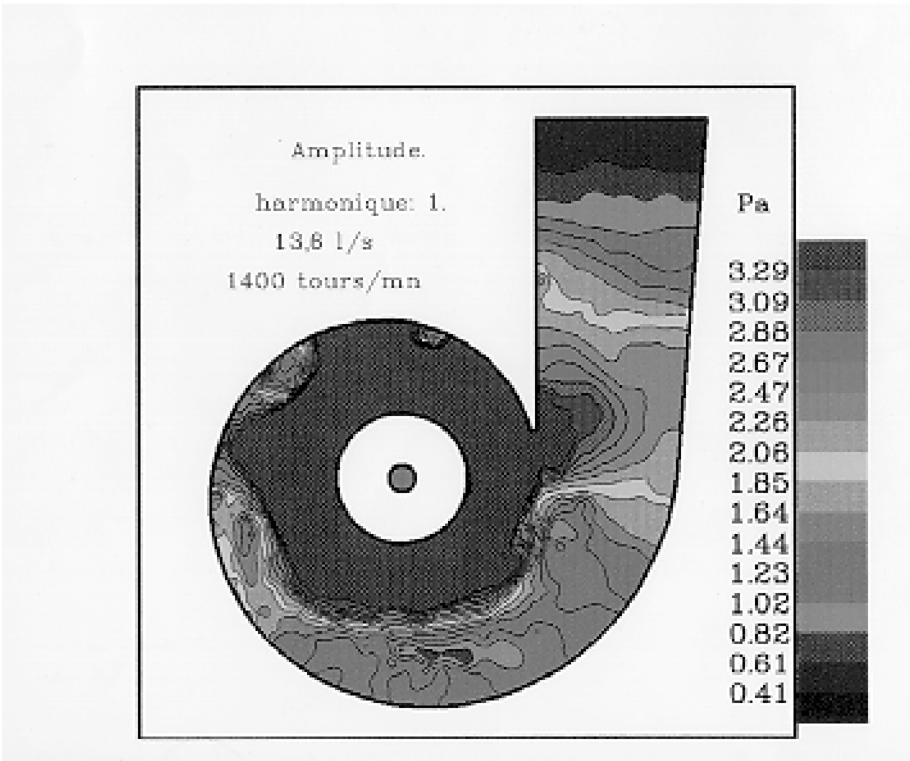

Figure 6 : Map of the first harmonic amplitude (experiment).

In the spectrum there are three main harmonics of blade passing frequency. On Figure 5 and Figure 6 there is a comparison of amplitude maps (computed and measured) for the first harmonic of blade passing frequency.

The characteristic feature of unsteady pressure in the pump volute is lower pressure zones at blade exit edges and rotating with the impeller. Harmony computation also shows such zones.

Furthermore, the pressure amplitudes stand in good agreement with the experimental data. The mismatch is mostly below 2 dB.

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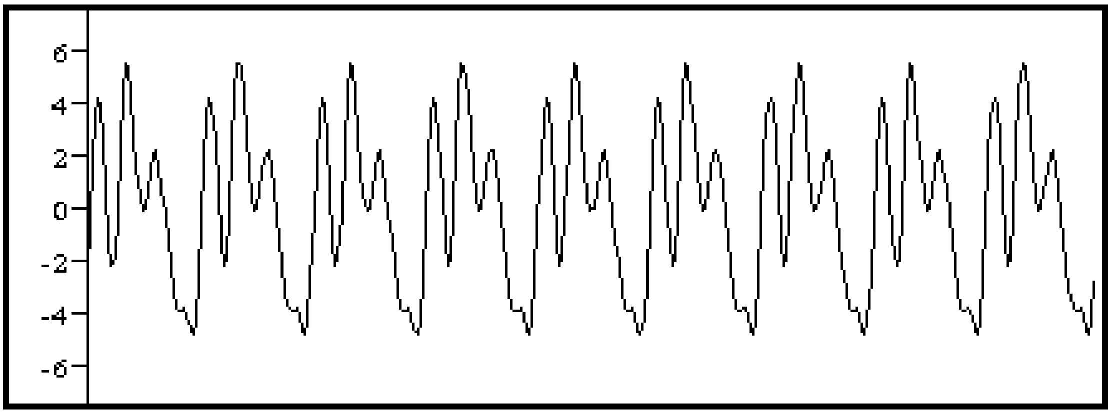

Figure 7 : Pressure pulsations in the volute [Pa]: Harmony computation.

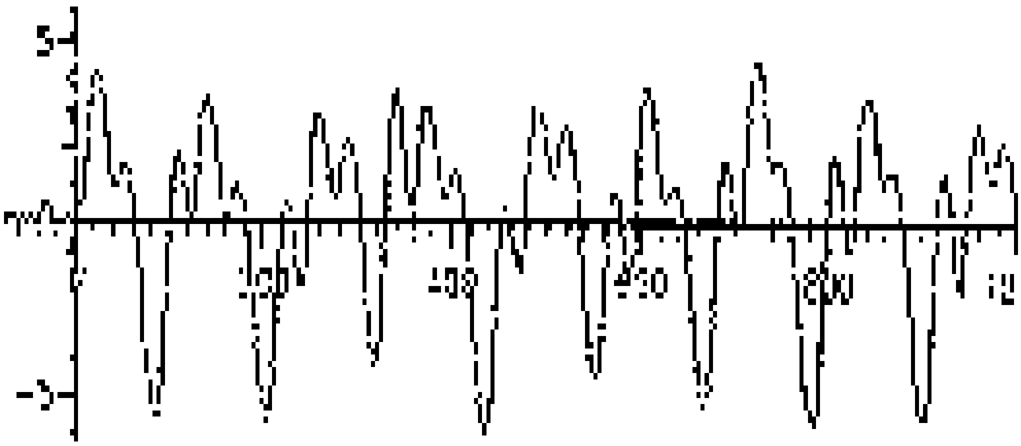

Figure 8 : Pressure pulsations in the volute [Pa]: experiment.

Such agreement in amplitude map gives a possibility to gain as a right amplitude as a shape of the pressure signal. On Figure and Figure the computed and measured pressure signals at a point in the volute are shown. Open-end-condition at the diffuser exit enforces pressure pulsation resonance on the third harmonic of blade passing frequency.

4. EXIT IMPEDANCE EFFECT. COMPUTATIONAL PREDICTION

The same pump was analysed for the case of an infinite exit pipe to determine the exit impedance effect on pressure pulsation within pump working cavity. It was found that in the most part of the volute exit impedance practically did not affect the pulsation amplitude. Near the volute throat pulsation had almost the same amplitude but different shape of the signal as it can be seen from Fig 9.

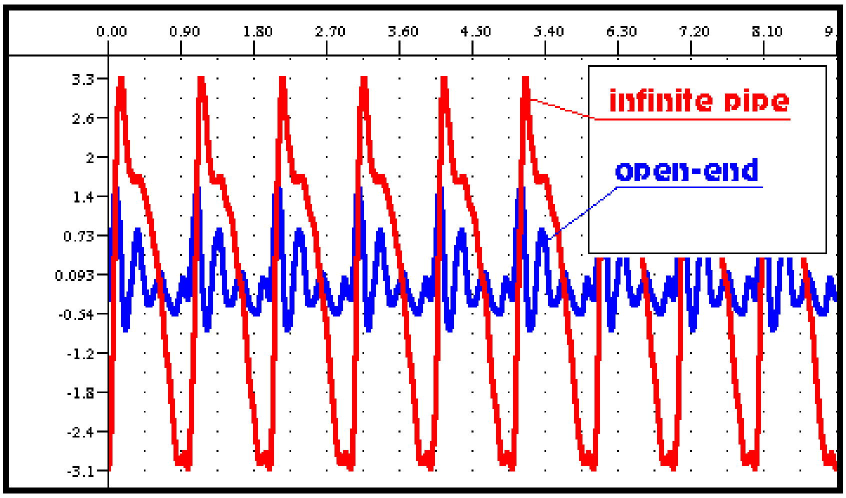

Figure 9 : Computed pressure pulsation in the volute throat section of air pump model

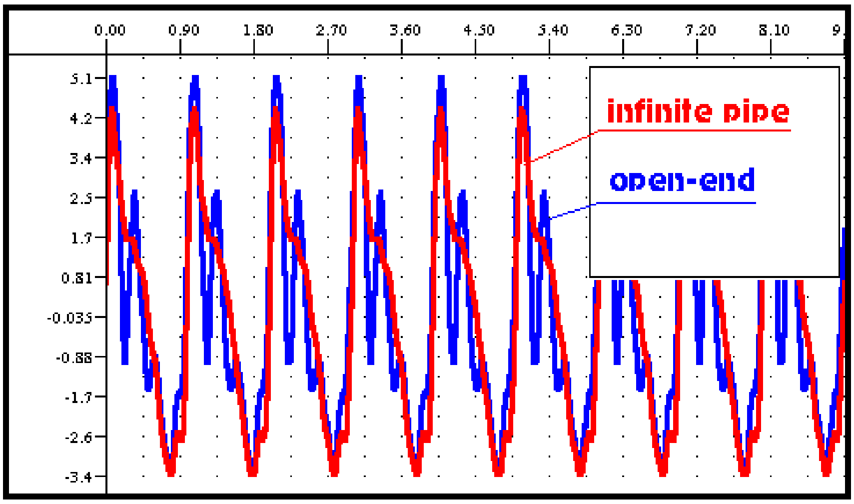

Figure 10: Computed pressure pulsation in the volute throat section of air pump model.

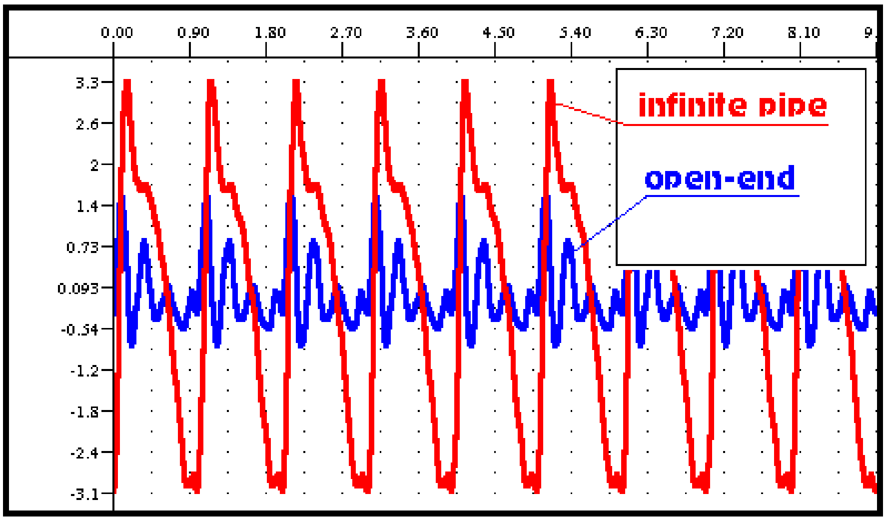

Figure 11 : Computed pressure pulsation at the air pump exit.

The biggest difference between these two cases can be found certainly at the pump exit, Fig 11, where a considerable difference could be seen in both the amplitude and the shape of signals.

5. LOCAL WALL IMPEDANCE EFFECT. COMPUTATIONAL PREDICTION

A computational study was undertaken to see a qualitative effect of damping coating of conical diffuser wall for the centrifugal pump LPRE RD180 [4].

Local wall complex specific impedance was defined by

$$Z = 1.5 - i(1.5N\Lambda)$$

where $\mathbf{\Lambda}$ is a relation of impeller radius to main BPF wave length and N – harmonic number. Computation was made for 4 harmonics of the main blade passing frequency.

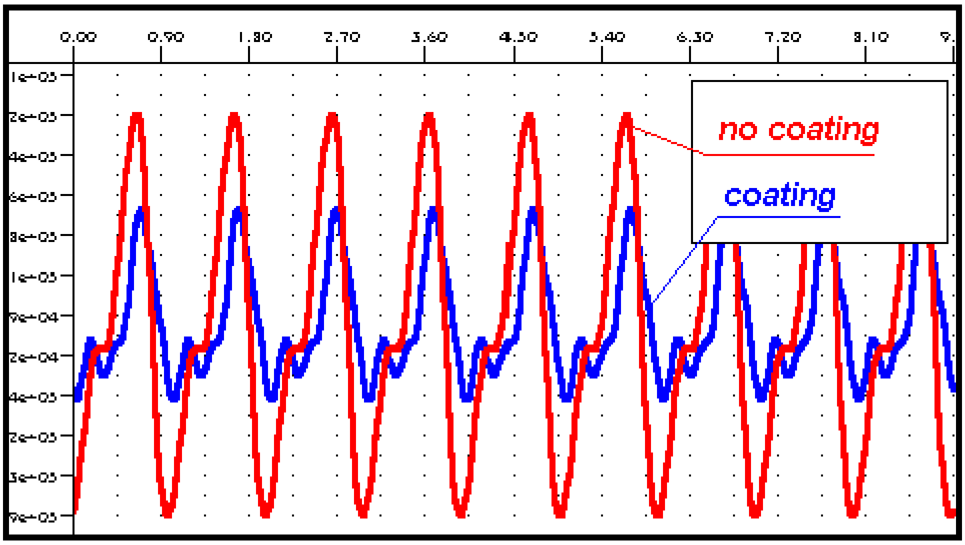

Figure 12 : Coating effect in the middle section of conical diffuser.

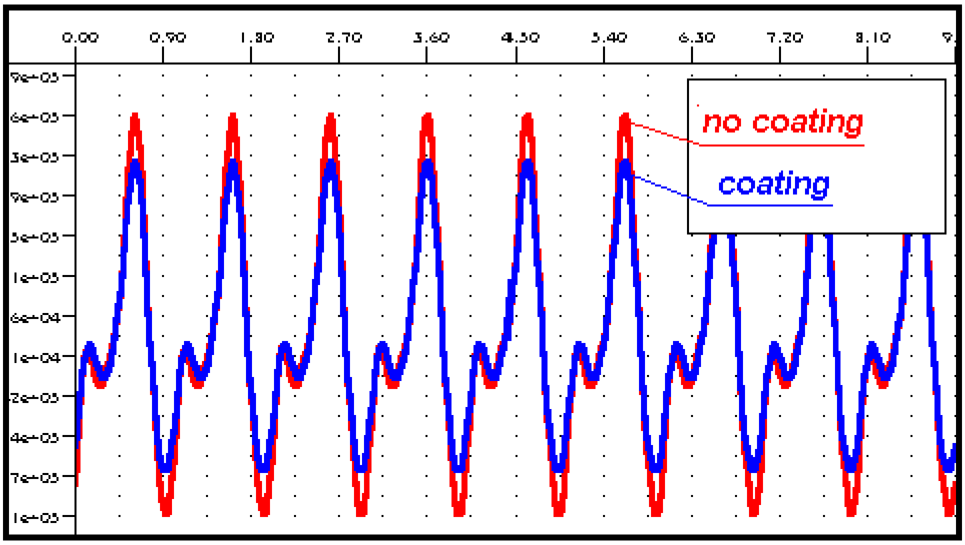

Figure 13 : Coating effect in the volute throat.

As the coating with complex impedance placed only into the conical diffuser this practically did not effect pressure pulsation amplitude into the volute until the throat domain were the effect is not considerable as one can see on the Figure 13.

Middle section of the conical diffuser is more influenced by the wall coating. It resulted in more then 2 times factor of reduction of pressure pulsation amplitude as shown on Figure 12.

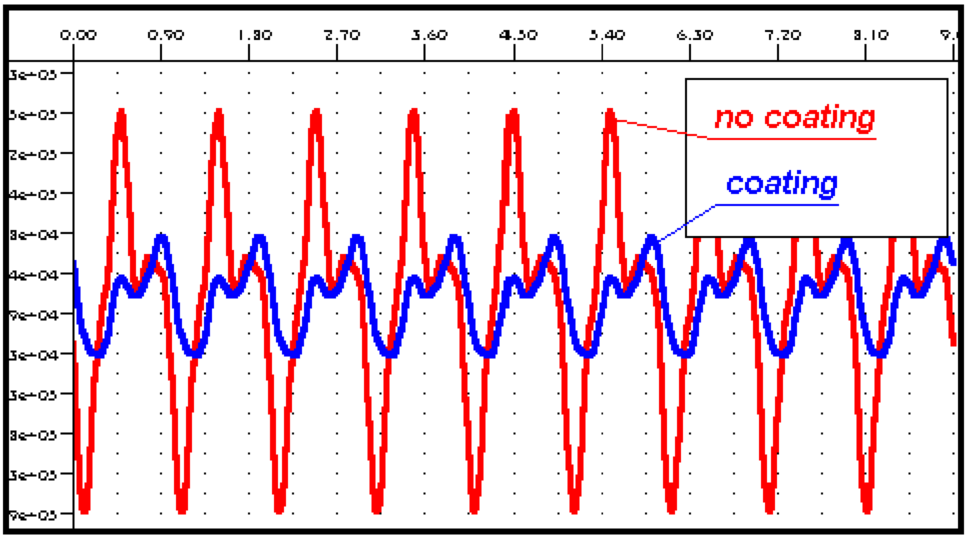

Figure 14 : Coating effect at the pump exit.

The largest effect of coating on the pressure pulsation amplitude was found at the pump exit. It resulted in a reduction of more then 3 times of amplitude as shown on Figure 14.

6. DYNAMIC LOADING

The dynamic loading of a centrifugal machine exerted by the flowing fluid depends much on its geometry. The loading was analysed for a water pump of 240 m^3/h delivery at 1200 RPM. Fig 15 shows the computed pressure pulsation distribution for the first-harmonic amplitude of this pump.

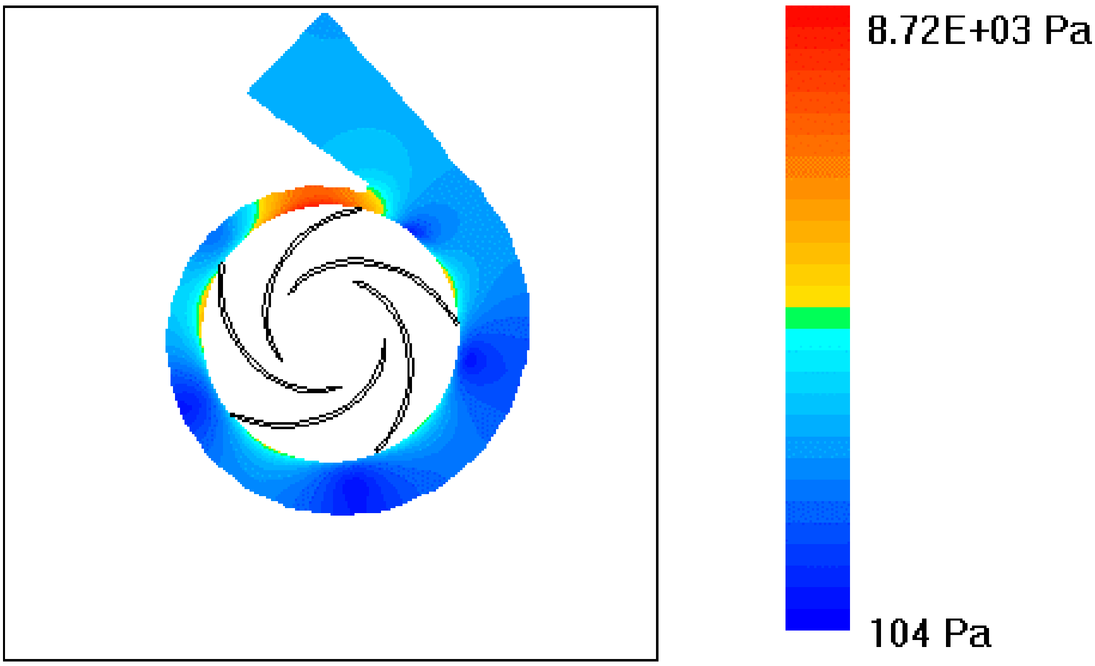

Figure 15: Amplitude of first harmonic of pulsations in the water pump

Figures 16 and 17 show the computed dynamic loading acting on the cutwater for three values of spacing between the impeller exit and the cutwater edge : 20 mm (reference value), 10 and 4 mm.

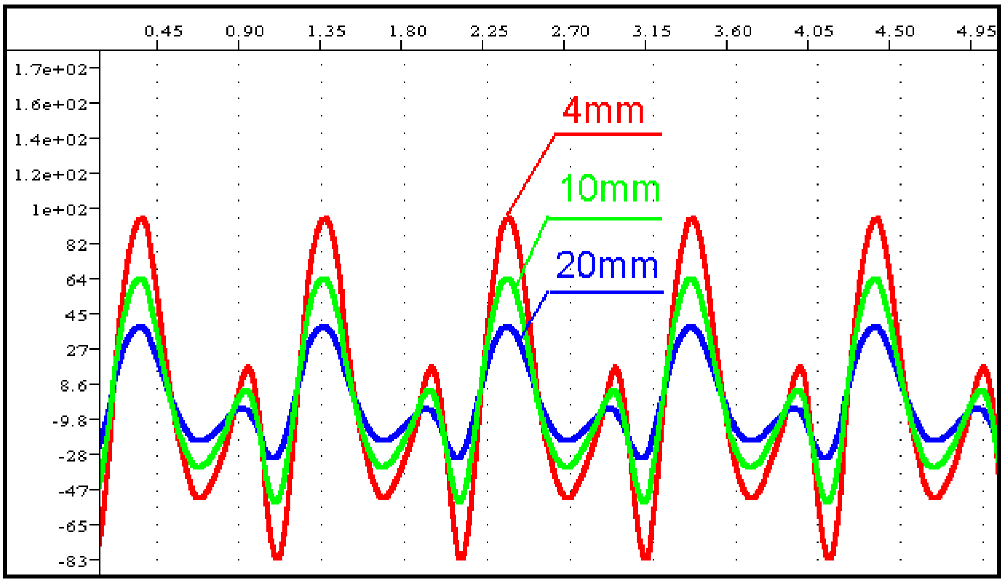

Figure 16: Tangential dynamic load [N] on the cutwater

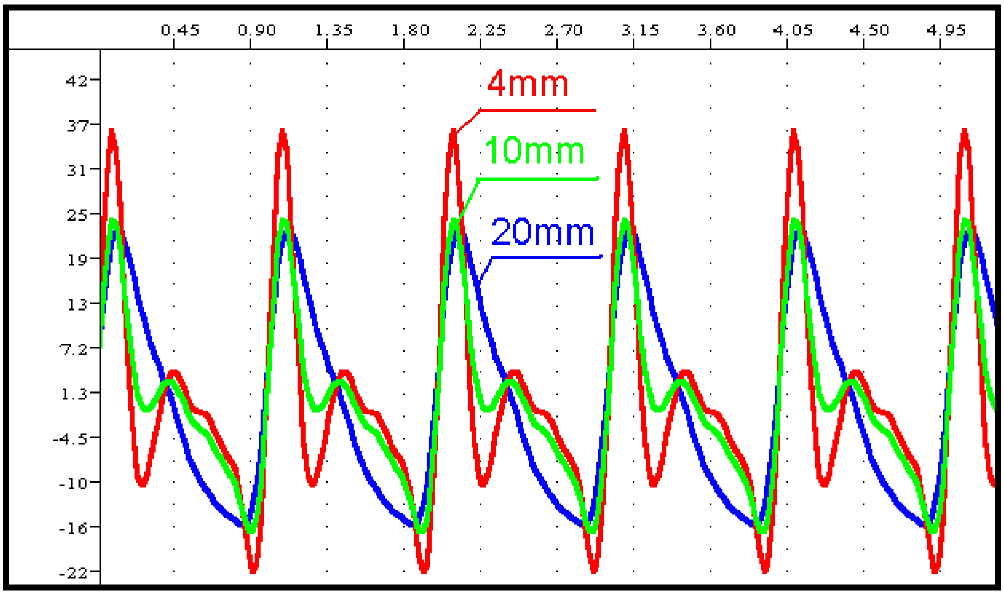

Figure 17: Radial dynamic load [N] on the cutwater

It can be seen that the extreme values of the loading change a lot with the cutwater clearence.

CONCLUSIONS

Exit impedance condition affects mainly pressure pulsation in the pump diffuser. The same result is obtained for the influence of a damping coating of the diffuser wall.

The dynamic load on the cutwater depends much on the spacing between the impeller and the cutwater.

References

[1] D. Dourneau, A. Veigne Leukam, J.P. Barrand, G. Caignaert. Effect of the line on the pressure fluctuations in the impeller and the scroll of a centrifugal pump tested with air. Fan noise - An International INCE Symposium, Senlis -- France. CETIM, 1992.

[2] Harmony MK I. Software package for pressure fluctuation analysis in the volute casing of a centrifugal pump. Theoretical manual and users' guide. IRI / CETIM, 1997.

[3] J. Tourret etc. Hydraulic Noise Generation Studies in Centrifugal Turbomachine Through Visualization of the Non Stationary Pressure Field in the Volute and in the Impeller. ASME/ Reprinted From FED-Vol.128, Experimental and Numerical Flow Visualization. Editors: B. Khalighi etc./Book No. -- 1991.

[4] S.F. Timouchev, K.P. Illichov, L.M. Koshelev. Numerical and experimental study of pressure pulsation in high-speed LPRE centrifugal stage. 4th French Congress on Acoustics, Marseille -- France, 1997.