Successive Model-Updating of the dynamic behaviour of casing bodies on a practical example of an axial piston pump

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Summary:

These days, generally all new products are developed first with virtual models rather than physical prototypes. This also involves the acoustic design, which demands valid simulation models.

Model Updating is a more and more widespread method to generate numerical models for structural dynamics based on experimental results. The paper describes the process of model updating for a casing body of an axial piston pump, which is assembled by screwed connections.
The prerequisites to match experimental data with numerical results and the process of updating based on the modal assurance criterion (MAC) are discussed. The model is updated by solving an optimization problem in several steps to identify the unknown parameters individually. The deviation of the corresponding eigenfrequencies between measurement and simulation could be reduced below 3 %.
The paper concludes with the experimental identification of relevant boundary conditions, which have to be included in a valid FE model besides the appropriate modelling of the structure itself.

Keywords:

Model Updating, Experimental Modal Analysis, Structural Dynamics, Screwed Joints

1 Introduction

Simulation results of structural dynamics are highly dependent on the quality of finite-element (FE) models. Although single parts in free-free boundary condition can be described fairly well by the standard material parameters only, more effort is needed if several components or boundary conditions are included in the FE-Model.
The default values of standard FE contacts couple assemblies in a very stiff way, which does not satisfy the dynamic contact behaviour. Hence the dynamics of assemblies must be set in contrast to experiments.
Model Updating defines the process of comparing data from an experimental modal analysis (EMA) with the corresponding results from a numerical modal analysis (NUMA), finding the differences and modifying the input parameters to adapt the model to the measured results. Therefore the updated model is adjusted to the real world model.

Before starting the updating process, some aspects should be discussed in order to obtain a reasonable physical model:

- Usually, the initial model already describes the reality quite well. This is also important in order to correlate results with experimental data, what would not be possible if the model was too far off.
- The deviation between virtual and real models is caused by the uncertainty of one or more parameters as e.g. a deviation in geometry, material, stiffness values or boundary conditions. The parameters may have different tolerances and sensitivities; some are described by literature values and others are only roughly known.
- Measurements always imply uncertainty, and different physical models yield to different results, too. It might be necessary to average a set of measurements.
- It is important to compare the same extent of the virtual with the physical model. If physical boundary conditions have an influence on the results or additional parts are included in the measurement, both have to be considered in the virtual model as well.
- The number of input parameter (unknown variables) should be as low as possible but as high as necessary. The updating process converges faster with a lower number of parameters, therefore it is recommended to start with a lower number and increase the number of parameters if necessary.

As a practical example, the housing body of an axial piston pump is updated with experimental results. The pump represents a typical mechanical structure, by which the driving mechanism is covered with a casing body.

2 Axial Piston Pump

2.1 Functionality

Axial piston pumps have usually an odd number of pistons, which run in a cylinder block. The kinematics of the pistons is defined by the swash plate, whose angle can be changed to adjust the pump flow rate. Due to the pivoted swash plate, the rotation of the driving shaft and the cylinder causes an axial movement of the pistons. The extension of the pistons sucks the oil into the cylinder, and the retraction discharges it. The control disk separates the suction port and the high pressure port.

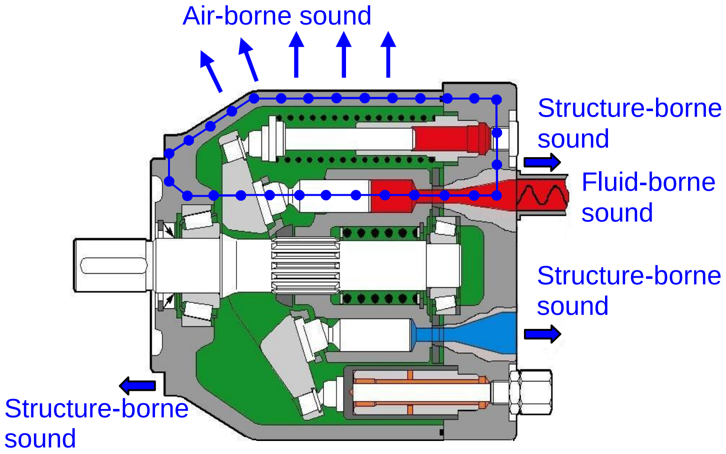

Fig. 1: Schematic longitudinal section of an axial piston pump

As a result of the discrete number of pistons, the resulting force of the pistons is oscillating. Because of the fact that the casing body of the pump is located within the flow of forces, it is excited by the oscillating piston force. As seen in Fig. 1, the vibrating housing generates the air-borne sound directly, but also transmits structure-borne sound to connected structures. The fluid-borne sound is caused by the discontinuous flow rate of the piston pump.

2.2 FE-Model

As mentioned before, the casing body is located within the flow of forces and is responsible for the radiated sound of the pump. Thus it is the most important component in an acoustic FE-simulation and its dynamic behaviour has to be validated thoroughly. Fig. 2 shows the components of the casing body, which are assembled by screwed connections.

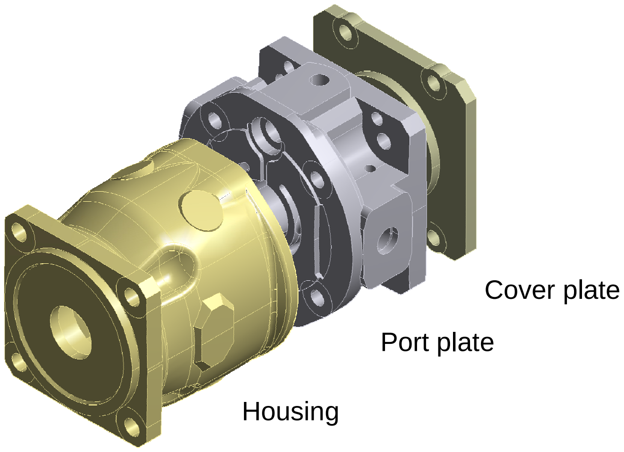

Fig. 2: The three components of the casing body

The FE-model couples the components by bonded contacts, which have a linear-symmetric behaviour. Ansys allows to parameterize the normal and tangential stiffness separately. The stiffness is defined as a specific stiffness per area in units Force/Length³. Research on the parameterization of the contact stiffness is given in [1-3].
It is generally known that the joint clearance of screwed plates depends on several parameters as plate thickness, fastening torque or the distance to the screw. The interface areas may even start to open, especially in case of thin plates. This was considered in the model by separating the total contact area into a screw zone and a joint zone, as shown in Fig. 3. The parameters of these zones may be set individually. The diameter of the screw zone was evaluated on the basis of the deformation cone of screwed connections [4].

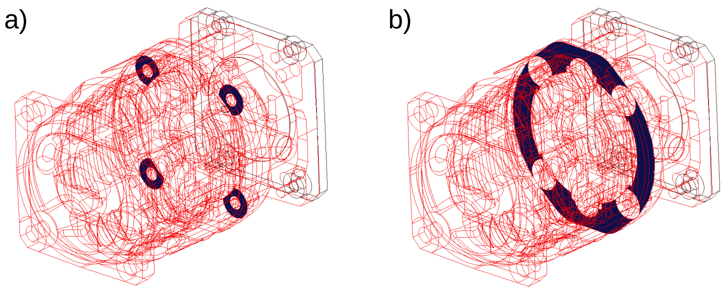

Fig. 3: Separation of the contact area into a screw zone (a) and a joint zone (b).

CAD geometries often have a high level of detail which lead to fine FE meshs and high solution times. For simulations in structural dynamics, geometric discretization with all model details is not as important as for other simulation disciplines, e.g. fatigue testing in which a very fine mesh is needed. Therefore all minor details as rounded edges and imprintings were removed before importing the geometries into the FE pre-processor (Fig. 4). The number of nodes could be reduced by 67 %, whereupon the simulation results hardly changed.

a) b)

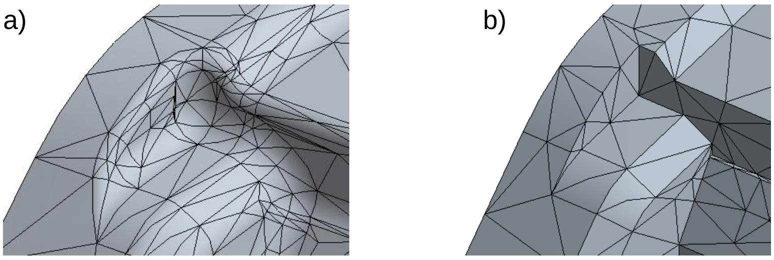

Fig. 4: Local reduction of details: mesh before (a) and after (b) the reduction

3 Experimental Modal Analysis

The EMA was proceeded by the roving hammer method. All measurements were done in a free-free boundary condition, which can be realized in a FE model, too.
The coordinates of the measurement points were taken from the CAD geometry, as well as the origin of the global coordinate system. This ensured the identification of corresponding nodes in the FE model within a certain tolerance during the following correlation process.
In general it is useful to measure with the help of multiple reference sensors to resolve all modes, including repeated roots or local modes. There are numerous papers describing the basics of EMA and the major points, such as [5-7].
For the later correlation process it is also important to measure equivalent configurations because additional components which are measured, though not included in the FE model, will influence the vibrancy and lead to inaccurate results.

4 Model Correlation

A major mistake in the comparison of experimental and numerical modal results is often made by simply comparing the eigenfrequencies. This might work for very simple structures such as plates, but for most structures these results are incorrect because modes can be switched and thus have a different order. Not all simulated modes may be resolved by the measurements, too or the FE model might miss some modes because it is improperly modelled or parameters are set in the wrong way.
For a proper modal correlation, similar mode shapes have to be allocated, so that the differences of the corresponding eigenfrequencies give a conclusion about the correlation quality.

4.1 Modal Assurance Criterion

Comparing mode shapes manually can be an elaborative process, but there are several criterions defined to assist finding similarities of mode shapes [8]:
- modal assurance criterion (MAC)
- coordinate modal assurance criterion (COMAC)
- frequency response assurance criterion (FRAC)
- coordinate orthogonality check (CORTHOG)
- frequency scaled modal assurance criterion (FMAC)
- partial modal assurance criterion (PMAC)
- scaled modal assurance criterion (SMAC)
- modal assurance criterion using reciprocal modal vectors (MACRV)

The most commonly used criterion is the MAC, which is defined in the following way:

$$MAC_{EMA,NUMA} = \frac{\left[\{f_{EMA}\}^{*}\{f_{NUMA}\}\right]^{2}}{\{f_{EMA}\}^{*}\{f_{EMA}\}\cdot\{f_{NUMA}\}^{*}\{f_{NUMA}\}} \qquad (1)$$

The MAC evaluates the orthogonality of modal shapes by applying the displacements of the eigenvectors. The eigenvectors are allowed to have different scaling factors, so that residual vectors from an EMA can be compared with mass-normalized vectors from a NUMA, too.

The MAC results in values between 0 and 1. The literature states that values between 0.9 and 1 mean identical shapes, values between 0.6 and 0.9 show a high similarity and values less than 0.6 have no coincidence.
Fig. 5 shows a sample MAC matrix with typically high MAC values in its diagonal and values close to zero next to the diagonal. The lower modes are often resolved better than higher modes and thus have higher MAC values. Fig. 5 also shows that mode 3 and 4 are switched.

The 5th measured mode corresponds only with a low MAC to the 6th FE mode. Even though the MAC is below the literature value of 0.6, the modes can still be the same as no other modes correspond to these modes. Low MAC values may be caused by a bad resolution of the measurement or by local modes to which other parts contribute with residual noise.

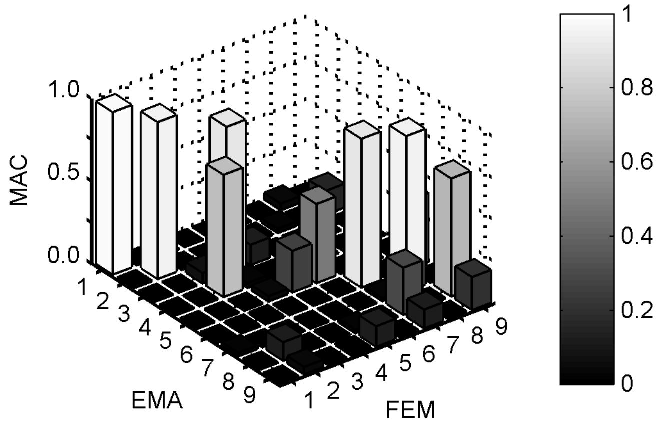

Fig. 5: Sample MAC matrix

Table 1: Corresponding Mode Pairs

EMA	FEM	MAC
1	1	98 %
2	2	94 %
3	4	89 %
4	3	74 %
5	6	48 %
6	7	90 %
7	8	94 %
8	9	71 %

To find all corresponding mode pairs, including local modes with lower MAC values, the following procedure has been adopted:

- find the maximum MAC value for each row and each column
- if a MAC is the highest MAC in its row as well as in its column, the MAC represents a successful correlation

The corresponding mode pairs for the sample MAC matrix were shown in Table 1.

4.2 Prerequisites for matching FE results and measurement data

It is not possible to compare results from EMA and NUMA instantly in general. Node numbers and result coordinate systems have a different setup, which can be seen in Fig. 6.

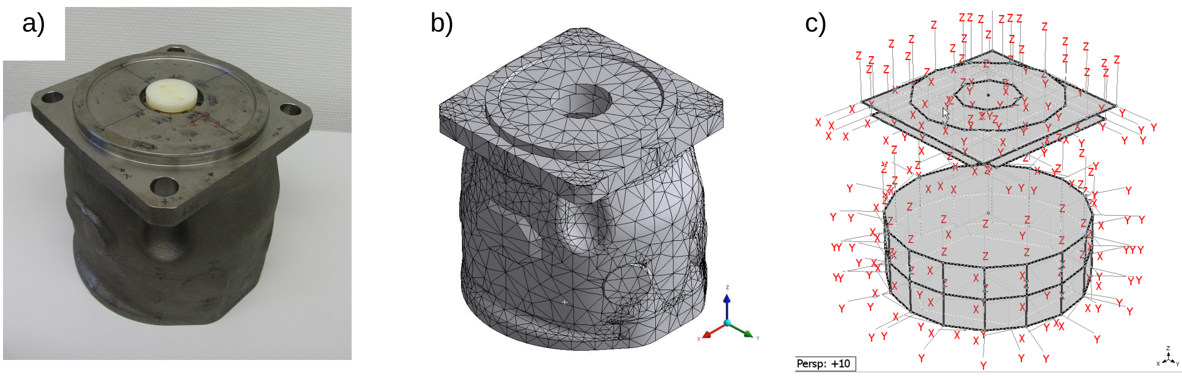

Fig. 6: Different views of the housing component: a) photo, b) FE mesh, c) measurement structure

The eigenvectors used for the MAC comparison must have corresponding entries for each degree of freedom (DOF) in the same order. Each DOF is defined by the position and the direction of the measurement.

There are several methods to expand the test model or to reduce the analytical model to the measured DOFs [9]. These methods involve the transformation of system matrices and thus are more complicated to handle. In this paper, the analytical results are truncated and mapped to the measured DOFs but without reducing the analytical model.

Depending on the complexity of the structure, several steps have to be applied to match the data between the FE model and the test model:

- check the origin of the global coordinate systems
- find the corresponding nodes in the FE model
- read the FE results in the same coordinate direction as the measured DOF
- truncate unmeasured DOF (e.g. tangential to the surface)

4.3 Formulation of an optimization problem

Model updating can be understood as an optimization problem with input and output parameters, an objective function and an optimization model. It is possible to solve the optimization problem by any parameter optimization software. In this paper, OptiSLang was used.

4.3.1 Input parameters

The input parameters in the optimization cycle are used to update the analytical model. Depending on the configuration of the model the following parameters were used as input parameters:
- young's modulus of each part
- normal and tangential contact stiffness of the screw zone
- normal and tangential contact stiffness of the joint zone

The density of the model's material was initially set by adjusting the weight of each part to the real components.

4.3.2 Output parameters and objective function

After applying the MAC criterion to the modal results as described in 4.1, a table according to Table 1 is created, including the deviations of the corresponding eigenfrequencies.
Based on experience, the following objective function was formulated to consider the m corresponding mode pairs in the objective function:

$$\frac{\sum_m \frac{|\Delta f_m|}{f_{m,EMA}} \cdot MAC_m{}^2}{\sum_m MAC_m{}^2} \rightarrow \min \qquad (2)$$

The weighting factor MAC² ensures that mode pairs with a higher similarity get a higher weight than mode pairs with a lower coincidence.

4.3.3 Optimization algorithm

Due to the procedure of matching the corresponding mode pairs, the objective function is nonlinear and discontinuous. By changing input parameters, the number of corresponding mode pairs might change and give a discontinuous objective function.
In the beginning of the updating process, it is senseful to use an explorative algorithm as an evolutionary optimization. Especially if the initial parameters are only roughly known, the entire parameter space is covered in order to find the global minimum. If the model is already within the region around the global minimum, a direct or gradient based method is suggestive.

5 Successive model updating

The casing body has several unknown parameters to identify during the model updating process. As the material parameters are already well described in literature, the contact parameters are expected to underlie a greater variation. Therefore the model updating is proceeded successively:
- correlation of single parts to identify the young's modulus
- correlation of partial assemblies with one screwed joint

This bottom-up process allows solving for one unknown during each model updating step. After updating these steps independently, all configurations were included in one global optimization loop, together with the last configuration in order to allow a final fine-tuning:
- correlation of the entire casing body

Fig. 7 displays the MAC matrices before and after the model updating. It can be seen that the MAC values have risen and more modes correlate.

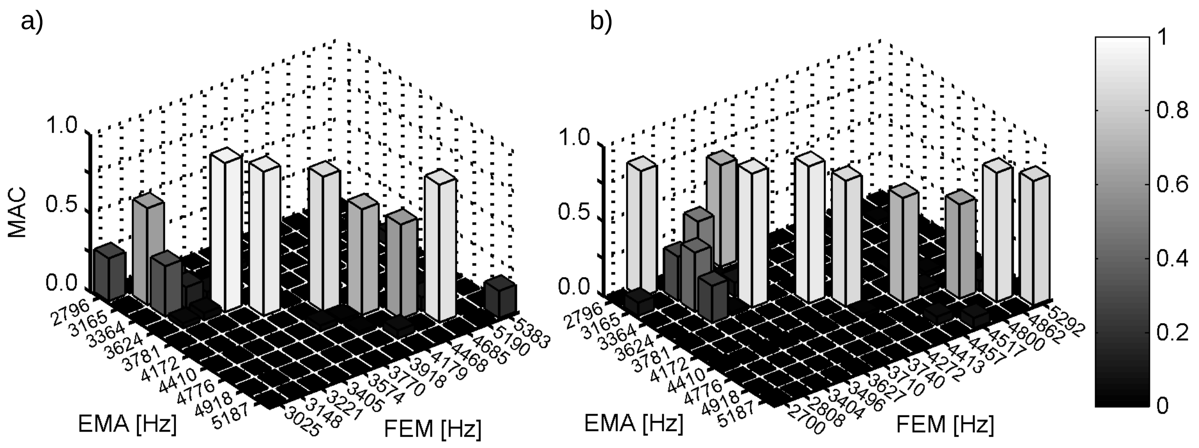

Fig. 7: MAC matrix of the casing body before (a) and after (b) model updating

A complete overview over the optimization results is given in Fig. 8. The deviation of corresponding eigenfrequencies could be reduced from partly more than 20 % to less than 3 % for the single components as well as for the assemblies. Only some few modes show a larger difference, which is caused by low MAC values and therefore a minor consideration in the global objective function.

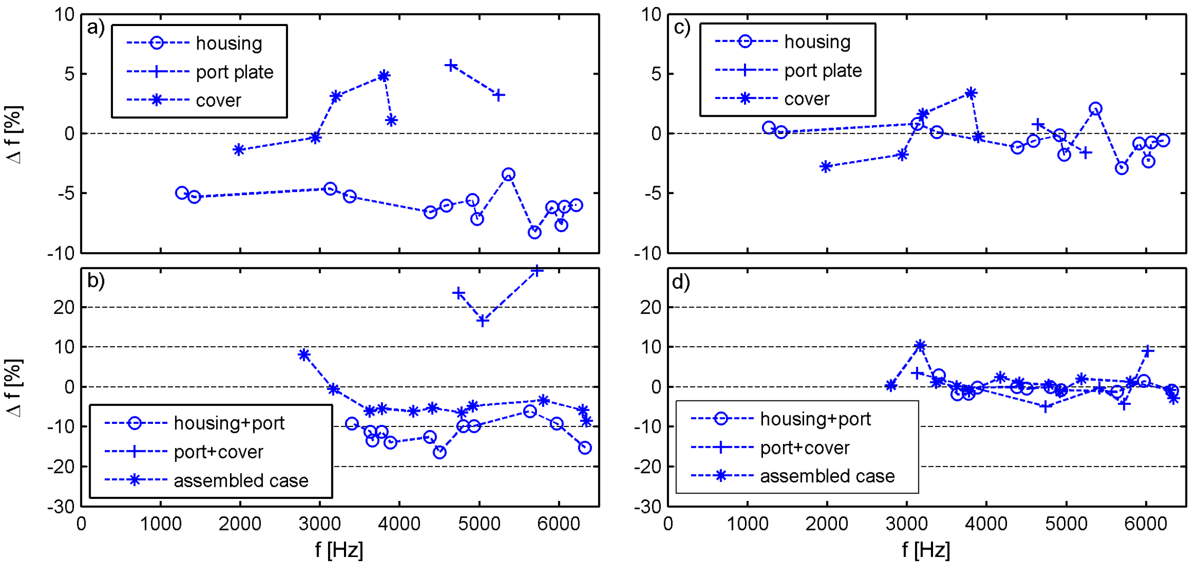

Fig. 8: Deviation of corresponding eigenfrequencies before (a, b) and after (c, d) model updating

The change of the input parameters from the initial to the optimized values can be seen in Table 2. As presumed, the material parameters were only slightly modified, whereas the joint parameters had a major change. Another remarkable point is that the stiffness in the joint zone between port plate and cover plate was reduced to zero, but the other joint stiffness made a major rise. This fact was further investigated by a static simulation in which the bolt forces were applied to the structure. The results can be seen in Fig. 9. The thinner cover plate is highly deformed, so that the interfaces of the joint zone open and the contact stiffness drops to zero. The thicker port plate is hardly deformed and the contact is still closed.

Table 2: Change of input parameters during model update

Material properties	Density	Young's modulus
Housing	-2 %	+12 %
Port plate	-5 %	-8 %
Cover plate	-3 %	-3 %
Joint stiffness	Screw zone	Joint zone
Housing / port plate	+110 %	+110 %
Port plate / cover plate	+170 %	-100 %

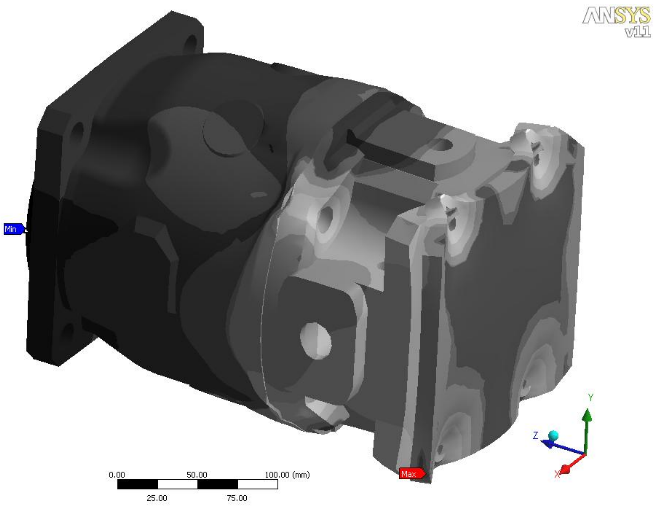

Fig. 9: Static displacement simulation: bolt forces applied to the casing body

6 Boundary conditions

Beside the appropriate parameterization of materials and joints, boundary conditions (BC) may highly influence the structural dynamics of the casing body and therefore have to be considered in a FE model for acoustic simulations. The following describes a method to identify the influence of BC and if they have to be considered in an FE model or if certain BC may by neglected. Applicable boundary conditions for axial piston pumps are

- support (fixed flange)
- hydraulic hoses
- hydraulic hoses (outlet hose pressurized)
- support (damped flange)

To compare the influence experimentally, the level of structure-borne sound was measured reciprocally by applying the roving hammer method with fixed sensors inside the pump [10]. The experiment was performed with an empty casing to isolate the effects from operational noise.
The results are displayed in Fig. 10. The influence of each boundary condition can be seen, including their effects on amplitude and frequency. The results conclude that all boundary conditions have an effect related to the reference measurement, except for the pressurized outlet hose which is identical with the unpressurized hose.

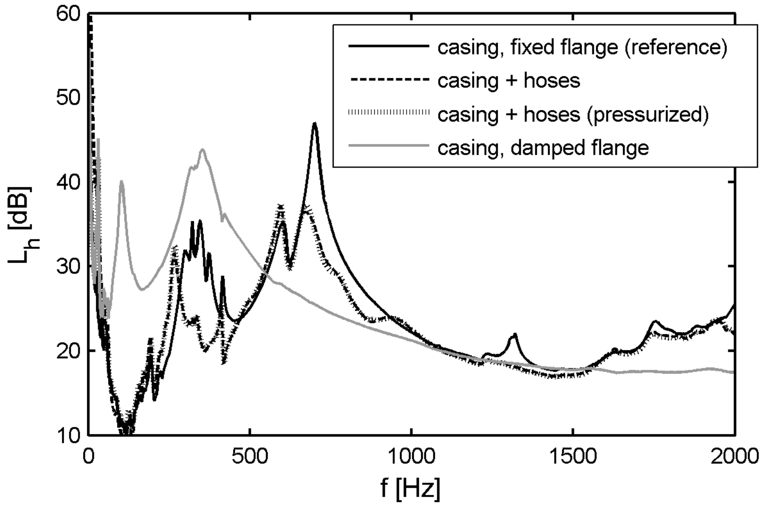

Fig. 10: Influence of miscellaneous boundary conditions on the level of structure-borne sound

7 Conclusion

The process of model updating was described on a practical example of the casing body of an axial piston pump. The paper showed how the screwed connections were defined and correlated with experimental results in order to achieve a valid structural dynamics model. The prerequisites to match analytical results and experimental data were discussed, and an optimization problem has been formulated to update the model iteratively. The deviation of the corresponding eigenfrequencies between measured and simulated modes could be reduced from more than 20 % below 3 %.
In addition to the updating process, an experimental method has been proposed to identify whether applicable boundary conditions are relevant and thus have to be considered in a FE model or not.

The paper demonstrated the importance to adapt dynamic models to real measurements, which has to be carried out by a systematic model updating process rather than manual parameter estimation.

8 References

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