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Vibrational analysis of cracked cantilever beam

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ABSTRACT

A damage is one of the vital characteristics in structural analysis because of safety cause as well as economic prosperity of the industries. Identification of faults in dynamic structures and components are a significant aspect in judgment creating about their overhaul and retirement. The existence of cracks which influence the performance of structures as well as the vibrational parameters like modal natural frequencies, mode shapes, modal damping and stiffness. In the present work, the effect of crack parameters (relative crack location and crack depth) on the vibrational parameters of a cracked cantilever beam are examined by different techniques using numerical method, finite element analysis (FEA), using ANSYS. Finite Element Method has been accomplished to derive the vibration signatures of the cracked cantilever beam. The results obtained analytically are validated with the results obtained from the FEA. The simulations of FEA have done with the help of ANSYS software. Similarly, Modal analysis is carried out using ANSYS software. Harmonic analysis is done to observe the resonating and anti-resonating peaks for the Un-cracked and cracked cases considered. The stresses induced due to crack are obtained from the static structure analysis done with the help of ANSYS software. The natural frequencies values obtained from all these approaches are observed and the final conclusions are drawn. With increase in crack depth there is an increase in natural frequency. When the distance of crack from fixed end is increased there is an increase in natural frequency.

Keywords: *Cantilever beam, Crack, Crack depth, Mode shapes, Natural frequency, MATLAB, ANSYS, CATIA*

1. INTRODUCTION

The cracks present in the structure interrupt the continuity of the assembly in most of the engineering structures like beam, columns in which geometrical properties can also be altered. Cracks caused due to fatigue stresses or stress concentration reduces the natural frequency and change mode of vibration due to local flexibility induced by the crack. All these effects due to concentrated cracks have been exclusively discussed in this literature. A crack is modeled by describing the variation of the stiffness matrix of the member in the vicinity of a crack. The presence of a crack in a structural member introduces a local compliance that affects its response to varying loads. The change in dynamic characteristics can be measured and lead to identification of structural alteration, which at the end finally might lead to the detection of a structural flaw. Yogesh D. Shinde et al. [1] have conducted experiments to find the natural frequencies of cracked beams for different crack locations and crack depths. The effects of crack depth and crack location on changes of natural frequencies of beams are determined. P. F. Rizos et al. [2] has determined the location of a crack in a beam of varying depth by the help of known lowest three natural frequencies of the cracked beam. The crack location id also determined by measuring its modal characteristics. Here crack is behaved as a rotational spring and graphs are plotted between spring stiffness and crack location for each natural frequency. The point of intersection of three curves gives the location of the crack. M. Kisa et al. **[3]** analyzed the vibrational characteristics of a cracked Timeshenko beam by compound mode synthesis. The beam is divided into two components related by a flexibility matrix which incorporated forces. The forces were derived from fracture mechanics expressions as the inverse of the compliance matrix is calculated using stress intensity factors and strain energy release rate expressions B Sayyad et al**.** [4] found the natural frequencies of a cracked beam by simuating crack as an equivalent spring. The results are verified by experimental study. T. G. Chondrous et al. [5] determined the relation between the change in natural frequency of vibration of a cantilever beam and the crack depth that appears at the built-in edge which is clamped by way of a weld. The variation in the natural frequency of the beam gives information for crack appearance and furthermore makes possible the estimation of its depth. The results have been used to determine the behaviour of beams with different boundary conditions and can be extended to formed structures with complicated geometries by the cracked section. P. Yamuna et al [6] have investigated the variation of natural frequency due to the presence of triangular crack in a simply supported beam. The results obtained shows that the lowest fundamental frequency of the beam without crack is higher than the lowest frequency obtained for beam with cracks. Lee [7] has used easy and effective method to identify the multiple cracks in a beam in which the crack is used as a mass less rotational spring. Here finite element method is used as for solving forward problem based on the Euler–Bernoulli beam theory and inverse problem is solved iteratively for the crack positions and dimensions by the Newton–Raphson method. Dayal R. Parhi etal [8] determined the location and quantification of the size of damage in beam type structure from vibration mode. To locate the crack and magnitude, the graphs are plotted between the frequency ratios versus relative crack depth. The first procedure is to determine the natural frequency of both cracked and un-cracked beam. The experiments were carried out on a cantilever beam with different cracked depth and location for calculating the natural frequency and mode shape H. Nahvi etal. [9] Determined experimentally to determine the crack location and crack depth of a cracked cantilever beam. To identify the crack, contours of the normalized frequency in terms of the normalized crack depth and location are plotted. The intersection of contours with the constant modal natural frequency planes is used to relate the crack location and depth. A minimization approach is employed for identifying the cracked element within the cantilever beam. The proposed method is based on measured frequencies and mode shapes of the beam. P. Srinivasa Rao et al. [10] has determined that there is a viable relationship between the modal natural frequencies and different crack depths by experimentally determining the natural frequencies of a cantilever beam with two open transverse cracks on universal vibration apparatus. The results are validated with ANSYS. Saavedra and Cuitino [11] have used to calculate the dynamic response of a cracked free-free beam and a U-frame after a harmonic force is applied. For calculating the equations of motion using different integration techniques like Taylor, Hilbert and Hughes which are applied using Matlab software platform. Chaudhari and Maiti [12] determined the presence of an 'open' edge crack normal to its axis using the concept of a rotational spring to represent the crack section and the Frobenius method to enable possible detection of location of the crack based on the measurement of natural frequencies Mazanoglu and Sabuncu [13] have presented an algorithm for crack detection from searching over the frequency map and minimizing the measurement errors. Also a statistical approach called recursively scaled zoomed frequencies (RSZF) is used for reducing the deviations. Zheng and Ji [14] have calculated the natural frequencies with a variable stiffness distribution along the length of the cracked beam by using improved

Rayleigh method. The method greatly simplifies the calculation of cracked beams with complicated configurations, such as a beam with several cracks, a cracked beam with concentrated masses, a beam with cracks close to each other, and a beam with periodically distributed cracks. Viola et al. [15] examined the effect of the crack on the stiffness matrix and mass matrix for a cracked Timoshenko beam. Here mass matrix is obtained from the shape function for rotational and translational displacements of the beam and detection of the cracks in beam using modal test data.

2. THEORITICAL FORMULATION AND MODAL ANALYSIS

In this paper it has been analyzed that when the crack is present in beam the reduced stiffness matrix can be found using Fracture mechanics theory. The existence of a crack results in a reduction of the local stiffness, and this additional flexibility alters also the global dynamic structural response of the system. In order to solve this paper, the FEM formulation is done for beams with open surface cracks. A relevant computer program is developed in MATLAB to find out the overall stiffness matrix, natural frequencies and non- dimensional frequencies.

2.1. Properties of the beam.

Table 2.1 Material properties of the beam

Parameters	Mild Steel Beam
Young's Modulus (E)	200×10^9 N/m^2
Density (ρ)	7850 kg/m^3
Poisson ratio (υ)	0.3
Length of beam (L)	0.5 m
Width of beam (b)	0.02 m
Height of beam (t)	0.02 m

2.2 Natural Frequencies of Un-cracked cantilever beams

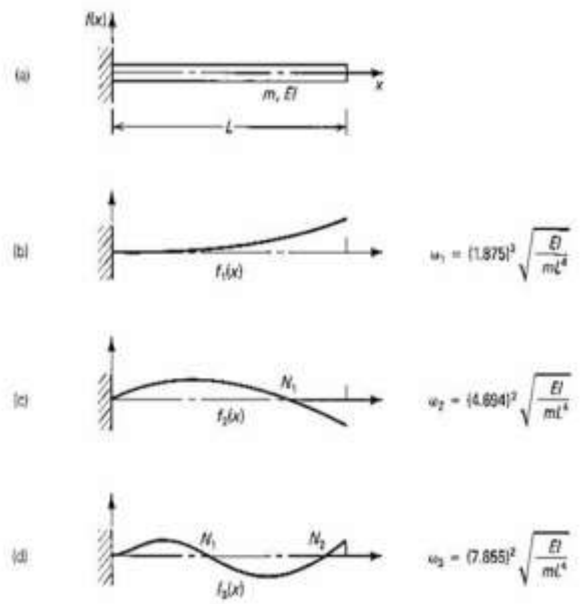


Figure 2.1 Mode shapes and natural frequency formulae for first three modes.

Here, λ_i = 1.875, 4.694, 7.855 for the first three natural frequencies of un-cracked cantilever beams. λ_i is the frequency parameter which gives the natural frequency values. By, changing the λ_i value for the respective modes it can find the natural frequency. The natural frequencies obtained by using the formula is given in table 2.2

Table 2.2 Natural frequencies of un-cracked cantilever beam

Material	Natural Frequency (Hz)		
	First	**Second**	**Third**
Mild Steel	65.223	408.77	1144.7

2.3. Natural Frequencies of the beam

a) 1st mode natural frequency

$$\omega_n=\lambda_i^2\sqrt{\frac{EI}{\rho A}},\quad \omega_1=1.875^2\sqrt{\frac{EI}{\rho A}}$$

$$I=\frac{bh^3}{12}$$

b=0.02
h=0.02

$$E = 200\times10^9\ \mathrm{N/m^2}$$
$$\rho = 7850\ \mathrm{Kg/m^3}$$
$$A = 0.02\times0.02\ \mathrm{m^2}$$
$$L = 0.50\ \mathrm{m}$$

$$\omega_1 = 1.875^2\sqrt{\frac{200\times10^9\times0.02^4}{12\times7850\times0.02\times0.02\times0.5^4}}$$
$$= 409.808\ \mathrm{rad/s}$$

$$\omega_1 = 2\pi f_1$$
$$f_1 = 409.808/2\pi = 65.223\ \mathrm{Hz}$$

b) 2nd mode natural frequency $\omega_2=4.694^2\sqrt{\frac{EI}{\rho A}}$

$$\omega_2 = 4.694^2\sqrt{\frac{200\times10^9\times0.02\times0.02^3}{12\times7850\times0.02\times0.02\times0.5^4}}$$

$$= 2568.377\ \mathrm{rad/sec}$$

$$f_2 = \frac{\omega_2}{2\pi}$$

$$f_2 = 2568.377/2\pi = 408.77\ \mathrm{Hz}$$

c) 3rd mode natural frequency $\omega_3=7.885^2\sqrt{\frac{EI}{\rho A}}$

$$\omega_3 = 7.885^2\sqrt{\frac{200\times10^9\times0.02\times0.02^3}{12\times7850\times0.02\times0.02\times0.5^4}}$$

$$=7192.36\ \mathrm{rad/sec}$$

$$f_3=7192.36/2\pi = 1144.7\ \mathrm{Hz}$$

2.4 Natural Frequencies of Cracked cantilever beams

The MATLAB platform is optimized for solving engineering and scientific problems. The matrix-based MATLAB language is the world's most natural way to express computational mathematics. Built-in graphics make it easy to visualize and gain insights from data. A vast library of prebuilt toolboxes lets you get started right away with algorithms essential to your domain. The desktop environment invites experimentation, exploration, and discovery. These MATLAB tools and capabilities are all rigorously tested and designed to work together. The change in stiffness for cracked cases are calculated for varying crack depths and crack location and crack angle using the below method.

$$K= -\lambda\frac{\Delta_2}{\Delta_1}$$

The values Δ_1, Δ_2 are obtained by solving the below matrices and α value is given by $\alpha = \lambda\beta$.

$$\Delta_1=$$
$$\begin{matrix} 1 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & -\cos\lambda & \cosh\lambda & -\sin\lambda & \sinh\lambda \\ 0 & 0 & 0 & 0 & \sin\lambda & \sinh\lambda & -\cos\lambda & \cosh\lambda \\ \cos\alpha & \cosh\alpha & \sin\alpha & \sinh\alpha & -\cos\alpha & -\cosh\alpha & -\sin\alpha & -\sinh\alpha \\ -\cos\alpha & \cosh\alpha & -\sin\alpha & \sinh\alpha & \cos\alpha & -\cosh\alpha & \sin\alpha & -\sinh\alpha \\ \sin\alpha & \sinh\alpha & -\cos\alpha & \cosh\alpha & -\sin\alpha & -\sinh\alpha & \cos\alpha & -\cosh\alpha \\ -\sin\alpha & \sinh\alpha & \cos\alpha & \cosh\alpha & \sin\alpha & -\sinh\alpha & -\cos\alpha & -\cosh\alpha \end{matrix}$$

$$\Delta_2=$$
$$\begin{matrix} 1 & 1 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & -\cos\lambda & \cosh\lambda & -\sin\lambda & \sinh\lambda \\ 0 & 0 & 0 & 0 & \sin\lambda & \sinh\lambda & -\cos\lambda & \cosh\lambda \\ \cos\alpha & \cosh\alpha & \sin\alpha & \sinh\alpha & -\cos\alpha & -\cosh\alpha & -\sin\alpha & -\sinh\alpha \\ -\cos\alpha & \cosh\alpha & -\sin\alpha & \sinh\alpha & \cos\alpha & -\cosh\alpha & \sin\alpha & -\sinh\alpha \\ \sin\alpha & \sinh\alpha & -\cos\alpha & \cosh\alpha & -\sin\alpha & -\sinh\alpha & \cos\alpha & -\cosh\alpha \\ -\cos\alpha & \cosh\alpha & -\sin\alpha & \sinh\alpha & 0 & 0 & 0 & 0 \end{matrix}$$

The effective mass of the beam is given by 0.2235ρL. The change in frequency is given by:

$\omega=\sqrt{\frac{K}{m}}$, the change in frequency obtained is subtracted from the Un-cracked beam frequency and the frequency of cracked beams are thus obtained. A MATLAB program is developed to solve all the cases and the results are tabulated in tables 2.3.

The table 2.3 gives the natural frequencies values for cracked beams with varying crack depths, crack locations of beams. The crack locations are varied from 0.10 m to 0.40m from fixed end and crack depth varied from 0.004 m to 0.012 m. It is observed that there is an increase in natural frequency values with increase in crack distance from fixed end. It is also observed that there is a decrease in natural frequency values with increase in crack depth.

2.5 MODAL ANALYSIS

Modal analysis is an important tool in vibration analysis. The primary and most important objective of modal analysis is to determine the natural frequencies and mode shapes of the structures. This chapter presents the calculations of natural frequencies of un-cracked and cracked beam structures using ANSYS software.

2.5.1 Modal Analysis Procedure in ANSYS

The step by step procedure for carrying out modal analysis in ANSYS 15 was given in the following and the detailed explanation of steps is presented in further sections.

1. Defining Material Properties
2. Meshing
3. Applying Boundary Conditions
4. Specifying Solution Analysis settings and Solving
5. Viewing Results

Table 2.3 Natural frequencies of cracked cantilever beam

Crack location (m)	Crack depth (m)	Natural Frequency (Hz)		
		First	Second	Third
0.10	0.004	65.320	407.587	1142.980
	0.008	64.541	406.870	1141.508
	0.012	63.875	406.180	1138.672
0.20	0.004	66.214	408.215	1143.267
	0.008	64.942	406.987	1142.618
	0.012	64.102	406.562	1139.051
0.30	0.004	66.854	410.651	1144.401
	0.008	65.024	408.014	1142.985
	0.012	64.512	407.555	1140.001
0.40	0.004	67.541	412.941	1145.342
	0.008	66.254	409.546	1143.789
	0.012	65.124	408.753	1141.106

2.5.2 Defining Material Properties

Pre-Analysis includes setting preferred analysis and defining material properties related to problem.

a) Start-Up & Pre-Analysis

Starting ANSYS15 Workbench by selecting the program from start menu

Start > All Programs > ANSYS 15> Workbench.

b) Set Preferences

Purpose of setting preferences is to filter quantities that pertain to this discipline only.

2.5.3 Meshing: The area is meshed by using Mesh200 elements. Mesh200 element is brick 8-noded element having 6 degree of freedom. There was a very fine mesh surrounding the crack region.

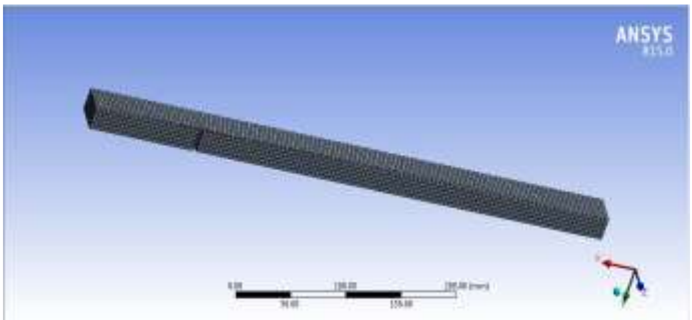


Figure 2.2 Meshing of elements for cracked beam model of crack depth 0.004 m and crack location at 0.10 m

2.5.4 Boundary Conditions

The physical setup involves setting boundary, initial conditions and analysis settings of problem. In present work modal analysis was done for cantilever beam of two different materials. The boundary conditions for all cases given in the following

1. Cantilever beam-A fixed support was placed at left end of the beam.
2. Deformations for required modes were selected and mode numbers were specified for each mode. No external load is given as this is free vibration analysis. Then this model is solved and results of mode shapes were obtained.

2.5.5 Specifying Solution Analysis Settings and Solving

Setting analysis solution option involves extraction required mode shapes in modal analysis for work. This can be done by choosing deformation to respective modes.

In order to request the deformation results *(right click) Solution > Insert > Deformation > Total.*

Selection of mode shapes numbers for respective mode shapes. The model was then solved to using solve option.

2.5.6 Viewing Results

After model was solved in ANSYS software the modal analysis results are obtained. The respective mode shapes of model can be viewed by clicking respective deformation. The numbered deformation gives consecutive mode shapes of model.

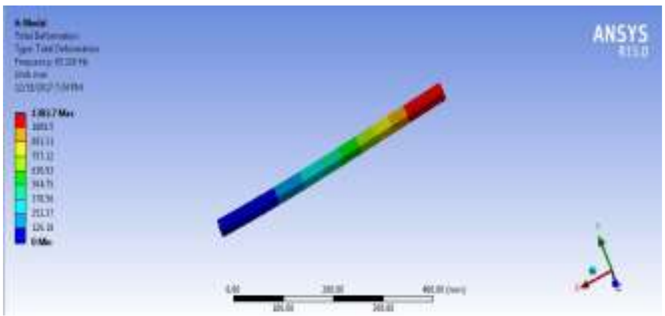
The mode shapes for un-cracked and cracked beams obtained in modal analysis using ANSYS are as follows

2.6 Mode Shapes of Un-Cracked Cantilever Beam

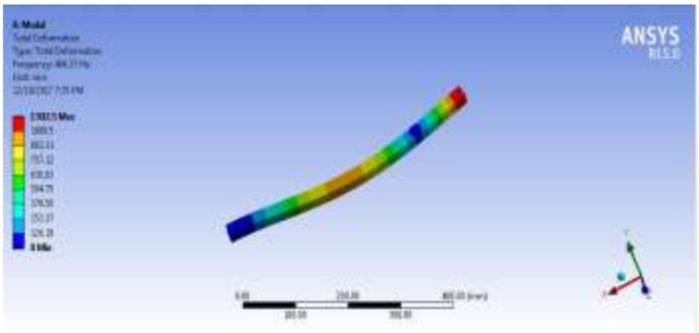
The mode shapes of un-cracked rectangular cross-section cantilever beam for three modes of vibration are shown in figure 2.3.The natural frequencies of un-cracked rectangular cross-section cantilever beam for three modes of vibration are given in table 2.4.

Table.2.4 Natural frequency of un-cracked cantilever beam

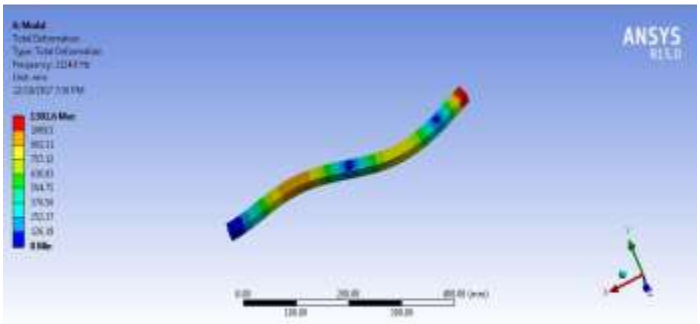
Mode of Vibration	Natural Frequency (Hz)
1	65.326
2	406.37
3	1124.8



a) First mode of vibration



b) Second mode of vibration

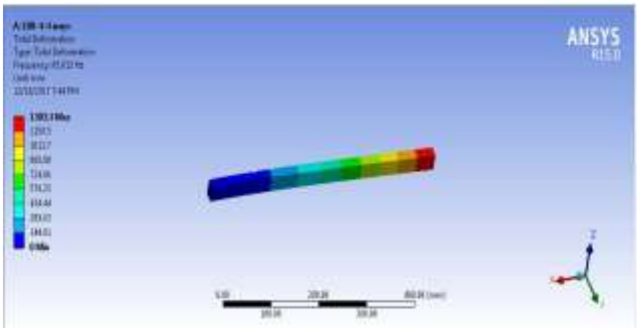


c) Third mode of vibration

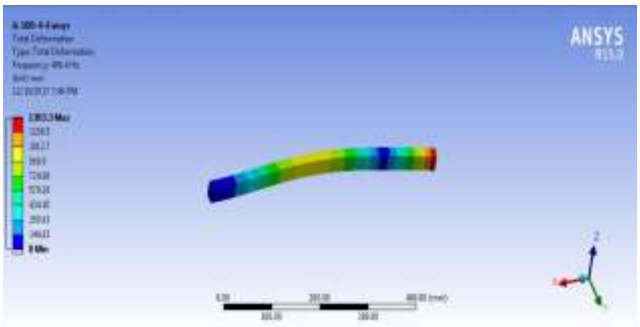
Figure 2.3 Mode shapes of un-cracked cantilever beam

2.7 Mode Shapes of Cracked Cantilever beam

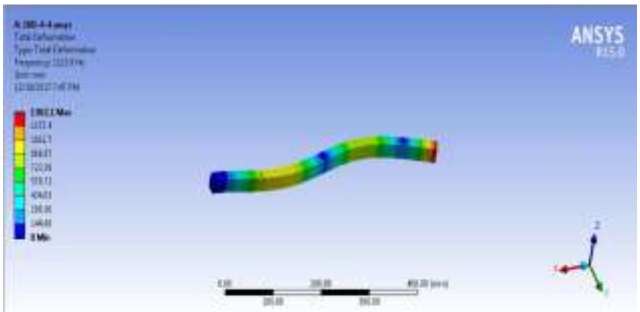
The mode shapes of cracked rectangular cross-section cantilever beam for three modes of vibration are shown in figure 4.4. The sample case considered in this analysis was cracked cantilever beam of crack depth 0.004 m and crack location at 0.10 m. Similarly, for all remaining cases of the cantilever beam model obtained results are tabulated.



a) First mode of vibration



b) Second mode of vibration



c) Third mode of vibration

Figure 2.4 Mode shapes of cracked cantilever beam with crack location at 0.1m and crack depth of 0.004m.

2.7.1 Modal Analysis Results for Cracked Beams

The natural frequencies for cracked rectangular aluminum beams are obtained by performing Modal Analysis in ANSYS. The natural frequencies for varying crack depths, crack angels and crack locations are obtained and tabulated in tables 2.5

Table2.5 Natural frequencies of cracked beam

Crack location (m)	Crack depth (m)	Natural Frequency (Hz) Third Natural Frequency (Hz)		
		First	**Second**	**Third**
0.10	0.004	65.013	406.4	1123.90
	0.008	64.350	406.35	1121.00
	0.012	63.347	406.18	1115.90
0.20	0.004	65.206	407.46	1125.62
	0.008	64.942	406.72	1124.09
	0.012	64.526	406.39	1118.10
0.30	0.004	65.324	408.95	1128.74
	0.008	65.286	407.82	1125.93
	0.012	65.209	406.64	1121.14
0.40	0.004	65.398	409.04	1132.47
	0.008	65.321	408.34	1129.80
	0.012	65.256	407.57	1125.28

3. HARMOIC ANALYSIS

A harmonic, or frequency-response, analysis considers loading at one frequency only. Loads may be out-of phase with one another, but the excitation is at a known frequency. In a harmonic analysis, Young's Modulus, Poisson's Ratio, and Mass Density are required input. All other material properties can be specified but are not used in a harmonic analysis because of the fact that modal coordinates are used;

a harmonic solution using the Mode Superposition method will automatically perform a modal analysis first.

3.1 Harmonic Analysis Procedure

In order to perform harmonic analysis in our problem ANSYS 15 Workbench is used. The main steps followed in performing harmonic analysis were discussed in the following

3.1.1 Build the Model

The type geometries employed for harmonic analysis was a beams with and without cracks. The Cracked beam comprises inclined cracks of varying depths and varying crack location and crack angles. The models for all cases of problems were developed using CATIAV5R20 package.

Defining material properties to the beam models using engineering data module. In a harmonic analysis, young's modulus, poisson's ratio, and mass density are required input.

Mesh the model using mesh control options. In our problem the model was meshed with Mesh200 elements. The meshed model has 5273 nodes and 840 elements.

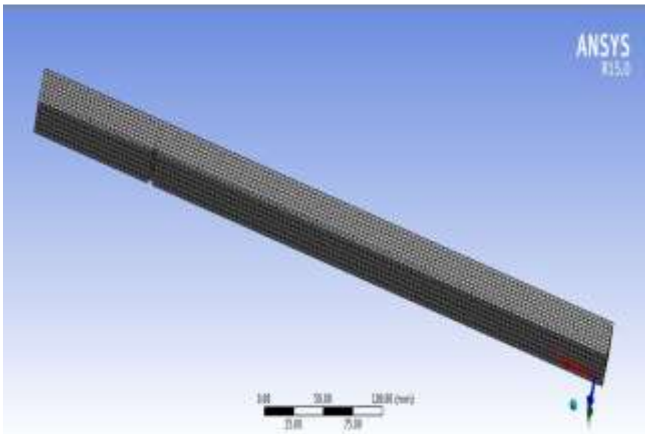


Figure 3.1 Meshing of elements for cracked beam model of crack depth 0.004 m and crack location at 0.10 m.

3.1.2Analysis Settings

All applied loads vary harmonically at the specified frequency. Specified in cycles per second (Hertz) by a frequency range and number of sub-steps within that range using Analysis Settings option. For example, a range of 0-500 Hz with 100 sub-steps gives solutions at frequencies of 5, 10,….500Hz. Same range with 1 sub-step gives one solution at 5 Hz. The In our problem Frequency range of rectangular cross sections beam models is 0 Hz to 300 Hz. Full method was implemented to perform harmonic analysis and it was set using analysis settings option. The harmonic analysis was performed beams of rectangular cross section cantilever beam conditions. Boundary condition for cantilever beam is a fixed support was placed at left end of the beam. A force of 1000 N was applied along y-direction for all cases of beam models.

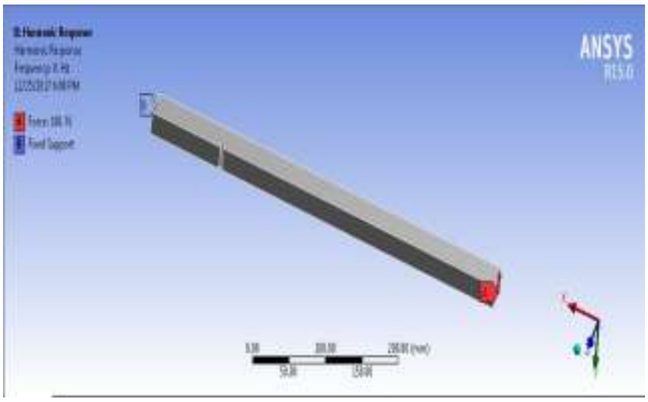


Figure 3.2 Loading conditions for harmonic analysis of beam model of crack depth 0.004 m and crack location at 0.10 m.

3.1.3 Solving and Results

In Solution phase the required frequency range is specified for harmonic analysis and the section of beam for viewing the results. In our problem range of frequency employed for rectangular cross section beam models is 0Hz to 300Hz. The harmonic response of models was set in Y-axis orientation for all cases of models. The model was then solved using solve button and results were viewed.

3.1.4 Frequency Response Functions (FRF's) of Aluminum Cantilever Beams

The Frequency Response Functions (FRF's) of rectangular cross-section cantilever beams with and without cracks of varying crack depth from 0.004 m to 0.0012 m are shown. Here are two of them.

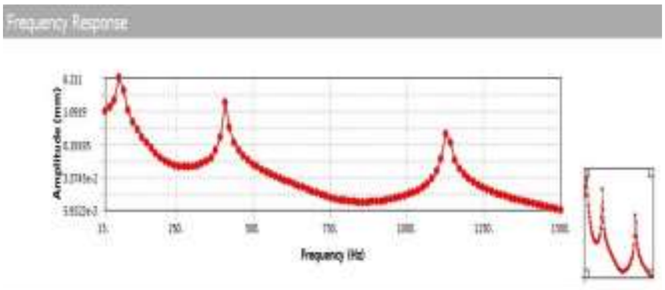


Figure 3.3 Frequency response of un-cracked cantilever beam.

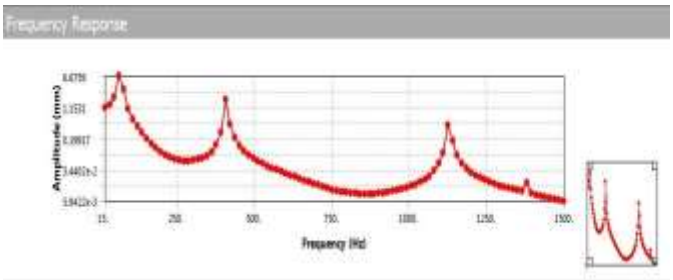


Figure 3.4 Frequency response of cracked cantilever beam of crack depth 0.004 m, crack location 0.10 m.

4. STATIC STRUCTURE ANALYSIS

A static analysis calculates the effects of steady loading conditions on a structure, while ignoring inertia and damping effects, such as those caused by time-varying loads. A static analysis can, however, include steady inertia loads (such as gravity and rotational velocity), and time-varying loads that can be approximated as static equivalent loads (such as the static equivalent wind and seismic loads commonly defined in many building codes).

4.1. Static structure Analysis Procedure

In order to perform harmonic analysis in our problem ANSYS 15 Workbench is used. The main steps followed in performing static structure analysis were discussed in the following

4.1.1 Build the Model

4.1.2Analysis Settings

All applied loads are used to determine the stresses acting on the beam and are also used to determine the total deformation acting on the beam. A static load of 100 N is applied on the free end. The stresses acting on the beam are evaluated.

4.1.3 Solving and Results

The equivalent stresses generated due to application of load are evaluated. The beams with different crack locations and different crack depths are used for this and the stresses are evaluated.

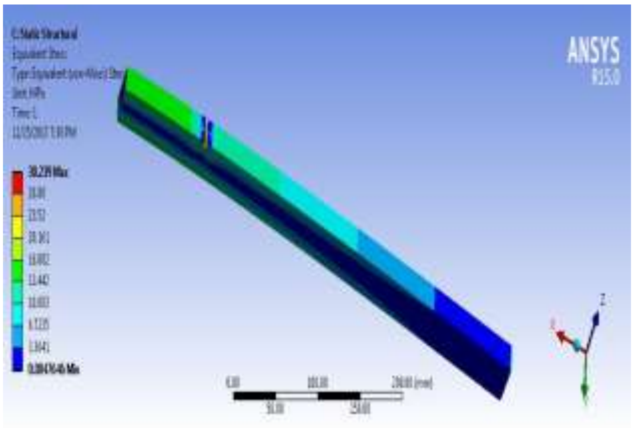


Figure 4.1 Equivalent stress obtained for static structure analysis of beam model of crack depth 0.004 m and crack location at 0.10 m.

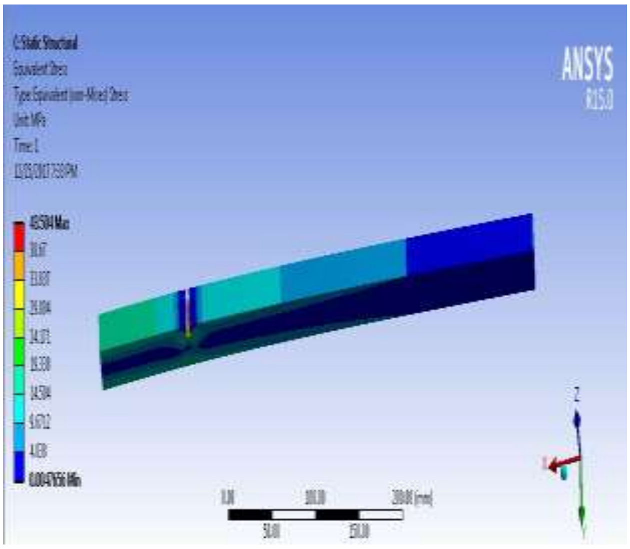


Figure 4.2 Equivalent stresses obtained for static structure analysis of beam model of crack depth 0.008 m and crack location at 0.10 m.

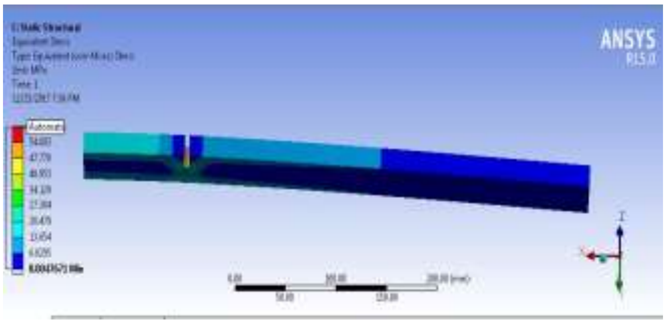


Figure 4.3 Equivalent stresses obtained for static structure analysis of beam model of crack depth 0.012 m and crack location at 0.10 m.

5. RESULTS AND CONCLUSION

In the present work the natural frequencies of the un-cracked and cracked beams are obtained by conducting modal analysis using ANSYS software. The theoretical natural frequencies of un-cracked and cracked beams are obtained from the formulae with the help of MATLAB software.

5.1 Comparison of Natural Frequencies of Un-cracked beams

The un-cracked beam natural frequencies obtained from ANSYS, theoretical calculations are tabulated below. The table 5.1 shows the variation of natural frequencies of un-cracked beams. It is clearly observed from the table 5.1 the natural frequencies for all the modes are all most coinciding for the two methods considered in this work.

Table 5.1 Natural frequencies of un-cracked cantilever beam

Natural Frequency (Hz)	ANSYS	Theoretical Calculations
First	65.326	65.223
Second	406.37	408.77
Third	1124.8	1144.7

5.2 Comparison of Natural Frequencies of Cracked beams

The comparison between the values obtained from ANSYS, Theoretical calculations for the beam are given from tables 5.3 to 5.5. The changes in frequency values can be observed from these tables. It is clearly observed from the table 5.3 to table 5.5 the natural frequencies for all the modes are all most coinciding for the two methods considered in this work.

Table 5.2 First natural frequencies of cracked cantilever beam

Crack Location (m)	Crack Depth (m)	Third Natural Frequency(Hz)	
		ANSYS	Theoretical
0.10	0.004	1123.90	1142.980
	0.008	1121.00	1141.508
	0.012	1115.90	1138.672
0.20	0.004	1125.62	1143.267
	0.008	1124.09	1142.618
	0.012	1118.10	1139.051
0.30	0.004	1128.74	1144.401
	0.008	1125.93	1142.985
	0.012	1121.14	1140.01
0.40	0.004	1132.47	1145.342
	0.008	1129.80	1143.789
	0.012	1125.28	1141.106

Table 5.3 Second natural frequencies of cracked cantilever beam

Crack Location (m)	Crack Depth (m)	Second Natural Frequency(Hz)	
		ANSYS	Theoretical
0.10	0.004	406.4	407.587
	0.008	406.35	406.870
	0.012	406.18	406.180
0.20	0.004	407.46	408.215
	0.008	406.72	406.987
	0.012	406.39	406.562
0.30	0.004	408.95	410.651
	0.008	407.82	408.014
	0.012	406.64	407.555
0.40	0.004	409.04	412.941
	0.008	408.34	409.546
	0.012	407.57	408.753

Table 5.4 Third natural frequencies of cracked cantilever beam

Crack Location (m)	Crack Depth (m)	First Natural Frequency(Hz)	
		ANSYS	Theoretical
0.10	0.004	65.013	65.320
	0.008	64.350	64.541
	0.012	63.347	63.875
0.20	0.004	65.206	66.214
	0.008	64.942	64.942
	0.012	64.526	64.102
0.30	0.004	65.324	66.854
	0.008	65.286	65.024
	0.012	65.209	64.512
0.40	0.004	65.398	67.541
	0.008	65.321	66.254
	0.012	65.256	65.124

5.3 Variation of natural frequencies with crack depth:

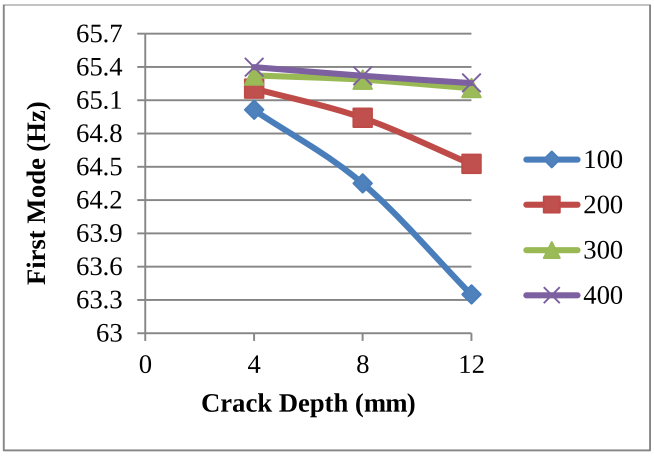


Figure 5.1 Variation of First Mode with crack depth for different crack locations.

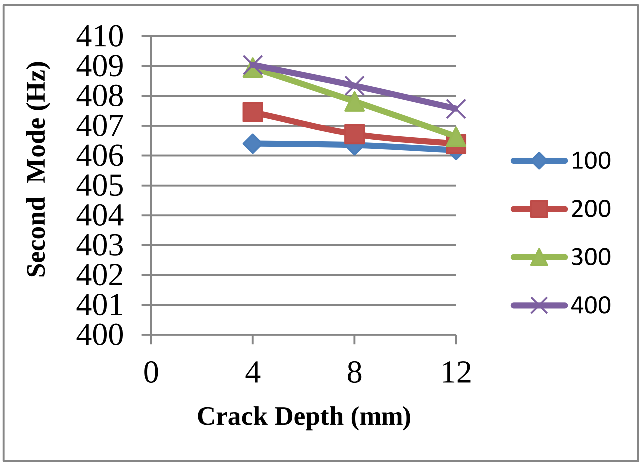


Figure 5.2 Variation of Second Mode with crack depth for different crack locations.

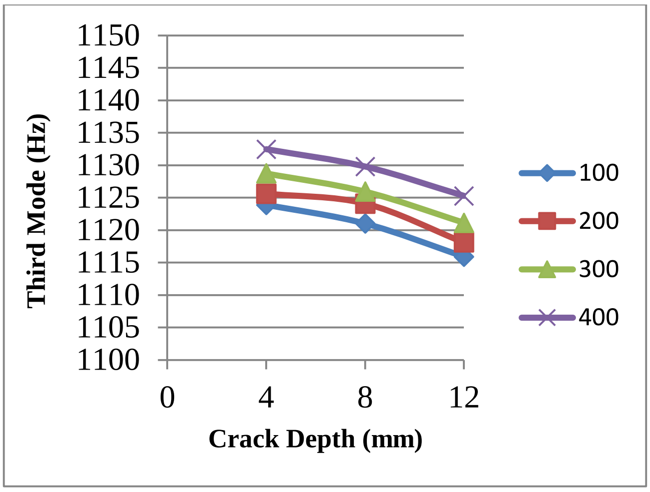


Figure 5.3 Variation of Third Mode with crack depth for different crack locations

5.3 CONCLUSIONS

Based on the results obtained from tables 5.1 to 5.4 for identification of crack on the cantilever beam structure, the following conclusions are drawn:

1. Due to the changes in the crack parameters (crack location and crack depth) there is always a significant change in the natural frequencies and mode shapes.
2. The results of the crack parameters have been obtained from the comparison of the results of the un-cracked and cracked cantilever beam during the vibration analysis. It is also observed that the values obtained from ANSYS and theoretical calculations are in good agreement.
3. When the crack location is constant, but the crack depth increases:
   The natural frequency of the cracked beam decreases with increase the crack depth. The amplitude at crack location decreases with increase the crack depth for each mode shape.
4. When the crack depth is constant, but crack location increases from cantilever end:
   The natural frequency of the cracked beam also increases. At particular crack location of a beam, the amplitude is less with respect to other beams having a different crack location.
5. The resonating and anti-resonating peaks are obtained from harmonic analysis. The natural frequency values obtained by the two methods are verified using harmonic analysis.
6. The stresses generated from static structure analysis clearly depicts that the stress obtained increases with increase in crack depths for a particular location.
7. The stresses generated decreases for different crack depths at a particular crack location.

6 FUTURE SCOPE

The work may be extended for multiple damage detection in bi-material and composite material elements. More robust hybrid techniques may be developed and employed for fault detection of various vibrating parts in dynamic systems such as railway tracks, overhead cranes, oil rigs, turbine shafts etc. The artificial intelligence techniques may be developed and integrated with the vibrating systems to make on line condition monitoring easier.

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