

COUPLED USE OF FEA AND EMA FOR THE INVESTIGATION OF DYNAMIC BEHAVIOUR OF AN INJECTION PUMP

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SUMMARY

A major concern for pump manufacturers is to guarantee certain limits of vibration velocity measured at the bearing housings of their pumps, e.g. 3 mm/s. For vane passing excitation the frequencies of interest depend on the rotational speed of the rotor and the number of impeller vanes, and are in general between 250 and 700 Hz (250 Hz at 3000 rpm and 5 vanes, and 700 Hz at 6000 rpm and 7 vanes). When a bearing housing has natural frequencies in this frequency range, they can easily be excited by resonance during operation and lead to vibration velocities above the allowable limits. A verified simulation model can be used for predictive and preventive analyses. As part of quality assurance, Sulzer Pumps has performed analytical and experimental investigations for an injection pump, focusing on the dynamic behaviour of the bearing housing. The output of this work constituted the input for the project presented here.

With the aim of quantifying the compliance of calculated natural frequencies and corresponding mode shapes with their measured counterparts, a series of correlation and sensitivity analyses has been performed by Sulzer Innotec using the specific tool. It was attempted, to update a finite element model of the whole pump using the algorithms integrated in the correlation software, selecting significant parameters and specifying their uncertainties.

First complex and then real modes have been extracted from operational displacement shapes using conversion features of the correlation software before further treatment. Due to the fact that experimental results were based on a modest number of measurement points only, their extrapolation was indispensable as the next step. The correlation between EMA & FEA has been visualised in form of MAC (modal assurance criterion) matrices. The iterative updating procedure has brought remarkable enhancement of the FE model, even for a limited number of modes. In particular, the agreement obtained for FEA/EMA at the eigenfrequency of 310 Hz proves to be practically sufficient. The step-by-step modification of the parameters (Young's modulus, density and spring stiffness for element groups) performed thereby can be viewed here.

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1: Introduction

Sulzer Pumps is a Swiss company manufacturing the world's largest injection pumps. To guarantee certain limits of vibration velocity (e.g. 3 mm/s), measured at the bearing housings of the pumps, is a major concern for pump manufacturers. Also for Sulzer Pumps Ltd, reliable predictions of operational vibrations are part of the quality assurance.

The so-called AIOC-pumps (Figs. 1 and 5) are compact injection pumps in production in Leeds, UK, for the Azerbaijan International Oil Company. Four of them, each one with a mass of 19'800 kg and a rated power at 27MW, will have the combined capability to inject up to 1.0 million barrels of sea water per day into the reservoirs at a pressure of 480 bar. The pumps, which are driven by Rolls-Royce RB211 gas turbines at a speed of 4800 rpm, are 50% greater in power than any other sea water injection pump ever built. Understandably, these pumps are products, the dynamic behaviour of which has to be investigated in detail prior to delivery.

As a matter of fact, for vane passing excitation, the frequencies of interest depend on the rotational speed of the rotor and the number of impeller vanes. In general, they are in the range between 250 and 700 Hz (250 Hz at 3000 rpm and 5 vanes, and 700 Hz at 6000 rpm and 7 vanes). When a bearing housing has natural frequencies in this frequency range, they can easily be excited by resonance during operation and lead to vibration velocities at the bearing housings above the allowable limits. A Finite Element (FE) model can be used for predictive and preventive analyses. However, the pumps are highly complex structures and it is indispensable, to prove the reliability of the predictions based on such simulations by means of state-of-the-art verification tools and methods.

Within the framework of a former project, which involved experimental modal analysis (EMA) and extensive computational parameter studies, Sulzer Innotec (SIT) had evaluated possibilities of simplifications during the modelling process, according to the motto «Keep it as simple as possible, but as complete as necessary!», and developed technical guidelines showing how to model bolted connections, seals, gaskets, etc.

Sulzer Pumps Ltd (S-PU) has applied these recommendations in the FE model of the AIOC injection pump and performed preliminary analytical calculations focusing on the dynamic behaviour of the bearing housings. On the other hand, S-PU has arranged experimental investigations (EMA on body in white of the pump) aiming to verify the predictions deduced from these simulations. SIT has many years experience in combined use of test and analysis for dynamic

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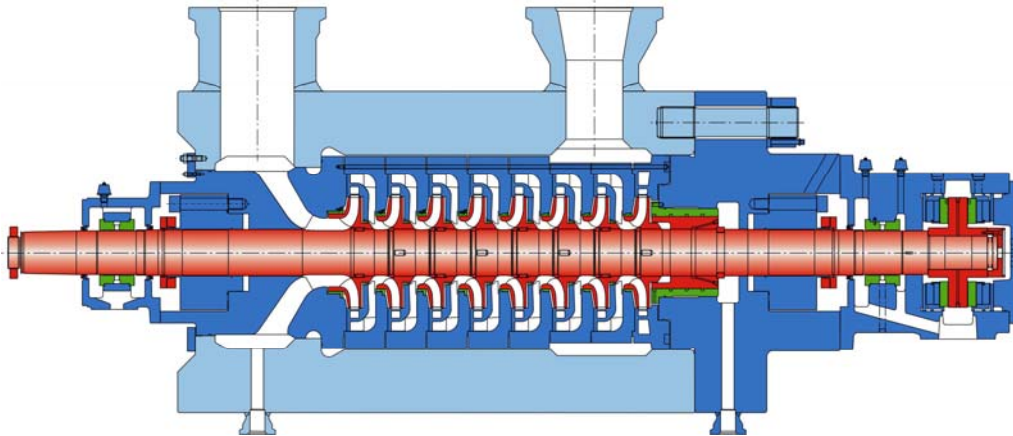


Figure 1: Longitudinal cross section of the AIOC sea water injection pump.

investigations and access to adequate software tools enabling quantifiable correlation between EMA and FEA results and having various FE model updating features.

Based on sensitivity and optimisation algorithms, the FE models can be modified using measured modal data as reference and selecting significant material- and stiffness-relevant parameters (density, modulus of elasticity, spring coefficient, thickness, etc.) as modifiable quantities for element sets or individual elements. The range of possible changes must be defined for selected parameters.

With this background SIT was asked by S-PU to evaluate the compliance EMA-FEA of a typical AIOC pump and perform a reasonable FE model updating using the specific software FEMtools. The step by step approach and selected results of this work are presented and discussed in the following.

2: Original FE model / Preliminary analysis

An ANSYS finite element model of the AIOC barrel pump HPcp 350-425/8s, originally created for design analysis purposes, could be used also for eigen-frequency analysis. This model (Fig. 2) has been built according to the rules which were worked out within the framework of a former parameter study. It consists of 107'000 elements with 173'000 nodes. The geometry is meshed with 10-node-tetra-hedron elements. Additionally, the model contains beam elements, simulating the bolts of the clamping pieces at the drive end, and springs. These are located at the following positions:

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- a) At the four feet, with different stiffness coefficients in three principal directions. At each foot the same stiffness values are applied, which were obtained by the model of a base plate and have to be adjusted by tests on the real pump and by correlation analyses.
- b) Between suction casing and barrel at the internal shoulder. The cartridge is tied against this shoulder by the bolts of the clamping pieces. The contact pressure at the shoulder is increased during operation due to internal pressure. The stiffness values are different in the normal and tangential direction and have to be adjusted by tests on the real pump.
- c) Between the last stage casing and the cover. There is a loose fit in between these parts, but it is radially guided. Consequently, only a normal stiffness is applied here, and no tangential. Also these stiffness coefficients have to be adjusted upon test results obtained on the real pump.

Whilst the first modes show rigid body motions of the pump, as expected, the modes at about 304 Hz and 376 Hz are the interesting eigenmodes related to the DE and NDE bearing housings. The corresponding mode shapes are shown in Figs. 3 and 4.

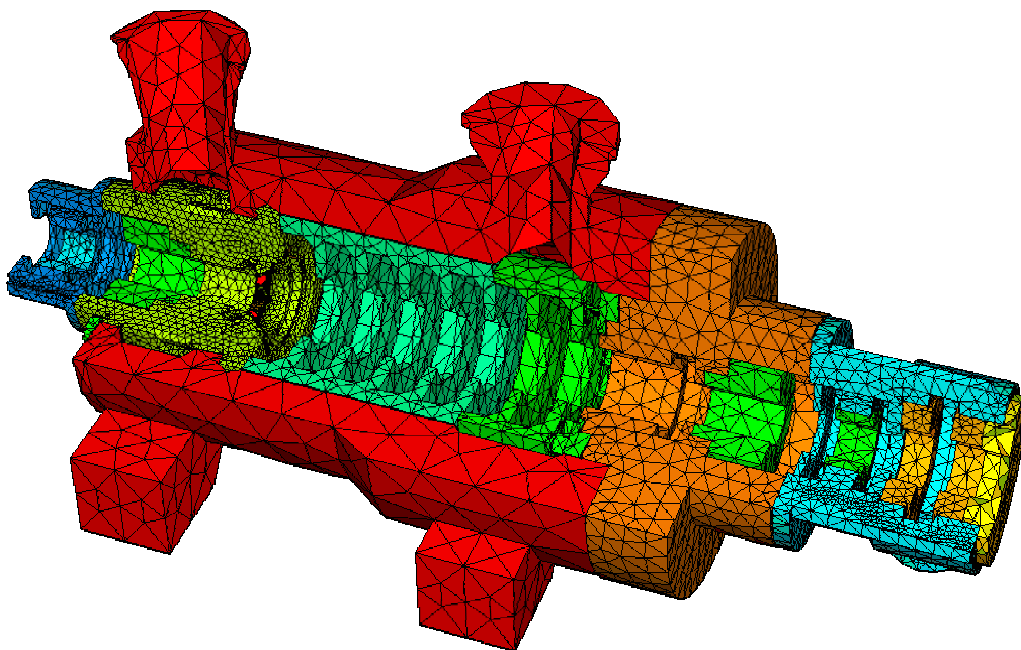


Figure 2: Section view of the FE model and its partitioning as preparation for the subsequent updating procedure. NDE (non drive end) of the pump is on the right hand side, DE (drive end) on the left.

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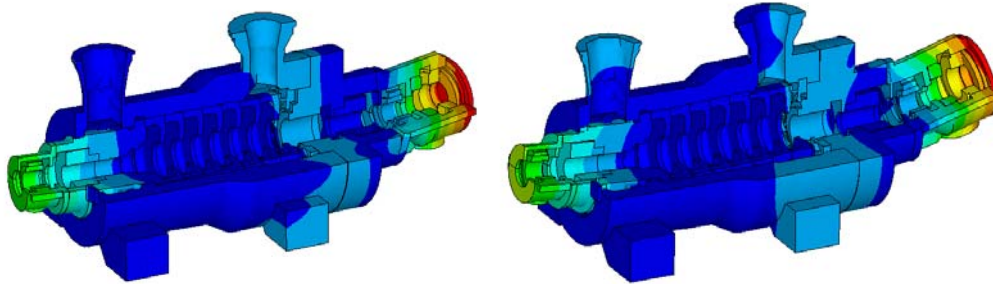


Figure 3: Horizontal and vertical NDE modes (both ends of the pump in phase), at 304.8 and 304.1 Hz respectively.

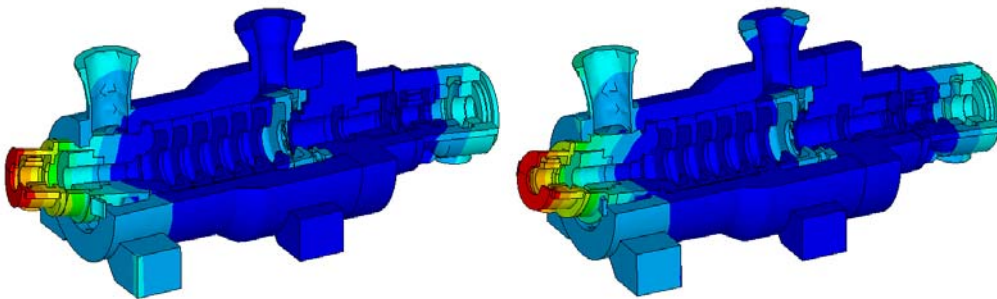


Figure 4: Horizontal and vertical DE modes (both ends of the pump out of phase), at 376.0 and 375.6 Hz respectively.

3: Experimental modal analysis

The experimental modal analysis has been performed at the pump mounted on its base plate. A number of fifteen tri-axial accelerometers were placed on the outer surface of the pump. In total 8 pairs of acceleration inputs in lateral and vertical directions were gathered at selected positions on DE and NDE bearing housings, inboard and outboard, top and side. Additional data have been measured at the DE and NDE pump feet, left and right hand sides each, on the sides of suction and discharge nozzles (the latter also in the axial direction of the pump) as well as in the vicinity of the excitation position in the middle of the pump. The instrumented hammer with a mass of 1.4 kg could just ensure the impact energy, needed to induce the random excitation covering the frequency domain of interest. The data acquisition system consisted of a 17-channel B&K signal conditioning unit and a notebook PC with the appropriate software. From the measurement data potential eigenfrequencies have been extracted later on, and corresponding operational deflection shapes "ODS". These shapes can be considered as acceptable alternatives to eigenforms, if they don't influence each other too much. A typical result is shown in Fig. 6.

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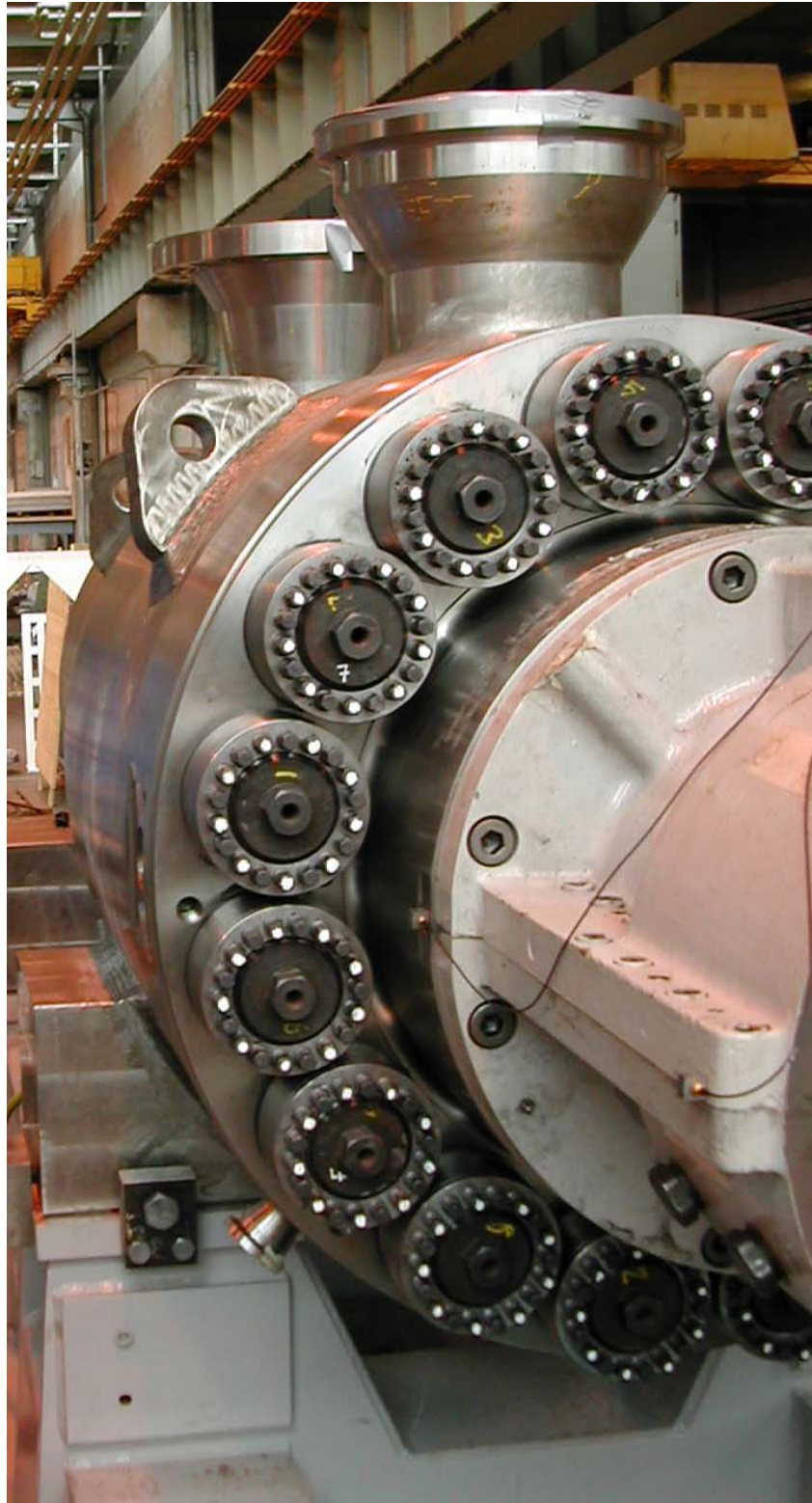


Figure 5: Side view of the AIOC pump ready for modal test. The modelling of the bolts, seals, bearings, flanges, etc. are challenging details for the analyst.

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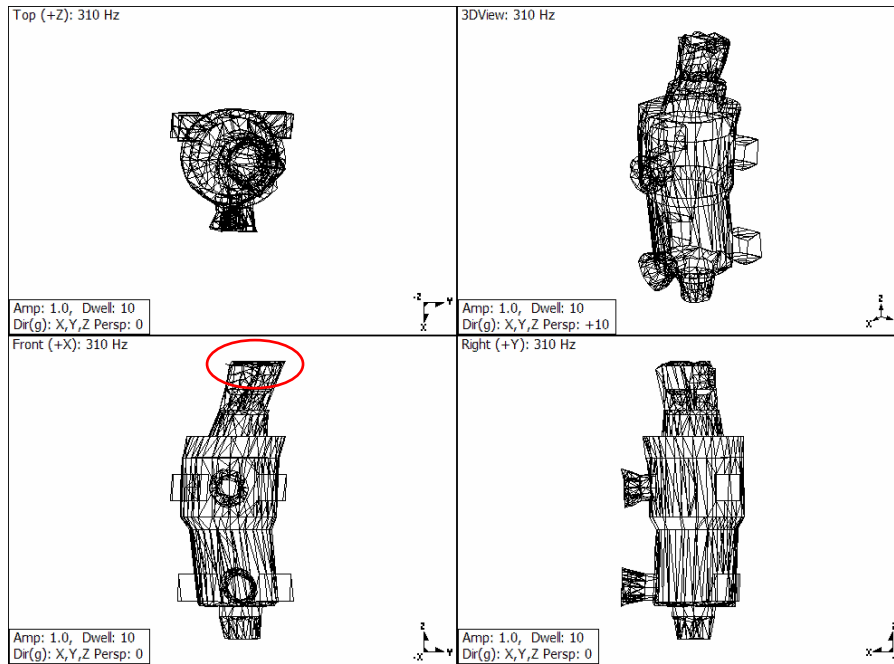


Figure 6: Typical result of the EMA: Eigenmode at 310 Hz, dominated by bending of NDE bearing housing. As shown clearly from the deformations, the lack of measurements in axial direction of the pump leads to a rather unrealistic representation of the mode shape.

4: Correlation and sensitivity analyses

The correlation analysis is performed by means of the software FEMtools, which can read in finite element models as well as the modal test data [1]. It is important to have the same coordinate system and origin, this was guaranteed by taking the finite element geometry into account already in the operations within the data acquisition system. The measurement grid and the eigenfrequencies with corresponding mode shapes in form of ODS were imported into FEMtools in universal file format. The original ANSYS FE model and the eigenmodes, obtained with 10-node-tetrahedron elements, could be read into the software also and used for correlation purposes. The comparison between test and analysis results is usually quantified by the so-called modal assurance criterion "MAC", which is defined by the eigenvectors of modal test and analysis for each identified pair of analytical and test mode shape [2]:

$$MAC_{ij} = \frac{\left(\left(\phi_t^T \right)_i \cdot \left(\phi_a^T \right)_j \right)^2}{\left(\phi_t^T \cdot \phi_t \right)_i \cdot \left(\phi_a^T \cdot \phi_a \right)_j}$$

ϕ_a represents here the analytical mode shape vector, reduced to the test degrees of freedom, and ϕ_t the test mode shape vector. The correlation between ANSYS FE model and the modal test data is shown in Fig. 7. The identifi-

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cation of node and measurement point pairs, which is essential for any correlation analysis, is visualised in Fig. 8 based on the extended test grid.

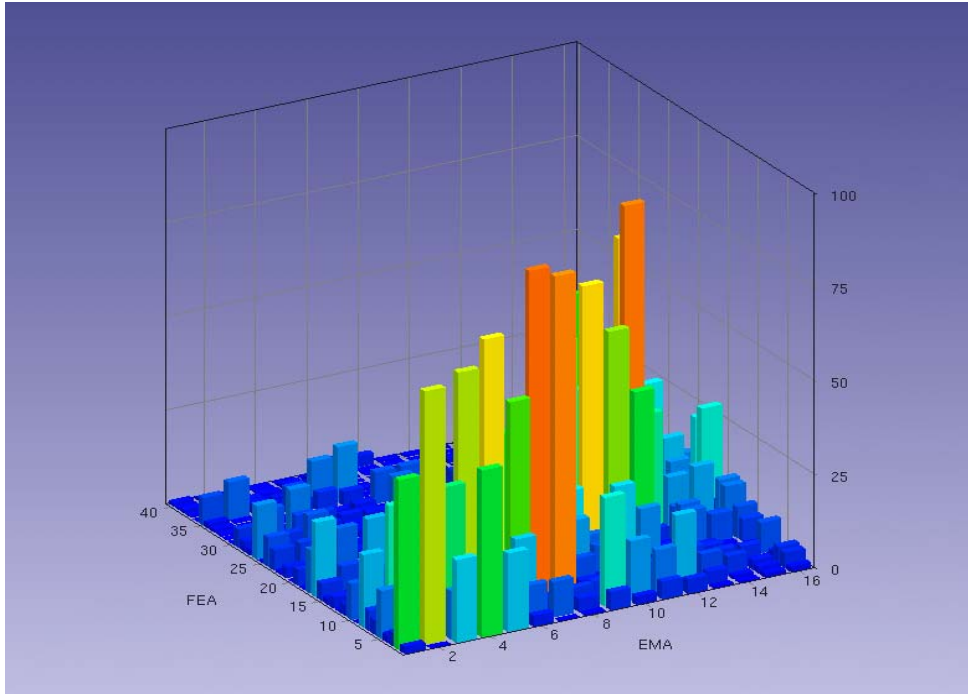


Figure 7: Correlation analysis: Evaluation of the MAC matrix obtained for test data and original ANSYS results.

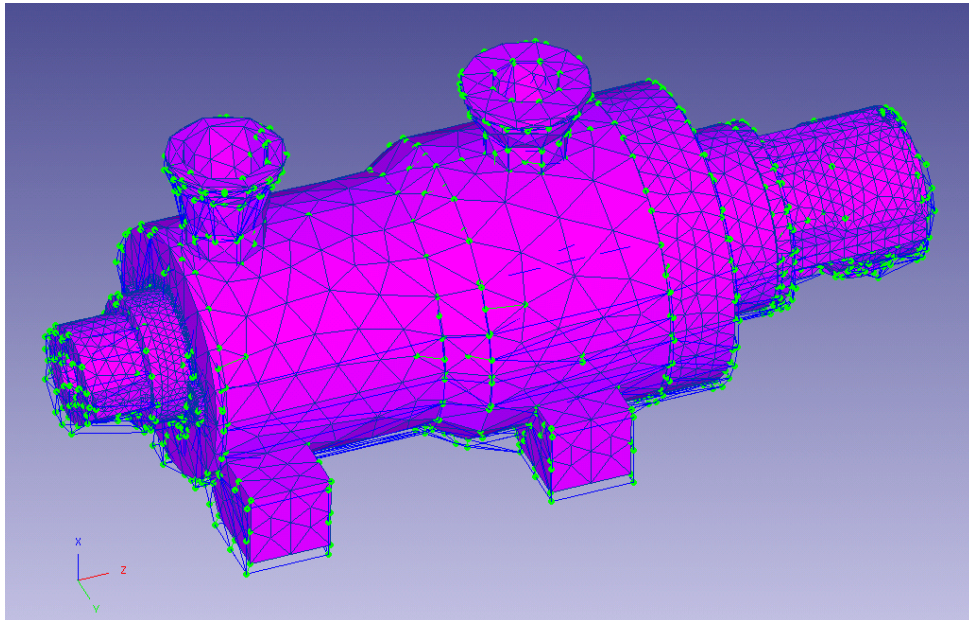


Figure 8: Pairs of the grid points used for the representation of measurement data and their counterparts i.e. corresponding nodes of the FE model.

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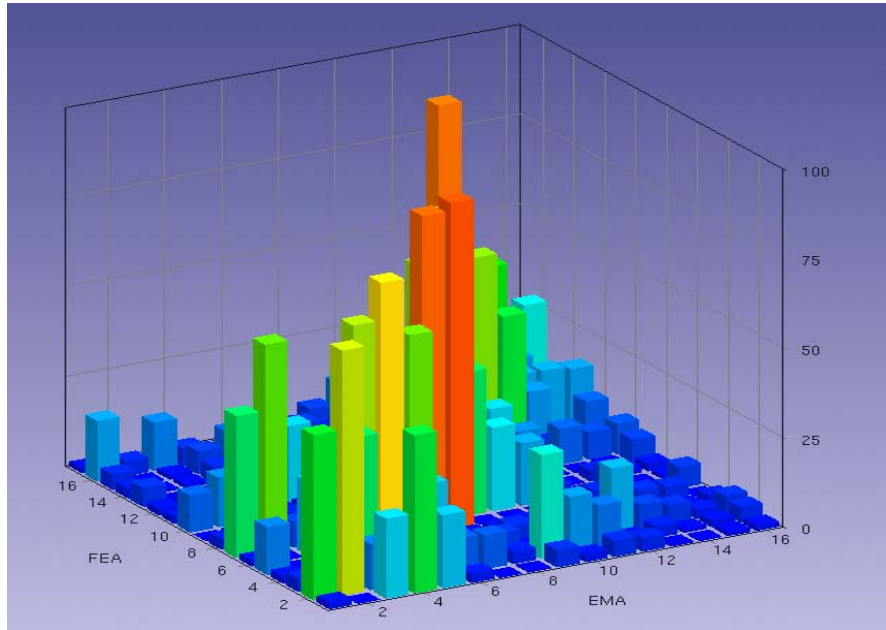


Figure 9: MAX matrix obtained with ABAQUS FE model consisting of 4-node-tetrahedron elements before updating procedure.

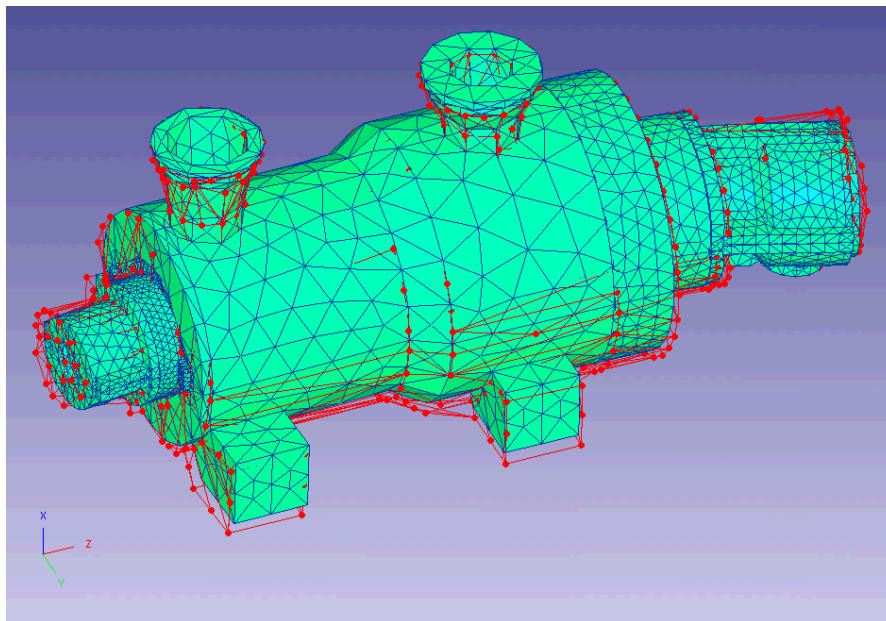


Figure 10: EMA-FEA pairing of the NDE bending mode (Test: 310 Hz versus analysis using FEMtools solver: 344 Hz) before updating procedure.

Due to the large size of the original FE model it was not possible to use the solver integrated in FEMtools for sensitivity and updating algorithms within the present version. To overcome this limitation the FE model has been trans-

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formed in an ABAQUS model replacing concurrently the 10-node-tetrahedrons by 4-noded ones. The MAC matrix and a representative mode shape pair obtained with this converted model are shown in Figs. 9 and 10 respectively. Naturally, most of the analytical eigenfrequencies were shifted about 10 % and even more with respect to ANSYS results due to the artificial stiffness increase caused by the different element formulations. However, the overall practically acceptable compliance between a certain number of EMA-FEA-mode pairs, necessary for any attempt of updating, is obviously still given.

In a next step the sensitivities of the significant eigenfrequencies (e.g. 310 Hz) to selected parameters of the FE model have been investigated. The pure bearing housing modes have been found to be not much sensitive on base plate stiffness. On the other hand, the modes, where the whole pump is involved, are quite sensitive on this parameter. The Young's modulus and density of cast iron and steel, which can vary in reality about 5 – 15 %, and stiffness coefficients of internal springs representing bolted interfaces between different parts of the pump, which exhibit inherently uncertainties up to 20 – 30 %, have been selected as reasonable parameters with some optimisation potential. They will be used within the subsequent step by step updating procedure.

As an example of sensitivity analysis results, the sensitivity distribution of the NDE bearing housing bending mode at 310 Hz to the modulus of elasticity is plotted in Fig. 11.

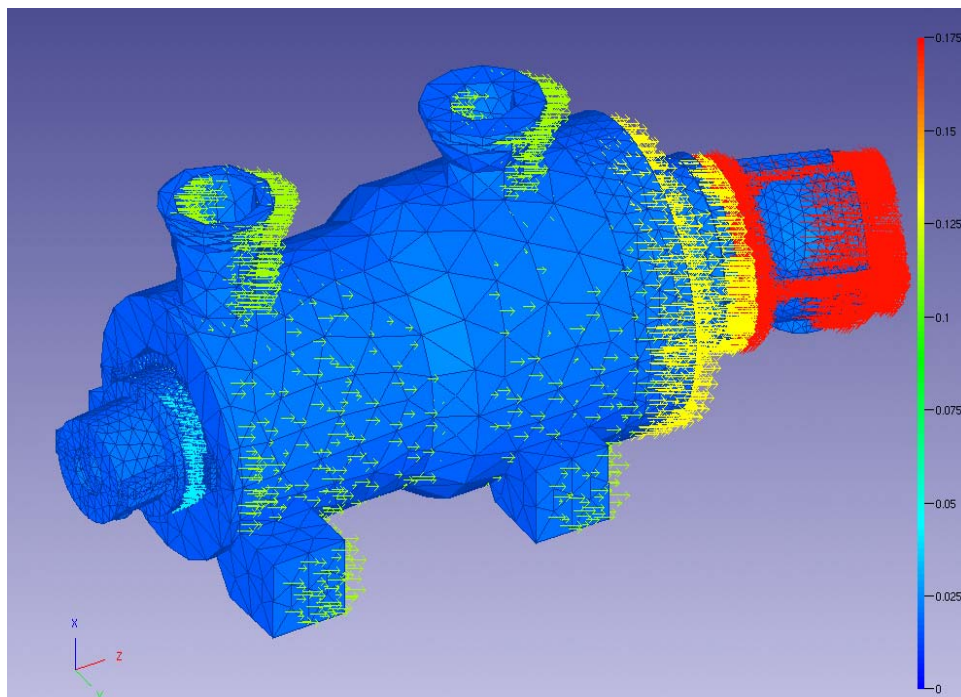


Figure 11: Results of the sensitivity analysis for the Young modulus related to the eigenfrequency of 310 Hz.

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5: Updating of the FE model / Results

The results of the iterative updating procedure, based on the parameters mentioned in the previous chapter, which was performed focusing on the DE and NDE bearing housing modes, are summarised in the following (Figs. 12 and 13). Evaluation of MAC matrix after the last updating is shown in Table 1.

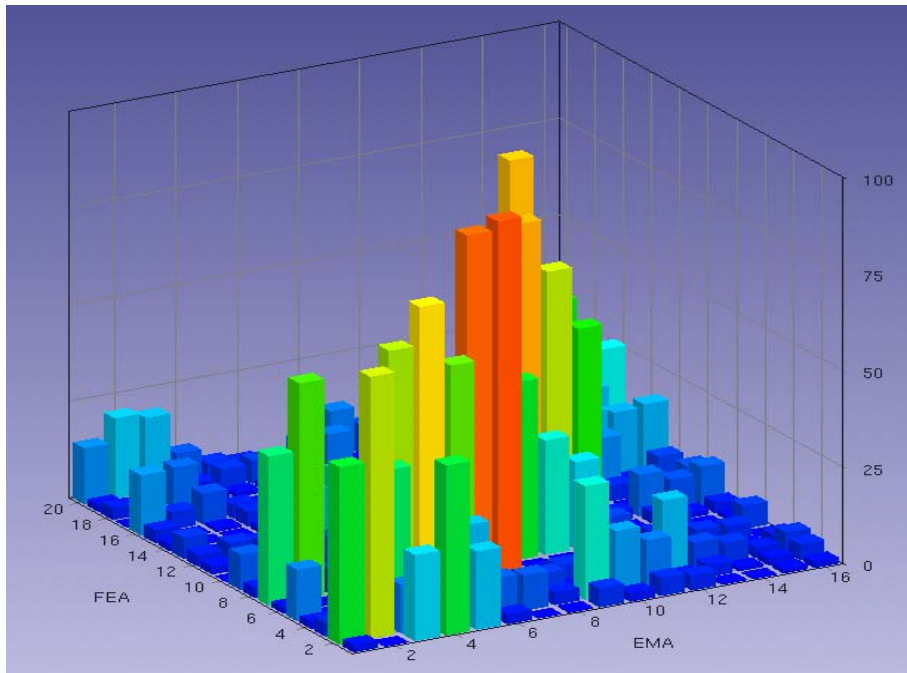


Figure 12: MAX matrix obtained with ABAQUS FE model after updating.

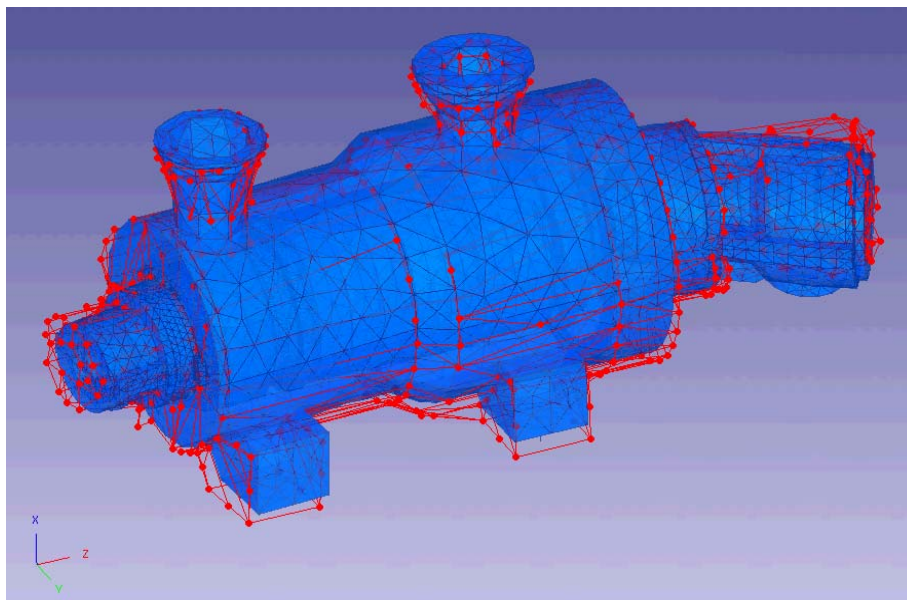


Figure 13: EMA-FEA pairing of the NDE bending mode (Both modes, from test and analysis, at 310 Hz) after updating.

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Pair #	FEA mode #	Frequency [Hz]	EMA mode #	Frequency [Hz]	Difference	MAC [%]
1	1	39.66	4	95.00	-58.25	45.5
2	2	49.47	2	9.50	420.73	60.8
3	4	76.80	3	76.75	0.07	45.5
4	5	98.75	5	98.75	0.00	77.9
5	6	156.90	8	157.00	-0.06	89.5
6	7	185.31	9	285.00	-34.98	40.9
7	8	185.91	6	104.00	78.76	31.3
8	10	310.00	10	310.00	0.00	80.2
9	17	555.17	13	385.00	44.20	78.7

Table 1: The MAC matrix indicates that only three pairs of EMA and FEA mode shapes correlate well with each other.

Only five local parameters of more than twenty selected ones were subject to automatic updating. The total modification of them range less than $\pm 10\%$.

6: Discussion / Conclusions

Already the first correlations EMA-FEA using the results of ANSYS as well as those obtained with the simplified ABAQUS FE model, were fairly good, considering that only "ODS" were available from the modal test. The real mode shapes have been extracted from these data artificially. On the other hand, they based on the data acquisition at a relatively small number of measurement points. The quality of updating is highly dependent on the significance and completeness of the test. The relatively large size and setup of the original FE model was a handicap for FEMtools. Nevertheless, the performance of the FE model could be improved at the end by modifying specific physical parameters for selected sets of elements in a reasonable range. The enhancement of correlation, evaluated in terms of MAC values, amounts about 10 %, at least for a few important eigenfrequencies. Further updating or optimisation of FE model is possible, but requires a new set of experimental data, covering more measurement points and mode shapes. The use of another model, consisting of much less elements, but of the type 10-node-tetrahedron, is also advisable. The combined test-simulation procedure can easily be applied in similar cases in practice.

REFERENCES

- 1 FEMtools Users' Manual, Version 3.0, 2003, DDS Dynamic Design Solutions, Leuven, Belgium
- 2 DASCOTTE, E: "Linking FEA with Test", Sound and Vibration, April 2004, Bay Village, Ohio, USA