Finite Element model validation of a large spinning facility

C. Schedlinski
ICS Engineering GmbH
Am Lachengraben 5, 63303 Dreieich, Germany
e-mail: **sched@ics-engineering.com**

Abstract

A new large spinning facility has been built in Germany representing a world class unique test facility for testing large rotating systems. In order to assess the capabilities of the system a validated Finite Element model is required. To obtain such a model a two-step procedure was applied:

In a first step, vibration tests were conducted focusing on the rigid body frequencies of the resiliently mounted system as well as on its basic elastic behavior (natural frequencies, modal damping, and mode shapes). In a second step, a Finite Element model was created and subsequently validated with the help of the obtained vibration test data.

The paper will present the testing approach and the model validation results obtained by application of computational model updating techniques. All in one the employed two-step procedure provided valuable insight into the dynamics of the facility next to a validated Finite Element model of high fidelity.

1 Introduction

A new large spinning facility (LSF) has been built in Germany. The LSF represents a world class unique test facility for testing rotating systems with diameters larger than 1.5 m up to 12,000 RPM.

In order to assess the capabilities of the LSF system, and to verify that specifications are met, a proper understanding of its dynamics is crucial. One key issue in this context is to have readily available a validated Finite Element model of the LSF system. The Finite Element model can furthermore be used to tune the parameters of test vehicles in order to ensure the right behavior under test. Also, environmental protection aspects under operational loads can be assessed to guarantee integrity of the building and the LSF system itself.

To obtain a validated Finite Element model of the LSF system a two-step procedure was applied:

In a first step, vibration tests were conducted focusing on the rigid body frequencies of the resiliently mounted LSF system (mounted on an array of air springs) as well as on its basic elastic behavior (natural frequencies, modal damping, and mode shapes). In a second step, a Finite Element model of the LSF system was created and subsequently validated by means of the obtained vibration test data.

Enormous challenges for the validation campaign were imposed by the very large overall mass of the system of approximately 1,600 tons and the gross dimensions of about 25 x 10 x 10 meters (length x width x height).

Thus, for modal testing a combined approach of ambient (or environmental) excitation and classical hammer excitation was selected, in combination with special analysis techniques (output only modal analysis). Also, the expected low frequencies of the suspension system (< 1 Hz) demanded for specialized sensors, capable to accurately measure practically down to zero Hertz.

This paper will show that the selected testing approach provided a rather consistent and reliable data base for subsequent Finite Element model validation. Furthermore, the model validation results obtained by application of computational model updating techniques will be presented. All in one the employed two-step procedure provided valuable insight into the LSF dynamics next to a validated Finite Element model of high fidelity.

2 System Overview

The LSF system in principle consists of four major parts: ferro concrete foundation, steel frame, vacuum vessel and drivetrain (figure 1). The foundation itself is mounted on multiple air springs in order resiliently decouple the facility from the building. The steel frame holds the test piece which is located in a vacuum vessel during operation while it is powered by the drivetrain.

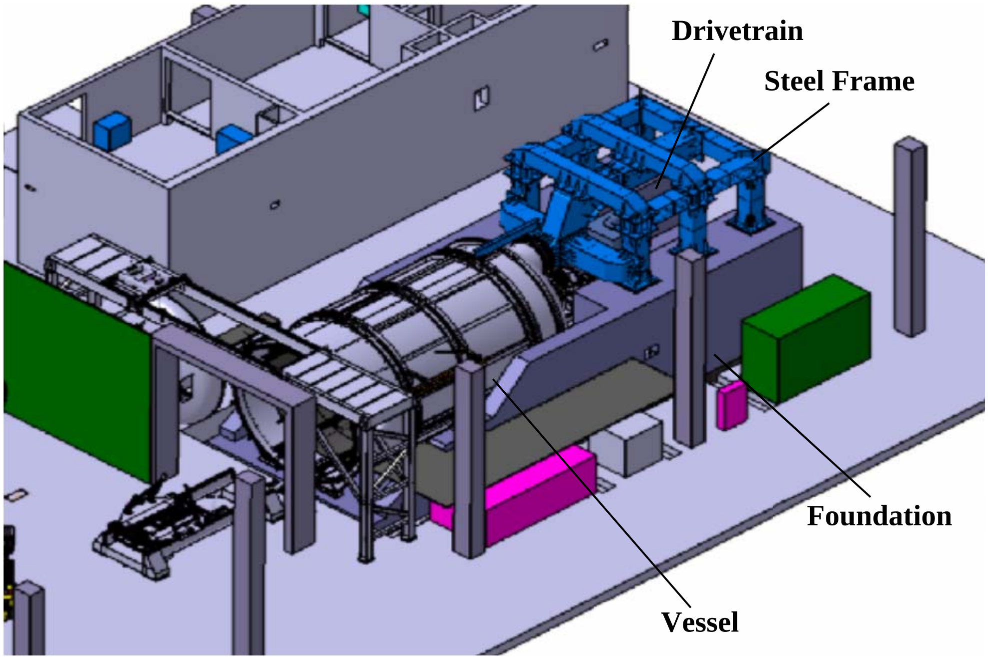

Figure 1: Survey of LSF system

3 Obtaining the required test data basis

3.1 Test strategy

In order to obtain the required test data basis for subsequent model validation dedicated modal testing was performed for the LSF system. A special challenge was that the large mass of the system of around 1,600 tons is usually prohibitive to classical shaker or hammer driven experimental modal analysis. Thus, ambient (or environmental) excitation[1] was used in addition to classical hammer excitation, in combination with special analysis techniques (output only modal analysis). Also, the expected low frequencies of the suspension system of the foundation (< 1 Hz) demanded for specialized sensors, capable to accurately measure practically down to zero Hertz.

[1] Ambient excitation is usually randomly generated by the environment of the system to be tested. Sources can for instance be pedestrians or road noise on the system itself or in the vicinity of the system. If the ambient excitation by the environment is not sufficient to properly excite the system, additional ambient excitation of the system can be introduced for instance by walking, jumping, or dropping weights on the system. A key feature of ambient excitation techniques is that neither location nor characteristics of the excitation need to be known.

Therefore three separate tests were designed in order to capture all relevant dynamic effects: an ambient excitation test of the foundation, an ambient excitation test of the foundation and steel frame, and a hammer excitation test of the steel frame. A survey of the tests is given in figure 2.

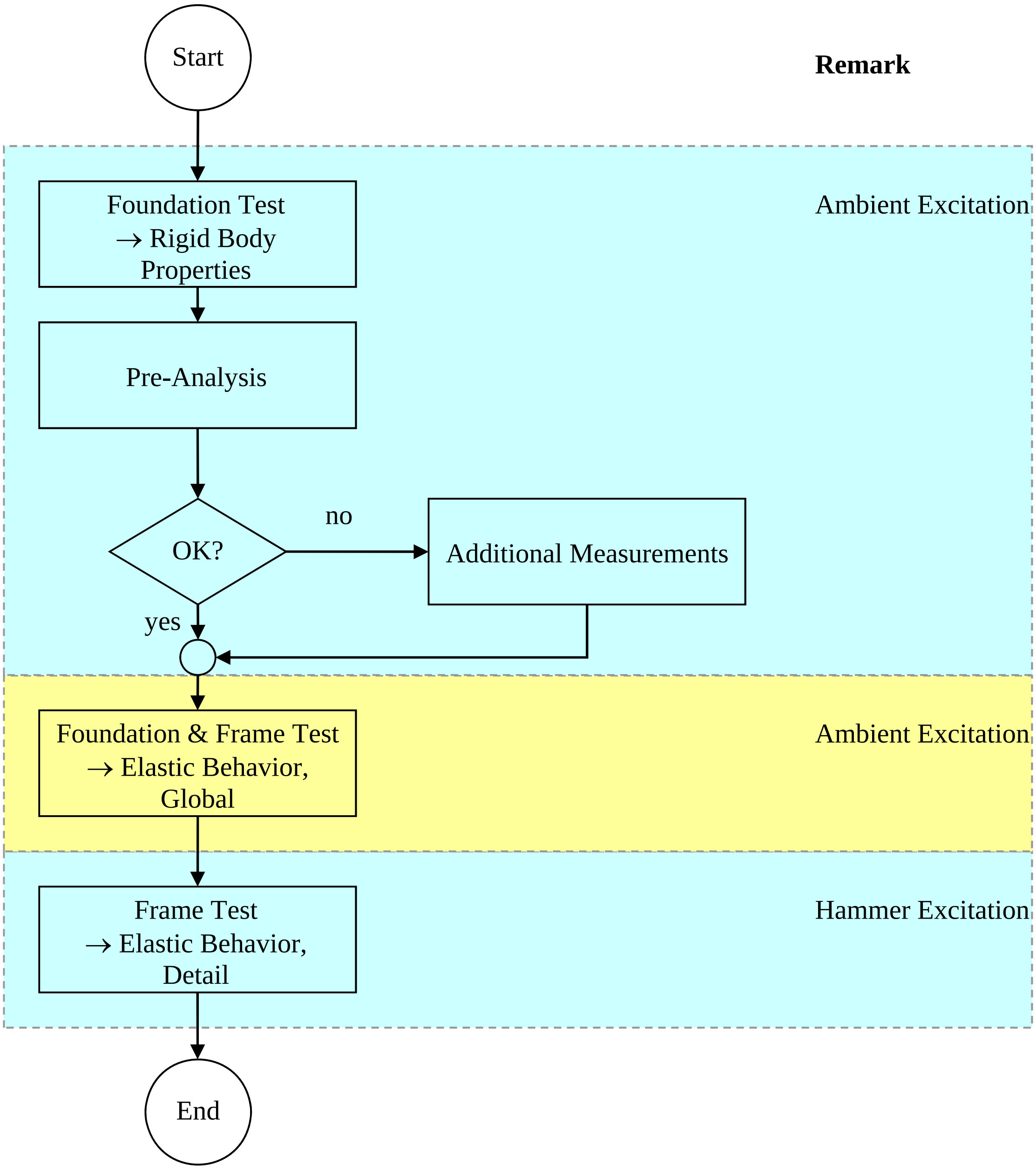

Figure 2: Survey of test procedure

The first test solely focused on the low frequent rigid body behavior of the suspended LSF system (rigid body frequencies and modes). Specialized sensors and ambient excitation were used to obtain the required information. The second test aimed at capturing the basic elastic behavior of the LSF system, especially the interaction of foundation and the steel frame. The data obtained here can later on aid to properly tune the foundation stiffness and the interface between foundation and steel frame. The third test should finally give detailed insight into the behavior of the steel frame itself.

3.2 Test Planning

Before testing, thorough test planning was performed, that primarily focused on a proper definition of measurement and excitation locations as well as on an assessment of the required measurement parameters. It utilized data from a Finite Element (FE) analysis, and enabled not only the test design but also considerably simplified the later correlation with the analytical results (Finite Element model and test model ‘match’). Test planning covered the following aspects (see also [1]):

- selection of relevant target modes
- selection of measurement degrees of freedom with respect to
  - essential test information
  - sufficient spatial resolution of the target modes (linear independence)
  - coincidence of measurement and FE model nodes
  - accessibility of the measurement nodes
  - redundancy of the measurement degrees of freedom
  - robustness of the test model
- selection of exciter positions (if possible, simultaneous excitation of all target modes)
- sufficient frequency resolution (for proper identification of modal data)

Test planning was conducted utilizing a special MATLAB® based software package (ICS.sysval, [2]). Main goal was to provide best possible test models and a priori test setup specifications in order to obtain highly reliable test data.

3.3 Test item

Two typical views of the test item are shown in figure 3 and figure 4. A special challenge was imposed by the gross dimensions of about 25 x 10 x 10 meters (length x width x height). Especially logistics and instrumentation (see also figure 5) had to be planned ahead very carefully in order to guarantee for a smooth and undisturbed flow of the test campaign.

Figure 3: Side view of test item

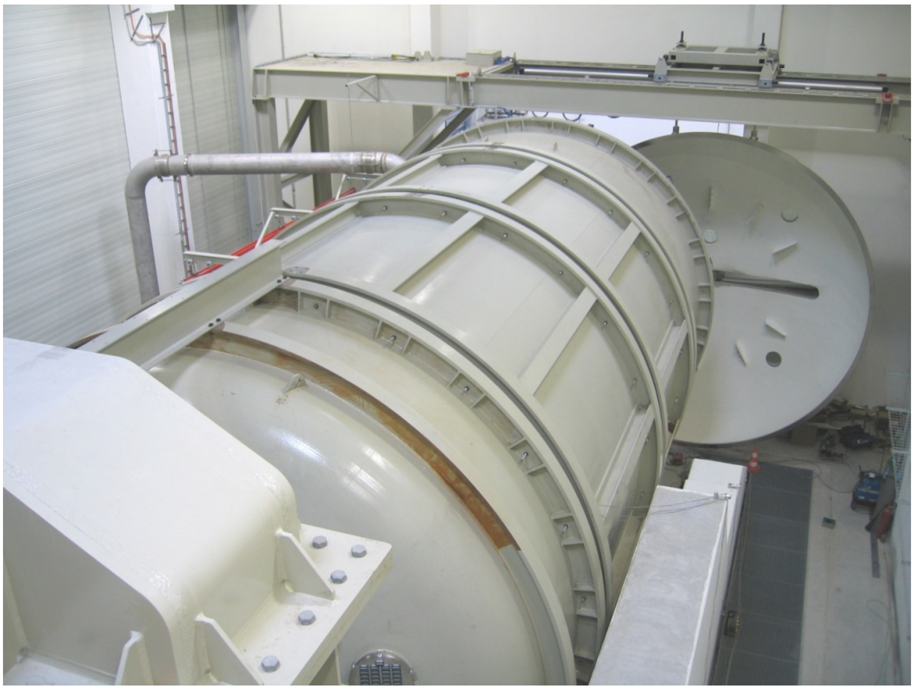

Figure 4: View from top on vessel

Figure 5: Mounting of sensors on top of steel frame

3.4 Ambient excitation testing

Ambient excitation testing was performed by ICS in cooperation with the BAM (Bundesanstalt für Materialprüfung). For ambient excitation testing sensors were attached to the LSF system according to the test planning results. To identify the lower rigid body frequencies and mode shapes of the foundation and its attachments, the relevant measurements degrees of freedom were measured with specialized accelerometers, capable of measuring down to very low frequencies. For the higher natural frequencies and mode shapes, electro dynamic velocity sensors (geophones) were used. In order to excite the LSF system four different excitation types with different frequency contents were applied:

- pure ambient (environmental) excitation
- impulse excitation by jumping (on foundation), three men simultaneously
- impulse excitation (on foundation) with falling weight of about 30 kg from about 1.5 m
- impulse excitation (on foundation) with 5.4 kg heavy sledge hammer

During the test campaign continuous time data were sampled with durations from several minutes (impulse excitations) up to 11.4 hours (pure ambient excitation). For the impulse excitation measurements multiple impulses were applied within the defined overall measurement time with delays from 20 to 80 seconds, depending on the used block size and sampling frequency.

Ambient excitation testing finally provided eight rigid body type modes of the system on air springs up to about 2.5 Hz. Also, seven elastic modes of the complete system (foundation, steel frame) could be identified up to about 40 Hz.

3.5 Hammer excitation testing

During hammer excitation testing it was found, that a rather good excitation of the entire LSF system could be achieved which was not expected a priori. Thus, to obtain additional and consistent information on the foundation as well, the original test model (steel frame only) was extended such that measurement nodes on foundation, vessel and drivetrain were added (figure 6).

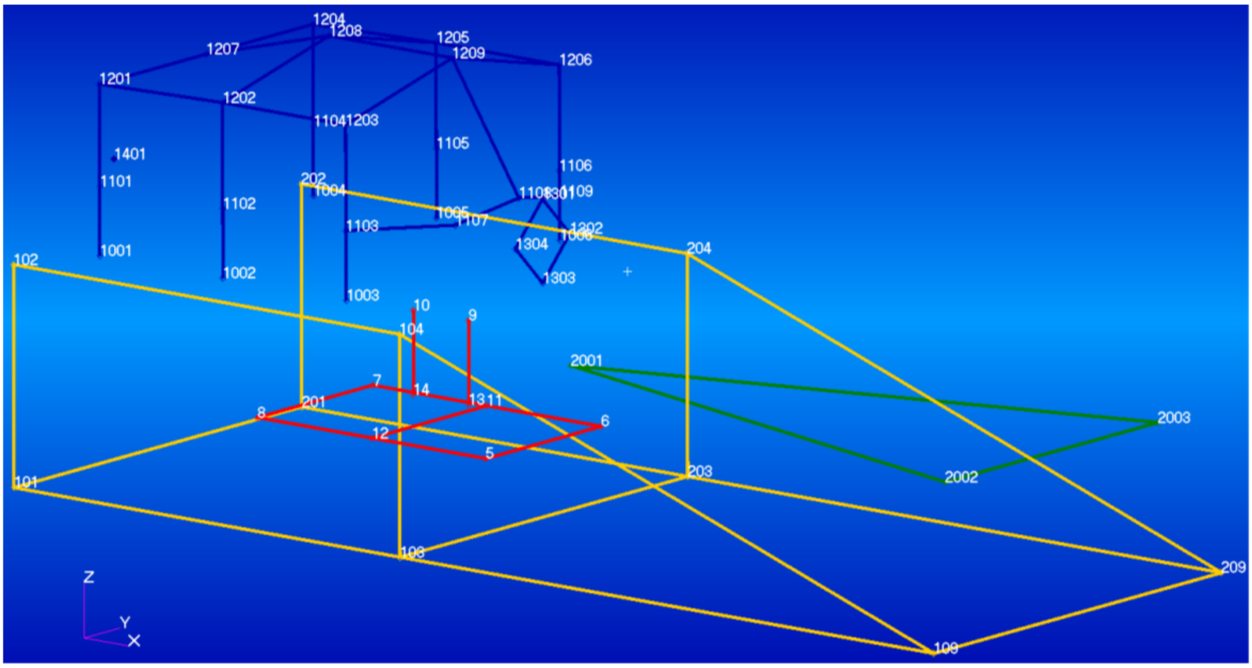

Figure 6: Extended test model for hammer excitation testing

Typical measured frequency response functions (imaginary parts) are show in figure 7. Clear and pronounced peaks with moderate damping can be observed in the entire frequency range of interest.

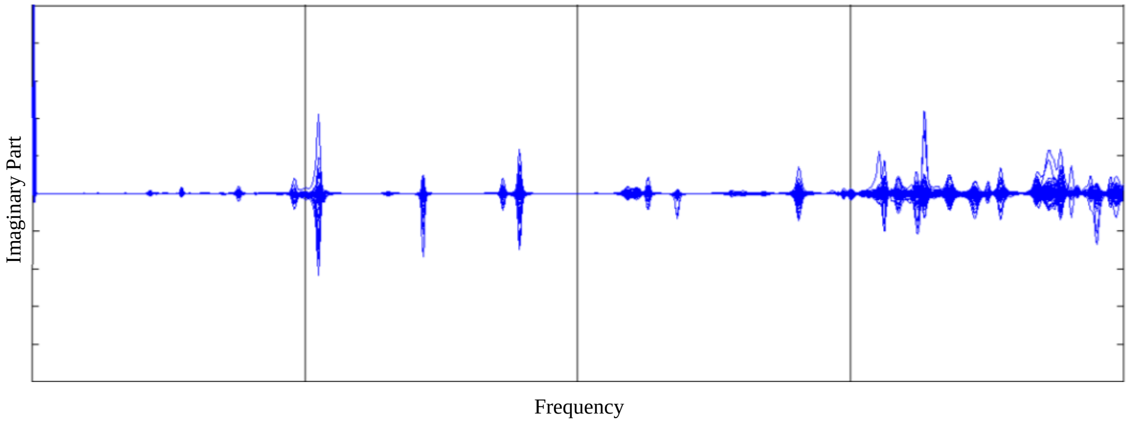

Figure 7: All measured frequency response functions for LSSF, typical hammer excitation measurement

The experimental modal analysis was primarily focused on the significant resonance peaks up to a defined frequency and provided about 20 elastic mode shapes of the LSF system. All in one, data with mostly good to very good confidence ratings could be extracted. Furthermore the corresponding mode shapes show plausible motions and are rather clean and undisturbed.

3.6 Comparison of ambient and hammer excitation

The corresponding elastic modal data identified from ambient and hammer excitation testing were compared. Table 1 lists the correlation results and figure 8 shows the MAC values of the mode shapes. All in one very consistent results could be obtained which increases the confidence in the identified modal data.

#	Ambient #	Hammer #	Freq. Dev. [%]	MAC [%]
1	9	1	-0.2	80.6
2	10	2	-0.6	92.3
3	11	3	0.1	92.3
4	12	4	0.2	98.6
5	13	5	0.2	49.3
6	14	6	0.1	94.8
7	15	7	0.1	84.7

Table 1: Correlation of ambient and hammer test

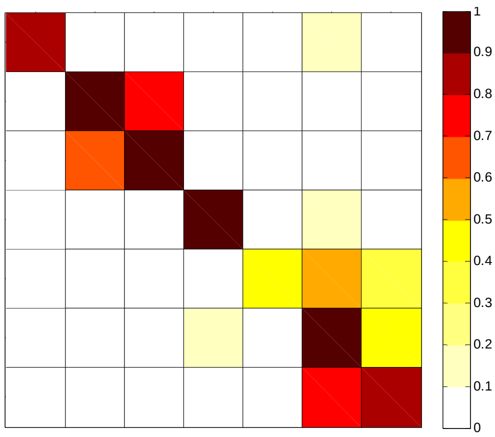

Figure 8: MAC matrix for ambient and hammer test

4 Model validation

4.1 Theory overview

The foundation for model validation is based on updating physical stiffness, mass and damping parameters by a parameterization of the system matrices according to equations (1) (see also [3-4]):

$$\mathbf{K} = \mathbf{K}_A + \sum \alpha_i \mathbf{K}_i \text{ , } i = 1 \dots n_\alpha \tag{1a}$$
$$\mathbf{M} = \mathbf{M}_A + \sum \beta_j \mathbf{M}_j \text{ , } j = 1 \dots n_\beta \tag{1b}$$
$$\mathbf{D} = \mathbf{D}_A + \sum \gamma_k \mathbf{D}_k \text{ , } k = 1 \dots n_\gamma \tag{1c}$$

with: $\mathbf{K}_A$, $\mathbf{M}_A$, $\mathbf{D}_A$ initial analytical stiffness, mass and damping matrices
$\mathbf{p} = [\alpha_i \; \beta_j \; \gamma_k]$ vector of unknown design parameters
$\mathbf{K}_i$, $\mathbf{M}_j$, $\mathbf{D}_k$ given substructure matrices defining location and type of model uncertainties

This parameterization permits the local adjustment of uncertain model regions. By utilizing equations (1) and appropriate residuals, which consider different test/analysis deviations, the following objective function can be derived:

$$J(\mathbf{p}) = \Delta\mathbf{z}^T \mathbf{W} \Delta\mathbf{z} + \mathbf{p}^T \mathbf{W}_p \mathbf{p} \rightarrow \min \tag{2}$$

with: $\Delta\mathbf{z}$ residual vector
$\mathbf{W}$, $\mathbf{W}_p$ weighting matrices

The minimization of the objective function (2) yields the desired design parameters **p**. The second term on the right hand side of equation (2) is used for constraining the parameter variation. The weighting matrix must be carefully selected, as for $\mathbf{W}_p >> \mathbf{0}$ no parameter changes will occur (see also [3]).

The residuals $\Delta\mathbf{z} = \mathbf{z}_T - \mathbf{z}(\mathbf{p})$ ($\mathbf{z}_T$: test data vector, $\mathbf{z}(\mathbf{p})$: corresponding analytical data vector) are usually nonlinear functions of the design parameters. Thus, the minimization problem is also nonlinear and is to be solved iteratively. One solution is the application of the classical sensitivity approach (see [4]). Here, the analytical data vector is linearized at point 0 by means of a Taylor series expansion truncated after the linear term. Proceeding this way leads to:

$$\Delta\mathbf{z} = \Delta\mathbf{z}_0 - \mathbf{G}_0 \Delta\mathbf{p} \tag{3}$$

with: $\Delta\mathbf{p} = \mathbf{p} - \mathbf{p}_0$ design parameter changes
$\Delta\mathbf{z}_0 = \mathbf{z}_T - \mathbf{z}(\mathbf{p}_0)$ test/analysis deviations at linearization point 0
$\mathbf{G}_0 = \partial\mathbf{z}/\partial\mathbf{p}|_{p=p_0}$ sensitivity matrix at linearization point 0
$\mathbf{p}_0$ design parameters at linearization point 0

As long as the design parameters are not bounded the minimization problem (2) yields the linear problem (4). The latter is to be solved in each iteration step for the current linearization point:

$$(\mathbf{G}_0^T \mathbf{W} \mathbf{G}_0 + \mathbf{W}_p) \Delta\mathbf{p} = \mathbf{G}_0^T \mathbf{W} \Delta\mathbf{z}_0 \tag{4}$$

For $\mathbf{W}_p = \mathbf{0}$ equation (4) represents a standard weighted least squares approach. Of course, any other mathematical minimization technique can be applied for solving equation (2).

In contrast to the assembly of the analytic stiffness and mass matrix, the generation of the analytic damping matrix is usually a difficult task. For treating system damping in an update process modal damping parameters can be utilized alternatively. For further discussions on this topic it is referred to the literature (see for instance, [3-5]).

Commonly, the eigenvalue and the eigenvector residuals are employed. Here, the analytical eigenvalues (squares of the eigenfrequencies) and eigenvectors are subtracted from the corresponding experimental results. The residual vector in this case becomes:

$$\Delta \mathbf{z}_0 = \begin{bmatrix} \lambda_{Ti} - \lambda_i \\ \cdots\cdots \\ \mathbf{x}_{Ti} - \mathbf{x}_i \end{bmatrix}_0 \qquad , i = 1,\ldots,n \tag{5}$$

with: λ_{Ti}, λ_i test/analysis vectors of eigenvalues
$\mathbf{x}_{Ti}$, $\mathbf{x}_i$ test/analysis mode shape vectors

The correlation between analytical data and test data is accomplished by means of the MAC value of the eigenvectors:

$$MAC := \frac{\left(\mathbf{x}_T^T \ \mathbf{x}\right)^2}{\left(\mathbf{x}_T^T \ \mathbf{x}_T\right)\left(\mathbf{x}^T \ \mathbf{x}\right)} \tag{6}$$

which states the linear dependency of two vectors $\mathbf{x}_T$, $\mathbf{x}$. A MAC value of one denotes that two vectors are collinear and a MAC value of zero indicates that two vectors are orthogonal.

The sensitivity matrix for the residual vector introduced in equation (5) is given by equation (7). The calculation of the partial derivatives can be found for instance in references [3-4].

$$\mathbf{G}_0 = \begin{bmatrix} \dfrac{\partial \lambda_i}{\partial \mathbf{p}} \\ \cdots \\ \dfrac{\partial \mathbf{x}_i}{\partial \mathbf{p}} \end{bmatrix}_0 \qquad , i = 1,\ldots,n \tag{7}$$

If real eigenvalues and eigenvectors are employed, the adjustment of damping parameters is not possible. The corresponding sensitivities equal zero since the real eigenvalues and eigenvectors depend solely on the stiffness and the mass parameters of the system.

4.2 Model validation strategy

Model validation is accomplished here through computational model updating (CMU) of physical parameters (stiffness and inertia parameters) of the FE model by minimizing the deviations between the identified and the analytical eigenvalues and mode shapes. It is presumed that all deviations between test and analysis are exclusively based on uncertainties of the FE model.

CMU is conducted by a special MATLAB® based software package (ICS.sysval, [2]). This software tool takes advantage of the analysis capabilities of MD/MSC.Nastran™, particularly the sensitivity module within 'Solution 200' (optimization), which enables the handling of large scale FE models. The necessary parameter changes are directly applied to the so called 'bulk data' section of the MD/MSC.Nastran™ input file. Typical parameters are for instance shell thicknesses, beam section properties, Young's moduli, and densities. However, virtually all physical parameters, which can be considered in an eigenvalue and eigenvector sensitivity analysis by MD/MSC.Nastran™, can be used for model updating.

After successfully updating the stiffness and inertia properties, physical or modal damping parameters can be adjusted subsequently by minimizing the deviations in the resonance regions between measured and simulated frequency response functions. However, this is not covered here.

A common difficulty in computational model updating is the proper choice of appropriate model parameters. Besides selection with engineering experience automated methods may be applied [6], which currently do not deliver a reliable prediction. Another possibility to select parameters is provided by a sensitivity analysis. Here the sensitivity matrix according to equation (7) is computed for a set of suitable parameters. In a subsequent investigation those parameters are identified, which have a significant influence on the analysis results. However, the sensitivity analysis does not supply any information on the physical relevance of a particular parameter but detects merely its potential to change the analysis results.

4.3 Model validation results

Model validation was conducted for the FE model of the LSF system as shown in figure 9. Utilizing the available EMA data an initial test/analysis correlation was performed in order to assess the quality of the initial model. The results are presented in table 2 and figure 10. Although the MAC correlation was already rather good, the correlation shows high frequency deviations. Thus further steps were taken to improve the situation.

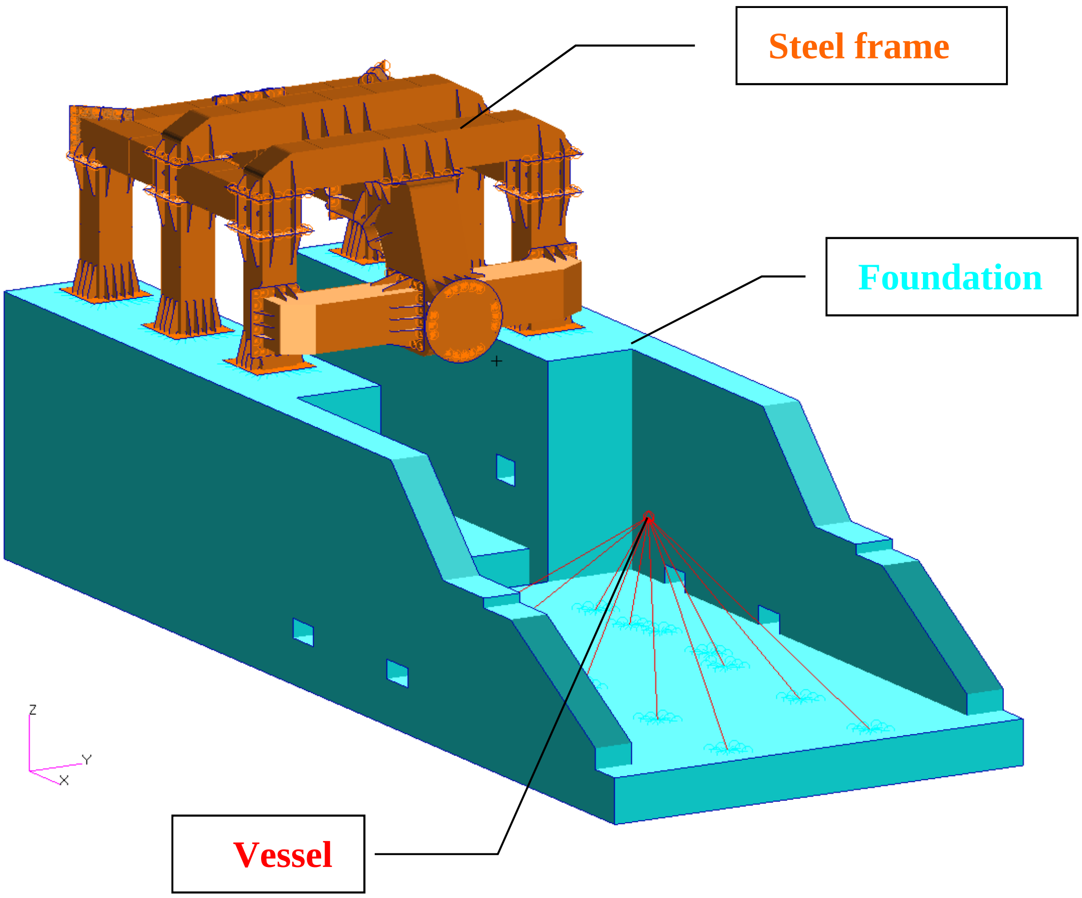

Figure 9: FE model of LSF system

#	EMA #	FEA #	Freq. Dev. [%]	MAC [%]
1	1	7	-27.38	90.50
2	2	8	-31.64	97.09
3	3	9	-29.77	93.49
4	4	10	-30.83	97.48
5	5	14	-18.86	95.06
6	6	13	-21.50	95.43
7	7	15	-18.61	82.74
8	8	16	-20.18	96.26
9	10	19	-17.92	96.32
10	11	20	-17.80	79.62
11	12	22	-14.09	73.69
12	14	25	-11.31	89.30
13	17	27	-13.25	82.94
14	18	26	-16.22	94.43
15	19	28	-11.55	77.02
16	21	32	-9.47	85.76
17	22	35	-6.42	83.08

Table 2: Initial test/analysis correlation for LSF system

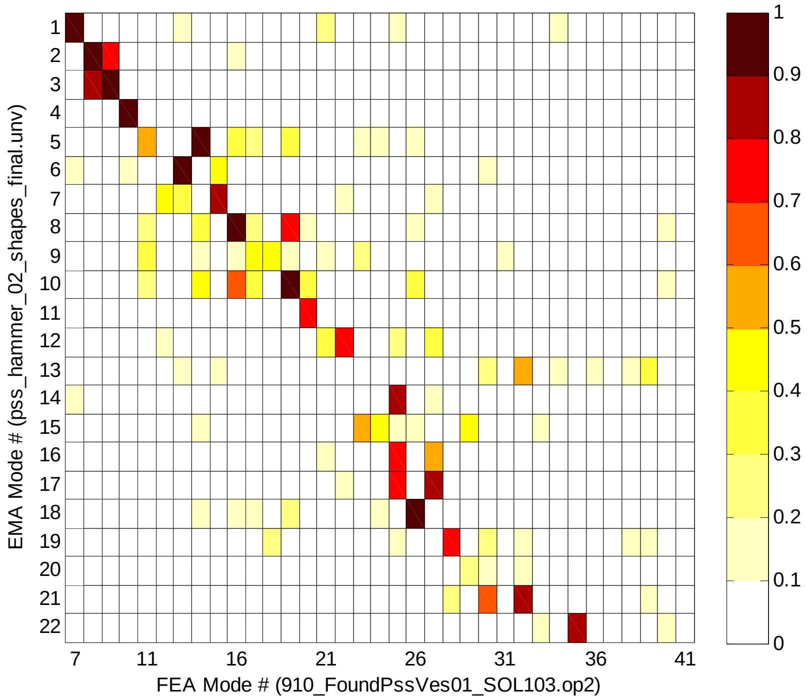

Figure 10: Initial MAC matrix for LSF system

To improve the test/analysis correlation, the initial model was at first remodeled before application of CMU. Then CMU was applied. After sensitivity analysis and parameter localization, several CMU runs were made with different promising parameter sets utilizing the ICS.sysval software. Finally six parameters were updated: figure 11 gives a survey of the model regions and properties taken into account during CMU. The obtained parameter changes were for some regions rather high. However, the corresponding regions were insecure with respect to their actual stiffness properties. To this extend the changes could be physically interpreted. Also, modifications to the FE model are not planned in the future, thus parameters can also be accepted that – to a certain extend – function as mathematical substitute parameters.

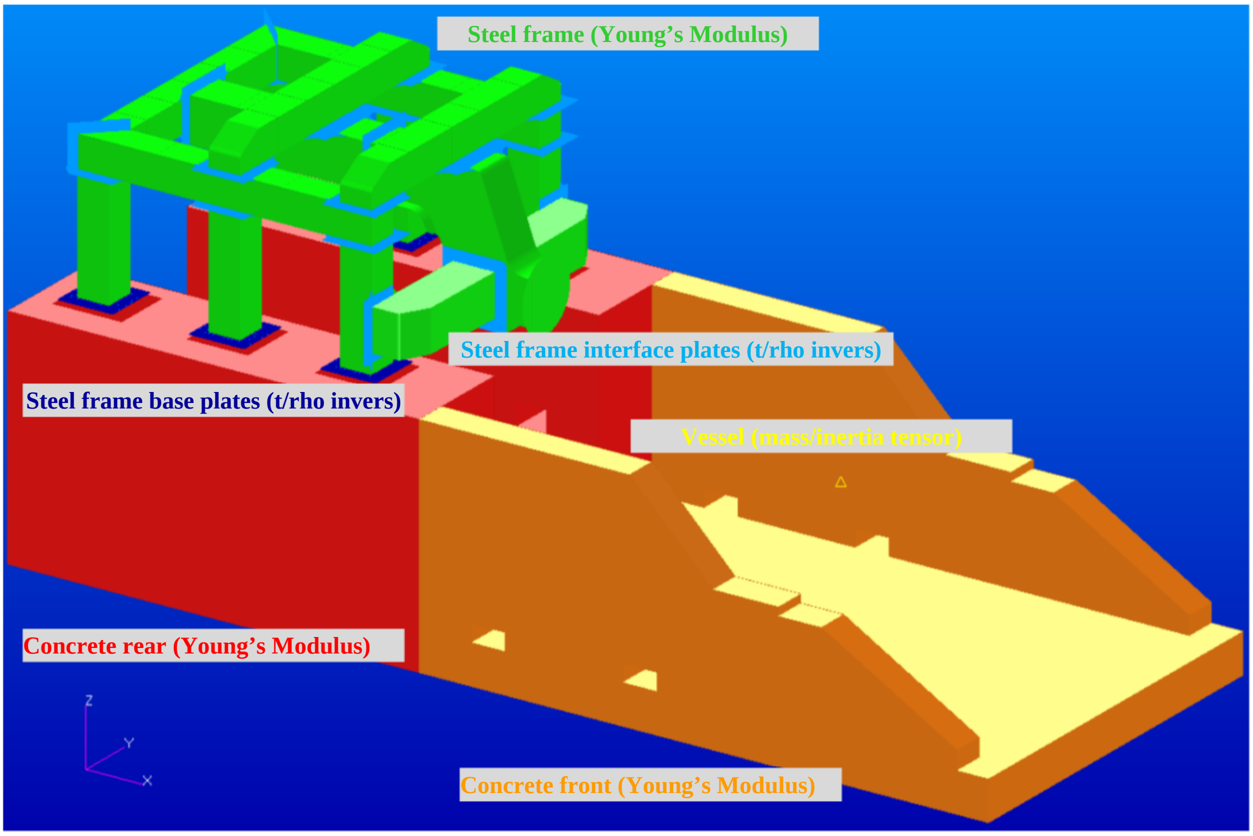

Figure 11: Overview of selected CMU parameters

The final correlation results for the LSF system after CMU are presented in table 3 and figure 12. It can be noted that the frequency deviations can effectively be reduced, and the MAC correlation is improved significantly. However, for the first two modes the percental frequency deviations are still not satisfactory. Considering the fact that the absolute deviations were smaller than 2 Hz, this is acceptable.

#	EMA #	FEA #	Freq- Dev. [%]	MAC [%]
1	1	7	-12.99	91.55
2	2	8	-8.55	97.97
3	3	9	-1.10	98.22
4	4	10	-3.08	99.15
5	5	13	-0.53	95.86
6	6	14	-0.89	97.27
7	7	15	-0.51	80.33
8	8	16	-3.39	97.94
9	10	19	-1.14	99.15
10	11	20	1.00	93.51
11	12	22	2.75	73.21
12	14	24	5.24	91.22
13	15	23	0.01	71.41

#	EMA #	FEA #	Freq- Dev. [%]	MAC [%]
14	17	27	3.58	94.99
15	18	26	-1.07	97.70
16	19	28	1.01	85.86
17	21	30	1.26	91.70
18	22	33	3.84	91.09

Table 3: Test/analysis correlation after CMU

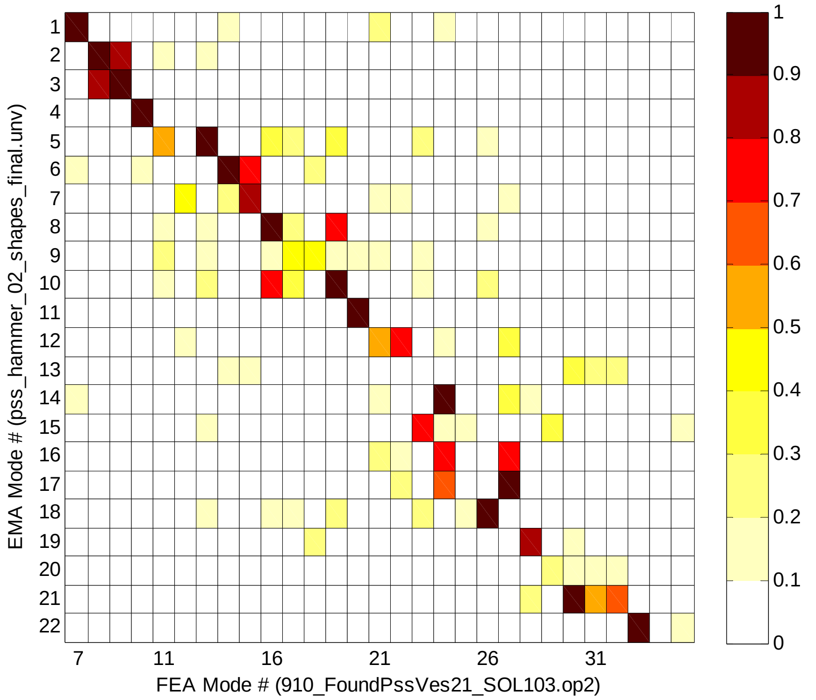

Figure 12: MAC matrix after CMU

4.4 Rigid body frequencies

The LSF system is mounted on an array of air springs, and the air springs are already incorporated in the FE model. Yet, the rigid body frequencies needed to be adjusted to the test frequencies obtained from ambient testing. This was accomplished by manually tuning the spring stiffnesses such that the lateral and vertical rigid body frequencies of the test are met. However, a clear pairing of the rigid body modes from test and analysis could not be achieved. A possible explanation may be that some test results were rather insecure because of excitation difficulties, especially for the lateral directions.

5 Summary and conclusion

This paper presented a two-step testing approach and model validation results for a new large spinning facility (LSF) that has been built in Germany. The employed two-step test procedure provided consistent test data and valuable insight into the rigid body and elastic dynamics of the LSF system. Model validation utilizing computational model updating techniques in addition yielded a FE model of high fidelity. All in one the combination of ambient and hammer excitation testing proved to be very effective for the analyzed large scale LSF system.

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