Study the Effected Parameters on Vibration Analysis of Cantilever Beam with a Bolted Joint

Dr. Nathera A. Saleh
Department of Mechanical Engineering
University of Basrah
College of Engineering
Nathera1971@yahoo.com

Zainab Asaad Hardan
Department of Mechanical Engineering
University of Basrah
College of Engineering
Zasaad56@gmail.com

***Abstract*- The main objective of the present paper is to study the vibration behavior of cantilever beam with a bolted joint of different lap's type (i.e. Single lap, Double lap) for free and forced vibration. The effects of various parameters such as beam configuration, preload, harmonic force magnitude and harmonic force positions on natural frequency, mode shape, and amplitude have been investigated. The experimental work carried out during this study included material selection, chemical composition test, tension test, preparation configurations of beam, free and forced vibration tests at per- torques range (6-60N.m) and at rotational speeds range (300-900RPM). Numerically, a general purpose finite element method (ANSYS Ver.16.1 package) has been used. The results show that the natural frequencies of single lap (1Bolt, 2Bolt) bolted beams were roughly equal to those of intact beam. But, in double lap bolted beam were slightly lower than those of intact beam with same profile. Moreover, as preload or pre-torque on the bolt increases, the natural frequency becomes constant for all types of beam configuration. For forced vibration, the vibration amplitude is largely dependent on magnitude and position of harmonic force. The validation results present a good agreement with numerical results as the largest margin of error is about (5%).**

***Index Terms*—bolted joint, preload, natural frequency, FEM, vibration amplitude, free vibration, forced vibration.**

I. INTRODUCTION

Bolted joint connections are generally employed when securing parts together in a variety of processing machines and mechanical structures due to its simple structure, ability to increase a clamping force , easily maintenance and inspection at low cost. These joints present further flexibility and damping to the over-all structural dynamics. However, bolted joints exhibit an intricate nonlinear behavior that may arise from, material properties, geometry or by contact conditions of joints. Mostly, the strength of the joint is indicated by preload force. In practical applications where joints are dynamically loaded with transient, harmonic, periodic or non-periodic loading and so on, joints frequently experience self-loosening (i.e. gradual loss of preload) with expanding service time. The two predominant reasons of failure of bolted joints exposed to cyclic loading are vibration induced loosening and fatigue. Revealing loosening of bolted joint at initial stage can obstruct failure of structure.

A vibration-based method that uses variations in natural frequencies of a structure is operative in revealing loosening of bolted joints as the method concentrates on a stiffness decreasing of a bolted structure. Many inclusive papers on joint and fasteners have well implemented for many years to understand joints and their effects. Experimentally, theoretically and modeling approaches for bolted joint structures are proposed in the literature.

Majority of the stated studies have concentrated on the linear and non-linear determination of the dynamic characteristics of the bolted joints, the dissipation of energy through joint, the uncertainty of the parameters, relaxation and effective control of the joint pre-load. Some of researchers attempt to model the dynamic behaviour of bolted joint under vibration through several physics- based models.

Oldfield et al. [1] employed an exhaustive finite element model to analyze the dynamic frictional contact of a bolted joint under harmonic loading. This model was then substituted with the Bouc–Wen model and a number of Jenkins elements through fitting the hysteresis loops created by the finite element method. They showed for a constant bolt preload, the increase in the torque that applied in the joint caused an increase in the energy dissipated per each cycle. They deduced that the higher the amount of slippage per cycle, the lower the average dynamic joint stiffness. Also, when the bolt preloads are high, a slight sliding is created at the joint interface and the two components approximated to a rigid connection.

Ahmadian and Jalali [2] - [3] proposed a non-linear model to represent the dominant physics such as micro and macro slip that involved in the bolted lap joints. They used for modelling the bolted joint interface both linear and nonlinear springs to characterize the softening phenomenon because of the slip and a damper to simulate the damping effects of the joint. A multiple scales method is employed to estimate the response of the bolted joint structure with a non-linear model exposed to external excitation. Also, they identified the joint interface's parameters through compared the obtained frequency function with the corresponding experimental counterparts [2]. They also developed the formulation of a non-linear generic element for modelling the bolted lap joints. This generic element model is composed of a generic damping matrix and a generic stiffness matrix that will be able to represent the non-linear constitutional relationships at a joint interface [3]. The incremental harmonic balance (IHB) method was used to get the dynamic response of the developed model for the assembled structure containing the generic joint interface element. Jalali et al. [4] used a force-state mapping approach to identify the parameters of a nonlinear joint model from time-domain acceleration records in steady-state vibration response to single-frequency excitation close to the first natural frequency. They confirmed experimentally that the determination of the viscous damping coefficients depends on the displacement amplitude. These identified parameters have demonstrated that the presence of both a damping coefficient and a cubic softening spring dependent on the amplitude of vibration.

Ahmadian and Rajaei [5] employed an Iwan –type model to predict the nonlinear effects of a frictional contact interface, they described the Iwan model by its distribution density function which is commonly identified by double differentiation of the experimentally obtained joint interface restoring force. They developed a more reliable procedure in identification of the Iwan model by relating the density function to the joint interface dissipated energy. They showed that the density function of the Iwan model is uniquely related to the dissipation of the contact interface and is identified with modest computational efforts.

Karim and Blanzé [6] showed that the controlling of the preload force at a bolted joint part of a structure reduces the vibration and noise of this structure. Two control laws have been proposed for vibration reduction, in the first law the critical frequencies of the bolted structure was taken as input parameters. But, in the second law there is no need for any information about the Eigen frequencies of the system or the displacement and is solved by a 3D finite element model of bolted joint. Both types of nonlinearity (friction model and unilateral contact) was taken into account.

Yaobin Li et al. [7] adopted the single-bolt fixed cantilever beam model to analyze the super harmonic nonlinear response of structure induced by the local nonlinearity in bolt joint connection structure. The implicit dynamic simulation method that based on the principle of displacement and energy convergence was utilized to analyze the phase trajectory curve. This study showed that super harmonic response generated when the excitation frequency is 1/2 of the resonant frequency under strong nonlinearity conditions. Xin Liao et al. [8] studied the transient behaviour of simple bolted joint beam in transverse direction by suggesting an analytical model based on phenomenological model using the fourth-order Runge-kutta method to compute the transient response. In addition, to describe nonlinear behaviors of bolted joint beams in shear vibration, a series Iwan model containing cubic stiffness term has been established. They observed that the natural frequency depends slightly on cubic stiffness term.

Most of experimental and EFM researchers focused on studying the preload effect on natural frequency and damping of bolted joint subjected to impact or shock loading. Toshimichi .F et al. [9] investigated the effect of axial bolt force, bolt position and surface roughness on natural frequencies of bolted joint through bending vibration experiments. Also, a numerical approach based on FEM has been proposed to evaluate natural frequencies and mode shapes of bolted joint by taking the interface stiffness effect due to surface roughness into account. They observed that the natural frequency of bolted joint structure is less than that of intact structure with same profile, and it is increases with higher bolt preload and become constant after a certain limit of bolt preload.

Kumarswamy et al. [10] employed the experimental and Finite Element Analysis (FEA) methods to analyze the transient behavior of bolted joint structures under impact or shock loads. They have developed computational modeling procedures that offer structural analysis of an improved physics-based shock model for combat vehicles. Several factors that caused effects in the response of bolted joint structures for shock loading such as preload, damping, and type of FE modeling and intensity of impact load have been studied. They showed that the bolt preload effect on the cantilever structure confirms that the increase in preload increases the natural frequency of the structures at higher modes.

Feblil et al. [11] estimated the joint loosening by revealing fluctuations of high frequency response with the health monitoring system that depended on an impulse response excited by laser ablation. Besides, they suggested a finite element model of bolted joints that use a three dimensional elements and taken the applied preload force and contact between components into account. This model show a good agreement with experimental results. Also, they demonstrated that different loosening cases and their locations could be successfully recognized through using the Recognition-Taguchi method as frequency response based approach. Loosening is usually exists in a bolted joint exposed to transverse or shear load and low velocity impact load which causes a decrease in structural stiffness eventually variations in natural frequency or separation of secured parts.

Weiwei Xu et al. [12] investigated the association between the natural frequency and structural damping with the bolt preload through impact hammer testing on a single lap bolted joint structure. They observed that the damping ratios increase with decrease preload, and natural frequency increases with increasing preload, but stayed approximately constant for preload larger than 30% in the bolt yield strength.

Pravin Kulkarni [13] studied the effect of preload on natural frequency of bolted joint subjected to low velocity impact loading. The effect of preload is carried out in experimental analysis and simulated by pre-stressed modal analysis. This study showed that as preload or tightening torque on bolt increases, the natural frequency goes on increasing up-to certain limit of tightening preload but after that there is a reduction in natural frequency because after certain limit of tightening, washer face of bolt starts cutting washer and ultimately reduces the stiffness of bolted structure and again goes on increasing with increasing preload. Kashinath H. and Mahesh P. [14] evaluated the variation of natural frequency with different preload conditions using FEM pre-stressed modal analysis in ANSYS 14.5 software. They concluded that small Variation in preload does not causes noticeable change in the natural frequency's values in each mode.

The main aim of the present paper is to investigate the effects of various factors such as beam configurations (i.e. single lap and double lap), preload, motor's position and motor's speed on natural frequency, mode shape, amplitude of a cantilever beam. Experimental and finite element analysis (ANSYS software 16.1) have been used to study this parameters.

II. EXPERIMENTAL WORK

A. Testing of mechanical properties

There is a wide range of tests and inspections carried out to ensure the materials and items that meet their specifications or are suitable for the purpose required. The following two tests are executed to ensure the mechanical properties of material used.

A.1. chemical composition test

The chemical compositions for both bolt and beam's alloys were analyzed by using optical emission spectrometer analyzer device to confirm the constituents of elements. The results for both alloys can be shown in Table I. The results were compared with the standard results in the ASTM Handbook of comparative word steel standards [16].

Table I the chemical composition of AISI 1020 and 1026 carbon steel in wt. %.

Element	(Bolt)1020 Chemical Composition %	1020ASTM [16] Results %	(Beams)1026 Chemical Composition %	1026 ASM[16] Results %
C	0.18	0.18-0.23	0.25	0.22-0.28
Mn	0.33	0.3-0.6	0.77	0.6-0.9
S	0.012	0-0.05	0.0217	0-0.05
P	0.026	0-0.04	0.043	0-0.04

A.2. tension tests

The tension test is also conducted in this study to evaluate many of mechanical properties of the Beam's material as shown Table II.

Table II the mechanical properties of plates from tension test

Material property	Medium carbon steel (Beam)	Grade 8.8 (Bolt)[16]
Modulus of elasticity , (E)	190Gpa	200Gpa
tensile yield strength	364Mpa	640Mpa
tensile ultimate strength	478Mpa	800Mpa
Density	7916 Kg/m³	7850 Kg/m³
Poisson's Ratio	0.3	0.29

B. preparation Beam configurations of bolted joint

In the present work, three types of beam configurations have been studied as follow.

B.1. single lap-1Bolt joint beam

A single lap-1Bolt joint beam consist of two beams fastened with a single M10 bolt as shown in Fig .1.

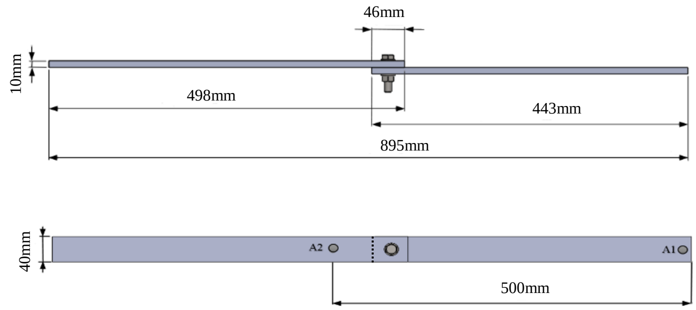


Fig.1 single lap joint with sensor location (front and top view)

B.2. Single lap-2Bolt beam

A single lap-2Bolt joint beam consist of two beams fastened with a double M10 bolt as shown in Fig .2.

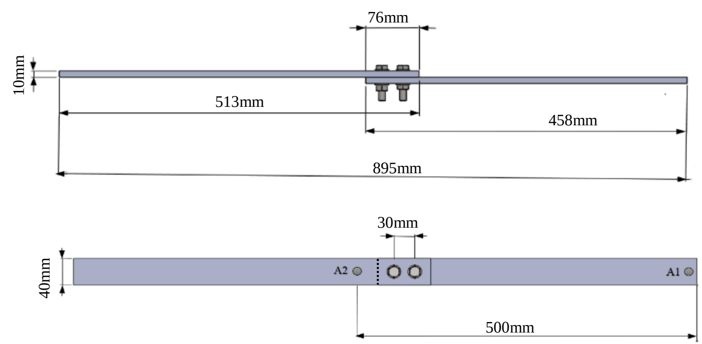


Fig.2 Double bolted lap joint with sensor location (front and top view)

B.3. double lap bolted joint

A double lap joint beam consist of four beams fastened with a double M10 bolt as shown in Fig .3.

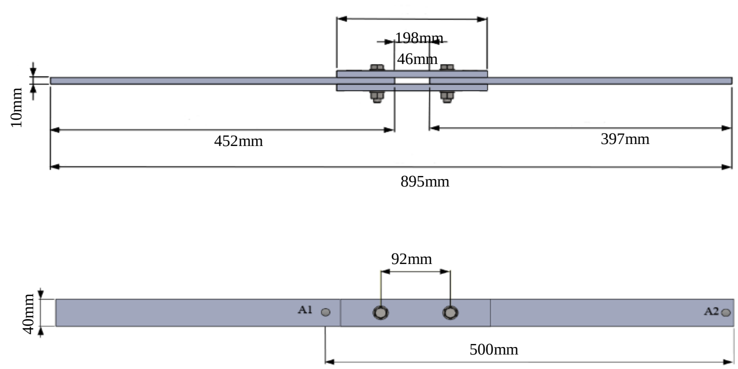


Fig. 3 Double lap joint with sensor locations (front and top view)

C. Description Experimental setup

The best way to study the bolted joint structures and their effect on vibration analysis is to select simple structure such as cantilever beams with three lap joints. For the purpose of comparison, in addition to these bolted structures, a structure with similar identical material properties and geometry to the bolted joint structures, but with no joint interface (intact structure) was employed as shown in Fig. 4. Thus, under the same boundary conditions and forcing the joint effect on vibration analysis will be determined by comparison the vibration effect of the jointed and monolithic structures.

Fig. 4 beam configurations

C.1. free vibration test setup

The free vibration test setup consists of bolted structures, accelerometers, Data acquisition system (DAS) to acquire the vibration signal from the accelerometer to encode it in a digital form. This signal is then processed (e.g., FFT), and displayed on the computer screen by means of analysis software, as shown in Fig. 5.

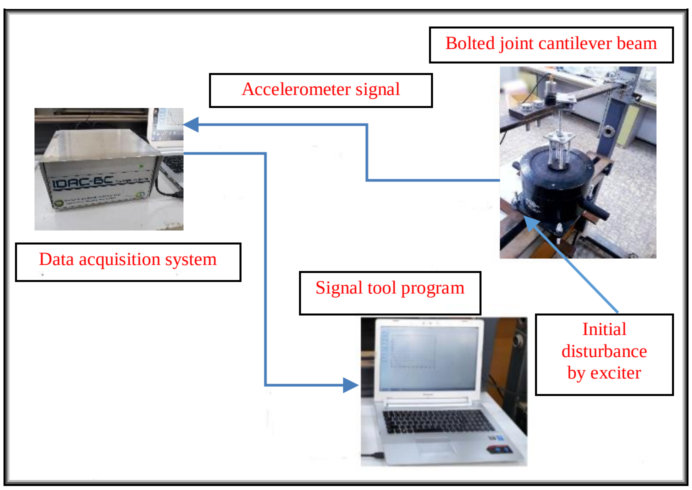
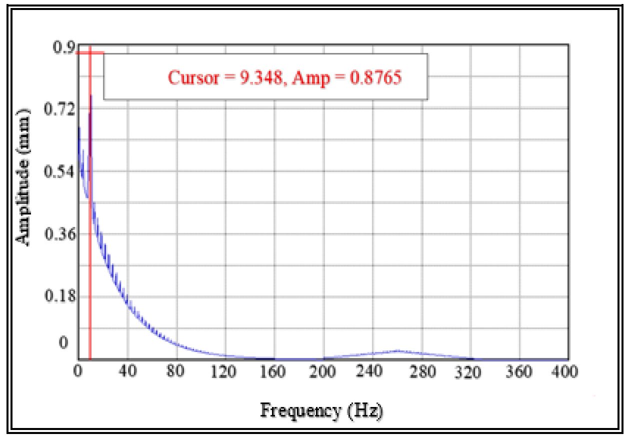


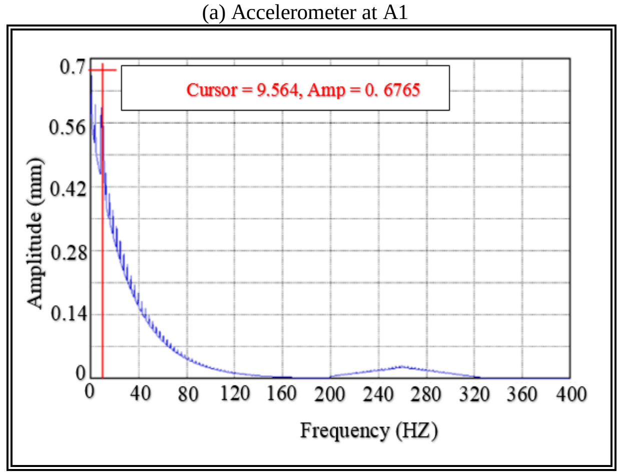
Fig. 5 experimental setup for free vibration

Calibrated torque wrench is used to apply the pre-torques on the bolted joint. The free vibration test was conducted on the beams with bolted joints for seven pre-torques of 6, 10, 25, 35, 45, 55, 60 N.m. The preloads on the bolt shank produced by these pre-torques are 3, 5, 12.5, 17.5, 22.5, 27.5, 30 KN respectively. The excitation is applied on the cantilever beam by initial disturbance by exciter. Two piezoelectric accelerometers were pasted on the cantilever beam. One (A1) is close to the free end and the other (A2) is 50cm away from the free end.

The outputs shown on computer screen after analysis of cantilever beam (monolithic beam) as shown in Fig. 6.



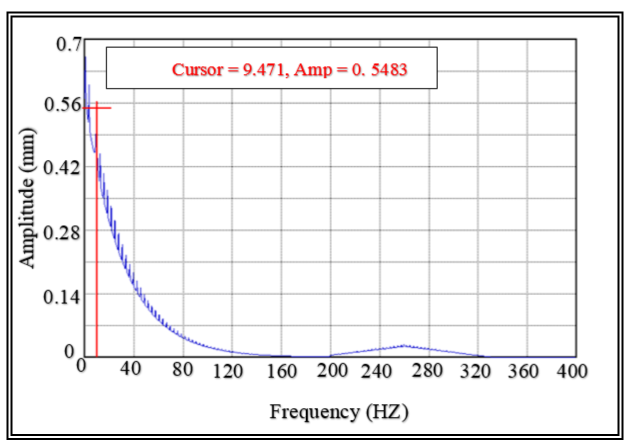
(a) Accelerometer at A1



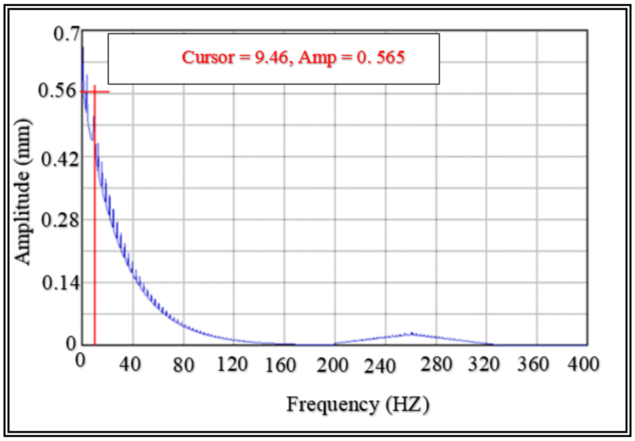
(b) Accelerometer at A2

Fig. 6 FFT of experimental data of intact beam

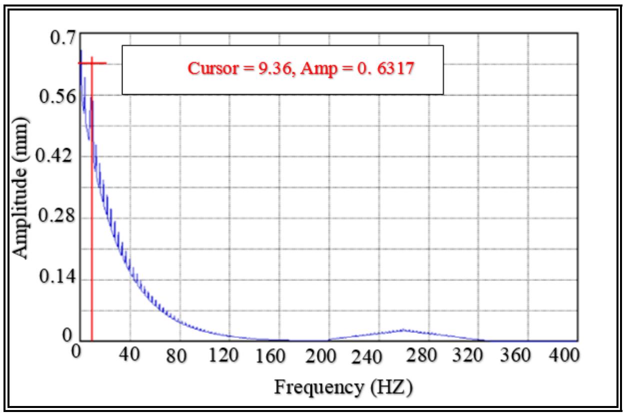
There is no a significant difference in natural frequency value between accelerometer's two sites so the following data will be taken on one site (A2). The outputs shown on computer screen after analysis of cantilever beam with bolted joint as shown in Fig. 7.



(a) Single bolted lap joint



(b) Double bolted lap joint



(c) Double lap bolted joint

Fig. 7 FFT of experimental data of cantilever beams with the three bolted lap joints

C.2. Forced vibration test setup

The experimental setup for forced vibration test is same as for free vibration but the excitation is applied by external rotating unbalance force as shown in Fig.8.

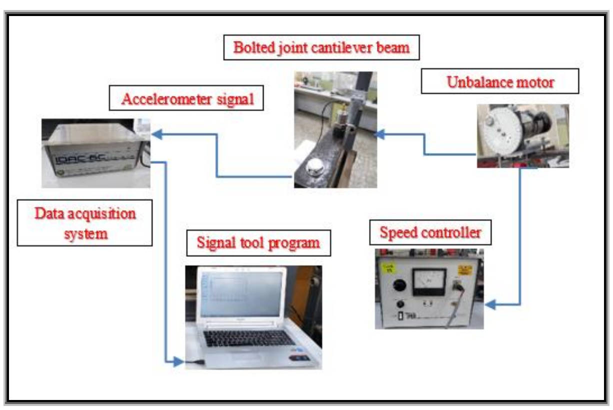


Fig.8 Experimental setup for forced vibration

The DC motor with out- of- balance dicks attached to the beam at three points A, B and C as shown in Fig.9 is used in this study to produce harmonic excitation. The unbalance in disks represented by an eccentric mass m with eccentricity e which rotates with angular velocity ω provides the harmonic force (meω²) that proportional to the square of angular velocity ω. The speed control unit can be used to adjust the exciting frequencies that produced by motor. Also, the range of exciting frequencies from 300 to 900 RPM will be applied.

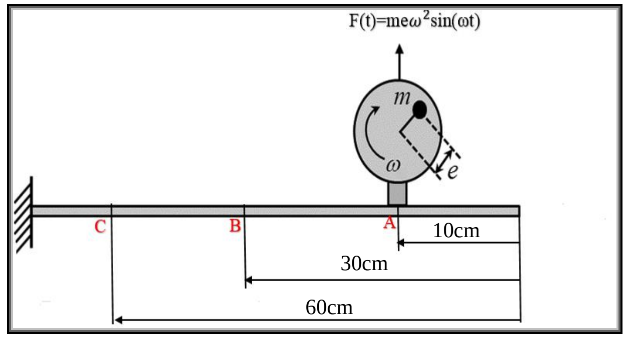
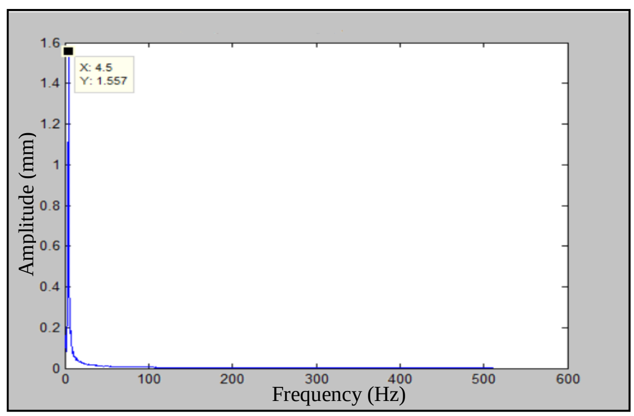
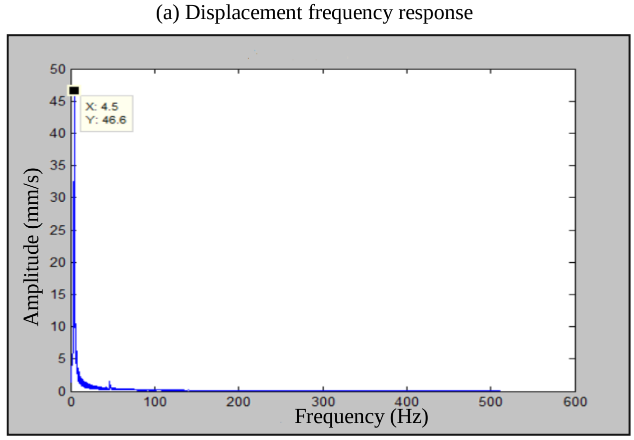


Fig.9 motor’s positions on the cantilever beam

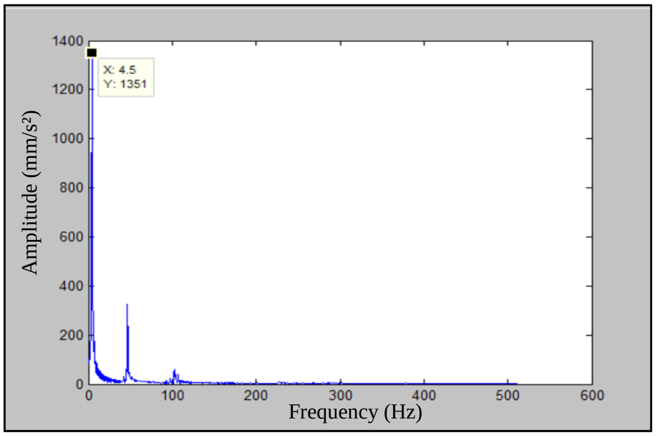
The output shown after analysis of cantilever beams (intact and jointed) under forced vibration as shown in Figs. (10-12) respectively. Where the letters x and y refer to frequency and amplitude. These three cases are performed with unbalance motor at point such as point A, at speed such as 300RPM and for three vibration response forms (displacement, velocity, acceleration) For the purpose of clarifying the method of taking results. The other results are shown in the section of results and discussion.



(a) Displacement frequency response

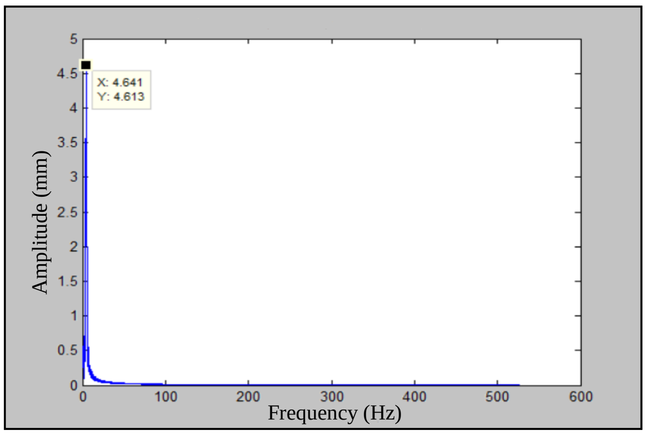


(a) Velocity frequency response

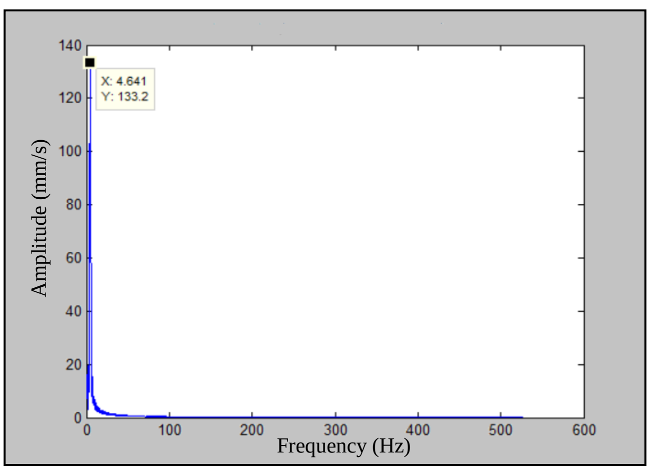


(c) Acceleration frequency response

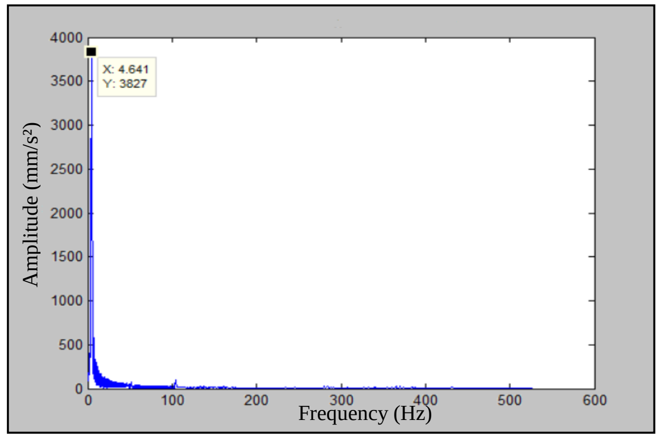
Fig.10 FFT of experimental data of single lap-1Bolt beam



(a) Displacement frequency response

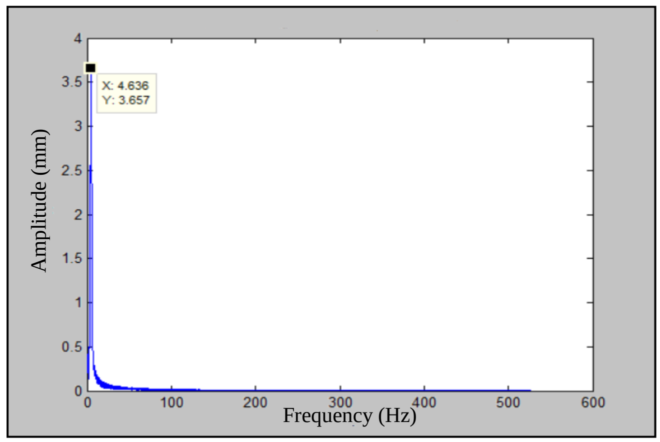


(b) Velocity frequency response

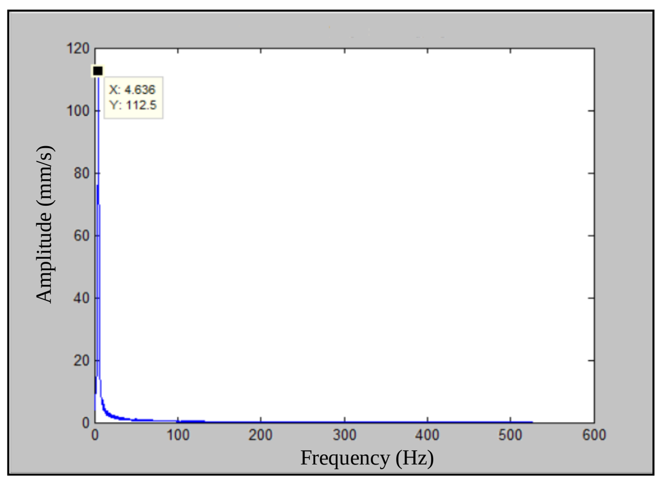


(c)Acceleration frequency response

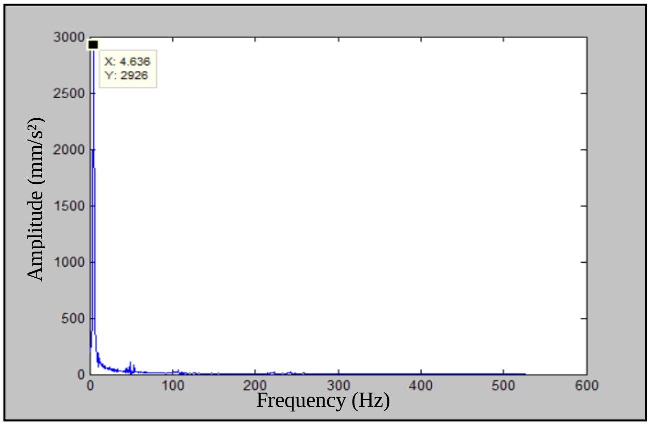
Fig. 11 FFT of experimental data of single lap-2Bolt beam



(a) Displacement frequency response



(b) Velocity frequency response



(c) Acceleration frequency response
Fig.12 FFT of experimental data of double lap beam

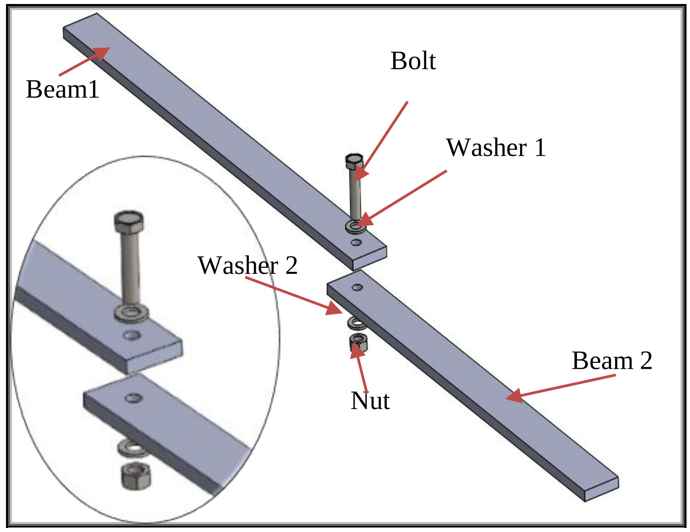


Fig.13 Bolted joint structure

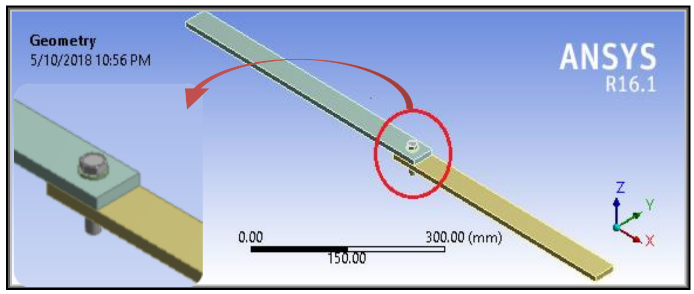
III. FINITE ELEMENT ANALYSIS OF BOLTED JOINT

To simulate and verified experimental results, Finite Element Analysis i.e. (Pre-stressed modal analysis and Pre-stressed harmonic analysis) of bolted structure is achieved.

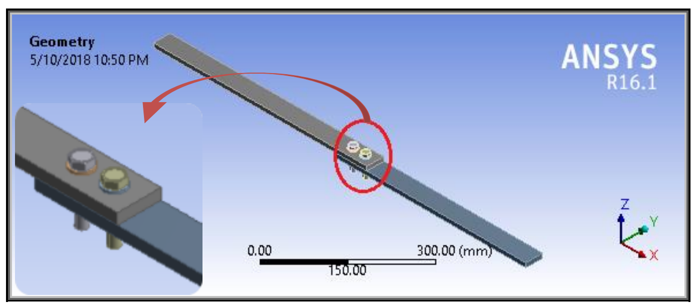
Three dimensional models of bolted joint were developed by using modelling software (Solid Work ver. 15) and standard commercial software ANSYS ver.16.1 package) to simulate the dynamic analysis of these joints.

A. Description of finite element model

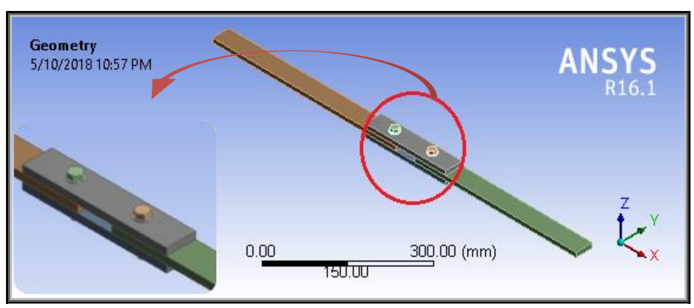
This section describes the case studies under the study. Three case studies have been designated with different lap joints, First case: single lap-1Bolt joint, Second case: single lap-2Bolt joint and Third case: Double lap joint. These three cases of bolted joint are modelled by 3D solid bolt with pretension load and contact elements. In general, the bolted joint structure has four major components: flat beams, hexagonal headed bolts, washers, nuts as shown in Fig .13. The assembled models of the bolted beams can be showed in Fig. 14.



(a) Single lap -1Bolt



(b) Single lap-2Bolt



(c) Double lap bolted joint

Fig.14 Assembled models in ANSYS workbench

B. Meshing of bolted joints

All the parts of bolted structure were meshed by using Tetrahedrons patch confirming method with Quad/ tri mesh type. Ten-node Solid elements (SOILD187) were used for meshing procedure. The element is defined by ten nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions .it has a quadratic displacement behavior and is well suited to modeling irregular meshes. The element has plasticity, hyper elasticity, creep, stress stiffening, large deflection, and large strain capabilities [adopted from ANSYS help]. The type and number of elements in the FE model has an important influence on the response .The quality of mesh is high with the maximum element size of 5mm and the minimum element size of 2mm with coarse relevance and span angle center. The numbers of solid elements for the three cases mentioned in the previous section are 7249, 8033and 19399 respectively. The typical finite element mesh that used in the current work are shown in Fig.15.

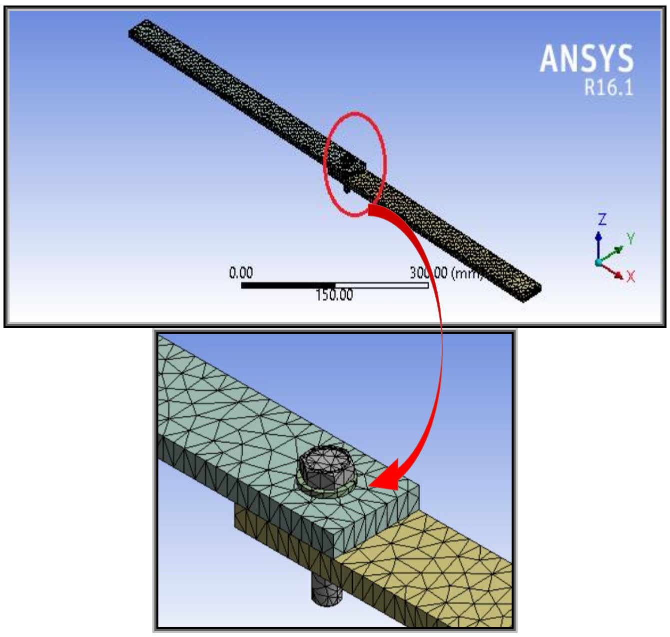


Fig.15 Meshing of bolted joint assembly

C. Contact condition

The strength of the bolted structure depends on Modeling of contact conditions. As the Forces are applied, the elements of contact and target faces are possible to penetrate in to each other. This penetration can be avoided by presenting proper contact type between interfaces of the assembly and appropriate formulation of solver. The contacts are described between the beam and bolt assembly as well as between the two beams at the lap joint. A surface-to-surface contact elements, which consists of contact elements (CONTAC174) and target segment elements (TARGE170) are used on the interfaces between the bolted surfaces. In the present work nine contact connections are founded for all contact surfaces in the finite element model as shown in Fig.16. Coulomb friction model was utilized as it’s propped through ANSYS workbench with a coefficient of friction (μ) equal to 0.2.

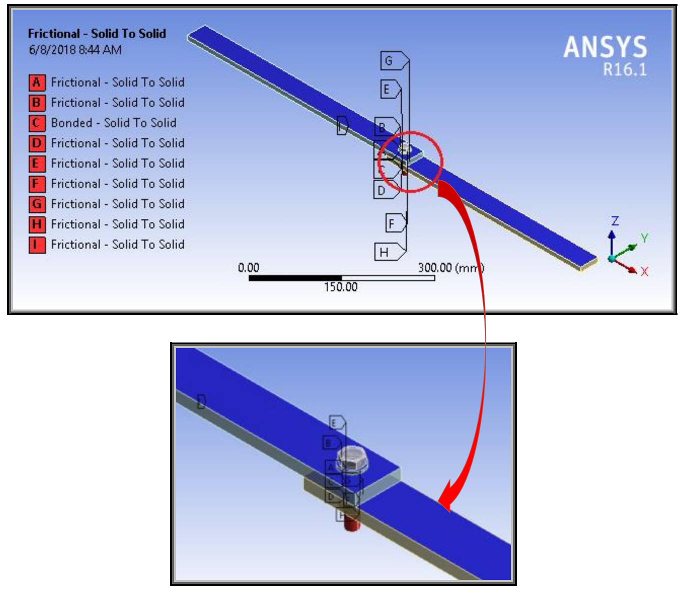
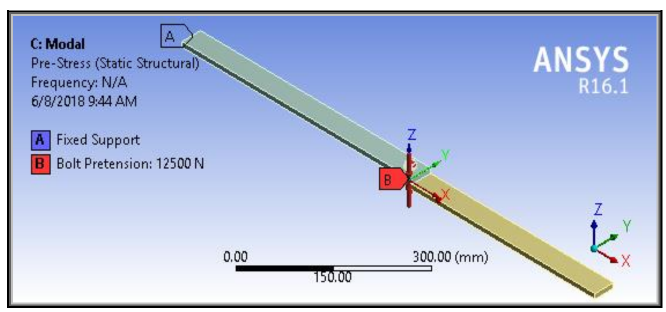


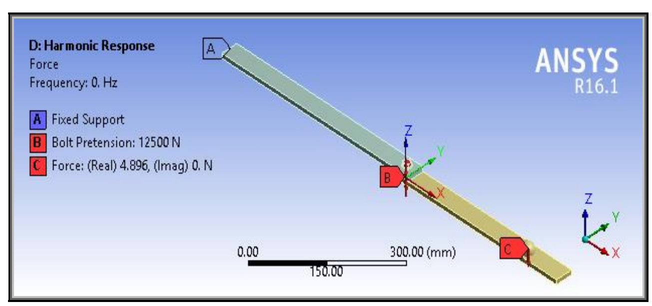
Fig.16 assembly with preferred contact

D. Applying boundary conditions to the bolted joint assembly

In the bolted beams, all degrees of freedom of all nodes and elements of outer left side surface of beam-1 are fixed supported. Though stretching bolt and nut by tightening torque, clamping force is transferred from the washer face of bolt head and nut i.e. contact face to face of washer and fastened beam i.e. target face respectively. The contacts apply uniform distributed load on the target faces. The preload force is also applied on the cylindrical surface of the bolt shank along the axial direction which is equal in magnitude but opposite in direction of each other as shown in Fig.17.



(a) Modal analysis



(b) Harmonic analysis

Fig. 17 Boundary conditions for modal and harmonic analysis

E. Solution

After applying the boundary condition and applying external forces and bolt preload, the models have been solved for modal analysis (i.e. free vibration) and harmonic analysis (i.e. forced vibration).

F. Verification of finite element model

It is necessary for any finite model to confirm if the results obtained agree with other models .This proves that the model is real sufficient to arrest the residue of the results. So, the comparison between the present experimental results for all case studied as shown later and previous published experimental result of bolted cantilever beam with single lap under free vibration which is reported by Toshimichi F. et al. [9] is executed. Fig.18 shows the changes of natural frequencies of the beam due to preload force are compared with the result of Toshimichi F. et al. [9]. As seen from the figure agreements are good.

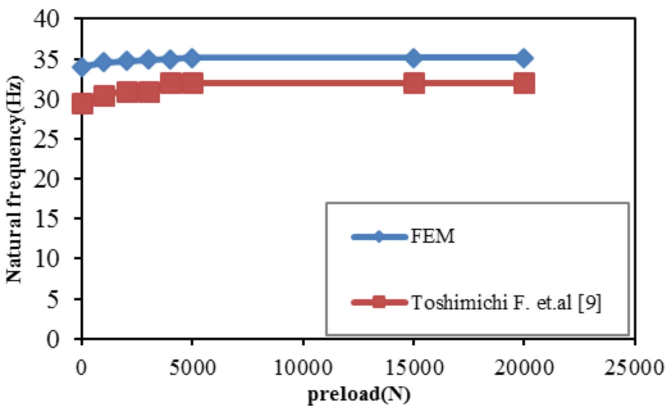


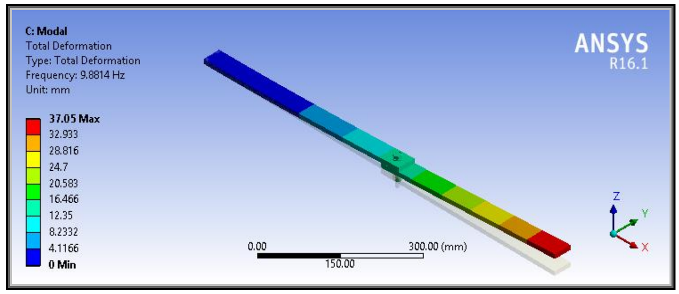
Fig.18 effect of preload on natural frequency

G. Case studies

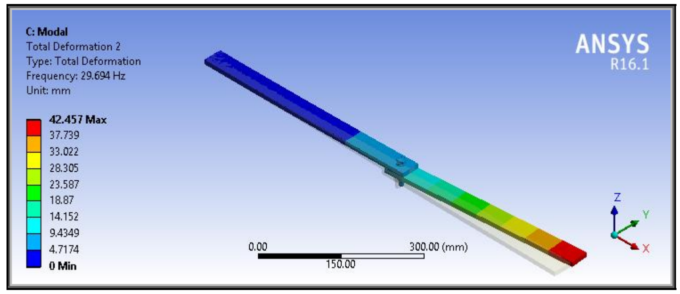
G.1 Free vibration

In many mechanical applications, the natural frequencies of vibration are interesting. This type of dynamic analysis is likely to be the most commonly and is indicated as an 'eigenvalue analyses'. Besides to these natural frequencies, the mode shapes of vibration which initiated at the natural frequencies are also of interest. The fundamental natural frequencies at preload 12.5KN of the three cases are found to be 9.8814, 10.116, 9.3646 Hz respectively.

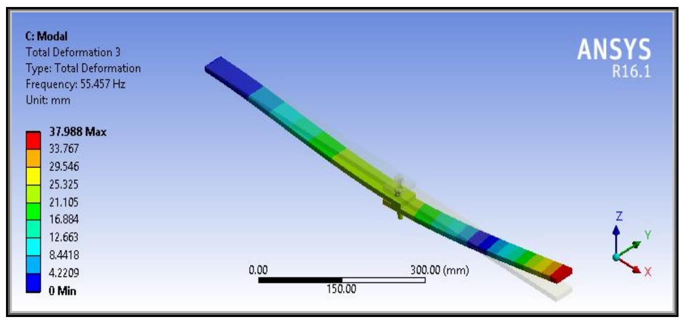
Vibrational modes are inherent properties of the structure and be influenced by damping, mass, boundary conditions and stiffness. Each mode has its own mode shape, natural frequency and modal damping ratio. The first three mode shapes of different beams configuration are shown in Fig.19.



Mode-1

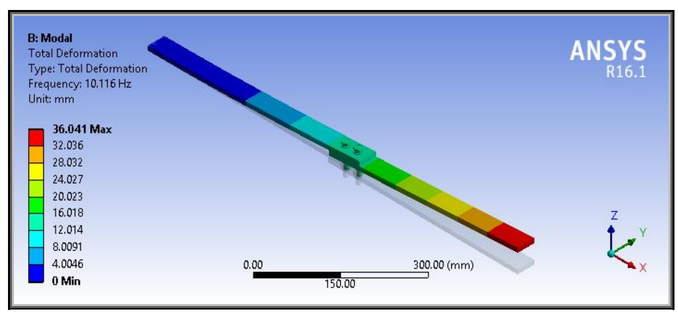


Mode-2

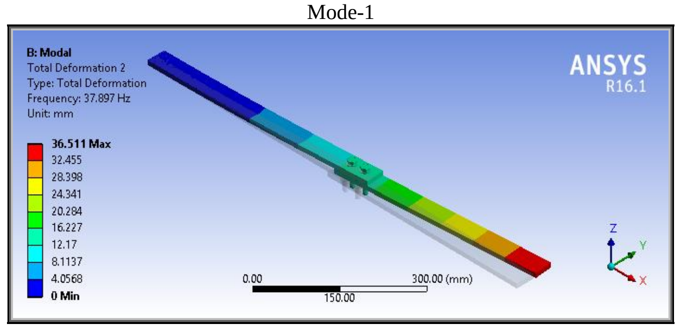


Mode-3

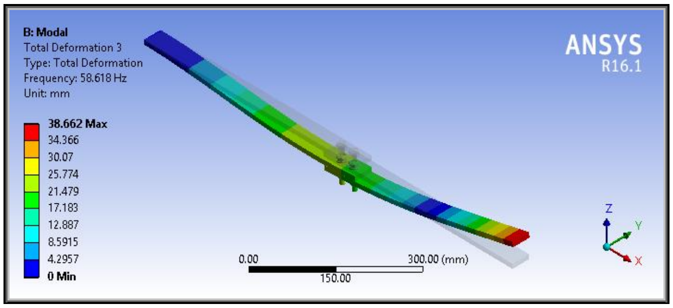
(a) Single lap-1Bolt



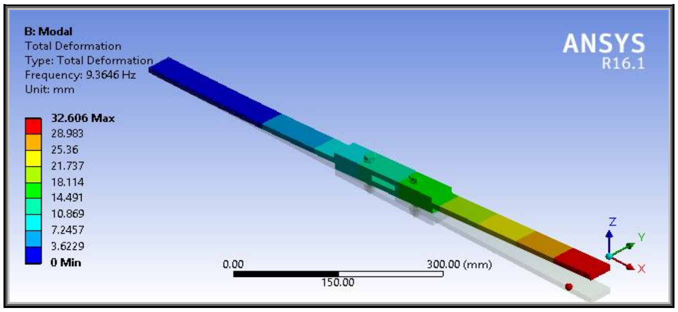
Mode-1



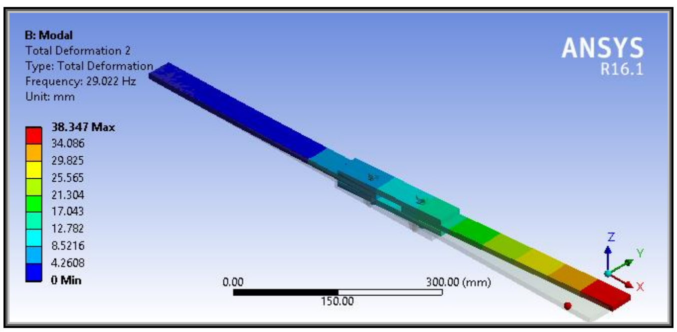
Mode-2



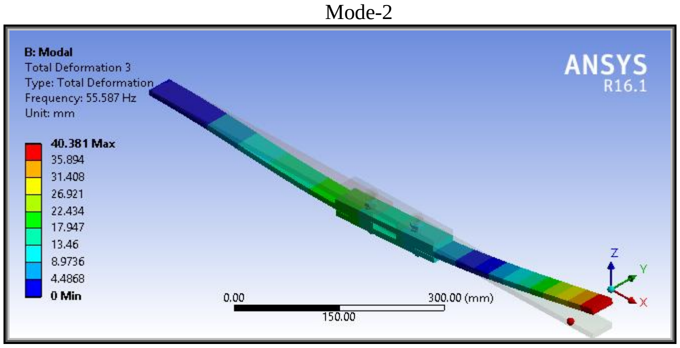
Mode-3
(b)Single lap-2Bolt



Mode-1



Mode-2

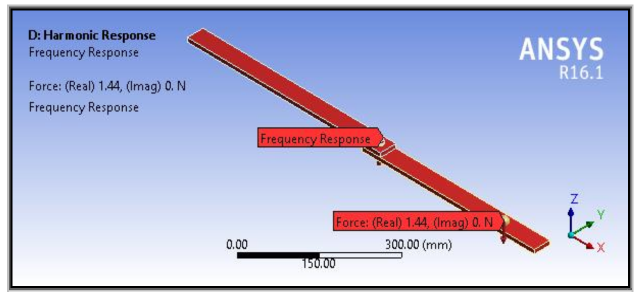


Mode-3
(c) Double lap bolted joint
Fig. 19 mode shapes for different beams configuration

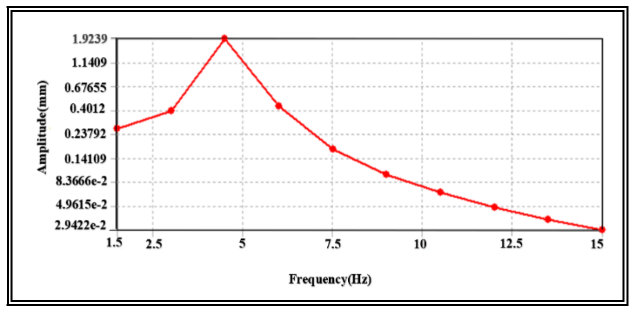
G.2 Forced vibration

When the external energy is provided to the system during vibration, a structural or mechanical system is supposed to undergo forced vibration. This external energy can be provided to the system through either an imposed displacement excitation or an applied force. The displacement excitation or applied force may be harmonic, non-harmonic, periodic, non-periodic, or random in nature. The response of system to harmonic excitation is called harmonic response or vibration amplitude [15].

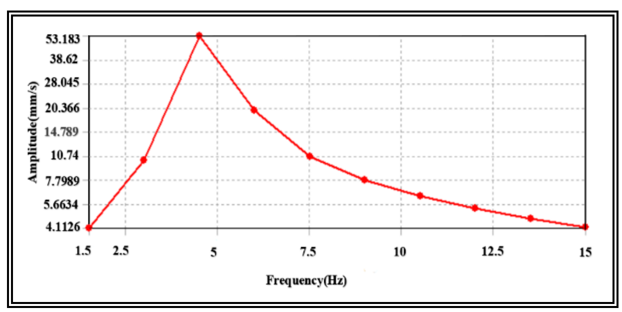
To simulate the forced vibration in ANSYS software, the force is applied at point of the model and vibration responses are measured in this point. In this study, the method used to conduct harmonic analysis is the Mode-superposition method. This method employs the mode shapes and natural frequencies from modal analysis to describe the dynamic response of a structure to harmonic excitations. Also, it is used when the frequency band is limited. Fig. 20 shows the vibration responses (i.e. displacement, velocity and acceleration) at preload 12.5KN, motor position at point A and speed at 300 RPM for three cases.



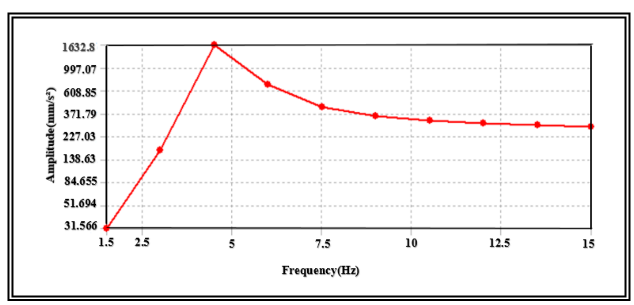
Harmonic response



Displacement frequency response

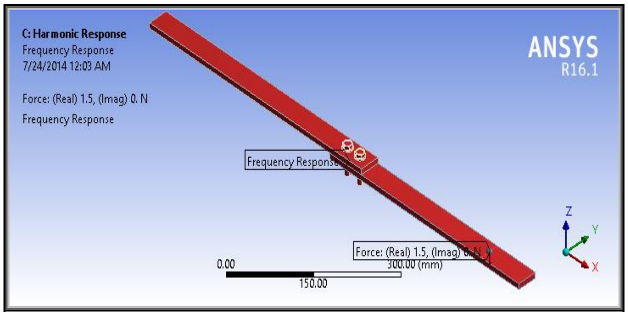


Velocity frequency response

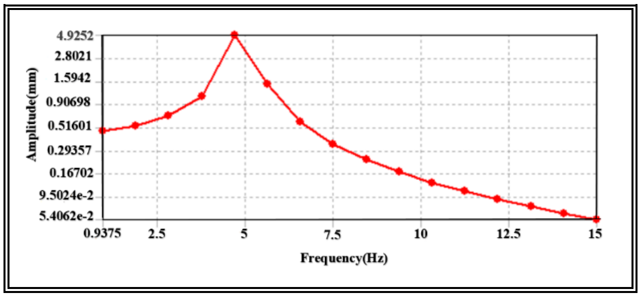


Acceleration frequency response

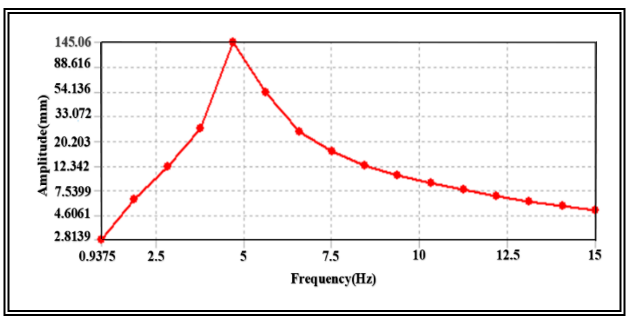
(a) Frequency Reponses for case 1



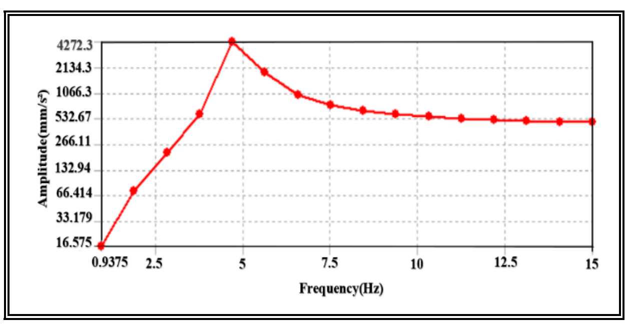
Harmonic response



Displacement frequency response

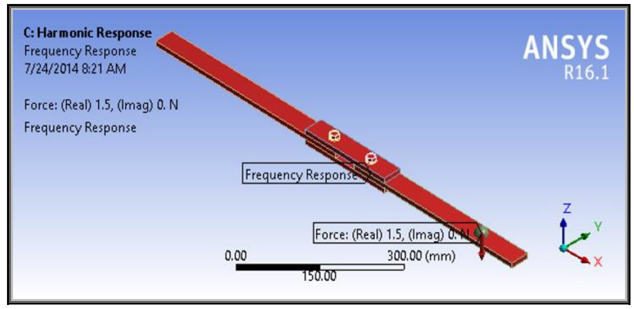


Velocity frequency response

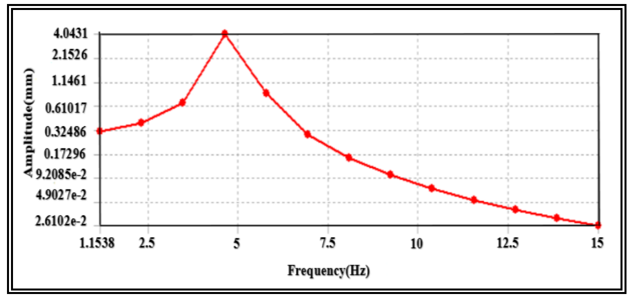


Acceleration frequency response

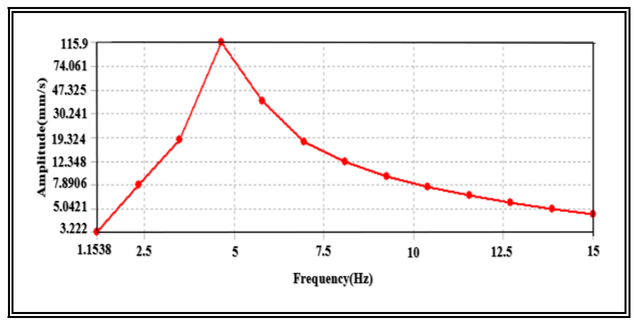
(b) Frequency responses for case2



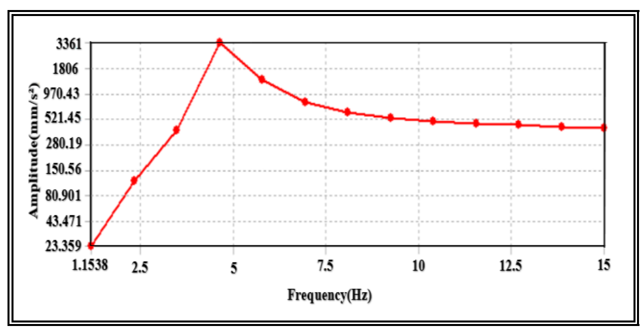
Harmonic response



Displacement frequency response



Velocity frequency response



Acceleration frequency response

(c) Frequency responses for case 3

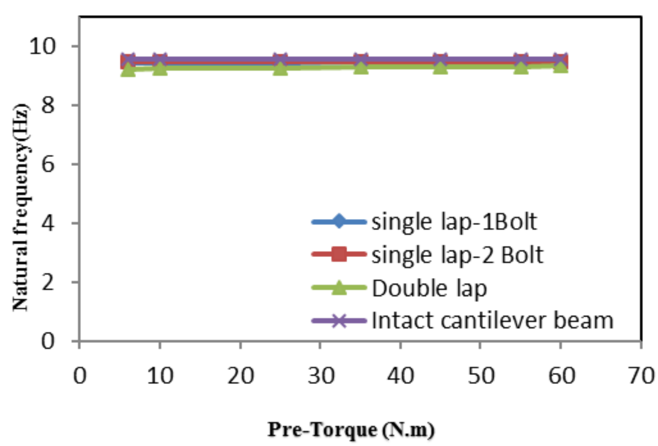
Fig.20 Frequency responses for different beams configuration

IV. RESULTS AND DISCUSSION

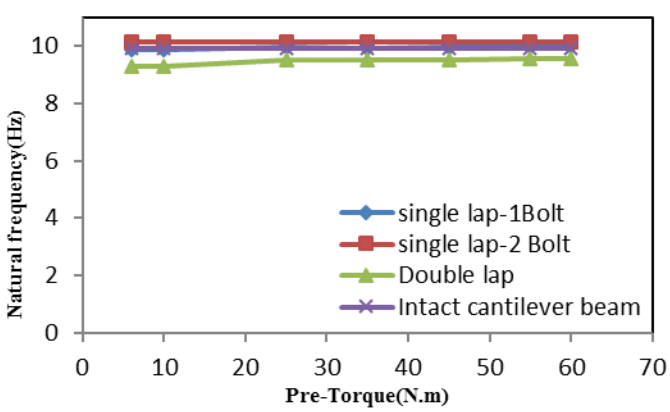
A. *free vibration case study*

A.1 Effect of beam configuration on natural frequency and vibration amplitude

The experimental and EFM investigations are shown in Fig.21. The natural frequencies of bolted joint beams with single lap (1Bolt, 2Bolt) are roughly equal to those of intact beam with same profile. But, in the case of double lap bolted beam the natural frequencies are slightly less than those of intact beam with same profile. This behavior may be attributed to the effect of mass increase in double lap beam case than other beams .Also, frictional slipping between connected members which can be considered as a primary damping mechanism at the joint. Small regions of the interface area will be released and subject to micro slip when the force is applied to a joint. This motion arises without any relative displacement between the members and only on a very small interface region. By increasing the force applied to the joint, larger parts of the interface will be released, which finally leading to macro slip with relative displacement between the members. This lead to energy losses occurs over the joint interface which will be more in the case double lap bolted beam.



(a) Experimental investigation

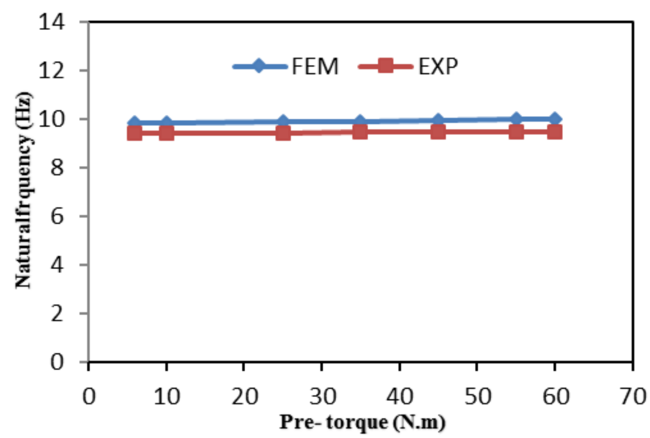


(b) EFM investigation

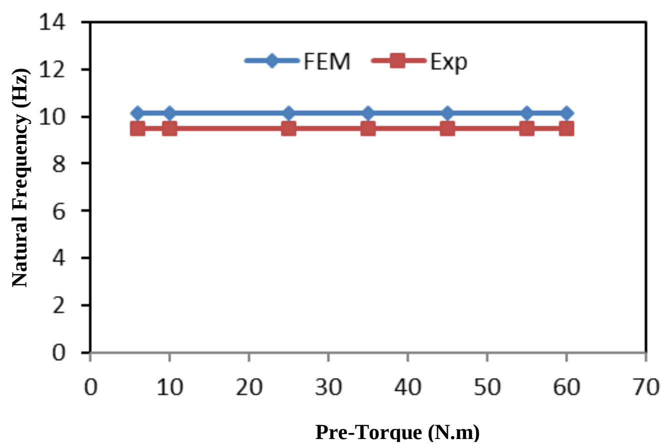
Fig.21 Effect of beam configuration on natural frequency

A.2 Effect of preload on natural frequency

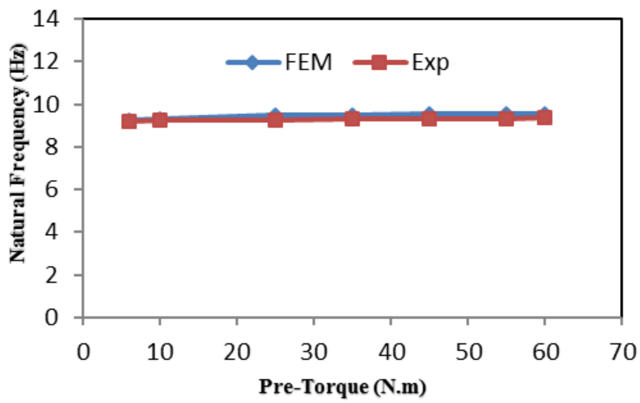
Fig.22 displays the values of the natural frequency versus the pre-torque for different beam configuration. In general, it can be seen from this figure that as preload or pre-torque on the bolt increases at the range (6-60N.m), the natural frequency becomes constant. This behavior may be due to the relationship between the joint stiffness and natural frequency. An increase of up to half in the natural frequency in case 1 can be observed before this range. This behavior can be interpreted to that the preload may not provide a sufficient stiffness to make two cantilever beams to be as an intact beam. In case 2, because the additional stiffness provided by two bolts the natural frequency remains constant before and after this range. Also, it can be noted that in case 3 there is a slight increase in natural frequency at the range (6-25N.m) after this range goes to a constant value. The suggested numerical method is validated by comparing the fundamental natural frequencies obtained by FEM with the experimental results as shown in Fig.22.



(a) Single bolted lap joint



(b) Double bolted lap joint

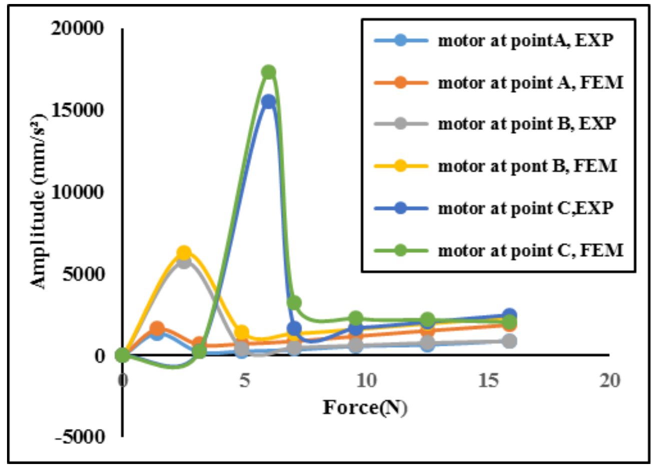


(c) Double lap bolted joint

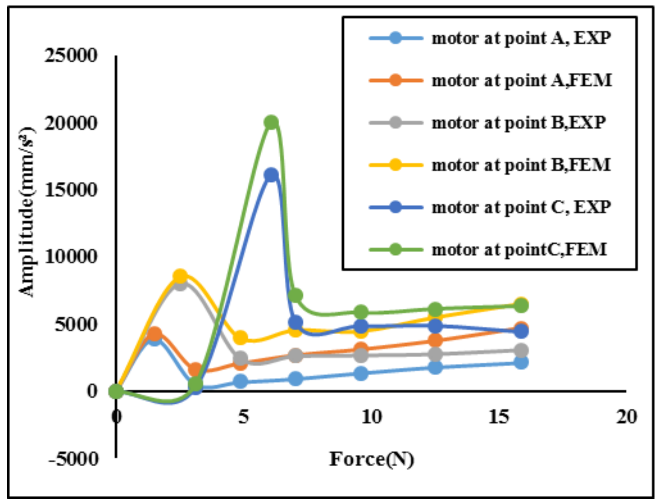
Fig.22 Effect of preload on natural frequency for different beam configuration

B. Effect of magnitude and position of harmonic force in forced vibration case study

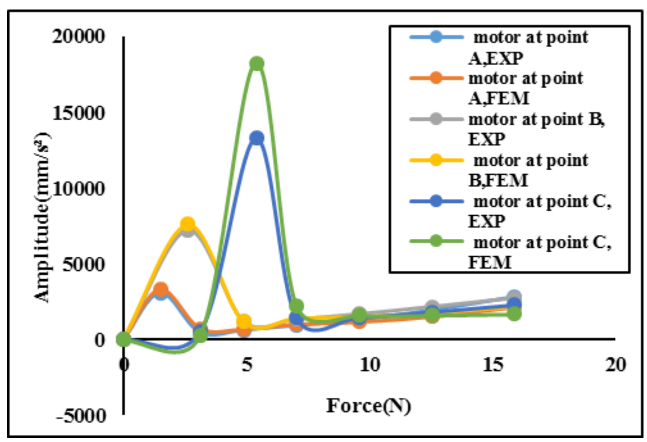
Harmonic force can be changed by controlling the angular velocity of motor (ω) and maintaining mass and it is eccentric position constant. Also, the position of harmonic force is changed by varying the position of motor along of the beam .Due to existing similarities in the behavior of acceleration, velocity and displacement responses, only the effect of magnitude and position of force on vibration amplitude of acceleration response is considered as shown in Fig. 23. This figure shows that for all types of beam configurations, the vibration amplitude is largely dependent on magnitude of harmonic force. As this force increases, the vibration amplitude goes on increasing. At a specific value of harmonic force, the amplitude reaches it is maximum value (resonance). But after that there is a reduction in vibration amplitude because after resonance all the stored energy is depleted and again goes on increasing with increasing force. In addition, it can be observed that when the motor position near to the fixed end (i.e. at point C), the vibration amplitude (at resonance) for all beam configuration be more than other points. This is due to a difference in damping relative to type of joint and position of motor. The bolted joint's damping differs with the excitation frequency. When the frequency of excitation increases, the damping value decreases.



Single lap-1bolt



Single lap-2bolt



Double lap

Fig. 23 effect of magnitude and position of harmonic force on vibration amplitude

VI. COCLUTIONS

Several important points were concluded from this study:

1. Good agreement between the numerical and experimental results with error range of about (3%-5%).
2. The natural frequencies of single lap (1Bolt, 2Bolt) bolted beam are roughly equal to those of intact beam , but, in double lap bolted beam they are slightly less than those of intact beam with same profile.
3- As preload or pre-torque on the bolt increases at the range (6-60N.m), the natural frequency becomes constant for all types of beam configurations.
4- The vibration amplitude is largely dependent on magnitude of harmonic force. The higher the harmonic force, the greater the vibration amplitude.
5- The vibration amplitude for all beam configuration is very large at motor position (x=60cm) from free end.

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VIII. BIOGRAPHIES

Nathera A. Saleh was born in Basrah – Iraq. She received her degree in Mechanical Engineering (Applied Mechanics); B.Sc. in 1993, M.Sc. in 1996, and PhD in Mechanical Engineering (Applied Mechanics-Biomechanics) from Basrah University/Engineering College in 2006. She is currently an Associate Professor at the Mechanical Engineering Department, Basrah University to teach and supervise undergraduate, M.Sc. and PhD students. Dr. Saleh research interests are fracture mechanics, vibration, buckling, composite material, biomechanics and finite element method.

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