Optimisation of the Dynamic Behaviour of a Machine Tool Mounting Device

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Summary:

Aiming to investigate the vibration phenomena occurring occasionally at the platforms of tool holders of type GPS240, Sulzer Innotec has performed experimental and analytical modal analyses under the contract of Mecatool AG, Flawil, Switzerland. The study focused on possible local re-design of the device in order to ensure a resonance-free operation.

In a first step the significant eigenfrequencies with corresponding mode shapes were obtained by means of an experimental modal analysis (EMA). Subsequently, the dynamic behaviour of the platform was simulated using an ABAQUS FE model. The comparison of the eigenfrequencies based on FE calculations with their experimental counterparts proved in general quite satisfactory correlation.

Sensitivity analyses, performed in the following using the verified FE model and the specific software "FEMtools", indicated some improvement potential by added stiffeners in close neigbourhood of the "beams" used for lateral fixation of the platform. Consequently, the original FE model has been adapted according to a re-design proposal and was subject to re-analysis. The eigenfrequencies obtained with this modified model confirmed the effectiveness of the proposed reinforcement measure: The significant modes have been improved substantially about 20 percent.

The paper presents the typical systematic procedure and details of the combined use of experimental and analytical methods (EMA and FEA, respectively) for structural dynamic evaluation purposes and discusses its successful applicability to similar cases in the mechanical engineering practice.

Keywords:

Eigenfrequency analysis, Experimental modal analysis, Optimisation, FE model updating

1 Introduction

For safety and precision reasons, it is of crucial importance, that machine tool equipments don't exhibit eigenfrequencies in the vicinity of operational frequencies. Obviously, the surface quality and precision of the workpieces depend primarily on a vibration–free mounting tool. In the present case of a machine tool platform (Figure 1), used for high speed milling of workpieces, an "inconvenient" eigenfrequency domain about 1100 Hz had to be avoided with respect to excitations observed under operational loads in some cases. Furthermore, this should be ensured for a wide range of workpieces with masses up to 100 kg. As part of this project Sulzer Innotec was asked to investigate a reasonable reinforcement measure which could shift the first eigenfrequency of the device considerably.

In a first step, the relevant eigenfrequencies and corresponding mode shapes of the "original" platform have been measured by means of an experimental modal analysis. Secondly, the observed dynamic behaviour has been reproduced by a FE simulation. A subsequent sensitivity analysis, performed with the specific software FEMtools, indicated that the areas around the webs, where the platform is fixed laterally, are the most efficient ones to obtain the desired reinforcement effect. Based on this result, the customer has worked out a modification proposal. Its efficiency could easily be proven by means of a further FE calculation taking into account the new stiffening ribs, as well as repeating the experimental modal analysis on a new prototyp. This paper illustrates the significant steps of the eigenfrequency tuning procedure which is typical for combined use of experimental and analytical techniques, as well as for close cooperation between customer, analyst and testing specialist.

2 Experimental Modal Analysis

To identify the dynamic parameters of the tool, an experimental modal analysis was performed using excitation by an impulse hammer and considering a limited number of measurement points, each instrumented with three-dimensional accelerometers (Figures 2 and 3).

The results of the experimental modal analysis have shown that

- the device exhibits rigid body modes below 1000 Hz,
- the frequencies of its elastic modes are highly dependent on the stiffnesses of fixations and guides,
- the damping ratios vary between 0.02 and 0.03.

The first three modes shapes, evaluated by means of the frequency response functions, measured without workpiece dummy on the device, are illustrated in Figure 4.

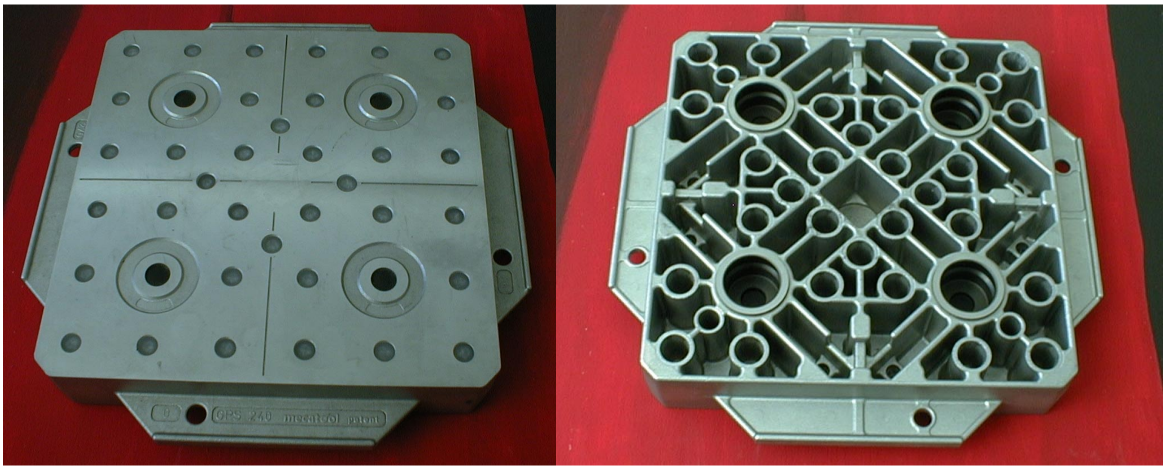


Fig. 1: Top and bottom view of the machine tool mounting device.

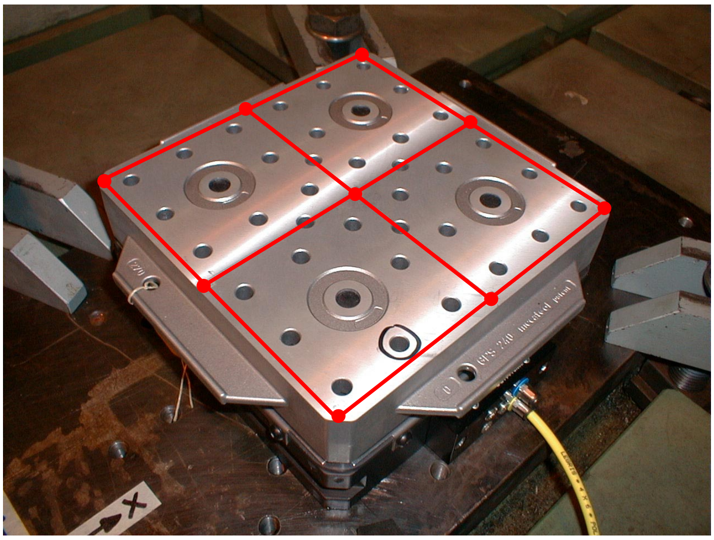


Fig. 2: Measurement point grid

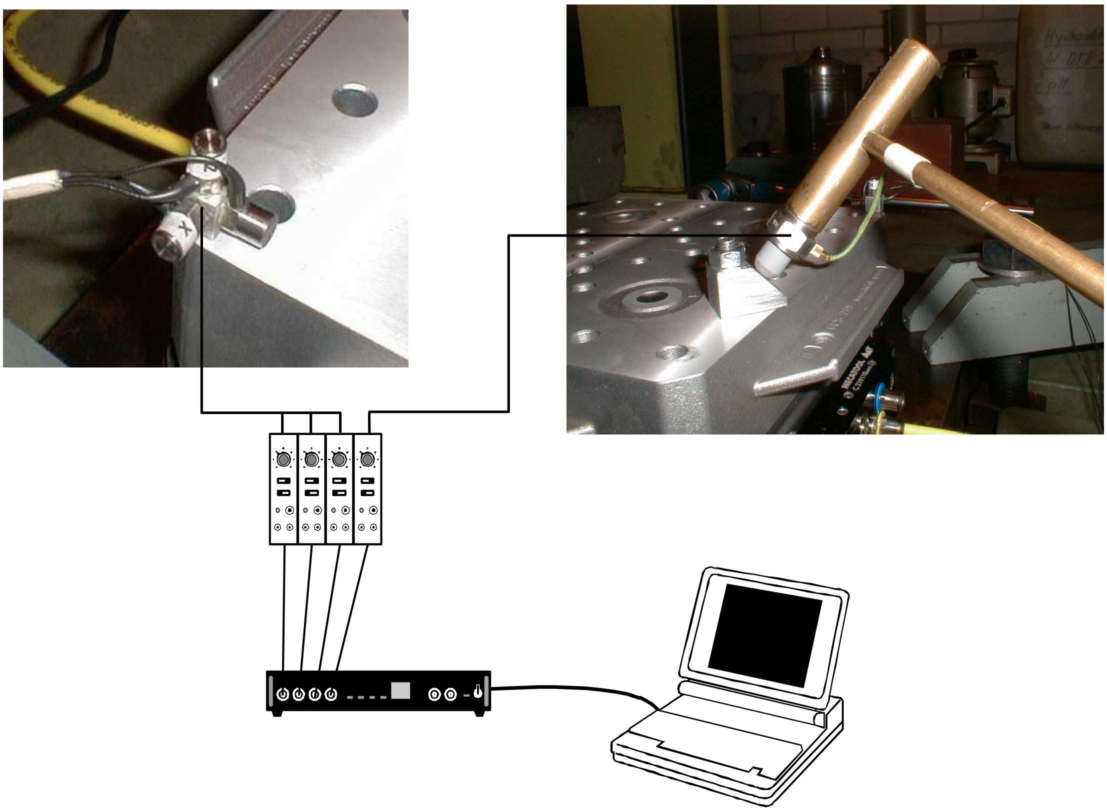
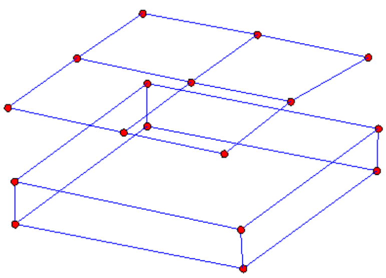
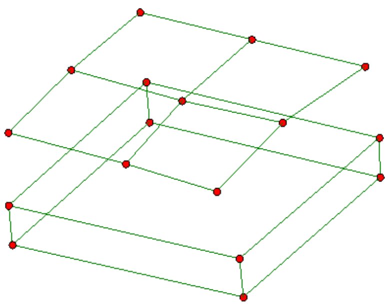


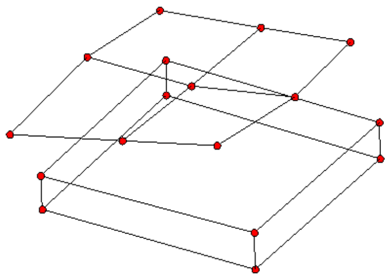
Fig. 3: Setup/Instrumentation of the experimental modal analysis (EMA)



1: 862 Hz



2: 1099 Hz



3: 2874 Hz

Fig. 4: Eigenmodes obtained by EMA

3 FE Modelling

The structural dynamic behaviour of the platform was simulated using an ABAQUS FE model consisting of approximately 17000 elements and 27000 nodes. The modelling of the rather complicated geometry by means of HEXA elements of type C3D8 could be achieved making use of the CAD model in IGES format, considering its symmetry planes (Figure 5) and, in addition, dividing the modular volume into eight horizontal segments appropriately (Figure 6). The weight of the platform has been measured as 4.5 kg. Based on the mass data extracted by ABAQUS the density of the model has been adjusted as 3.2276 kg/dm^3. The FE model with the boundary conditions applied is shown in Figure 7.

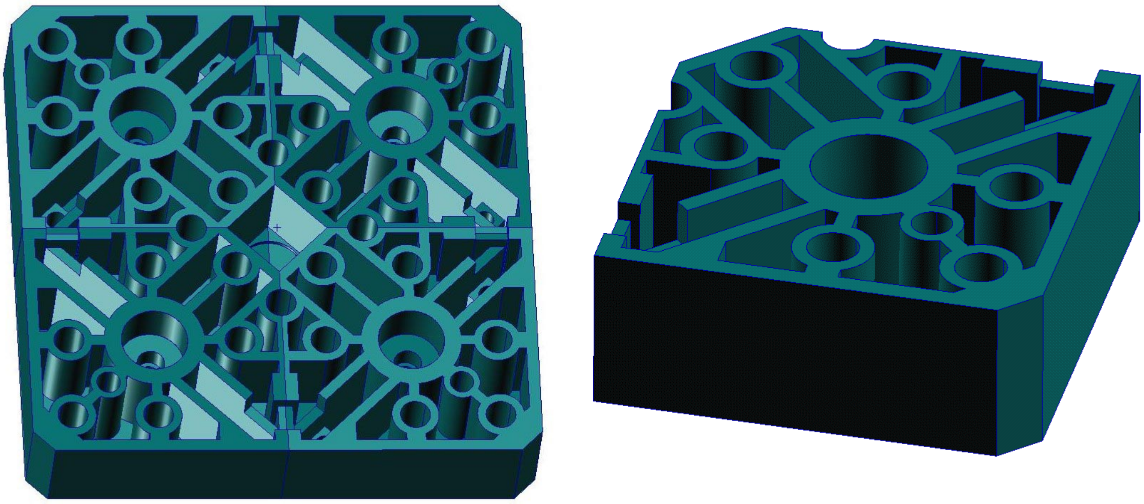


Fig. 5: CAD model with its obvious and hidden symmetry planes.

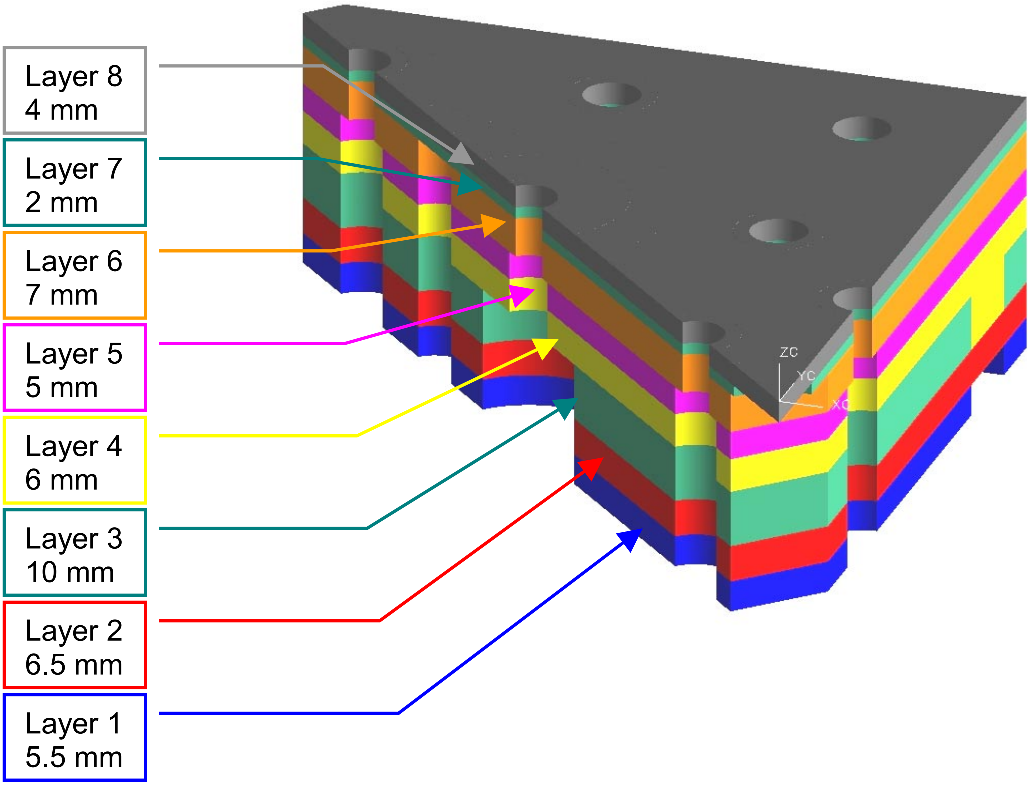


Fig. 6: Mesh generation procedure by proper subdividing of CAD geometry module.

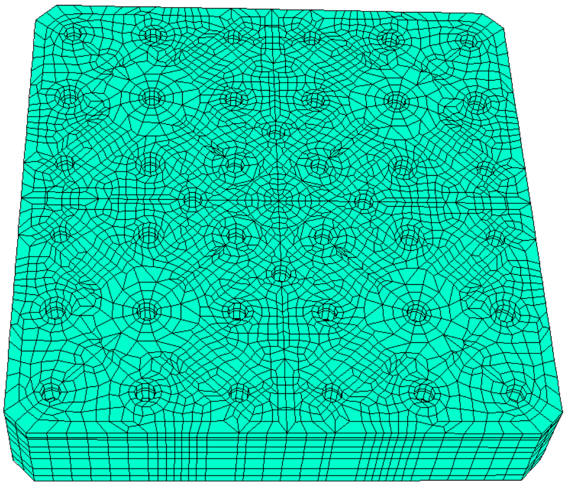


Fig. 7: FE model with boundary conditions (top and bottom view).

4 Results of Eigenfrequency analysis

The FE calculations using power sweep and Lanczos algorithms resulted practically in the same eigenfrequencies and mode shapes. These are given in Figure 8 below. As can be seen there, the first one corresponds to a diagonal translation, the second to a rotational one, both accompanied by remarkable deformation in the neighbourhood of fixations. The third mode can be described as “saddle”, the fourth as symmetrical bending and fifth (not shown here) as the “warping” mode.

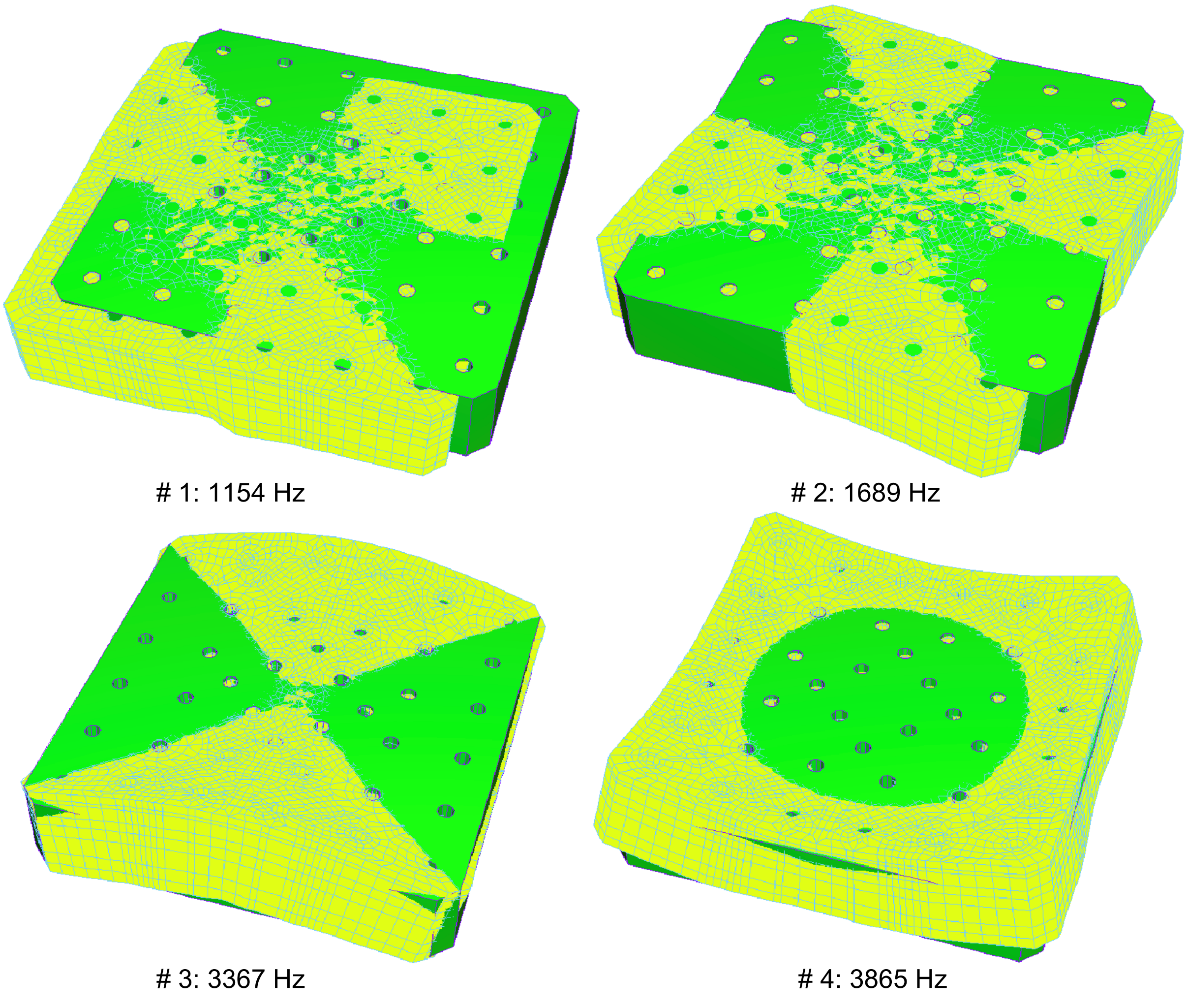


Fig. 8: Selected eigenmodes of the device, calculated by FE analysis.

5 Correlation EMA/FEA

A comparison of calculated modes with their measured counterparts is very helpful in general to verify the quality of an FE model for dynamic simulation purposes and to detect any possible improvements. In the present case, it was remarkable that the calculated mode # 2 (rotation about the vertical axis) was not represented under the measured ones. This could, however, easily be explained by the fact that the restriction of possible lateral movements by means of "balls" inside the bearing bush was not considered in the FE model (Figure 9). On the other hand, the coincidence of the eigenfrequencies # 1 (1099 Hz versus 1154 Hz), which were of special interest, has been judged to be quite good, even when the overall correlation of the modes is of rather moderate quality. The latter has been evaluated by means of a MAC (Modal Assurance Criterion) matrix based an the measured and calculated eigenvector pairs (EMA and FEA respectively) # 1 & # 3 using FEMtools and plotted as shown in Figure 10.

Fig. 9: Bearing details.

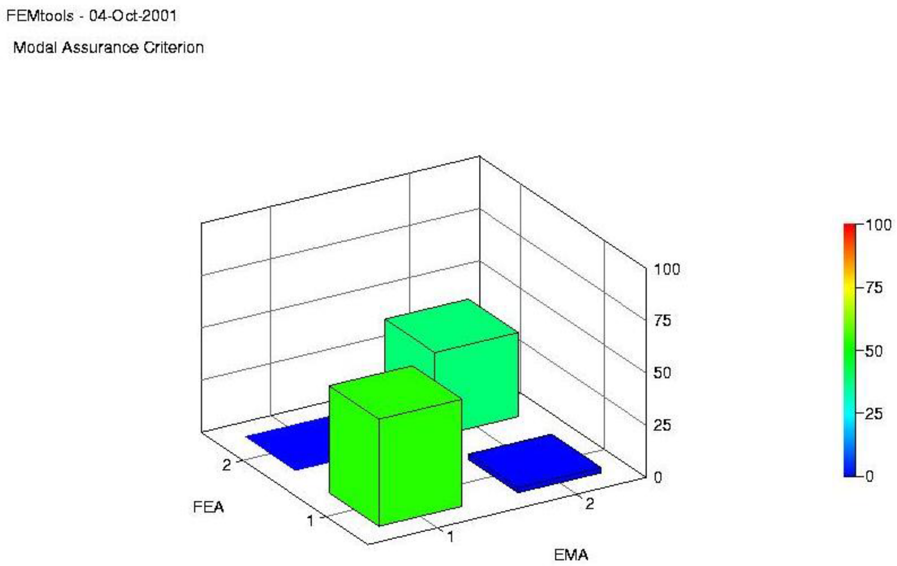


Fig. 10: MAC matrix indicating the moderate coincidence of measured and calculated eigenmodes.

Considering the limited number of measurement points and related data, and the already reasonable agreement for practical purposes, no further attempt was made to update the FE model and achieve an even better correlation by using the sensitivity and updating algorithms of FEMtools.

6 Sensitivity Analysis

Having created a structural model with verified dynamic behaviour, parametric studies were performed to find out structural modification options which could increase the eigenfrequency # 1 significantly. The sensitivity analysis for mode # 1 performed using local stiffnesses in terms of modulus of elasticity as parameter has shown clearly that the eigenfrequency mentioned above is highly sensitive to stiffness changes in the region around the fixations (Figure 11).

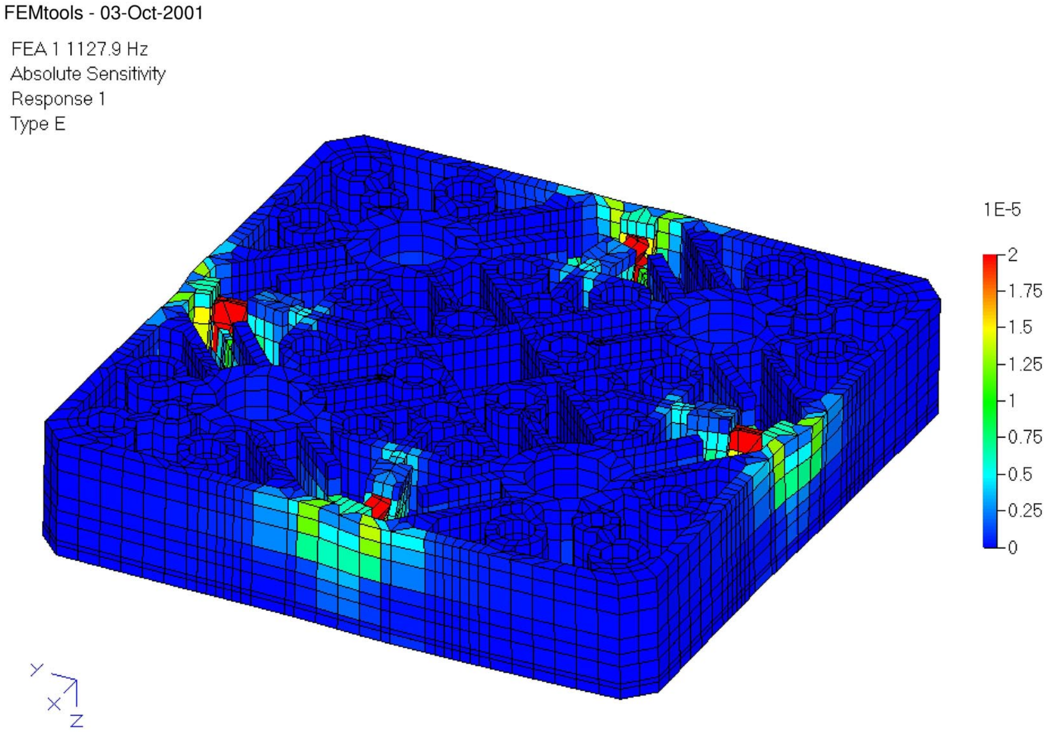


Fig. 11: Result of sensitivity analysis for mode # 1 with respect to stiffness changes.

7 Re-design and Re-analysis

Following the information gained from the sensitivity analysis, the customer has worked out a possible design change and provided Sulzer Innotec with a new CAD geometry. In a further step, the modelling has been adapted and the calculation has been repeated, aiming to quantify the contribution of the

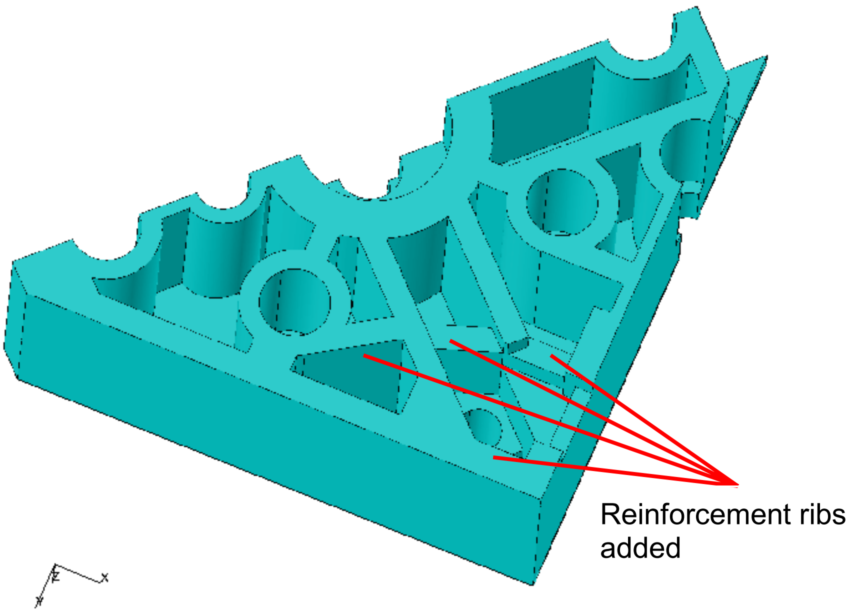


Fig. 12: Modification of the geometry based on the predictions from sensitivity analysis for mode # 1.

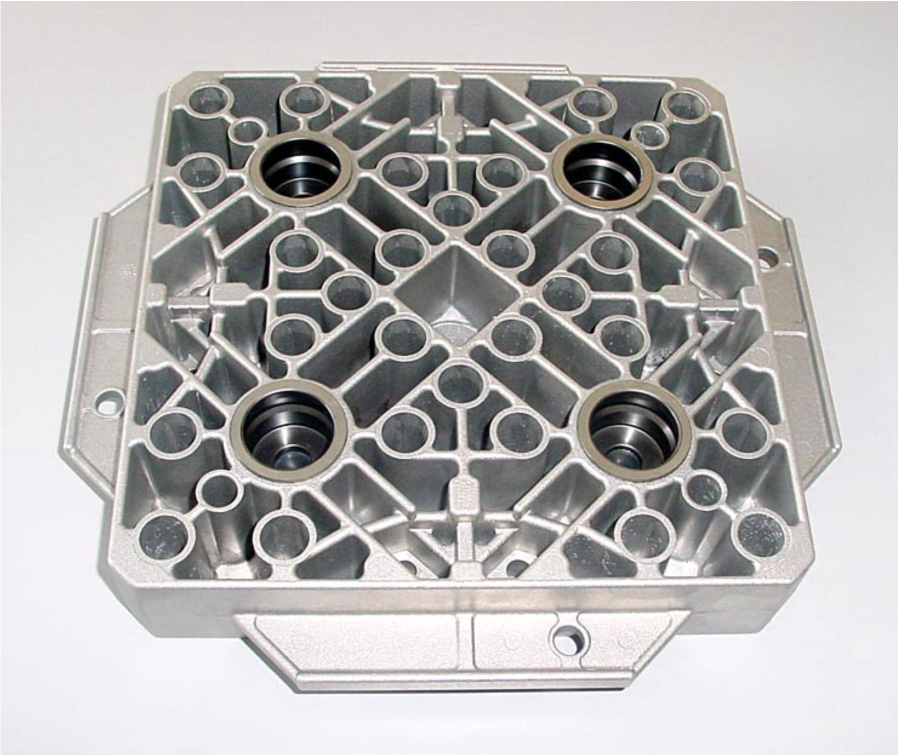


Fig. 13: Platform GPS 240 after design modification.

additional ribs (Figure 12). Figure 13 shows the new design of the product, modified by ribs providing a substantial increase of stiffness with respect to eigenfrequency # 1.

8 Discussion/Conclusions

The results obtained with the new FE model containing the reinforcement measure planned were quite satisfactory: According to the prediction based an this FE analysis, the eigenfrequency # 1 is shifted about 20 % and amounts to 1406 Hz, now. On the other hand, interestingly, the influences of the structural modifications on the higher modes are difficult to observe or are relatively modest. However, at a second glance it becomes clear that those modes do not exhibit any primary displacements & curvatures near modified regions and therefore practically no significant sensitivity is related. Table 1 below gives an overview on the eigenfrequencies before and after local modifications. One can easily see here that there are slight improvements at the modes # 3 and # 4, while # 2 remains unchanged. Remembering that only the mode # 1 was focused in the investigation, the results confirm the solution desired.

Mode No.	Description	Eigenfrequency of the original platform [Hz]	Eigenfrequency of the platform with additional reinforcements [Hz]
# 1	diagonal translation	1154	1406
# 3	"saddle"	3368	3368
# 4	symmetrical bending	3865	3950
# 5	"warping"	3953	4074

Table 1: Comparison of eigenfrequencies before and after structural modification.

The combined use of experimental modal analysis and finite element analysis to achieve the optimal design for machines and components subject to dynamic loads can be realised as schematically described above. Obviously, not every idea, although attractive and reasonable at first glance, proves to be a good one, if the issue is a substantial structural modification. Not seldom the procedure is not as straightforward as shown here. Working with prototypes and trials is generally time consuming and costly. However, working with prototypes and trials is generally more time consuming and costly. The evaluation efforts can be shortened drastically, when a simulation model, verified by experimental modal analysis, is available for use in parametric studies.

The main aim of this paper was to draw attention to how to built and verify such a model and how to use it for parametric studies on possible structural improvements for similar cases.

9 References

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