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EXPERIMENTAL MODAL TEST OF THE LABORATORY MODEL OF STEEL TRUSS STRUCTURE

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Abstract

The experimental modal analysis is often used to validate the accuracy of dynamic numerical models. It is also a good tool to obtain valuable information about current condition of the structures that could help to determine residual lifetime. The quality of modal testing results is highly dependent on the proper estimation of the natural frequencies from the frequency response function. This article presents the experimental modal test of the laboratory steel structure in which the natural frequencies and mode shapes are determined.

Keywords:

Experimental modal analysis;
Mode shapes;
Fourier transformation;
Frequency response function;
Natural frequencies.

1. Introduction

The laboratory model of steel space truss structure was constructed in the laboratory of the Department of Structural Mechanics and Applied Mathematics. Its main purpose is to use it for the calibration of the assembled measured system, especially in the case when a new device is expected to be applied as a part of the measured system. The proper synchronization between the old and new parts of the measured system is crucial to obtain valuable measurement of the mechanical dynamic signals [1]. Inaccuracies in the measurement can be easily recognised if the measurement is done on the well know structure such as the laboratory model. The laboratory model is also used to teach students the basic principles of experimental modal analysis [2]. Being successful in both applications expects that the modal parameters of the structure have been determined with the adequate accuracy. For this purpose, the measurement of the presented laboratory structure was done. The results are also compared with the results obtained from the numerical model to ensure that the estimation is correct. This comparison shows that there are also peaks in the frequency response functions which cannot be considered as natural frequencies of the steel structure.

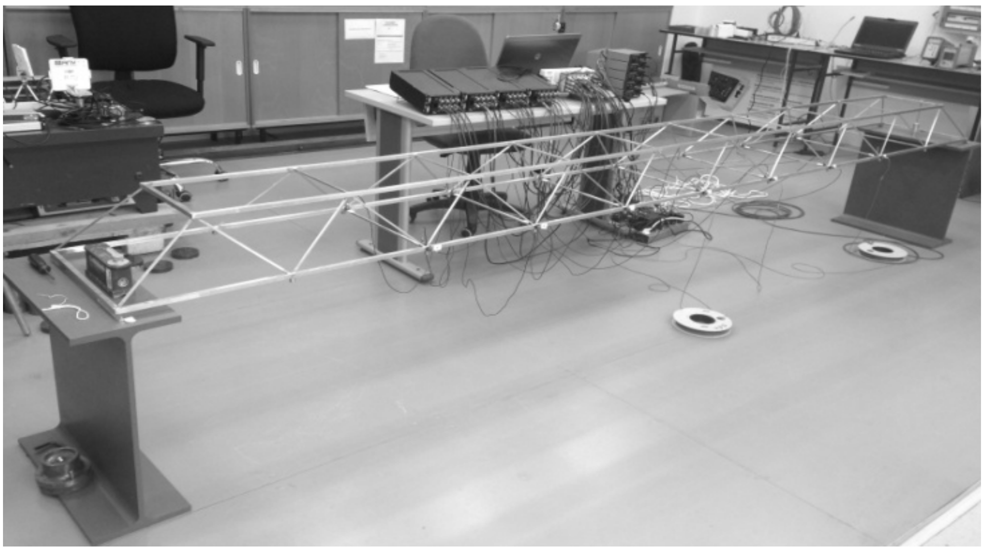

Fig.1: Laboratory model of the steel space truss structure.

2. Description of the laboratory and numerical model

The laboratory steel space truss structure is made of stainless steel with the Young modulus 200 GPa and the density 8000 kgm^{-3}. The structural elements with rectangular cross-sections are joined together by welding. The structure stands on two steel profiles that are considered as two rigid bodies. There are supports for the truss structure on the top of both. One of them is a steel triangle which is considered as a hinge that only allows the structure to rotate. The second type of the support that is located on the next site is made of two small rollers fastened to the truss structure. This type of support allows the structure to rotate but also to elongate or contract (Fig. 2).

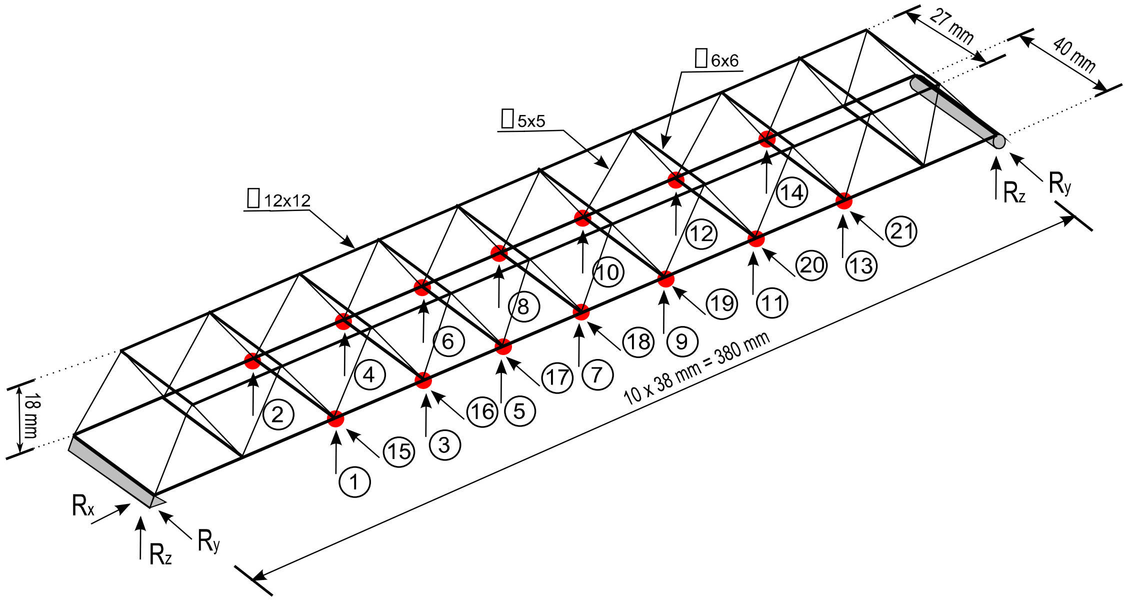

Fig. 2: The geometry of the steel truss structure with the location of the measured points.

The accelerometers that are used to measure the response of the structure are located at the joints between the diagonal braces and bottom chords. There are measured vertical acceleration at 14 points and horizontal acceleration at 7 points. The structure is excited with a modal hammer by hitting it at the degrees of freedom 20. This horizontal direction of excitation is used with regards to the torsional mode shapes that were identified by a numerical solution.

The numerical model has the same dimensions as the real laboratory model structure. There are beam elements with appropriate cross-sections used in the model. The isotropic material performance is comparable to the stainless steel material as it has the same Young modulus. The weight of the welds in joints is taken into account by increasing of the density of the material to the 8100 kgm^{-3}. The weight is also increased with the mass elements that define the weight of accelerometers and cables. There is expected that the weight of each accelerometer with cable is 15 g. The boundary conditions cover the ideal behaviour of real supports so their imperfections are neglected as well as their possible nonlinear behaviour.

3. Experimental modal analysis

The natural frequencies and mode shapes are estimated via the common techniques that are used in the Experimental modal analysis. The scheme of the used measured equipment and accelerometers is shown in the Fig. 3.

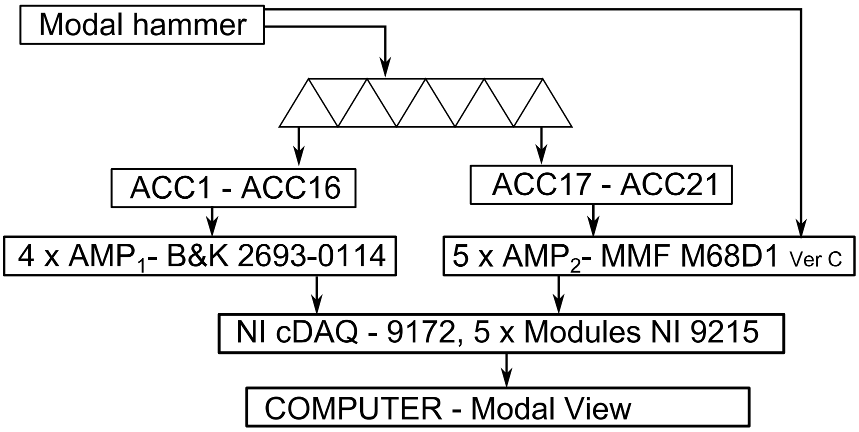

Fig. 3: Scheme of measured system used to measure the response of the structure excited with the modal hammer.

The proper measurement of the input and output is important to obtain considerable results. For this purpose, there is paid special attention to excitation of the structure with the impulse in which only one hit is detected. Examples of valuable measured signals of force excitation and the obtained acceleration response of the structure are shown in the Fig. 4.

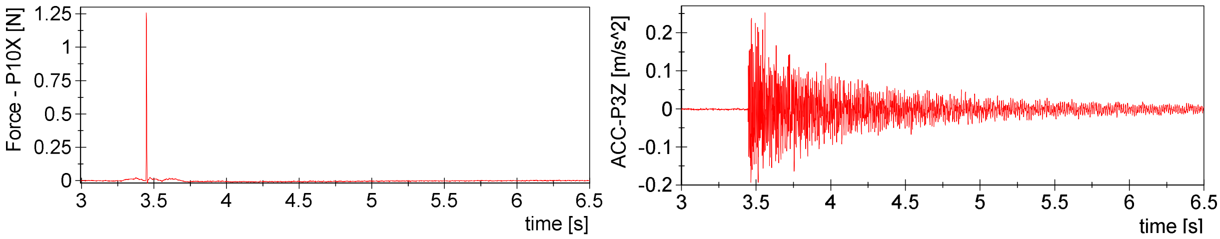

Fig. 4: The example of measured signal from the modal hammer and one of the accelerometers located on the structure.

Sampling rate is set to 256 and the block size to 1024. It means that four second of the measured signal is used to obtain frequency response function which is cross power spectrum (Sxy) of the input and output divided by the auto power spectrum (Sxx) of the input. It is used to estimate the natural frequencies (Fig. 5). The determined frequency response function is also used to estimate mode shapes for the relevant natural frequencies.

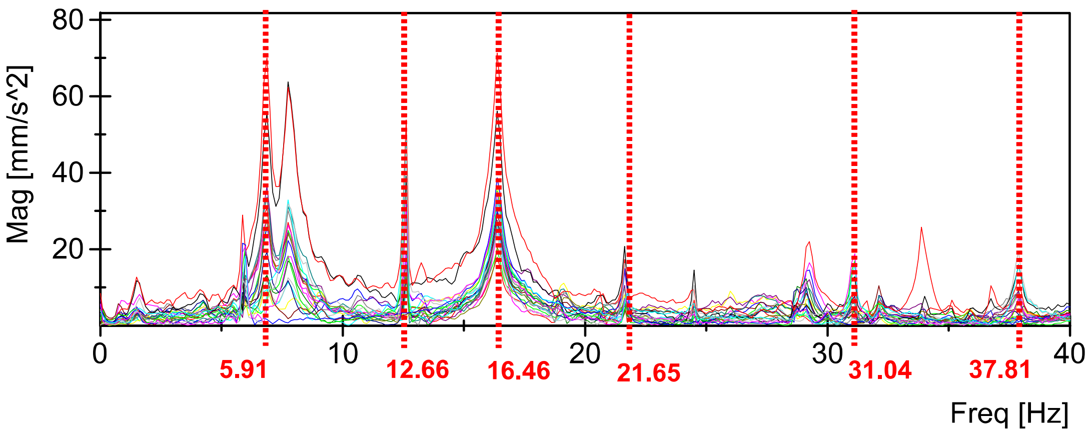

Fig. 5: Estimation of the natural frequencies from the Frequency response function.

4. Comparison of the results

The results from the measurement are compared with the results from the solution. The natural frequencies obtained from the measurement have a little bit smaller values than the frequencies calculated from the numerical solution. It means that the numerical model is stiffer than the laboratory model. There is also identified a peak in each frequency response function with the value 16.46 Hz that is not identified in the numerical solution as one of the natural frequencies.

Table 1: The comparison of the results of numerical solution and results of the experimental modal analysis.

Number of the identified frequency	Results of the experimental modal analysis	Results of the numerical solution	Description of the mode shape
1	5.91	5.97	Torsional and horizontal bending
2	12.66	12.97	Torsional and horizontal bending
3	16.46	not identified	-
4	21.65	21.65	Torsional and horizontal bending
5	28.15	29.89	Torsional
6	31.04	31.30	Torsional and horizontal bending
7	36.32	36.32	Torsional and horizontal bending

The mode shapes identified from the experiment (Fig. 7) are also compared with the numerical solution (Fig. 6). There are compared only torsional and horizontal bending-torsional modes with regards to the excitation which is only done in the horizontal direction. The vertical bending modes are not exactly identified from the modal test so they are not presented in the comparison. The excitation would have been changed to the vertical direction if the vertical mode shapes had been identified properly.

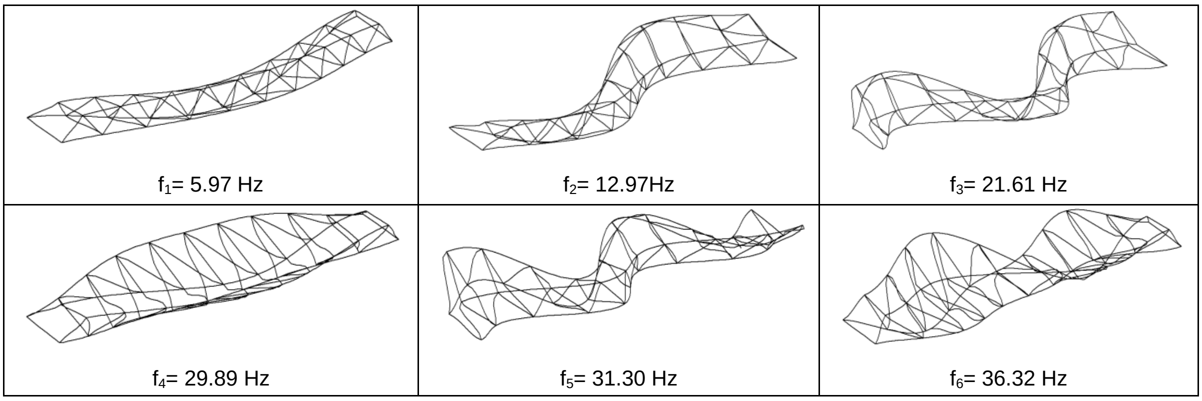

Fig. 6: The mode shapes and frequencies obtained from the numerical solution.

The comparison shows that the mode shapes are comparable between the numerical solution and the results from the modal test so the identified natural frequencies can be considered as being identical. There is also the mode shape obtained from the experimental modal test for the peak in the frequency response functions with the value 16.46 Hz presented (Fig. 7). In this case, the phase sifts identified in the measured points have different values in comparison with other identified mode shapes.

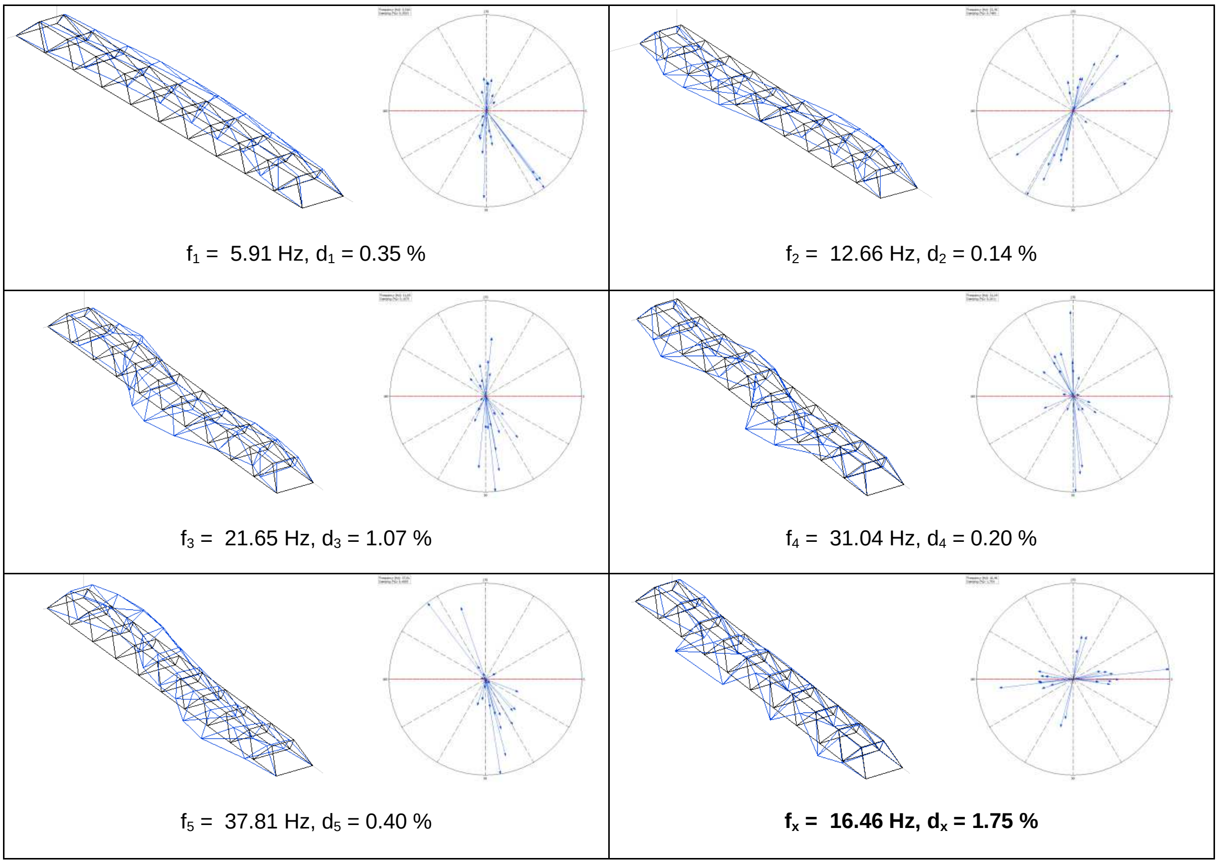

Fig. 7: The mode shapes obtained from the experimental modal test with the diagrams which show the phase shifts identified in each measured point.

5. Conclusions

The experimental modal test in which the hammer is used to excite the structure is an effective method to identify the natural frequencies as well as mode shapes. The presented investigation shows that the quality of the identified mode shapes depends on the location where the structure is excited by the hammer. In the presented case only the excitation in horizontal direction is done so the vertical mode shapes are not identified properly. The identification of natural frequencies is not so sensitive to the location of the excitation because there are also identified the natural frequencies for the vertical mode shapes. There is also a peak value of frequency identified in the frequency response functions which is not comparable with any natural frequency obtained from the numerical solution. This inequality will be a goal of the next investigation and it is not possible to explain why it is there at this moment.

Acknowledgement

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