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A FEM APPROACH TO PREDICT ACOUSTIC RESONANCE IN MULTISTAGE CENTRIFUGAL PUMP

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ABSTRACT

Acoustic resonance can develop within the hydraulic channels of the pump, especially in the long crossover channel of a multistage centrifugal pump. This phenomenon occurs when the hydraulic channel length is the same of the sound wave length produced by the Blade Passage Frequency (BPF). The matching between the BPF and the wave reflection from a stage to the next one generates an amplification of pressure pulsation and eventually a higher vibration level in the pump. This paper presents a Finite Element Method (FEM) approach to predict the acoustic resonance in multistage centrifugal pumps. The 1-D method used to calculate the equivalent length through the channel, assuming it as a straight pipe, demonstrated in some cases to be unsuccessful to predict the vibration problem related to the acoustic resonance phenomenon occurred in the centrifugal pump test bench with complex 3-D geometry, so the study of a new method approach started. The most important parameter to evaluate is the equivalent channel length which is the length of the sound wave path within a complex 3D geometry. FEM analysis performed by ANSYS® for the calculation of harmonic resonance of hydraulic channels has been used to predict the possible acoustic resonance in long crossover channels of API 610 BB3 (multistage centrifugal pump, axially split, between bearings) centrifugal pumps. The use of Acoustic Analysis coupled with the Harmonic Analysis implemented in ANSYS® allows to simulate the wave propagation and reflection in a complex 3-D channel and obtain the harmonic response of the machine. Several simulations on BB3 pumps, already tested in the past, were launched to validate the tool. The analysis showed a very good agreement between the FEM results and the vibration reports of many pumps. The new procedure was able to verify

both the pumps that showed vibration problems and the pumps without problems. A detailed case study demonstrated also that the accuracy of the method of analysis can catch the change of behavior of the system when a variation of fluid temperature is measured. After this series of simulations and comparative analysis, the new approach has become the standard procedure to detect vibration problems related to the acoustic resonance during the design phase and to avoid possible issues in the witnessed performance test.

INTRODUCTION

In the past, pressure pulsation in multistage pumps has always had a minimal consideration if compared with other dynamic behaviors. One of the first approaches to calculate the natural frequency of the long crossover in multistage centrifugal pump was [1]. Through experimental tests, the active length, necessary to calculate the acoustic resonance frequency, was determined and a correlation between geometry and active length was estimated.
With the introduction of system able to work at different speed through VFD (Variable Frequency Driver), the ability to predict the presence of problems related to resonance phenomena has become necessary. A wider range of velocity increased the possibility to match one of the natural frequencies of the system (i.e.: mechanical natural frequency of bearing housing, acoustic natural frequency of piping or crossover channel [2].
In particular, for acoustic resonance phenomena in long crossover centrifugal pumps, the low accuracy of prediction (up to 20%) is related to the uncertainties of the fluid characteristics and to the correct definition of the active length of the channel [3].
The correct definition of fluid characteristics is very important because in some cases a small change in the working conditions (e.g. temperature of fluid) can produce a significant effect on the system [4].

CASE STUDY 1

A 3 stages, 3000 rpm, 8x13 BB3 (Axially split, multistage, between bearing) double suction pump was developed for the Ammonia Project.

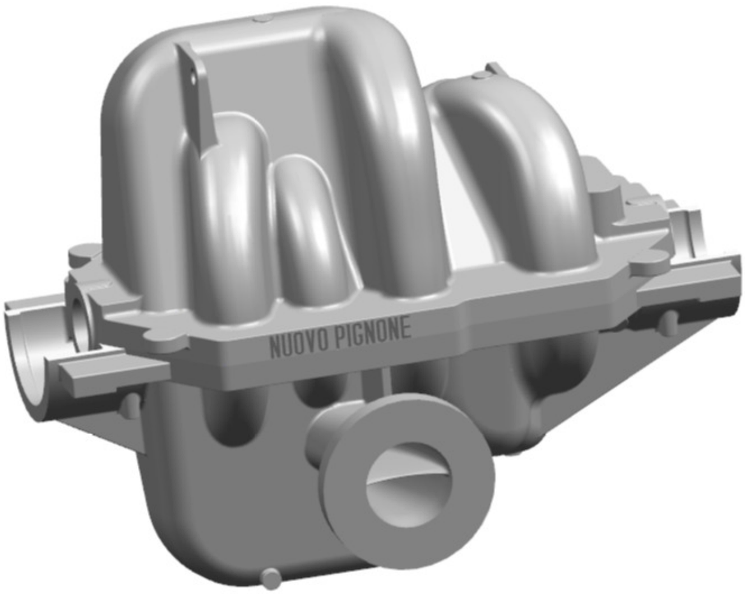

Figure 1 3D model of BB3 centrifugal pump

During the test phase, vibration measurement showed high values of the overall direct amplitude at the vertical DE at 3000 rpm, as shown in the Figure 2 , particularly at the minimum continuous flow (Q/Qbep=0.4). The amplitude peak was at the 9X, which is the BPF of the series impellers.

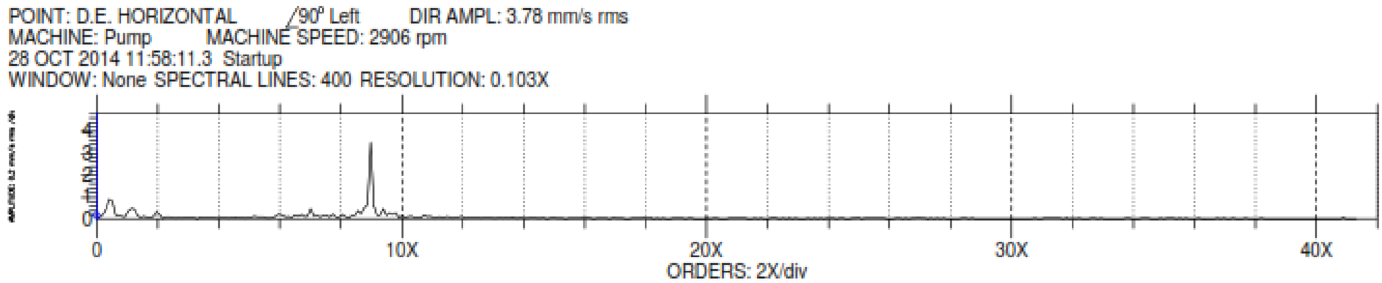

Figure 2 Spectral Lines at MCSF (Q/Qbep = 0.4) @ DE-H

A test at higher rpm was carried out to verify that it was not a hydraulic forcing, and the vibration level was within tolerance. Besides, dynamic transducers (Kistler 4075A50, 0 - 50 bar, 0 - 19 kHz) were mounted on the channels between the first and second stage and between the second and the third stage to identify any possible acoustic phenomena of the hydraulic channels (Figure 3 Figure 4).

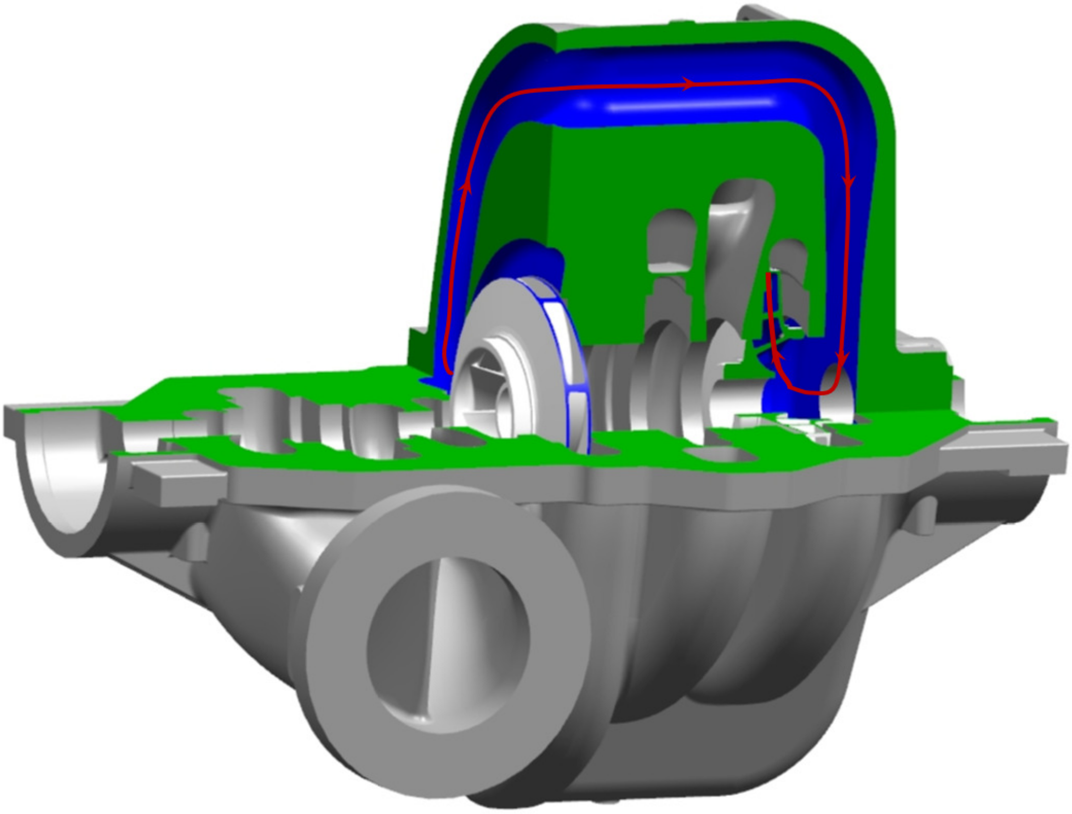

Figure 3 3D model cut view

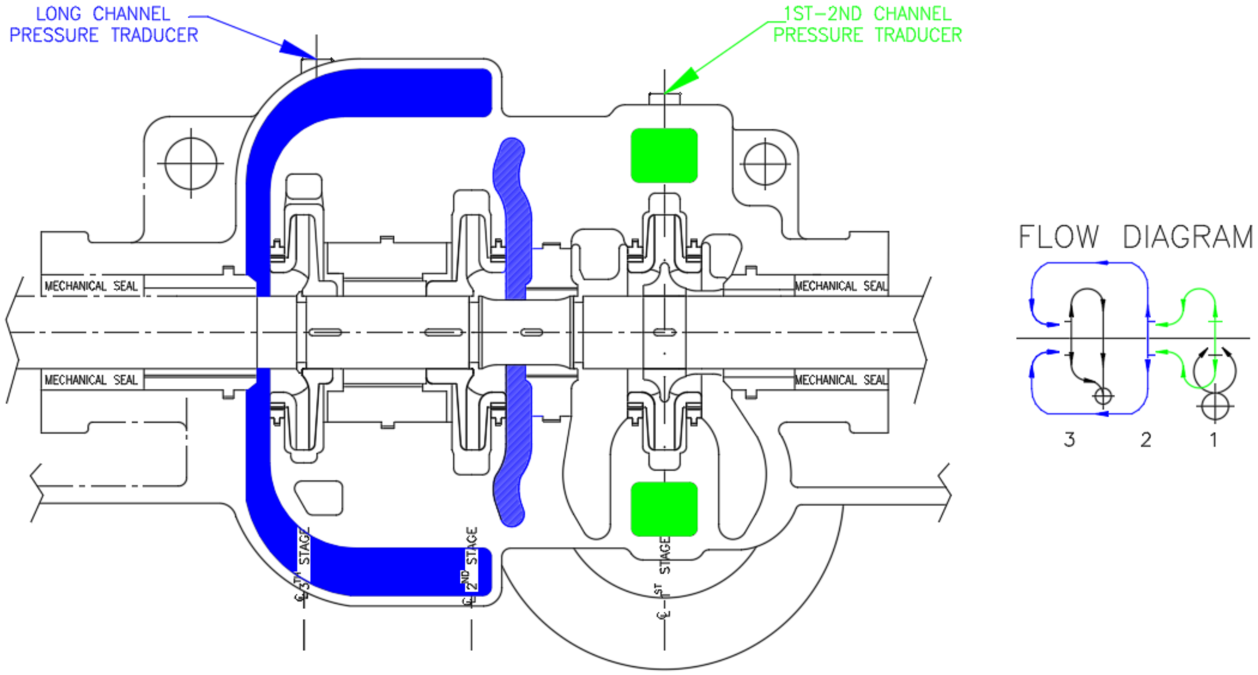

Figure 4 Cross Section of Case Study 1 pump

In the Figure 5 Figure 6 Figure 7 and Figure 8 are shown the pressure oscillations at 1st-2nd (green) stage and 2nd-3rd stage (blue) for different speed.

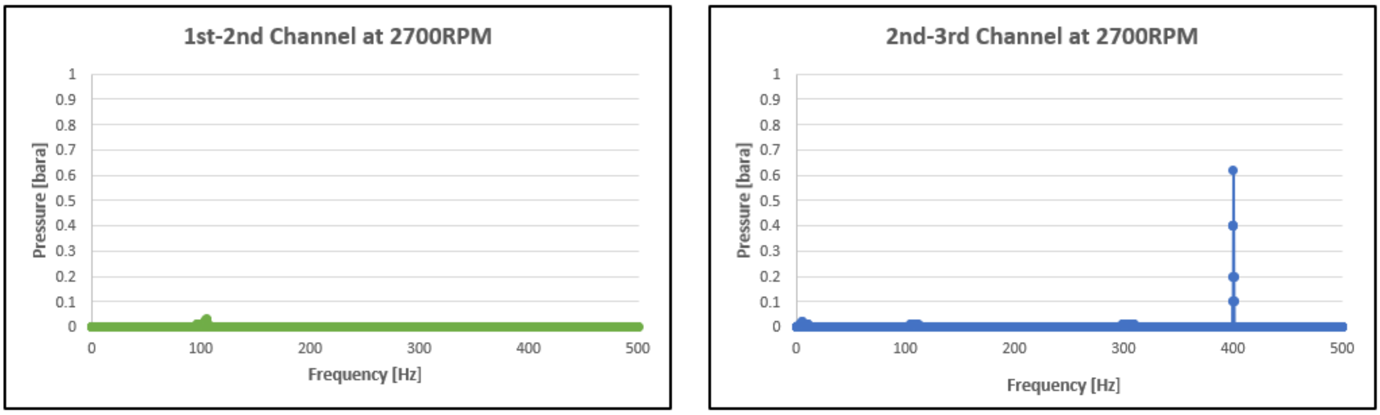

Figure 5 FFT (Fast Fourier Transformer) of pressure signals @ 2700 rpm

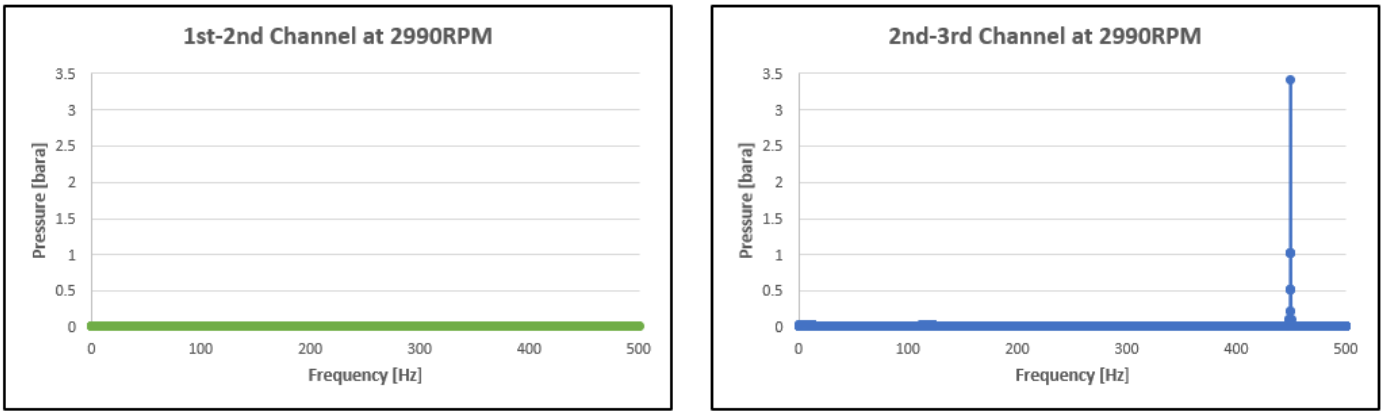

Figure 6 FFT (Fast Fourier Transformer) of pressure signals @ 2990 rpm

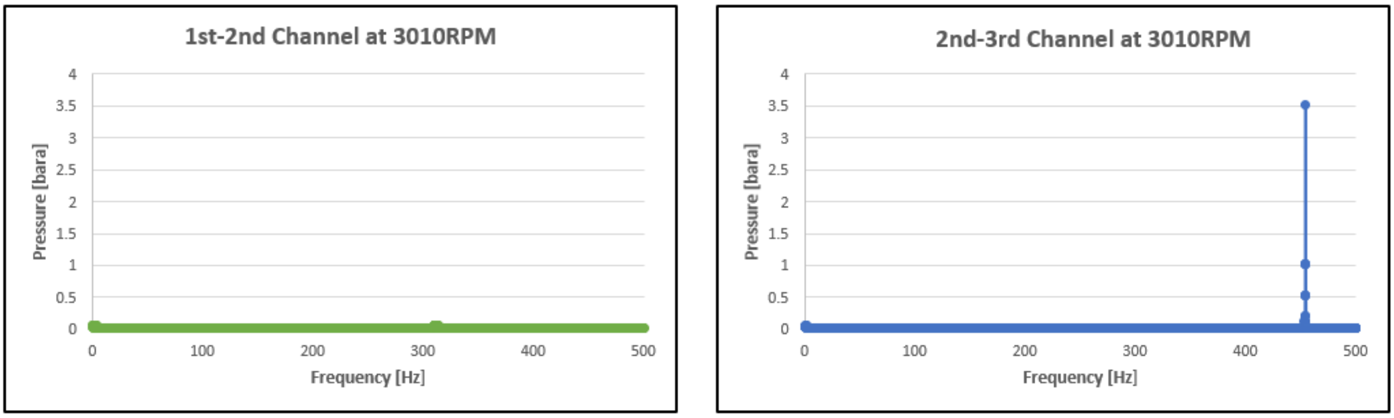

Figure 7 FFT (Fast Fourier Transformer) of pressure signals @ 3010 rpm

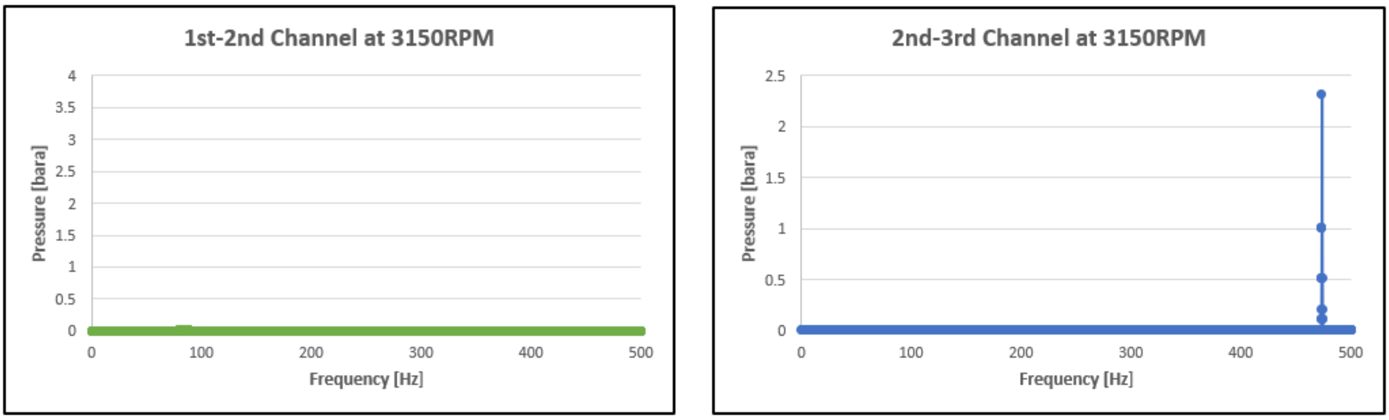

Figure 8 FFT (Fast Fourier Transformer) of pressure signals @ 3150 rpm

The pressure diagram of the two channels showed that only in the long channel a pressure peak is present, and this peak is greater when the speed is close to 3000 rpm, corresponding at a BPF equal to 450 Hz. A recap of the pressure measurements is visible in Table 1.
These results show an acoustic resonance phenomenon exists, and that this behavior was not predicted by the 1D tool. As per [5], the equivalent length to be compared with the wave length was calculated as a closed loop including distance around impeller periphery

and across next impeller eye as per Figure 9 :

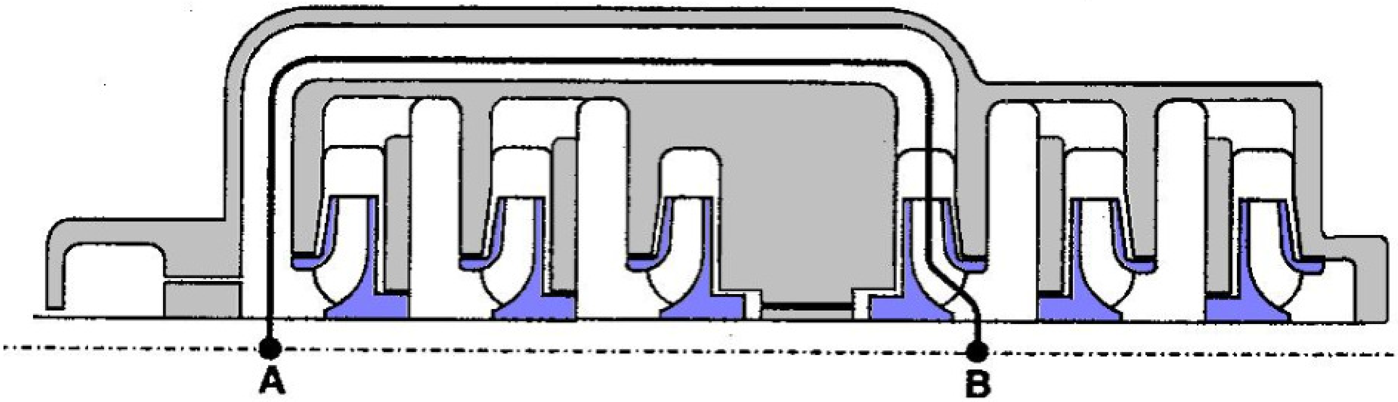

Figure 9 Equivalent length across long crossover channel (4)

Speed RPM	BPF (2nd stage Impeller) Hz	Pressure Peak-to-peak bar(a)
2700	405	0.6
2990	449	3.4
3010	452	3.5
3150	473	2.3

Table 1 Pressure peaks in long channel at various speeds

The implemented solution was the modification of the second stage impeller blades number from 9 to 7 (upstream of the long channel) and the vibrations totally returned to tolerance.
Once the specific problem was solved, it was clear that was necessary to accomplish a more accurate procedure to predict these phenomena.

FEM ANALYSIS OF CASE STUDY 1

Due to the geometric complexity of the channels, which cannot be simplified with a linear cylindrical duct, a new 3D approach was preferred to the old 1D method. The 3D modeling of the hydraulic parts was coupled to an FEM analysis, taking advantage of the Ansys® acoustic model [6].
Ansys® performs an acoustic analysis where it simulates the generation and propagation of pressure waves in a fluid considered at rest with only small pressure changes respect to mean pressure, properties of either the coupled acoustic-structural interaction (FSI) or the uncoupled pure acoustic wave in the given environment. In the FEM acoustic analysis, the rigid wall boundary is a natural boundary condition. The program assumes that the fluid is compressible but allows only relatively small pressure changes with respect to the mean pressure. The solution is the deviation from the mean pressure, not the absolute one. Also, the fluid is assumed to be non-flowing.
This analysis is available for modal, time-harmonic, and transient acoustic study. An acoustic analysis usually involves modeling the acoustic phenomena in an acoustic fluid and in a structure.

The analysis starts from the 3D modeling of the channel (Figure 10). The implemented model includes:

- Series collector (the fluid volume between the collector casing and the external surface of the impeller disks);
- Long crossover channel;
- An axial symmetric volume obtained rotating the impeller meridional plane around the axis of rotation of the pump (this volume reproduces the vanes of the impeller at the ending of long channel);

The Acoustic Extension allows to perform an acoustic analysis on Ansys® Workbench through the Harmonic Response toolbox

(Figure 11). The 3D model is imported, and the analysis implemented on Mechanical of Ansys® Workbench (Figure 12).

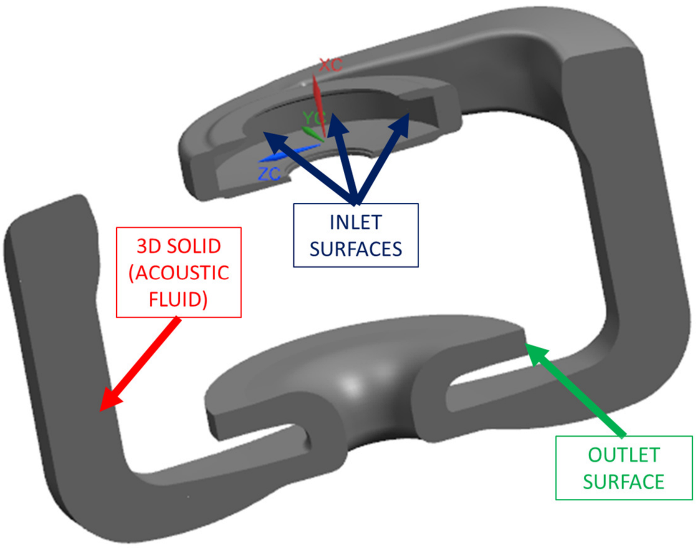

Figure 10 Section of the long channel 3D model in Ansys® Workbench

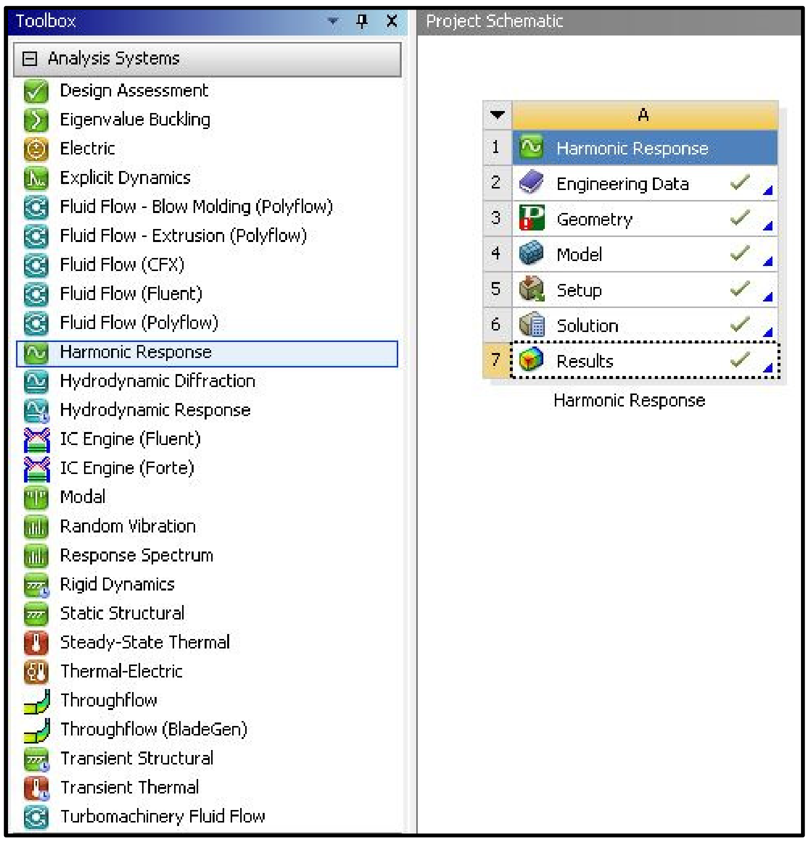

Figure 11 Harmonic Response Toolbox

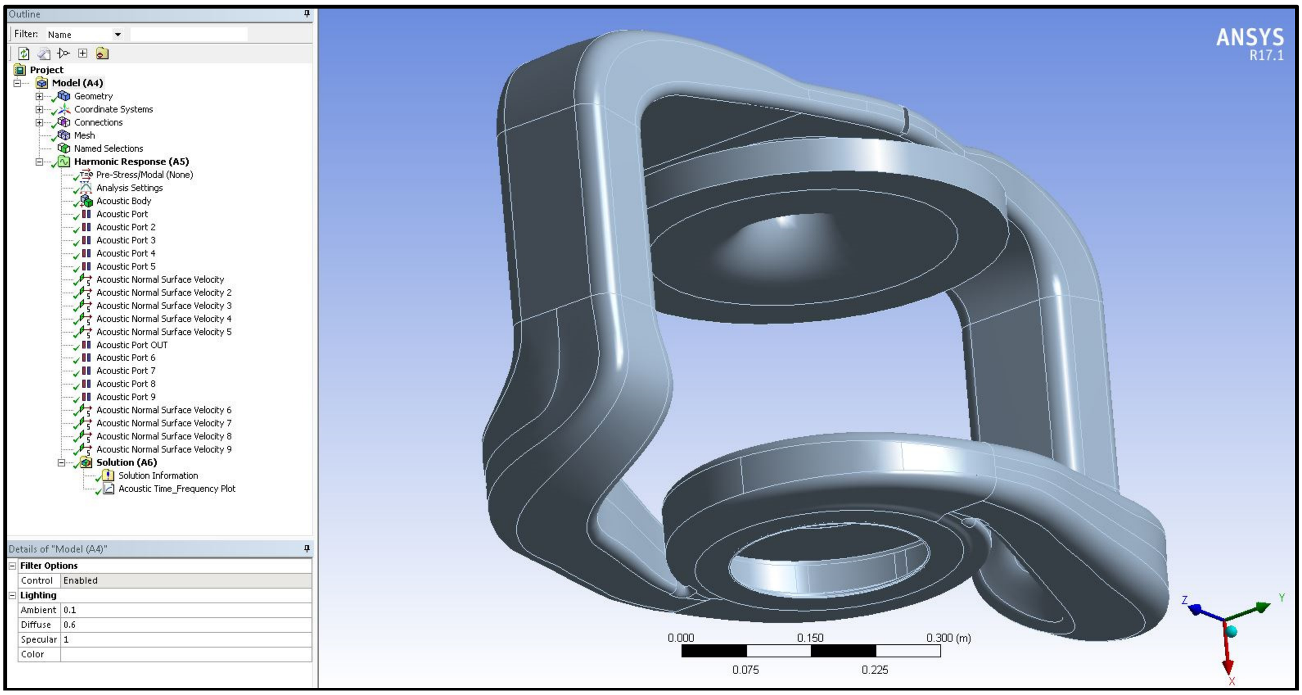

Figure 12 3D model of the long channel in Ansys® Workbench

In order to perform the analysis a series of object have to be created:

- Acoustic Body is the fluid volume on which the fluid properties, as density, sound speed, dynamic viscosity and compressibility, shall be defined (Figure 13);
- Acoustic Ports are the surfaces used to define the inlet and outlet of the fluid volume. There are as many surfaces as impeller vanes for the inlet and one for the outlet. The inlet areas are located at the exit surface of the first impeller, while the outlet is located at the impeller trailing edge of the rotating meridional plane;
- Acoustic Normal Surface Velocities are the input on the inlet surfaces. For a harmonic analysis, a complex normal velocity to the surface is defined by the amplitude and phase angle, so at the inlet ports the inputs have different phase in order to take into account the rotation of the impeller. The program solves for the pressure on the normal velocity excitation surface. Normal velocity excitation generates a wave propagating into the acoustic domain.

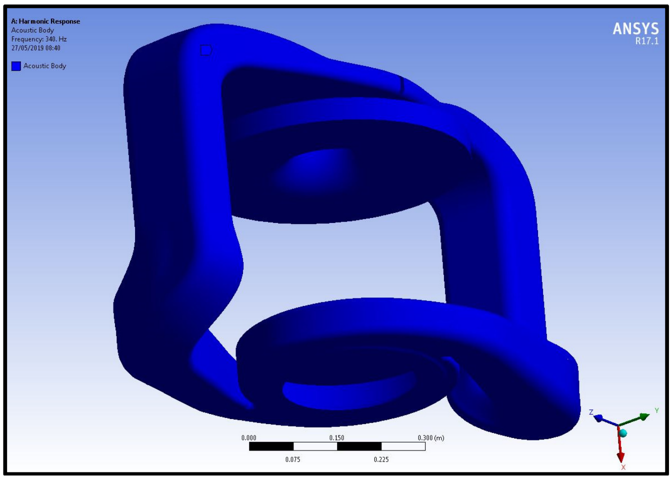

Figure 13 Acoustic Body

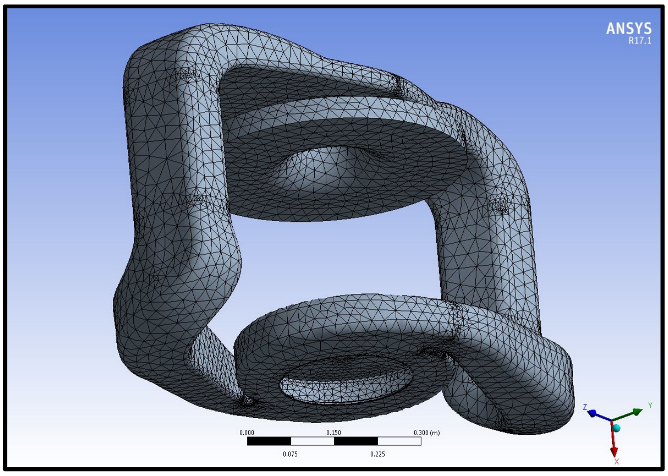

Figure 14 Mesh of the 3D model

The boundary conditions are:

- A series of inlet surface, corresponding to the outlet area of each impeller vanes upstream the long crossover channel, where the impulsive force applies (the number is equal to the number of impeller blades)
- An outlet surface, corresponding to the trailing edge of the impeller downstream the long crossover channel, to measure the frequency and the amplitude of the fluid answer.

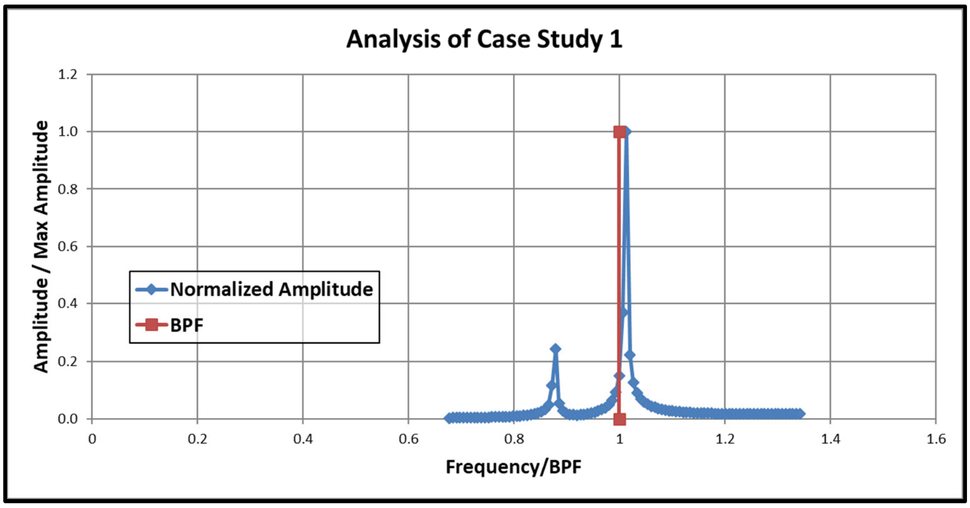

Figure 15 Comparison of Amplitude values between FEM analysis and test results

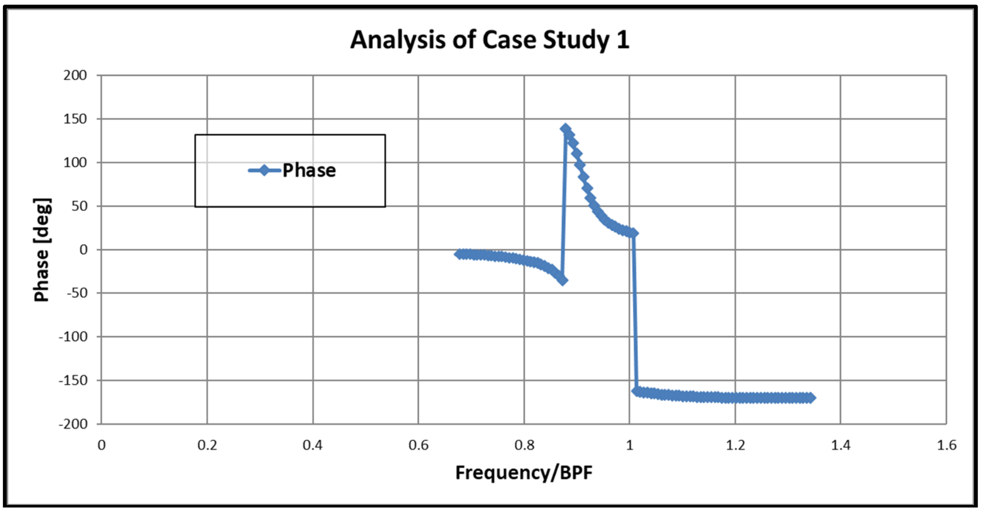

Figure 16 Comparison of Phase values between FEM analysis and test results

The mesh was built in a Ansys® WorkBench interface (Figure 14) with Adaptive as Size Function, Coarse as Relevance Center. The Element size is a function of the channel dimension and after a sensibility analysis was found that a good solution between numerical precision and solution speed is to use a ratio = 0.3 between Element Size and Max Impeller Diameter. The mesh had 250000 nodes and 170000 elements.
In the Harmonic Response Analysis of Ansys® Workbench the Acoustic Time Frequency plot has been used to check the behavior of the channel. A wide range of frequencies where imposed to verify the presence of acoustic resonance.
All the simulations were performed on a HP Z800 Powerstation, Quad-Core Intel Xeon Processor 5500 Series with Intel 64 Architecture, with 24 GB (6x4 GB) DDR3-1333 ECC Registered RAM 2-CPU and Windows 7. The simulations, running on parallel distribution with 8 processors, didn't need more than 1 hour.
The result of the analysis is that a peak of resonance appears at a frequency very close to the BPF, as shown in the Figure 15 (The

change of phase in Figure 16 also indicates that a resonance phenomenon is present).

CASE STUDY 2

After the identification of the new method of calculation and the verification of the effectiveness on the old project simulation, the last step was to prove its predictability by the simulations of all the pumps that would be tested in the following months at shop test bench. All the pumps to be later tested, were taken as a sample for calculation. In one case the acoustic resonance frequency was close to BFP, a BB5 (Double casing, radially split, between bearing pump) 4x11/7 stages.

The interception of the BPF with the resonance frequency has become necessary to validate the calculation. This intersection can occur in two possible ways:

1. varying the rotation speed and consequently the BFP;
2. varying the resonance frequency by modifying the sound speed of the fluid.

The first option was not achievable as the complete train, (i.e. pump + coupling + motor), was at fixed speed. The second option was implemented by varying the water temperature. This was feasible because the test bench was a closed circuit with a cooling system that allows to change the fluid temperature in a limited range.
Several simulations have been completed with different properties of the fluid (i.e.water) with rigid wall (Table 2):

Temperature [°C]	Pressure [bar]	Density [kg/m3]	Sound Speed [m/s]
30	30	996.94	1514.1
40	30	993.48	1534
50	30	989.3	1547.8
60	30	984.46	1556.3
70	30	979.04	1560.3

Table 2 Sound speed and density variation with temperature [7]

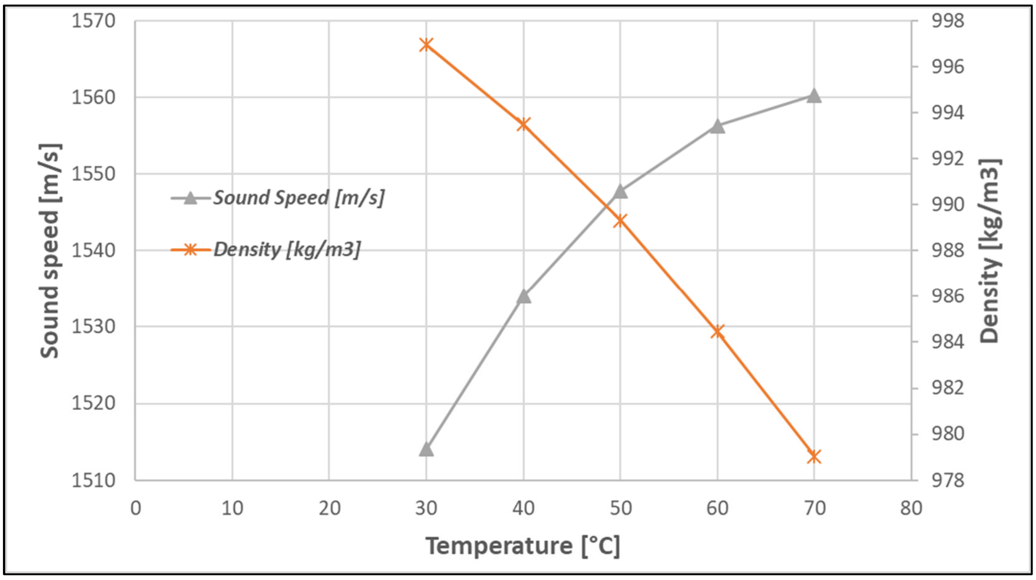

Figure 17 Sound speed and density variation with temperature @ 30 bar (6)

From the simulations it was found that as the temperature T increased, the resonance frequency in the long crossover channel approached the BPF. With a process temperature of about 60 ° C, the acoustic resonance overlapped the BFP.
After the simulations were completed, there remained the unknown factor of the damping of the system that could make the resonance

appreciable. Moreover, since it was a BB5, (i.e. a barrel pump), it was not possible to measure the pressure in the long channel with dynamic transducers, but only with the vibration probes on the bearings. Anyway, it was the only chance to verify the occurrence of the resonance issue and validate the new calculation sensitivity.
The BB5/7 stages pump was tested in two test bench configurations:

- Twater = 25°C
- Twater = 60°C

The resonance phenomenon was highlighted with the variation of the vibration levels. The results of the test were consistent with what was observed in the simulations, as shown in the graphs (Figure 18), where AR is the Acoustic Resonance frequency relative to BPF and Amplitude is vibration amplitude.

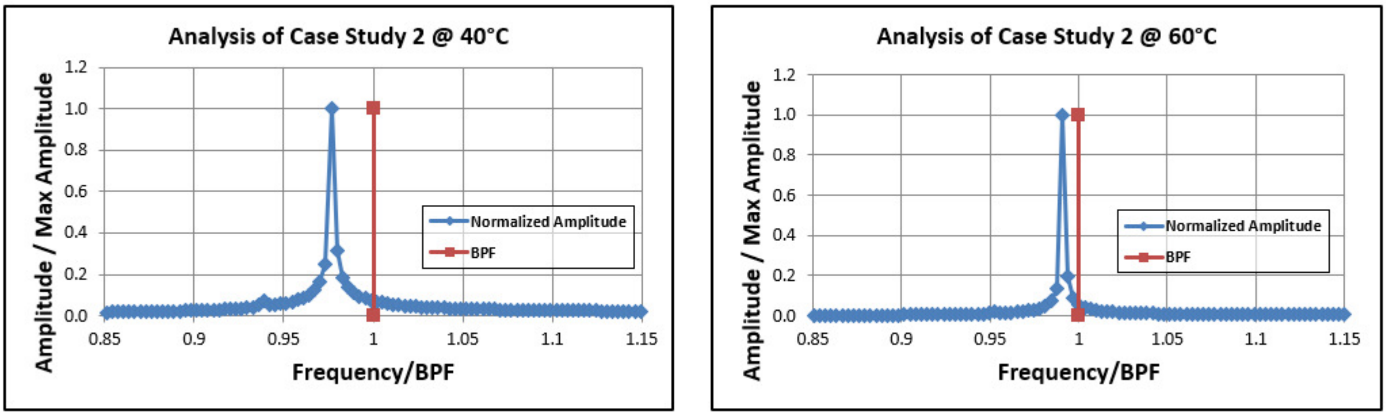

Figure 18 AR (Acoustic Resonance) approaching to BPF with the temperature increasing

VALIDATION OF NEW METHOD OF ANALYSIS

In order to have a more complete validation of this method, the analysis of older pumps with vibrations data reports available where performed. From the historical database of pump tests, in particular for multistage pumps with a 3D CAD model available, the most complete reports were selected. For each of these pumps a simulation was performed to compare the experimental results to the numerical analysis, considering water at 25°C as process fluid, coherently with the test configurations.

	Testing issue	Analysis issue	API pump	Peak Frequency / BPF	Speed [rpm]	Nr of blades	L3/L2 %	L/Deq
Test 3	NO	NO	BB3/7 Stages	1.035	2915	7	6	61
Test 4	NO	NO	BB3/4 Stages	1.427	1490	7	14	27
Test 5	NO	NO	BB3/10 Stages	0.861	5700	6	0.1	110
Test 6	YES	YES	BB3/6 Stages	0.972	2980	7	12	34
Test 7	NO	NO	BB3/6 Stages	0.893	2985	7	7	50
Test 8	NO	NO	BB3/4 Stages	1.273	1490	8	12	30

Table 3 Summary of analysis

The pumps analyzed and listed in Table 3 were BB3 (i.e. multistage pump, axially split, back to back impellers) with different operating speed, number of stages, and number of impeller blades. In the Figure 19 Figure 20 are shown the typical cross sections of a 4 and 7 stages pump, with highlighted in blue the long channel.

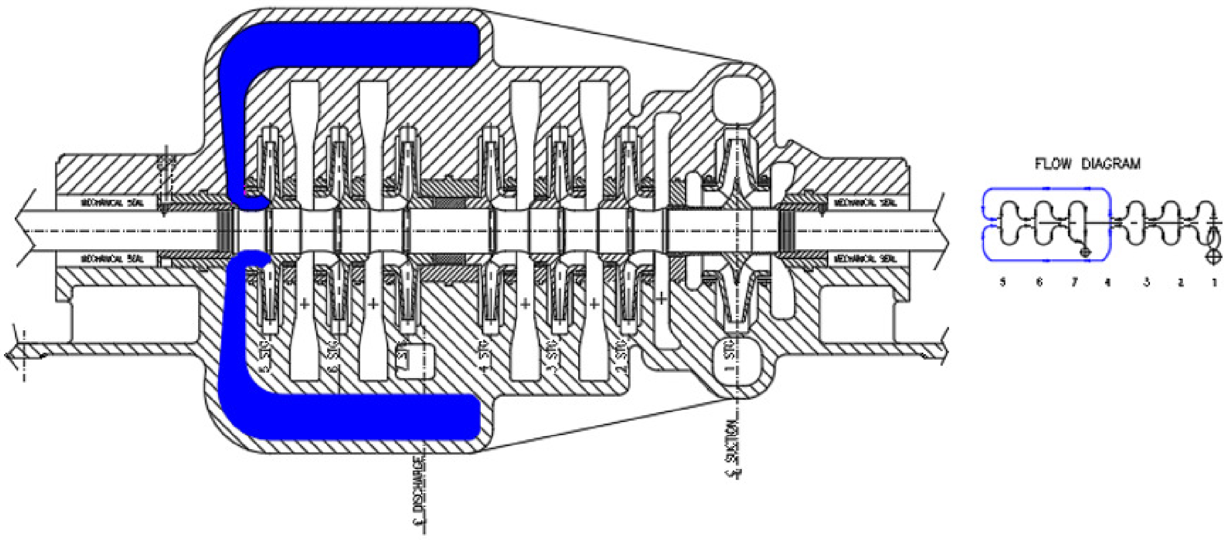

Figure 19 Cross Section of 7 Stage Pump (Test 3)

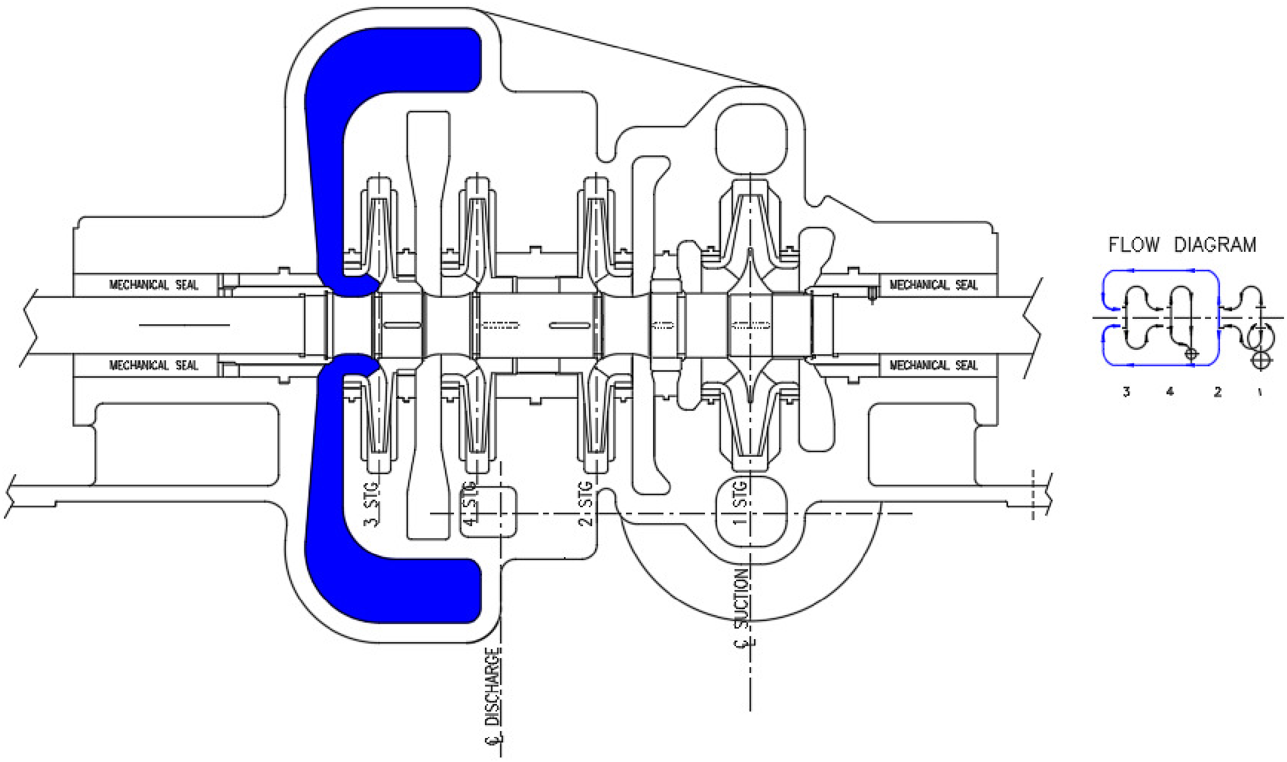

Figure 20 Cross Section of 4 stage pump (Test 8)

Moreover, as shown in Table 3, the percentage ratio L3/L2 % between the equivalent length, defined with both 2D and 3D methods, have been analyzed. It has been noted that the greater the L3/L2%, the lower the L/Deq ratio. This ratio has been defined with the dimensions shown in the Figure 21 .

Where:

Deq is the diameter of a circular section with area A (Figure 21), calculated as $\sqrt{\frac{4\times A}{\pi}}$;

L2 is the equivalent length calculated using the 1-D method;
L3 is the equivalent length calculated using the 3-D method;
L is the interstage channel length.

The difference between the L2 and L3 causes the incorrect estimation of the acoustic resonance with the 1-D method.

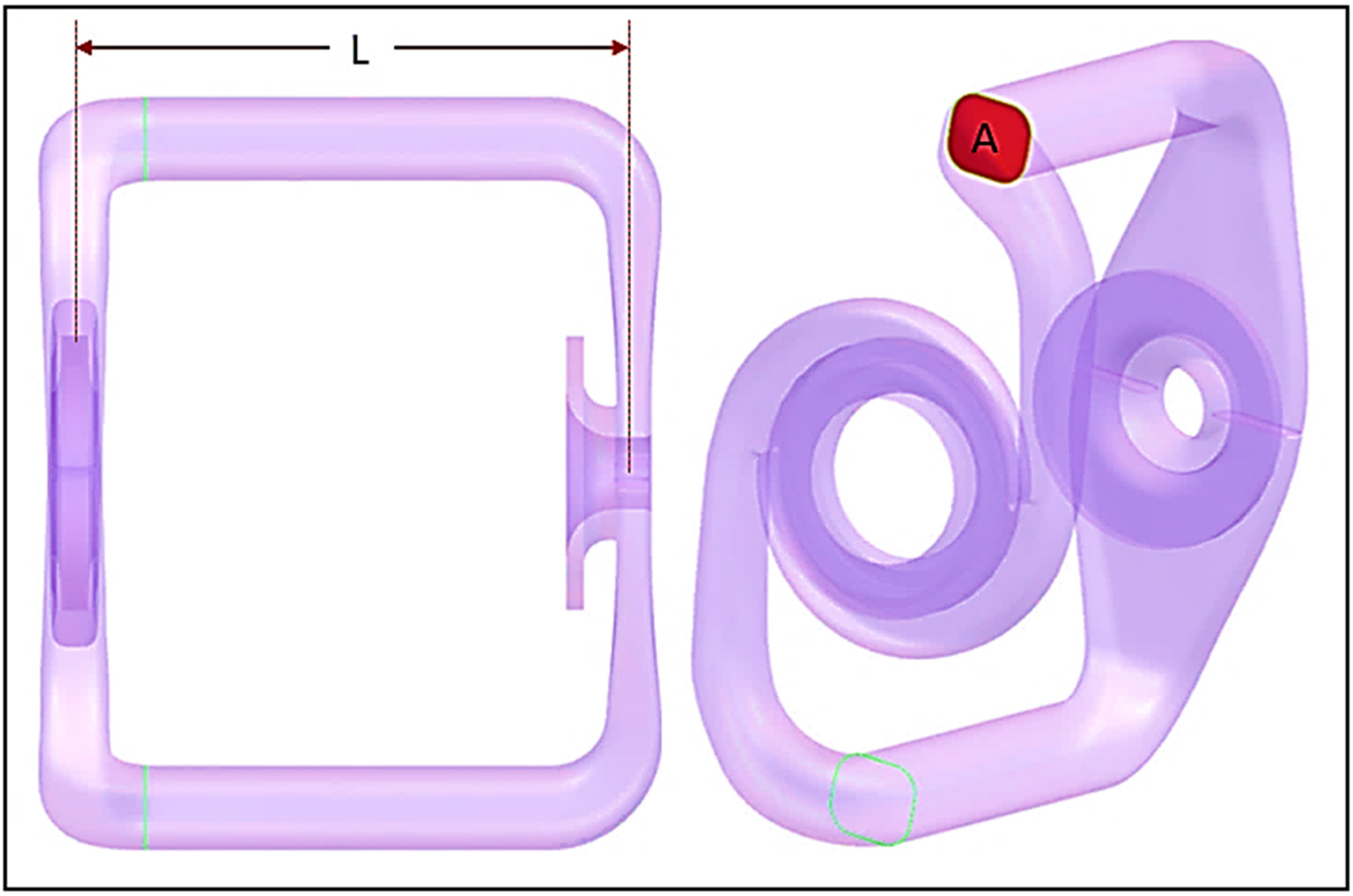

Figure 21 Long Channel Dimensions

In Figure 22 and Figure 23 the velocity spectrum plots (values measured in mms/s) for the Test 3 (i.e. BB3 pump, 7 stages) at BEP and MCSF respectively are shown. During the performance test, the acquisition of vibration shows a peak (lower than the API 610 threshold limits and for this reason acceptable) of vibration at 7X (all the impellers have 7 blades) at every flow with quite similar values indicating that the acoustic resonance phenomenon is present.

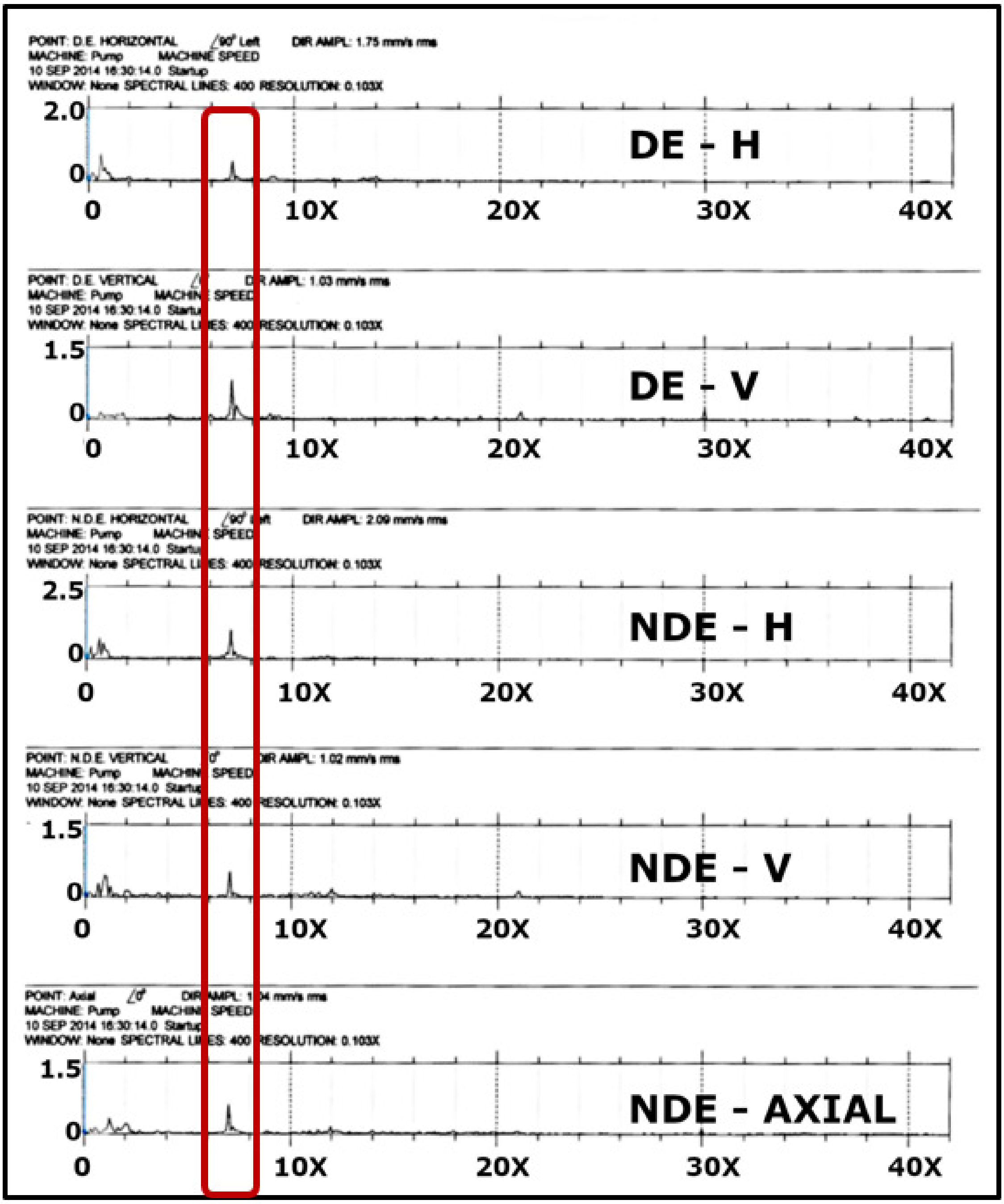

Figure 22 Spectral lines at BEP in each controlled direction (Test 3)

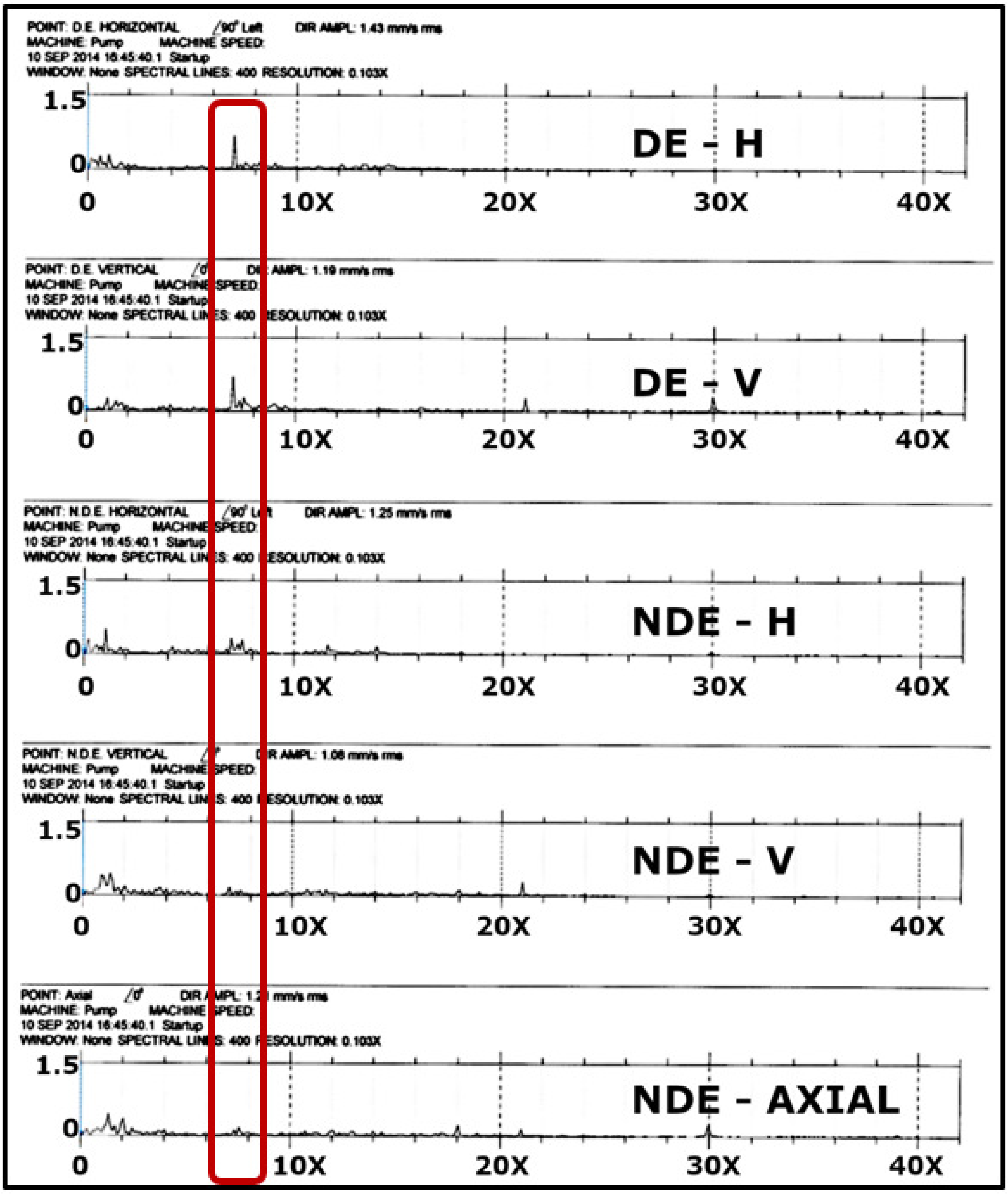

Figure 23 Spectral lines at MCSF (50% of BEP) for each controlled direction (Test3)

The numerical analysis predicts a possible acoustic resonance for the Test 3 (Figure 24 a), even if the experimental test doesn't show any vibration above the limits, probably for a damping effect. The ratio between the BFP and the peak frequency obtained by the numerical analysis is less than 5% that is a margin acceptable to consider the acoustic resonance close to the operating conditions.
For Test pump 4 (Figure 24 b) the analysis said that the AR phenomenon was quite far from the operating condition. This result is coherent with the vibration level measured in test bench.

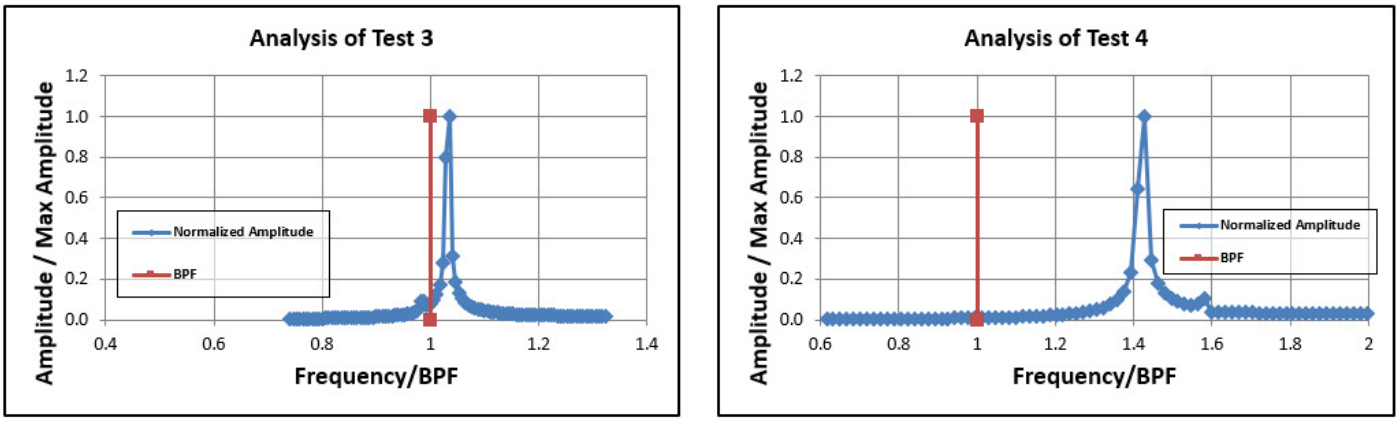

Figure 24 Amplitude diagram for Test 3 (a) and Test 4 (b)

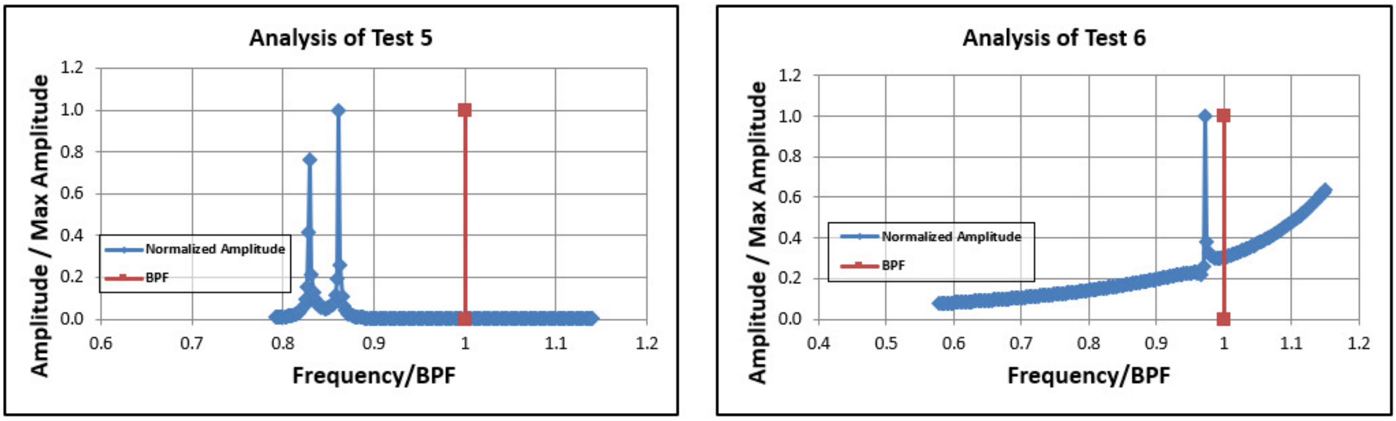

Figure 25 Amplitude diagram for Test 5 (a) and Test 6 (b)

The analysis predicted for Test 6 (Figure 25 b) a peak of acoustic resonance. During the test of the pump, a waterfall diagram (Figure 26) was performed. At line 7X (7 is the number of blades for all the impellers and at a frequency close to 350Hz (7 x 50Hz) a peak (in red) is shown, as predicted by the FEM analysis.

In Figure 27 a and b, the last simulations are shown. For Test 7 and Test 8 no resonance was observed during the experimental tests and the analysis results shown an Acoustic Resonance peak far from the BPF.

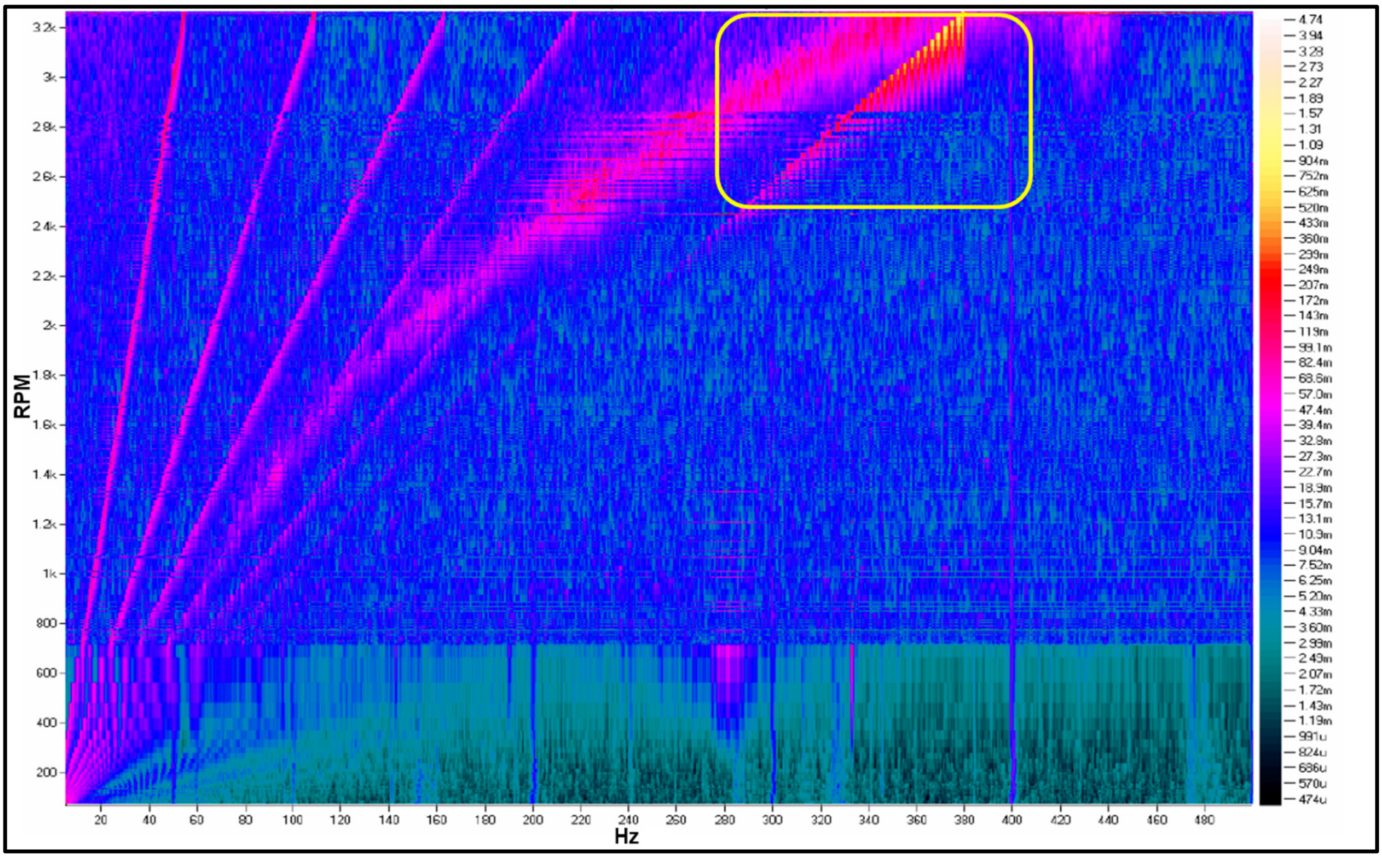

Figure 26 Waterfall diagram for Test 6

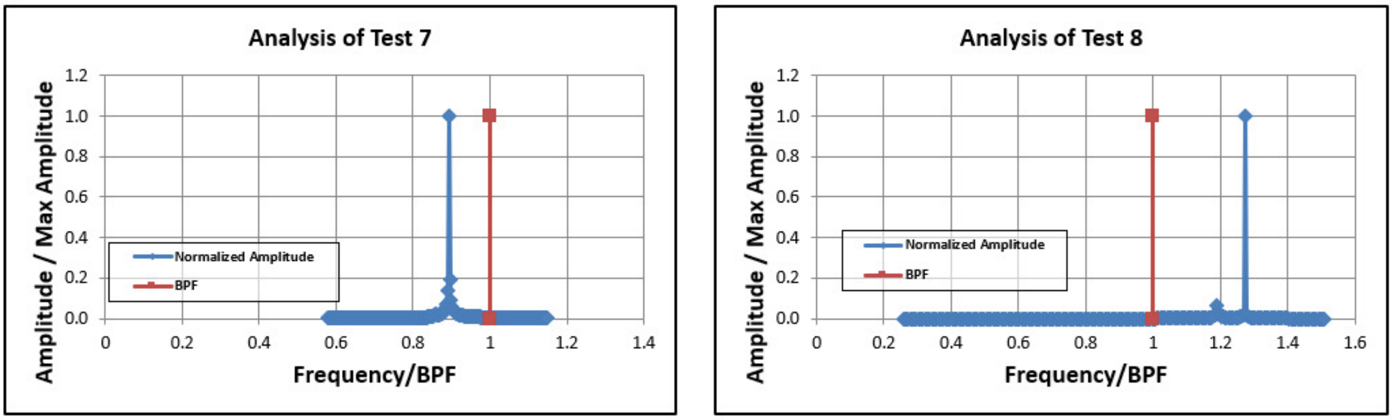

Figure 27 Amplitude diagram for Test 7 (a) and Test 8 (b)

CONCLUSIONS

An anticipation of a possible issue on field, not intercepted during the development phase of a new machine, provided the opportunity to improve the design procedure. The need of the market to design and operate centrifugal pumps in a range more distant from the comfort zone of the designers has shown the limits of a 1D tool suitable for standard machines.

It was therefore necessary to develop a new method of calculation and prediction of potential acoustic resonance that took into account the even more complex hydraulic channels that could not be approximate with a simple 1D geometry.

Once the method was developed and the predictability on the pump was checked, a campaign of simulations and indirect comparisons with the bearing vibration data was implemented to confirm and verify that the reliability of the new procedure was greater and suitable for machines with different Ns, speeds, sizes and number of stages.
Analyzing the different simulations, it has been noticed that the error in length calculation increases when the L/Deq decreases. In particular, for larger L/Deq values, the long channel can be compared to a straight pipe, while for smaller values, the three-dimensional reflection effects become significant.
A fundamental factor in the development of this new method is the increase in computational capacity. The procedure moved from a simple spreadsheet where it was possible to implement a simple one-dimensional geometry to a FEM calculation that represented the real geometry in all its three-dimensional complexity.
The comparison between the simulations and the tests of old pumps performed in the test bench of which the vibration values were known, has shown the excellent predictability of the new calculation, which was then included in the design procedure.

NOMENCLATURE

API – American Petroleum Institute
AR – Acoustic Resonance
BEP – Best Efficiency Point
BFP – Blade Passage Frequency
CAD – Computer Aided Drafting
Deq – Equivalent hydraulic diameter of long channel section
DE-H – Drive End Horizontal
DE-V – Drive End Vertical
FEM – Finite Element Method
L – Interstage length
L2 – 1D equivalent long channel length
L3 – 3D equivalent long channel length
MCSF – Minimum Continuous Stable Flow
NDE-H – Not Drive End Horizontal
NDE-V – Not Drive End Vertical
NS – Specific speed

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