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Single side damage simulations and detection in beam-like structures

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Abstract. Beam-like structures are the most common components in real engineering, while single side damage is often encountered. In this study, a numerical analysis of single side damage in a free-free beam is analysed with three different finite element models; namely solid, shell and beam models for demonstrating their performance in simulating real structures. Similar to experiment, damage is introduced into one side of the beam, and natural frequencies are extracted from the simulations and compared with experimental and analytical results. Mode shapes are also analysed with modal assurance criterion. The results from simulations reveal a good performance of the three models in extracting natural frequencies, and solid model performs better than shell while shell model performs better than beam model under intact state. For damaged states, the natural frequencies captured from solid model show more sensitivity to damage severity than shell model and shell model performs similar to the beam model in distinguishing damage. The main contribution of this paper is to perform a comparison between three finite element models and experimental data as well as analytical solutions. The finite element results show a relatively well performance.

1. Introduction

Structural health monitoring (*SHM*) has been a research focus in the past decades since more and more structures are sophisticated with the development of science and technology. As failure of the structure functionality can be caused by wear, corrosion, delamination and so on, various *SHM* techniques are developed, for instance, vibration based techniques [1-5].

Vibration based methods have been processed a booming developing period since the appearance of computer in 1970s. Modal analysis has become an important tool in *SHM*. Detailed theoretical and experimental modal analysis can be referred to [6]. Besides, techniques such as acoustic emission, ultrasonic, X-ray and so on are also developed with the high developing technology. However, experiments may confront two problems in real engineering, one is the availability in testing experiments, and another one is that the higher cost required for the experiments.

Due to the high cost of experiment in real engineering structures, numerical analysis has been exploited as another useful but affordable tool in *SHM*, and many other directions. Various commercial programs have been developed and used. Numerical analysis might be divided into three stages: preprocessing, simulation and postprocessing. In scientific field, real engineering structures are basically studied with some representative structures, such as beams, frames, and so on. As for *SHM*, numerical analysis and experiment verification are the commonly used approaches.

Even lots of research work have been conducted on beam-like structures, difficulty in detecting the damage still emerges, and double side saw cut in beam has been more often studied. However, single side damage may be more frequently confronted. And although a large amount of beam-like structure simulations have been carried out, few of them discussed the comparison of different finite element models, for instance, using solid element, shell element and beam element.

In this study, in order to show the performance of solid, shell and beam elements in beam-like structures, a real steel beam is studied using ABAQUS [7]. Afterwards, a saw cut is introduced into the side of the beam with varying depths for simulating damage severity, henceforth, natural frequencies are extracted and analyzed under each damage scenario and compared to the experimental results.

2. Theoretical background

Considering an elastic system, for example, a loaded beam, the dynamic equilibrium equation of damped vibration can be expressed as

$$M\ddot{x} + C\dot{x} + Kx = f(t) \tag{1}$$

where M, C and K are the system's mass, damping and stiffness matrices, which are assumed to be symmetric and positive definite, and $f(t)$ includes all possible types of time dependent loading [6].

As to modal analysis determining the dynamic characteristics of the systems, damping and external loading forces are assumed to be zero in order to achieve a simplest harmonically vibrating model. And therefore to vibration system, characteristic equation becomes

$$\det(K - \lambda M) = 0 \tag{2}$$

where each eigenvalue λ_i corresponds to a resonant frequency ω_i with $\lambda_i = \omega_i^2$, $i = 1,...,n$.

Then mode shapes can be calculated by substituting the resonant frequencies into the following equation

$$(K - \lambda_i M)\phi_i = 0 \tag{3}$$

In regard to the sensitive to structural damage, natural frequency change is firstly taken into account for detecting damage. Herein, an indicator for describing the natural frequency decrease due to the damage effect can be expressed as

$$FD_i = \frac{\omega_i^u / 2\pi - \omega_i^d / 2\pi}{\omega_i^u / 2\pi} \times 100\% \tag{4}$$

where ω_i^u means frequency under intact state, and ω_i^d frequency under damaged state.

In addition, modal assurance criterion (*MAC*) is also frequently used; for mode shape before and after damage, it can be expressed as

$$MAC(\phi_i^d, \phi_i^u) = \frac{\left|(\phi_i^d)^T (\phi_i^u)\right|^2}{\left((\phi_i^d)^T (\phi_i^d)\right)\left((\phi_i^u)^T (\phi_i^u)\right)} \tag{5}$$

where ϕ_i^u represents the *ith* mode shape under intact state, ϕ_i^d means the *ith* mode shape under damaged state.

3. Test specimen and experimental setup

A steel beam with length, height, and thickness of 1004×35×6 mm was selected to analyze, and is shown in Figure 1. The physical properties of the beam are, density 7917 kg/m^3 and Young's modulus 185.2 GPa. And the beam with rectangular cross-section was analyzed under free-free conditions (transverse bending). For the convenience in analyzing the beam, 23 equally spaced nodes (1-23) for translation response measurements were considered. For each node, one accelerometer is used for capturing the dynamic response.

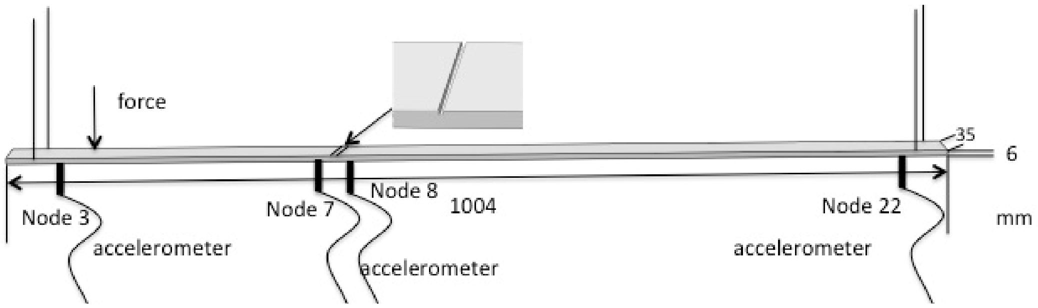

Figure 1. Schematic representation of the beam.

Afterwards, the damage was introduced into the beam by a saw cut, split into eight scenarios. Basically, a single cut, with varying depths, between node 7 and 8 (Figure 1) was introduced into the beam in order to create eight damage scenarios, as summarized in Table 1.

Two inextensible cables simulating "free-free" support conditions were used to sustain the test beam. A fan was used to generate wind to simulate the varying operational and environment conditions. On the other hand, a Brüel & Kjaer 4809 shaker was used to excite the beam at node 3, pseudo-randomly, and a Brüel & Kjaer 2706 power amplifier was used to amplify the excitation force signal. The force was transmitted through a stinger and measured by a Brüel & Kjaer 8200 force transducer. In terms of data acquisition, the responses were measured by 23 piezoelectric CCLD accelerometers at each node. The signals were fed into the Multi-channel Data Acquisition Unit Brüel & Kjaer 2816 (PULSE) and analyzed directly with the Labshop 6.1 Pulse software from the attached laptop (Dell series 400).

For each measurement, time domain responses were obtained, as well as frequency response functions (*FRFs*) and coherence functions from 30 averages, in order to get rid of some operational influences. In the original undamaged structural condition, 20 measurements were conducted, namely ten with and ten without artificial wind. For each damage scenario, five measurements were obtained with and without artificial wind. Finally, the frequency analysis of the beam was carried out in a frequency range of 0-800 Hz (3200 lines) with Hanning windows applied upon force time series as well as response acceleration time series.

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Table 1. Cut properties of each damage scenario.

Damage scenario	Width (mm)	Depth (mm)
1	1.5	0.8
2	1.5	1.0
3	1.5	1.3
4	1.5	1.6
5	1.5	2.2
6	1.5	3.0
7	1.5	4.0
8	1.5	4.8

4. Numerical analysis

In this section, test specimen is illustrated in the first step, and later the numerical models for simulating the experimented specimen will be given, finally simulation for modeling the cut will be also addressed.

4.1 Model description

For simulating the experimental beam, three models: solid (C3D20R), shell (S4R) and beam (B31) models are used for the analysis. Figures 2 and 3 show solid model and shell model, respectively.

For solid model, C3D20R element is used, which is a general-purpose quadratic brick element, with reduced integration (2x2x2 integration points). It takes reduced integration by using a lower-order integration (eight integration points) to form the element stiffness matrix that will reduce running time. The mass matrix and distributed loadings use full integration. And for shell model, a general-purpose linear four-sided shell element - S4R is used. Finally, for beam model, a general 2-node beam model B31 is utilized.

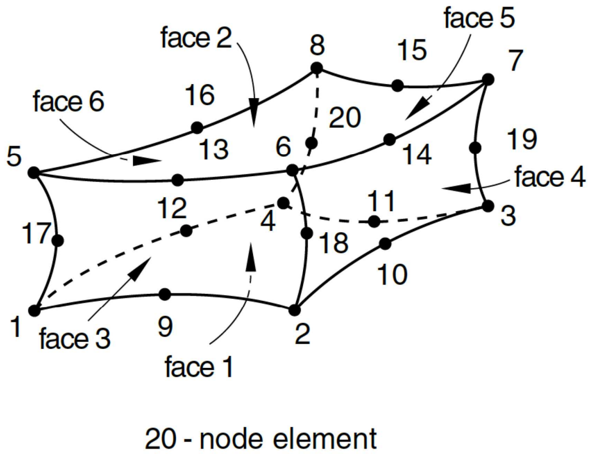

Figure 2. Solid model.

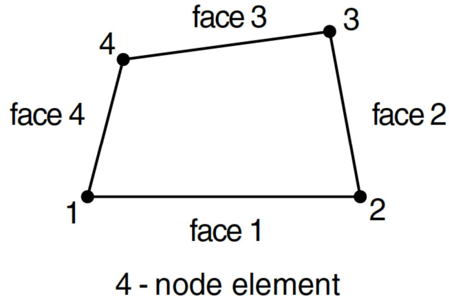

Figure 3. Shell model.

4.2 Saw cut damage simulation
For simulating the saw cut damage, the aforementioned three models are also used.

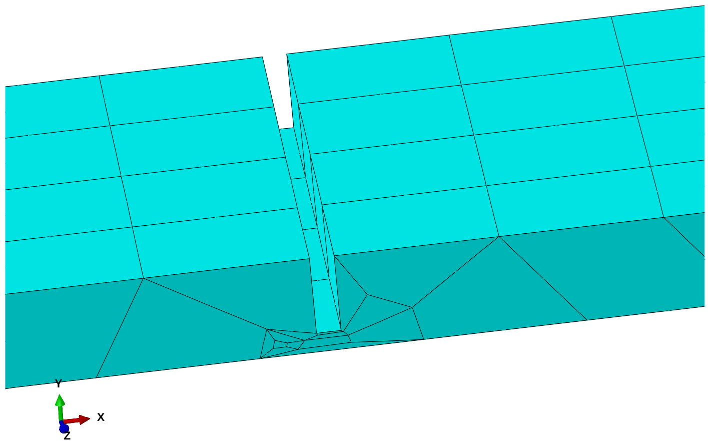

Figure 4. Zooming saw cut with solid model.

For solid model, it exactly describes the cut showing in Figure 4 in a local zooming scale. One can find that cut is indicated with solid element in exact physical saw cut geometry.

For shell model, in this study damage was simulated with stiffness reduction corresponding to the saw cut in section, with ignoring the mass reduction influence, as it is found that the thickness change will greatly affect the simulation results.

For beam model, Figure 5 shows the section under intact state, and Figure 6 shows the section under damaged state, from which one can find the difference clearly. And after occurrence of saw cut, the horizontal symmetrical line is moved downward. In order to overcome this difficulty, a generalized section is used as indicated in Figure 7. From the characteristic properties: area, the moment of inertia and so on, the generalized section can regenerate a section for the simulation.

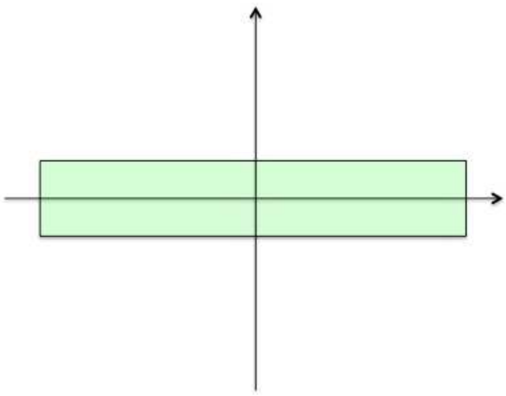

Figure 5. Section under intact state.

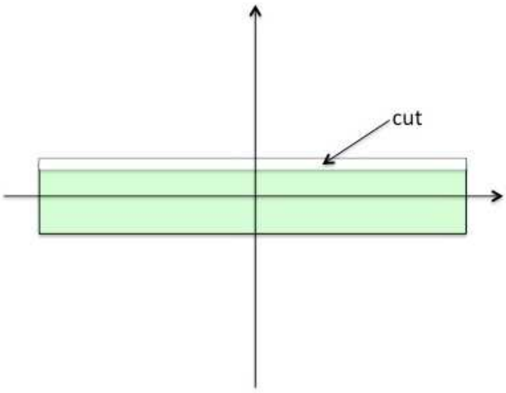

Figure 6. Section under D1 state.

5. Results and discussions
In order to illustrate the applicability of the performance of the three models, the aforementioned simulation results are analyzed as follows:

5.1 Natural frequencies

In this section, a captured natural frequencies comparison is proposed Table 2 for different models and experimental results as well as analytical values.

From Table 2, firstly, one can find that all the captured frequencies from analytical values and three models are close to the experimental results. The error percentages are all less than 1.375%, and this suggests that the experiment and simulations are well conducted. Secondly, Solid model shows better performance than shell model and beam model, as the captured natural frequencies are more close to experimental and analytical values as well. And similarly, shell model performs better than beam model. Note that solid model will consume much more time than shell model and beam model.

Table 2. Natural frequencies (Hz) extraction under intact state.

Mode order	1	2	3	4	5	6
Experiment	29.616	82.250	161.250	267.250	397.500	553.000
Analytical	29.593	81.573	159.916	264.349	394.892	551.544
(%)	0.078	0.823	0.827	1.178	0.656	0.263
Solid model	29.591	81.564	159.890	264.290	394.760	551.290
(%)	0.084	0.834	0.843	1.200	0.689	0.309
Shell model	29.591	81.561	159.880	264.250	394.680	551.130
(%)	0.084	0.838	0.850	1.215	0.709	0.338
Beam model	29.580	81.503	159.700	263.820	393.830	549.620
(%)	0.122	0.908	0.961	1.375	0.923	0.611

Figure 7 shows the captured natural frequencies difference and analytical values compared with the experimental results. It is clear that all are under 1.4%, which means that in this case, the experiment is well conducted, and the three models can all well describe the beam-structure. And one can also find that from analytical solution, solid model, shell model to beam model, the errors of each mode increase, which confirms that analytical solution considers the beam-like structure to be continuous elastic system, and finite element models discrete the structure and approximate the results.

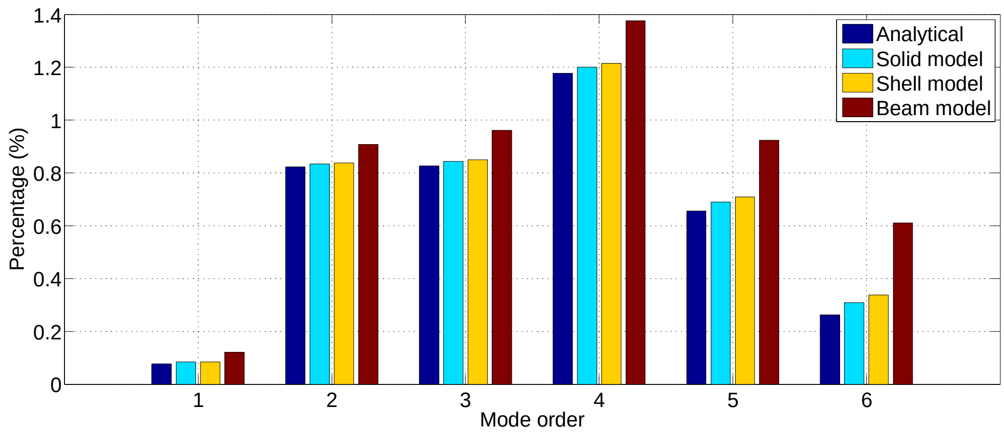

Figure 7. Frequency difference to the experimental value.

5.2 Frequency decrease

During damage detection process, the natural frequency identification before and after deterioration is the most commonly used indicator, which can reveal the characteristic change caused by the deterioration. Figure 8 shows the first six natural frequencies of the three aforementioned models and experiment results with respecting to each damage scenario and intact condition, respectively. For comparison reason, the vertical axis value means the natural frequencies divided by the experimental value under intact state.

From Figure 8, one can find that (i). all the three models show clear decrease as the damage increases, while for light damage scenarios, the natural frequencies vary not too much; (ii). in all three models, the natural frequencies in mode three and six show little differences in comparison with other modes, this is resulted from that the saw cut is close to the node of third and sixth mode shape, then the node value is almost zero, and recalling Equation (3), it will be clear that the corresponding natural frequencies will be relatively stable with little change. (iii). apart from mode three and mode six, to the other four modes, one can see that the natural frequencies captured by solid model are more sensitive to saw cut, as in damage scenario five (D5), the natural frequency captured by solid model shows more difference than that of shell model and beam model. And this may confirm that solid model performs better than shell model and beam model. (iv). for shell model and beam model, the difference between captured natural frequencies is little, this means that for beam-like structures herein presented, shell model and beam model show the same capability in characterizing the structure.

5.3 MAC value decrease

Figure 9 shows the *MAC* value of the three aforementioned models from mode one to six. From Figure 9, it is clear that (i). as damage increases, the *MAC* value decreases; (ii). from D0-D5, the *MAC* value decrease little, this suggests for small damage cases, the *MAC* can endure more challenge in detecting the damage; (iii). comparing the three models, one can find that solid model decreases earlier and greater than other two models. Besides, comparing Figure 9 with Figure 8, one can find that (i). similar to natural frequency change, to mode three and six, unlike other modes, the *MAC* value also change very small. This can be explained by Equation (3), each natural frequency is corresponding to the mode shape; (ii). generally, the natural frequencies change much more than the *MAC* value, this suggests that the natural frequency is more sensitive than *MAC* of mode shapes.

6. Conclusions

A steel experimented beam is simulated and analysed with three models: solid model, shell model, and beam model. Firstly, under intact state, the solid model performs better than shell model and shell model performs better than beam model in capturing the natural frequencies; while under damaged states, solid model performs better than shell model and shell model shows similar performance as beam model in distinguishing the saw cut by natural frequencies change. Secondly, to saw cut simulation, in shell model it can be interpreted by equivalent stiffness reduction in the saw cut section, however, this might reduce the capability in detecting the saw cut; and in beam model, the saw cut can be represented by generalized section.

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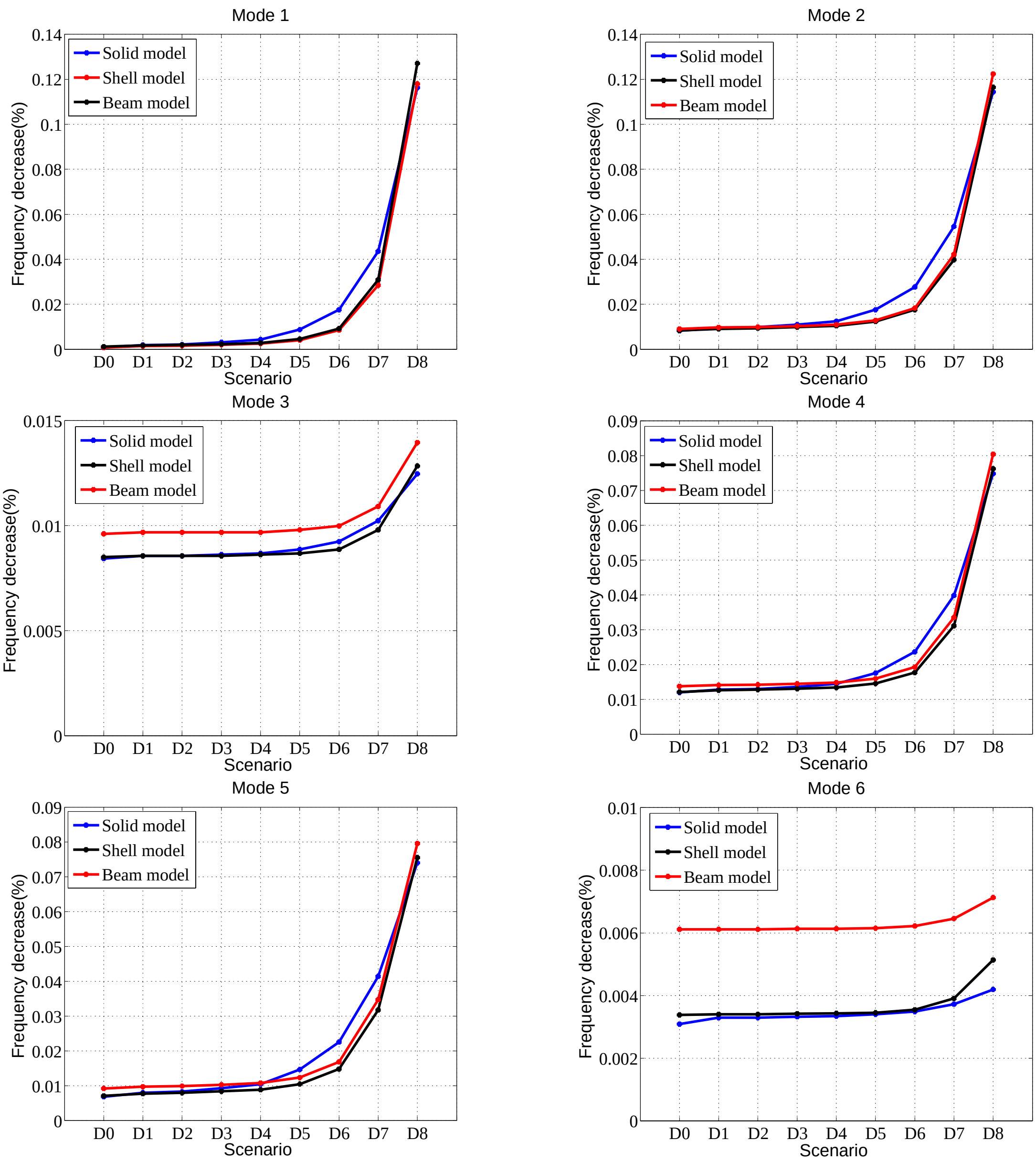

Figure 8. Natural frequency change of mode one to six for the three models.

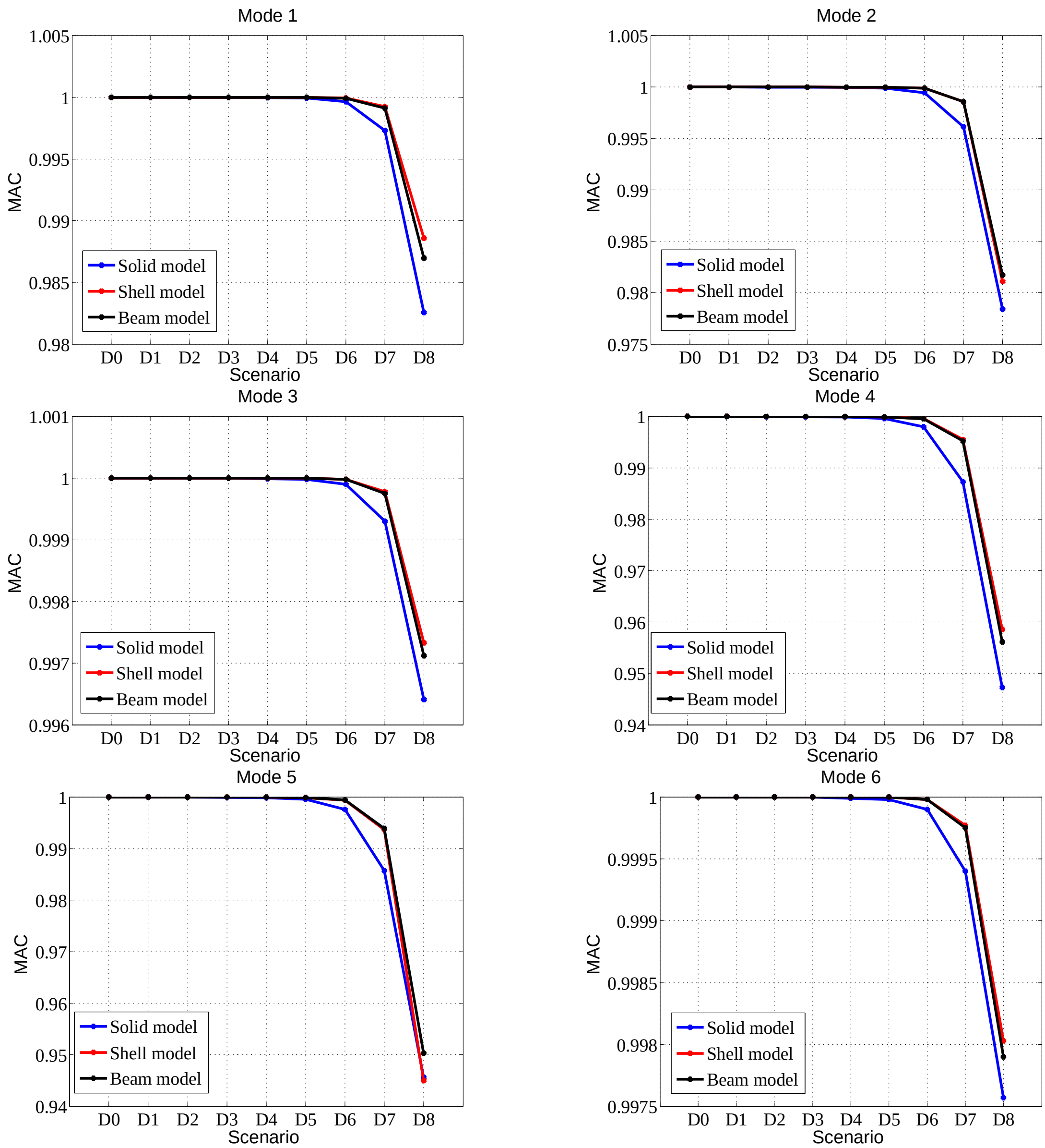

Figure 9. MAC value of mode shape from mode one to six for the three models.

Reference

[1] Farrar CR, Doebling SW, Nix DA. Vibration-based structural damage identification. *Philosophical Transactions of the Royal Society*; A 2001: 356(1778): 131-149.

[2] M. M. Abdel Wahab, G. De Roeck (1999). Damage detection in bridges using modal curvatures: application to a real damage scenario. *Journal of Sound and vibration*; 226 (2): 217-235.

[3] Montalvao D., Maia N.M.M., Ribeiro A.M.R. (2006). A review of vibration-based structural health monitoring with special emphasis on composite materials. *The Shock and Vibration Digest*; 38 (4): 295-324.

[4] Yun-Lai Zhou, E. Figueiredo, Nuno M. Maia, R. Perera (2015). Damage detection and quantification using transmissibility coherence analysis. *Shock and Vibration*; Volume 2015, Article ID 290714.

[5] Yun-Lai Zhou, E. Figueiredo, N. Maia, R. Sampaio, R. Perera (2015). Damage detection in structures using a transmissibility-based Mahalanobis distance. *Structural Control and Health Monitoring*; DOI: 10.1002/stc.1743

[6] Maia, N. M. M. and Silva, J. M. M. (1997). *Theoretical and Experimental Modal Analysis*. Research Studies Press, Ltd.

[7] ABAQUS/CAE, (2004), User's Manuals, Version 6.4.1, Hibbitt Karlsson & Sorensen, Inc., USA.