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Seismic Qualification of an Electrical Cabinet: Comparison of Analysis and Test Results

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ABSTRACT

In this paper, the seismic behaviour prediction for a safety-related electrical cabinet with respect to its stability by analysis is compared with the results of a successive test that was performed with the same cabinet. Based on a detailed finite element model of the cabinet, its modal properties were numerically determined. A response spectrum modal analysis (RSMA) calculates the cabinet's maximum responses (e.g. stresses) for the given response spectra ($D = 5$ %) at the installation site. After completion of analysis, the whole cabinet was build and tested on a triaxial shaking table. The paper compares e.g. modal properties of the FE model with results from the resonance search runs performed during the seismic test. Differences in global and local eigenfrequencies and damping values are found and discussed. Further comparisons are made based on transfer functions for two sensor locations. The result confirms the perception that numerical analysis leads to results that are generally more on the 'safe side', especially when it comes to qualification of components for NPPs with respect to stability/integrity. By testing the conservativities that can be more 'squeezed out', thus higher seismic loads can usually be qualified by this method.

INTRODUCTION

Electrical cabinets are often essential equipment for the safety of nuclear facilities. Thus the cabinets have to successfully pass several qualifications. One of these qualifications is to prove their resistance against seismic events. For this purpose, several methods are available (Henkel 2005, Ries 2016). The most frequently used methods are analysis and testing. Other viable qualification means are similarity considerations or experience-based methods. If only the structural stability has to be proved, usually proof by analysis or similarity considerations are performed. If additionally the cabinet's operability during / after an earthquake has to be shown, the qualification has to be done by performing a test. In principle there are two types of tests that can be run. The most obvious test is putting the whole cabinet on a shaking table. Another approach would be designing a test that only shakes the cabinet's safety-related electric modules. However, to keep time and costs low, usually only one type of qualification is performed. This means if a cabinet's stability is qualified by analysis, it will usually not be tested anymore and if it was tested, it will not be analysed any more. In the presented paper, a cabinet's seismic stability was intended to be qualified by analysis. After completion, an additional test was performed with the cabinet. This gives the opportunity to compare results from the preceding analysis with the following test. This means there was no option of tuning parameters of the finite element (FE) model in order to get a better agreement between model and test results.
The following chapter will introduce the considered cabinet. The subsequent chapter will address the FE model's modal properties and its seismic loading followed by a chapter describing the test and the test results. A comparison of modal properties determined during calculation and resonance search test run follows. In the last chapter, a conclusion with respect to comparison of calculation and test results is given.

CONSIDERED CABINET

The electrical cabinet is installed in a nuclear power plant (NPP). According to the cabinet's classification, its stability and operability during / after a seismic event has to be shown.
A scheme of the cabinet is depicted in Figure 1. Its dimensions are (WxDxH) 1204x804x1920 mm. The weight is approx. 950 kg. The cabinet is mounted in the building by welding it to anchor plates.

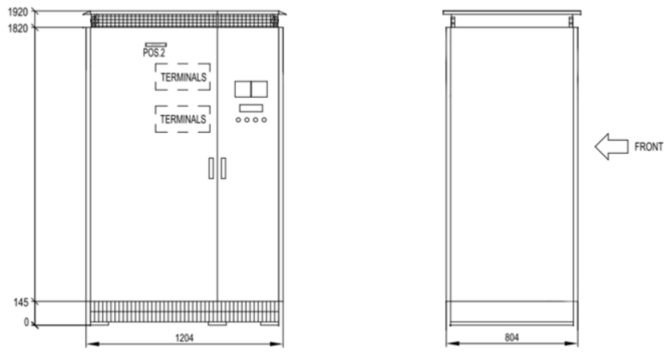

Figure 1. Scheme of cabinet

NUMERICAL ANALYSIS

The first step of the seismic qualification was to setup a numerical model of the electrical cabinet. For this purpose Finite Element (FE) analysis is used. As the seismic analysis was supposed to be performed by response spectrum modal analysis (RSMA), a linear elastic model had to be set up. By using the model's modal properties (eigenfrequencies, eigenvectors) in combination with the seismic loads (response spectra), the RSMA determines e.g. the maximum stresses, accelerations or section and reaction forces / moments in the cabinet's elements. The values are combined with the results from a dead load calculation. The seismic verification is accomplished by checking these values against allowable values.

FE modeling

The structural dynamic behaviour of the cabinet is modelled by using structural (lumped mass, spring), beam, shell and solid elements to form the whole cabinet. The elements are chosen and combined in such a way that they represent the cabinet's global and local load carrying behaviour.

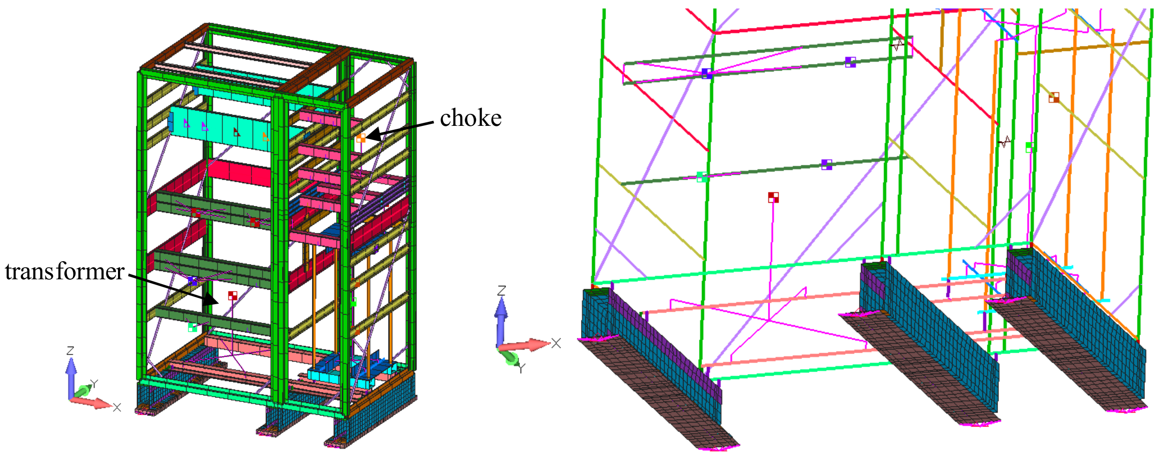

Figure 2. Finite Element Model of Cabinet: Overview (left), Detail (right)

The pre- and postprocessing of the model was done with the software Femap. The created FE model is shown in Figure 2. The left plot gives an overview of the whole model; the right plot shows details from the model's lower section. The main elements used for modeling are beam elements (cross-section not shown in right plot). To get the load path from the cabinet's frame into its three sockets (base profiles) and attached base plates down to the welding seams correctly, the base profiles and base plates are modeled in more detail by using shell elements. The welding seams at the ends of the base plates are idealized by boundary conditions. The built-in electric components are generally considered by lumped mass elements at their center of gravity, attached by rigid elements to the cabinet structure at their mounting points. This was e.g. done with the heavy transformer (approx. 300 kg, see Figure 2) arranged in the bottom part of the cabinet and a choke (approx. 90 kg, see Figure 2 & Figure 5) which was mounted in the upper region of the cabinet. This seismically unfavorable position of a heavy component had to be chosen due to space limitations in the cabinet.

Modal properties

For numerical analysis the software Stardyne was used. As the ZPA region of the enveloped spectra starts at about 70 Hz (see Figure 4), eigenmodes up to this frequency have been determined. Results from the modal decomposition are given in Table 1. The table gives selected modes up to 30 Hz. For eyample mode 1 with f_1 = 9.9 Hz is related to a global vibration of the cabinet in the horizontal X-direction (side to side). Mode 2 (f_2 = 14.8 Hz) is a global front-back motion (see Figure 3).

Table 1. Modal properties of cabinet (only modes with 'high modal weights')

Mode *[#]*	Freq. *[Hz]*	Modal weights MW_X *[N]*	MW_Y *[N]*	MW_Z *[N]*	Participation factors Γ_X *[-]*	Γ_Y *[N]*	Γ_Z *[N]*
1	9.9	4227.0	69.5	0.6	1.87	-0.24	-0.02
2	14.8	28.6	3010.7	0.1	0.12	1.24	-0.01
4	18.3	1557.8	103.7	1.6	1.16	0.30	-0.04
6	21.0	556.5	1.0	4.6	1.93	-0.08	-0.18
10	25.3	1713.4	156.4	93.5	-1.20	0.36	0.28
11	27.3	66.7	844.4	2.7	0.43	1.54	-0.09
13	28.8	10.2	2875.2	3.3	0.07	1.09	0.04

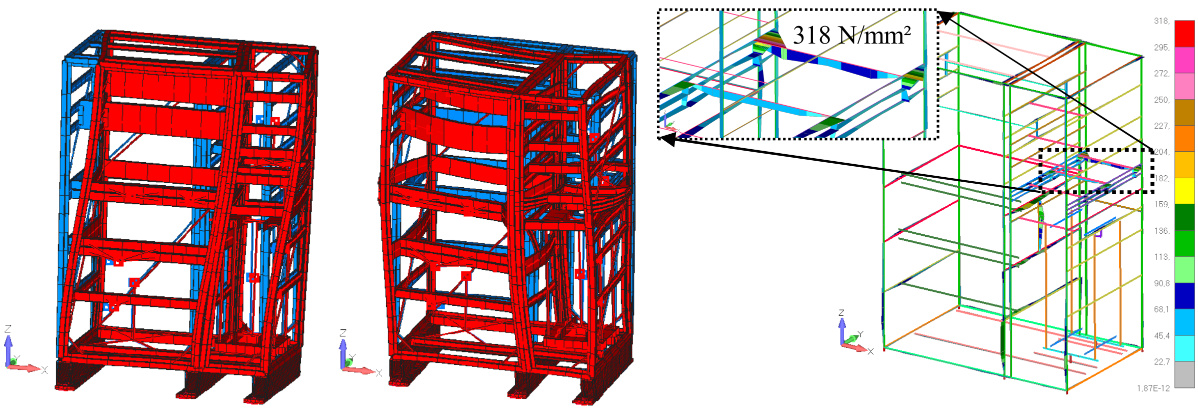

Figure 3. Mode Shape of mode #1 'side to side' (left); mode #2 'front to back' (center); comparative stress in choke's support (right)

Loading & Results

The seismic load acts simultaneously in the three directions of space. So the enveloping horizontal spectrum from Figure 4 is applied in each horizontal direction simultaneously together with the enveloping vertical spectrum. Usually no damping values are known for structures that are not built yet, but based on data that was available from former tests with similar cabinets, a value of D = 5 % was conservatively selected for the analysis. Furthermore, neither a response modification factor (R) nor an importance factor (I) was additionally considered. So for verification the given elastic floor response spectra have to be applied directly.
According to the applicable code and seismic specification, additional pre-factors for combining the seismic loads with gravitational load (dead load) have to be considered for load and resistance factor design (LRFD).
The cabinet is installed at several locations in one of the NPP's buildings. For the locations, two seismic loadings are defined: The seismic design response spectra (DRS) and the high frequency response spectra (HFRS). To reduce qualification steps, both spectra are enveloped, which is very conservative. The result for the horizontal and the vertical seismic loading used for numerical analysis is shown in Figure 4.

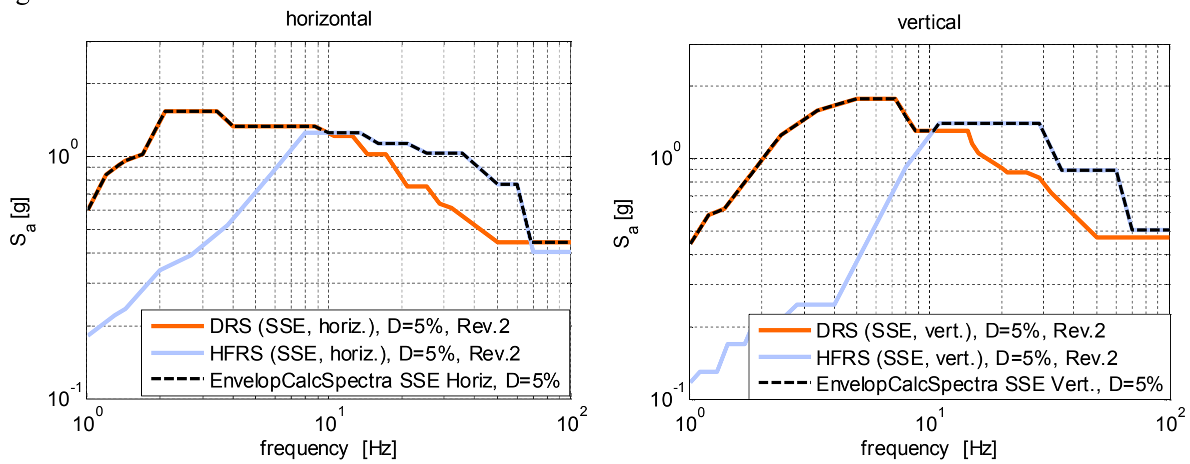

Figure 4. Enveloping spectra used for numerical analysis, D = 5 %

From the plots in Figure 4 it can easily be seen, that the HFRS peaks are lower than the peaks of the DRS, but the frequency content covers a wider range starting from about 10 Hz to its Zero Period Range, which begins at approx. 70 Hz. The enveloping spectra have a peak acceleration of 1.54 g (horizontal) and 1.77 g in vertical direction. The zero period acceleration (ZPA) of the enveloping horizontal spectrum is 0.44 g and approx. 0.51 g for vertical direction.

In the RSMA the contribution of the modes are combined by complete quadratic combination (CQC). The contributions of modes above 70 Hz are considered as missing mass.
Calculation shows that all displacements, comparative stresses (von Mises) and forces are well below the allowable values. Only in the supporting profiles of the choke (see Figure 2 right) the allowable values are significantly exceeded (usage of 195 %). Despite several viable improvements (model refining, possible design changes within given limits), this was the lowest usage that was achieved. With this result, it was not possible to verify the seismic resistance of the whole structure with the performed numerical analysis.
However, due to the profile's material (1.0226) which shows a highly ductile behavior, it could be assumed that the profile could withstand the seismic loading. This assumption was not further considered, as additional seismic tests with the cabinet were planned. So the improvement measures of the cabinet's weak spots were incorporated in the final cabinet design and a prototype for testing was built.

TESTING

By testing, besides stability/integrity, also the operability during / after a seismic event can be verified. The tests are performed on a threeaxial hydraulic shaking table according to IEEE 344 and the project specification.

Device Under Test

The device under test (DUT) is identical to the cabinet that will later be installed in the NPP. Figure 5 shows the DUT on the shaking table. The mounting conditions in the NPP are simulated by welding the cabinet's base plates to steel plates that are clamped to the shaking table.

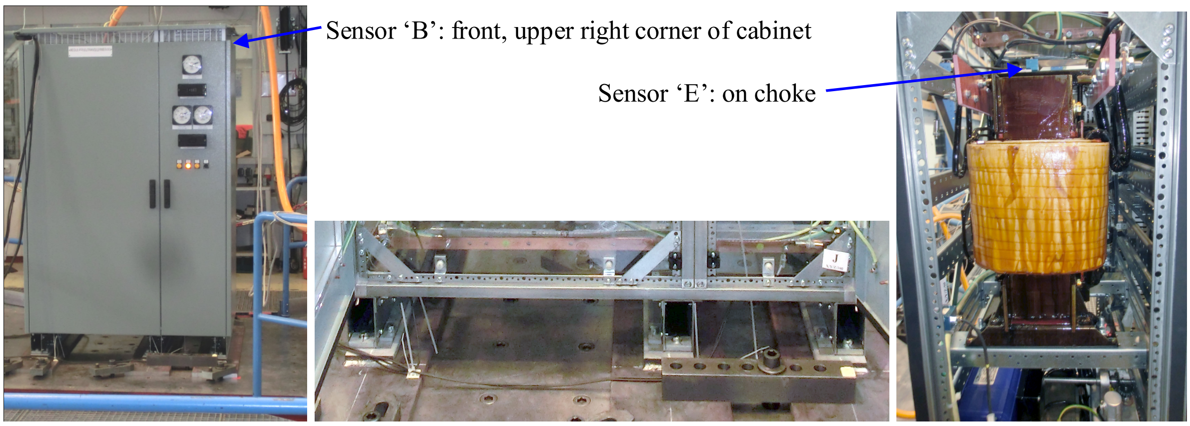

Figure 5. Cabinet (DUT) on shaking table (left); Detail mounting (middle); choke in cabinet (right); (coordinate system is identical to numerical analysis, see Figure 2)

During the test, the DUT can be electrically operated to check the safety functions. The DUT is equipped with several accelerometers to monitor its vibrations response.

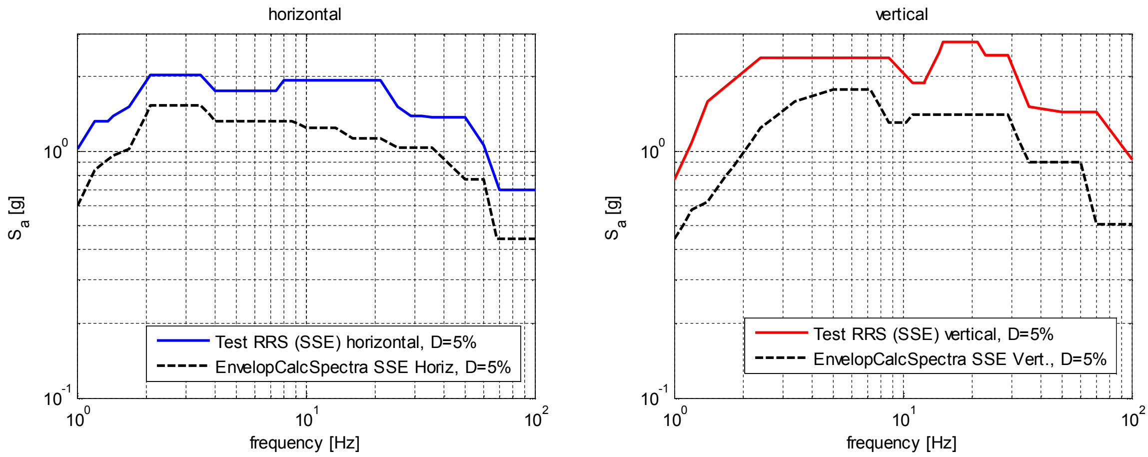

Figure 6. Comparison of spectra used for analysis and for testing (all margins included), $D = 5$ %

Seismic Loading in Test

The test has to be performed according to a newer revision of the project documents. As these documents include new spectra, the spectra used for numerical analysis (see Figure 4) cannot be applied.
A comparison of the calculation spectra with the 'new' test spectra is shown in Figure 6. The spectra contain all margins according to standards and specification.

Tests

As in the numerical analysis, the first step of the (mechanical) test sequence is the determination of modal properties. This is done be resonance search tests of each axis sequentially. Afterwards, five operational basis earthquakes (OBE) are performed by time history testing of all three axes simultaneously. The seismic loading of a OBE is half of the safe shutdown earthquake (SSE) level. The final step is testing at SSE level. The tests end with resonance search runs in order to check for changes in the modal properties due to the preceding seismic loadings.
The performed test sequence is:

- 3 x resonance search tests:
  - uniaxial sine sweep (separately in x-, y- and z-direction)
  - frequency range 1 to 70 Hz
  - amplitude 2 m/s²
  - sweep rate 1 octave/min.
- 5 x OBE tests:
  - time histories (3 axes simultaneously)
  - required response spectrum (RRS) is 0.5 times SSE-level from Figure 6
  - duration 30 s
  - check of operability of DUT
- 1 x SSE test (see OBE-test, as RRS the SSE-spectra found in Figure 6 are applied directly)
- 3 x resonance search test (see above)

As acceptance criteria, the test response spectra (TRS) have to envelop the RRS. In addition to checks before and after the tests, the operability during the seismic events has to be proved. Also the stability/ integrity of the cabinet structure has to be given.

Table 2. Global eigenfrequencies of the DUT before and after the seismic tests (source: test protocol)

Sensor	Resonance search before seismic tests		Resonance search after seismic tests	
	resonance [Hz]	critical damping *D* [%]	resonance [Hz]	critical damping *D* [%]
Bx	8.8	6	8.4	6
By	13.1	7	13.0	7

Based on the resonance search tests, the global eigenfrequencies and damping are determined from the resonances of the sensor B (triaxial accelerometer). For location of sensor B, see Figure 5. The results are shown in Table 2.
In x-direction (side to side) an eigenfrequency of 8.8 Hz is determined before the seismic test. After all tests the frequency drops slightly to 8.4 Hz, the damping value determined by the test laboratory stays constant at 6 %. In y-direction (front to back) the resonance starts at 13.1 Hz, after the test it is almost unchanged at 13.0 Hz. The damping value stays constant at 7 %.
As an example of the seismic loading the comparison of TRS- and RRS-spectra from the SSE-test are shown in Figure 7.

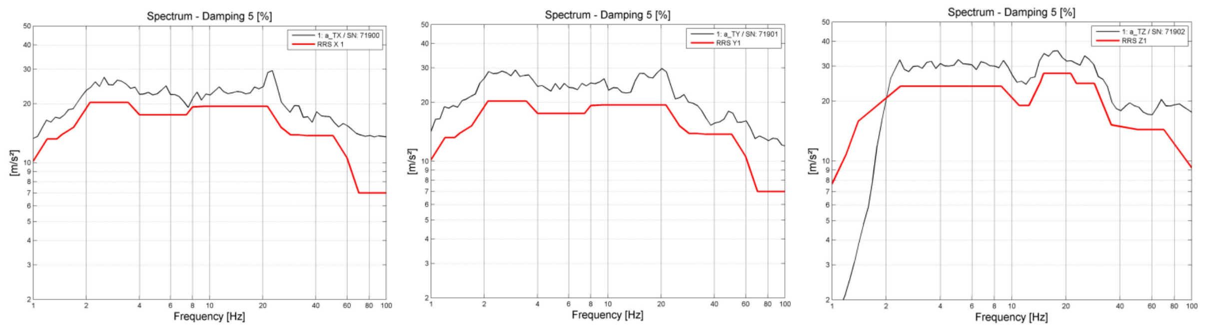

Figure 7. SSE-Test: Comparison of TRS and RRS in x-, y, and z-(vertical)-direction, D = 5%
(source: test protocol)

As the test met all acceptance criteria, the cabinet is seismically qualified for operation at its installation locations in the NPP.
Interestingly the numerical analysis showed a significant overloading in the choke's support profiles (see Figure 3). However the test showed no stability or integrity problem with these profiles – although the test loads were significantly higher than the seismic loads applied for numerical analysis (TRS > RRS > spectra used for numerical analysis; see Figure 6 and Figure 7). This example shows conservativism which is usually incorporated in a (linear elastic) numerical calculation used for verification of components.

COMPARISON OF NUMERICAL ANALYSIS AND TEST

The cabinet was numerically analysed before the test, so there was no chance of calibration of the FE model by using test data ('model updating'). Based on this, a comparison between numerical analysis results and test data could give an insight in the deviation of the FE model and the real structure.
In total 8 tri-axial accelerometers were distributed over the cabinet, so data based on acceleration time records can be used for comparison. For analysis, sensor 'B' (attached at the upper right corner at the frame of the cabinet, see Figure 5) and sensor 'E' (attached on top of the built-in choke, see Figure 2 & Figure 5) will be considered.
Firstly the global horizontal eigenfrequencies are compared. In the analysis step the lowest eigenfrequency in x-direction (side to side) was predicted to be 9.9 Hz (see Table 1), the resonance search runs reads 8.8 Hz (see Table 2). So the predicted value is +12.5 % higher. In y-direction (front to back), an eigenfrequency of 14.8 Hz was predicted. The measured value gives 13.1 Hz. Again, the prediction is higher (+12.9 %). This means the FE model is stiffer than the real structure with respect to its global eigenfrequencies. These higher frequencies bring along a reduction in loading, approx. -5 % in each horizontal direction (compare spectra in Figure 4).
Secondly a global modal damping value of D = 5 % was assumed for the whole structure. The evaluation of damping values from the resonance search test runs (see Table 2) indicates a higher damping value of 6 % respectively 7 %. This indicates that lower spectra could be used. Here it must be stated that the determination of damping is an estimation (see below). As usually only single degree-of-freedom methods are applied, the results depend strongly on the applied method an e.g. spacing of the modes.
An additional insight can be achieved by comparing transfer functions. From the resonance search runs transfer function (base acceleration to acceleration response at the sensor positons) were calculated. By using the original FE model these transfer functions can numerically be determined by harmonic response analysis. For this analysis a modal damping of 5 % for each mode was assumed in accordance with the 5 % spectra.
Figure 8 shows a comparison of the transfer functions for sensor B. In the left plot the comparison between the calculation (blue line) and test (red line) is plotted for the x-direction. It can be seen from the first resonances in the figure that the calculated lowest mode in x-direction (9.9 Hz) is slightly higher than

the measured frequency (8.8 Hz). Furthermore the calculated amplification is 18.7/8.5 = 2.2 times higher than the measured one. By determining the damping values using the 3dB-bandwidth method (e.g. Brandt 2011) better results are achieved. For the calculation a value of 5 % is determined from the transfer function, matching the modal damping used in calculation. From the measured red curve a damping of ~11 % is determined for the tested cabinet. Thus much more damping is involved in this mode in the real structure than in the model. The second resonance is at ~18.2 Hz, matching the 18.3 Hz eigenmode from Table 1. The overall shape of the transfer function of Sensor B in x-direction shows that the model envelops the measured results in the important low frequency region clearly.

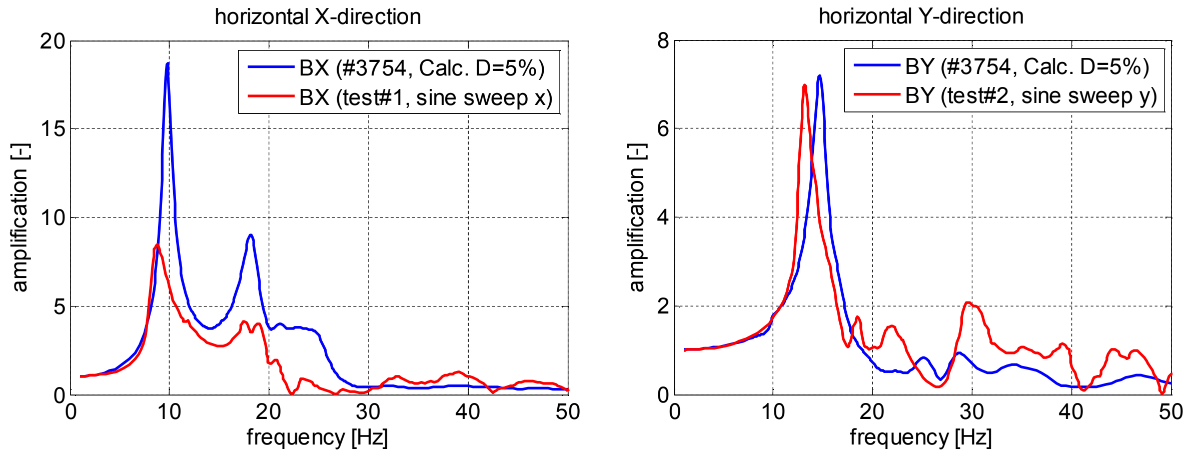

Figure 8. Comparison of transfer functions from calculation and test at Sensor B (cabinet frame, top): x-direction (left), y-direction (right)

When it comes to the y-direction (Figure 8, right plot) the effect of the slightly higher first resonance can be seen from the plot. The corresponding ratio of amplification values of the first resonance is 7.2/7 = 1.03 indicating almost same results. From this it can be concluded that the damping values of this mode almost match. Above 20 Hz no further distinct second resonance of the whole cabinet in y-direction up to 50 Hz is found. The 'performance' of the model at Sensor B in y-direction is assumed to be ok.

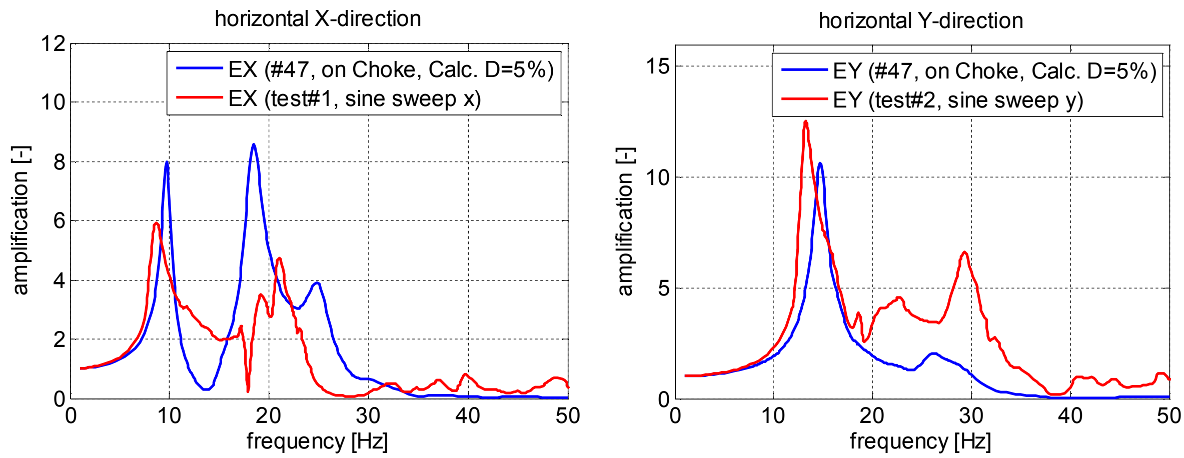

Figure 9. Comparison of transfer functions from calculation and test at Sensor E (on choke): x-direction (left), y-direction (right)

As addressed in chapter 'numerical analysis' a problem in the verification of the choke's support structure occurred. For this reason a comparison of transfer functions at sensor 'E' is made. Figure 9, left plot,

gives the picture for the x-direction. The slight difference in the first eigenfrequencies can be detected; the amplification factor of the model is higher than in the test: 8/5.9=1.36. The y-direction's second mode at 18.3 Hz is distinctive in the calculation but not in the test. However the model seems to capture the global characteristics and is 'enveloping' the test results in the frequency range up to 30 Hz.
In the y-direction of sensor E (Figure 9, right) we see the effect of the stiffer model in the higher first resonance. Interestingly the amplifications of the choke (sensor E) are higher than the cabinet's amplifications in y-direction (see sensor B, Figure 8, right) this means the choke performs more motion than the cabinet in this direction. This is due to the 'soft' support of the choke in the cabinet. When looking at the frequency range from 20 Hz to 40 Hz, the model does lack some dynamics. However, it has to be mentioned that the calculation results are given at the center of gravity of the cabinet whereas the measurements were taken from the top of the choke.

Further improvement / comparison steps

To improve the FE model the following steps could be performed:

- Experimental modal analysis (EMA) of resonance search runs, identifying eigenfrequencies, modal damping values and mode shapes based on curve fitting
- Comparison of eigenvectors from EMA and calculation by e.g. modal assurance criterion (MAC)
- Update of FE model

The updated model could then be used to perform additional calculations, the results of which could be compared with the test records. For example time history modal analysis simulation using modal damping values from the EMA and the time history records of the shake table motion during the SSE test. From these results in-cabinet response spectra at sensor locations could be determined and compared with in-cabinet spectra from the test.

CONCLUSON

From the comparisons above it is found that the FE model of the global horizontal dynamics is slightly too stiff, a maximum difference of +12.9 % in y-direction was found. Due to the shape of the spectrum this gives a reduction of ~5 % in seismic loading. When it comes to damping, the assumed D = 5 % seems to be a conservative value for the considered cabinet's global dynamics. Less conservative damping values could be determined by evaluating frequency response measurements – it has to be noted that the measurement has to be performed at an excitation level that is similar to the seismic loading, because damping increases as loading increases. Having in mind that the linear elastic FE model was set up without knowledge of the real cabinet's behaviour (blind forecast), the global model is assessed to be good. When it comes to local dynamics within the cabinet (e.g. choke), the model is acceptable.
The example showed that for the considered cabinet a qualification by analysis was not possible, but a qualification by test was successful – even with significantly higher seismic loads. This indicates that a calculation comprises more conservativities than an actual test. The reasons for this can be found in the fact that e.g. for seismic qualification of NPP components no ductility values are applied with the seismic load, conservative damping values are chosen and the allowable values do not allow plasticity. Thus the calculation is usually clearly on the 'safe side'. So with respect to stability or integrity usually higher loads can be proved by testing than by calculation.

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