Effect of friction on structural damping of beams

Dr. Abdul Amir Abdul Zahra
Technical institute of Kerbala

1- Summary:

The effect of friction on the structural damping of beams was investigated for a simple supported beam load centrally by an electric motor with a rotating unbalances. The friction generated by bolts was found to improve the structural damping to an extent depending on the location of the bolts, there number and the tightness torque.

In this research we have proved the validity of the continues beam theory and Raleigh's Method. We also had shown that frictional forces can be used to improve the vibration characteristics of mechanical system by properly selecting the bolts location and the magnitude of the tightness torque. A high torque does not necessarily means good vibration damping characteristics.

2-introduction:

In the past manufacturing procedures dictated that the production of machines to be made from small components. These components were joined together by bolts, rivets....etc. Also, these components were mostly covered of limited knowledge in the strength of materials and their alloys, and the computational methods were not advanced as in today's modern technology, especially with the accessibility of computer services to every design office.

The problems of vibration were not obvious at first. But with advance in the metal science and computation method, components were made smaller creating vibrations especially at certain frequencies excitation. This necessitated a study in the dynamical behavior of mechanical systems. With the advent of numerically controlled machine tools, it is now possible to machine a complicated piece of machinery from a solid block. Vibrations become worse than before due to reduction in the structural damping. This reduction was attributed to lack of friction between the various components that used to constitute the overall compound component.

The object of this research is to investigate the effect of friction on the frequency response of a structural element. A simply supported beam with a rectangular cross-section was selected as the structural element for the purpose of this work.

3- Structural damping:

Unbalance in rotating machines a common source of vibration excitation. Let us consider a machine with a rotating unbalance which is supported by a beam, as shown in Figure (1a). This can be represented by a spring-mass- despot system with a rotating unbalance as shown in Figure (1b). The rotating unbalance is represented by an eccentric mass (m) with eccentricity (e) which is rotating with angular velocity (w).

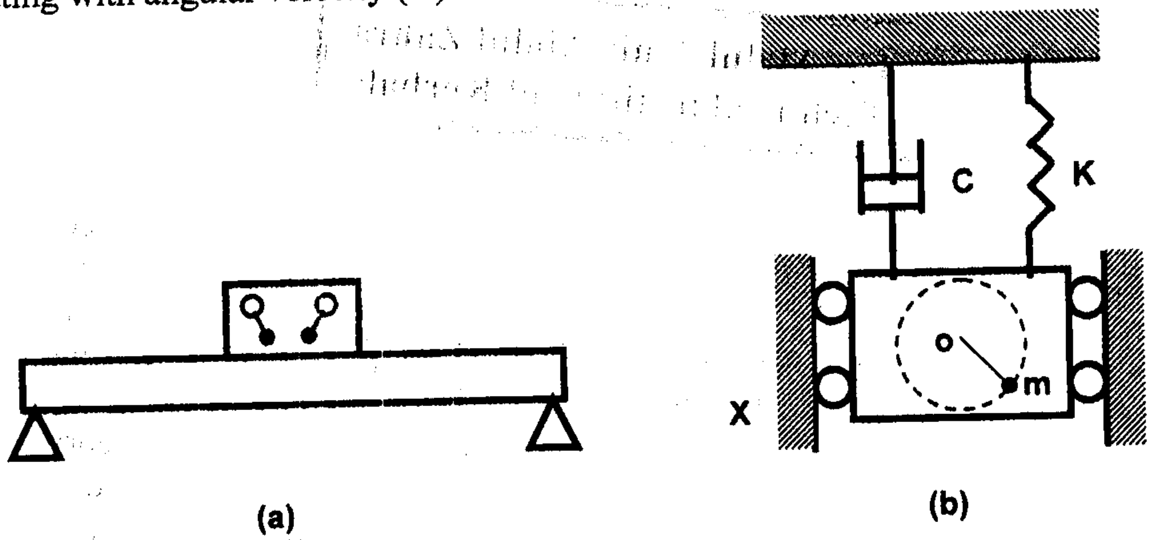

Fig. (1) a) Simply supported beam with a rotating unbalance.
b) Equivalent spring- mass- daspot system

The governing equation of motion can be obtained by applying Newton's second low of motion to the system of Fig. (1b).

$$x + e\sin(wt) \qquad \text{.....................(1)}$$

where (x) be the displacement of the non-rotating mass (M-m), and the equation of motion is then:-

$$(M-m)\ddot{x} + m\left(\frac{d^2}{dt^2}\right)(x + e\sin(wt)) = -kx - c\dot{x} \qquad \text{..............(2)}$$

which can be rearranged to

$$M\ddot{x} + c\dot{x} + kx = mew^2 \sin wt \qquad \text{.............(3)}$$

the solution to this equation is the sum of two parts, the complimentary function, which is the solution of the homogenous equation, and the particular integral.

We are interested in particular solution alone since the system is in steady state conditions. Equation (3) can be written in a dimensionless form, that is:-

$$\ddot{x} + 2\zeta w_n\dot{x} + w_n^2 x = \left(\frac{me}{M}\right)w^2 \sin wt \qquad \text{............(4)}$$

Where

W_n= natural angular frequency $=\sqrt{\frac{k}{M}}$ rad/sec.

ξ= damping ratio $= \frac{c}{2Mw_n}$

The solution of equation (4) is [1]

$$\frac{MX}{me}=\frac{\left(\frac{w}{w_n}\right)^2}{\sqrt{\left[1-\left(\frac{w}{w_n}\right)^2\right]^2+\left[2\xi\frac{w}{w_n}\right]^2}}\sin(wt-\theta)\text{.................}(5)$$

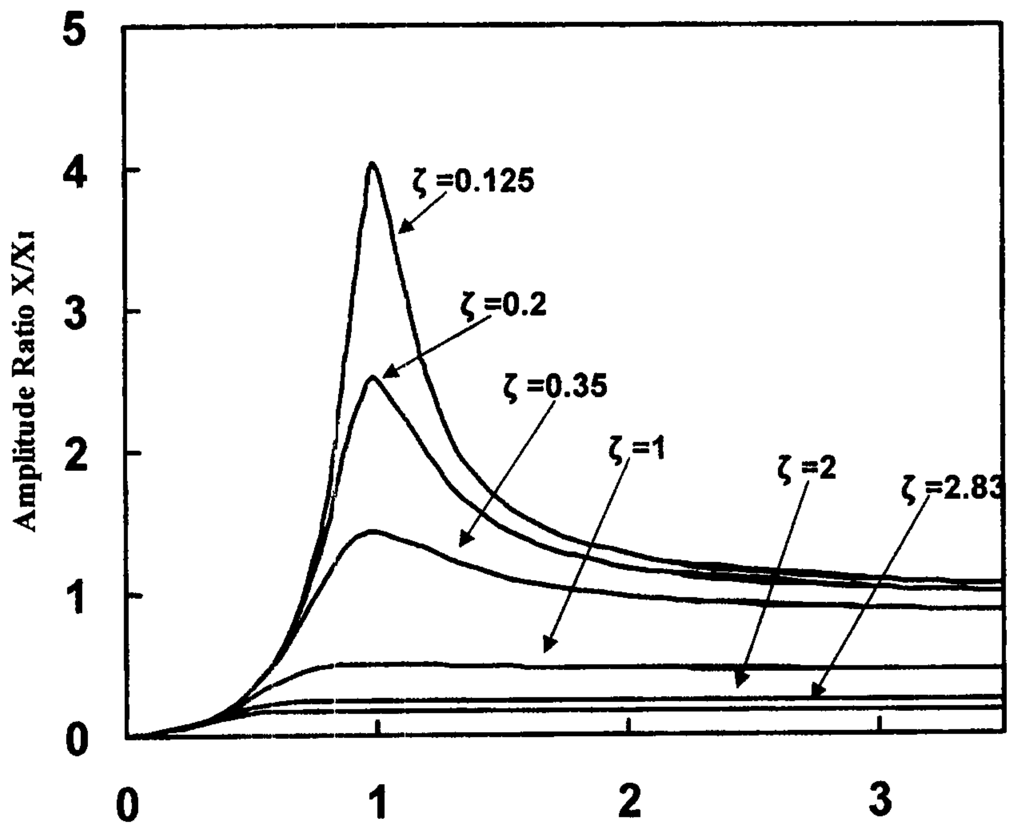

Fig. (2) Frequency response of a rotating unbalance mass expressed by equation (5).

Where

$$\theta = \tan^{-1}\frac{2\xi\left(\frac{w}{w_n}\right)}{1-\left(\frac{w}{w_n}\right)^2} \qquad (6)$$

Fig. (2) is a pot of the magnitude $\frac{MX}{me}$ versus, the frequency ratio $\frac{w}{w_n}$ for different damping ratio (ξ).

When structural material is cyclically stressed energy is dissipated within the material. Experiment shown that for most materials the energy dissipated per cycle of stress is independent of the frequency and proportional to the square of the strain amplitude[2].

That is

$$U = \alpha . x^2 \qquad (7)$$

It is an acceptable practice to replace damping forces of the real vibration systems by a force which is proportional to the velocity of vibration is,

$$F_d = \rho . \dot{x} \qquad (8)$$

Where ρ is the damping coefficient. This is possible if ρ is calculated so that the replacement system dissipates the same amount of energy per cycle as the real system[3].

It can be shown that the energy dissipated by a linear spring-mass-daspot system at resonance is given by [3]

$$U = \pi\rho\, w x^2 \qquad (9)$$

Equations (7) and (9) can be used to find an equivalent damping coefficient for a real structural member, that

$$\rho_e = \frac{\alpha}{\pi\, w} \qquad (10)$$

The structural damping force must then be, from equation (8) ,

$$F_d = \rho_e \dot{x} = \frac{\alpha}{\pi w}\dot{x} \qquad (11)$$

From equation (5) the steady-state amplitude of system excited by an unbalance rotating mass is:

$$x = \frac{mew^2}{\sqrt{\left[k - mw^2\right]^2 + \left[\frac{\alpha}{\pi w_n} w\right]^2}} \qquad (12)$$

which at resonance has the value

$$x_r = \frac{\pi}{\alpha}\left(mew^2\right) \ldots\ldots\ldots\ldots\ldots\ldots\ldots (13)$$

Since in the harmonic motion $\dot{x} = wx$

Then

$$F_d = \frac{\alpha}{\pi} x \ldots\ldots\ldots\ldots (14)$$

From equation (14), the damping force is proportional to the displacement. It is convenient then to express this equation in terms of the spring force (k.x) multiplied by a no dimensional factor (γ)an called the structural damping factor. Then

$$\gamma . k = \frac{\alpha}{\pi} \ldots\ldots\ldots\ldots (15)$$

and from equation (13)

$$\frac{kx_r}{mew^2} = \frac{1}{\gamma} \ldots\ldots\ldots\ldots (16)$$

With this substation in equation (12),

$$\frac{Mx}{me} = \frac{\left(\frac{w}{w_n}\right)^2}{\sqrt{\left[1 - \left(\frac{w}{w_n}\right)^2\right]^2 + \gamma^2}} \ldots\ldots (17)$$

which can be used in calculation the amplitude of a structurally damped system in the region of resonance.

Compared to the equation(5) for viscously damped system, with $w = w_n$, i.e. at resonance with small damping ratio, the value of

$$\frac{Mx}{me} = \frac{1}{2\xi} \ldots\ldots\ldots\ldots (18)$$

from which we obtain

$$\gamma = 2\xi \ldots\ldots\ldots\ldots (19)$$

that is the structural damping factor is twice the viscous damping factor.

4- Experiment rig and procedure:

The experimental set-up used consisted of:-

A:- Frame with two vertical I-beams and four U-beams for the bas. A specially designed brackets with beam-rings type 203EWB were manufactured. The surfaces which guides the bearings were machined with very high precision.

Two wooden blocks were fitted underneath the base to isolate the frame. See Fig (3).

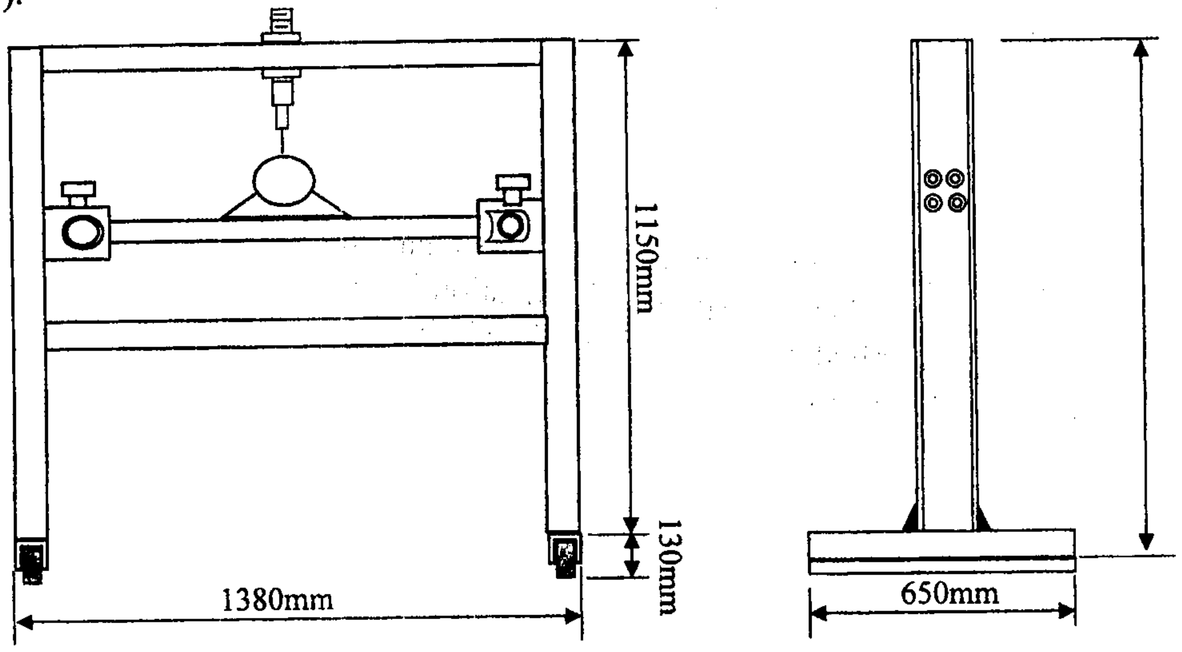

Fig. (3) Frame design

B:- Test beam:- Three beams were accurately machined to measure:-

1- 25 X 50 X1220 (mm)
2- 19 X 50 X 1220 (mm)
3- 6 X 50 X 1220 (mm)

The last two beams when put together they form a beam of the same dimensions as the first beam.

C:- Instrumentation :- An inductive type transducer element was attached between the unbalanced exciter frame and the rig' s frame. A special bracket was designed and manufactured to hold the transducer elements.

D:- Vibration exciter:- A synchronous electric motor type 126/53841 was used as a source of unbalanced excitation. The unbalanced rotating forces were generated by an unbalanced disc attached to the motor' s shaft.

5- Experimental results and discussion:

Fig (4) is a plot of the amplitude of vibration versus the member of repetition off the maximum of the exciting force per minute. The amplitude of vibration is small at low motor speeds since the maximum force mew2 is small. At resonance the amplitude become high (3.25 mm) and then drop to (0.25 mm) before it rises again. The second peak is close to the second harmonic but not coincident with it. From the experimental results, resonance occurs at 1675 rpm and double this speed is 3350 rpm which is definitely away from the second peak.

The exciter and transducer are positioned in the center, so the amplitude of vibrations is expected to be $\left|{me}/{M}\right|$ (that is 0.25 mm) the second harmonic since it corresponds to a node. An examination of the frequency response curve verifies this.

Therefore the second peak is attributed to a second natural frequency of the overall system including the frame.

A comparison between the calculated fundamental frequency of the solid beam and the experimental results show that theoretical results are 1.5% to 6% higher than the experimental depending on the value of **E**. The difference is due to the experimental errors and the assumption made the theoretical derivations. However, we concluded that the theoretical results matches the experimental values and this proves the continues beam theory and Rayleigh's method[4].

A comparison between fig. (4) and (2) at high frequencies yield a value of ${me}/{M}$=0.25mm. This value is used to normalize the high peak of fig. (4) which gives (M/me) =(3.12 / 0.25)=12.48. The structural damping of the beam can be calculated with the aid of equation (17). Structural damping of the solid beam=$\gamma=\frac{1}{12.48}=0.08$.

A combined beam of overall dimension equal to that of the sold beam was made by using two beams measuring 19X50X1220 mm and 6X50X1220 mm. Two frequency response tests were carried out with the thinner beam at the bottom and top. The results are presented in fig. (5) and (6). There is no difference between the maximum amplitude at resonance and the resonance frequency. Therefore it was decided to examine the friction effect with the thinner beam at the top only.

An examination of fig.(5) and (6) shown a reduction in the resonance frequency to 1360 r.p.m. and an increase in the amplitude of vibration at resonance. The reduction in the natural frequency is expected since the stiffness **$48EI/(2L)^3$** of the individual beams is less than that of the solid beam. The increase in the amplitude at resonance definitely indicate a reduction in the damping forces of the double beam. This is attributed to the reduction in the material's internal forces. Fig. (7) through (15) summaries all the frequency response results obtained for the combined beam with different number of bolts, locations and with various tightness torque levels. Fig. (16) is a plot of maximum amplitude of vibration versus bolt tightness torque. The structural damping ratio is calculated as indicated above and plotted versus the tightness torque as shown in fig. (17).

For a double beam fixed with six bolts located symmetrically at 0.153, 0.305, 0.455 m from the center line of the beam, the effect of adding bolts with 2 m.kg torque increase the structural damping by about 28%. For the tightening of the bolts reduces the damping ratio slightly due to the loss in frictional forces as a result of the reduced relative motion between the two beams. It also indicates that the major part of the damping forces are produced by the restraining effect of the added bolts rather than the frictional forces between the two beams surfaces.

The use of the two bolts only located symmetrically at 0.305m from the beam's center line shown an increase in the structural damping ratio by about 35%. As the torque increase on optimum point is reached with torque=3 m.kg. This means that the frictional forces are now more effective than with six bolts and as the tightness increases a paint is reached where the frictional forces effects become less effective.

The effect of the torque on the damping ratio is more prominent with two bolts located at 0.٣١٠m as shown in fig. ١٧. The maximum structural damping ratio has improved by about 58%. This improvement is attributed to the fact that the two bolts used located in a highly strained section of the beam.

The effect of the frictional forces on the damping is very clear. The increase in the tightness, torque reflects the same trend, namely a reduction in the damping forces due to the reduction in the relative motion between the beams surfaces.

In conclusion we have proved the validity of the continues beam theory and Raleigh's Method. We here also shown that frictional forces can be used to improved the vibration characteristics of mechanical system by properly selecting the bolts location and the magnitude of the tightness torque. A high torque does not necessarily means good vibration damping characteristics.

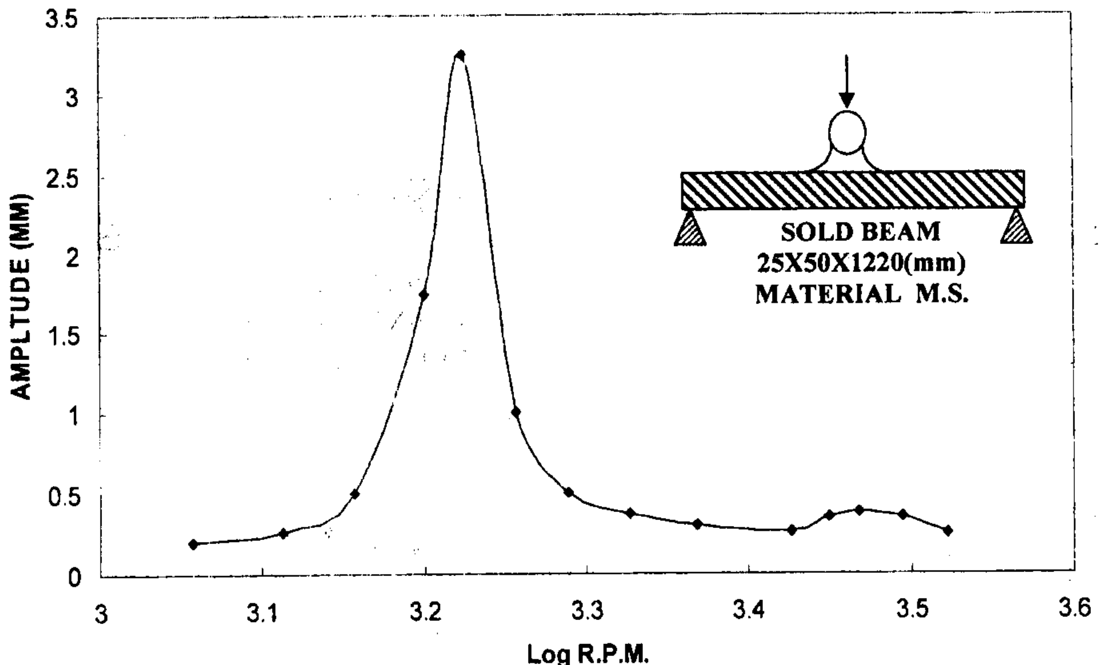

Fig. (4) Frequency response of a sold beam.

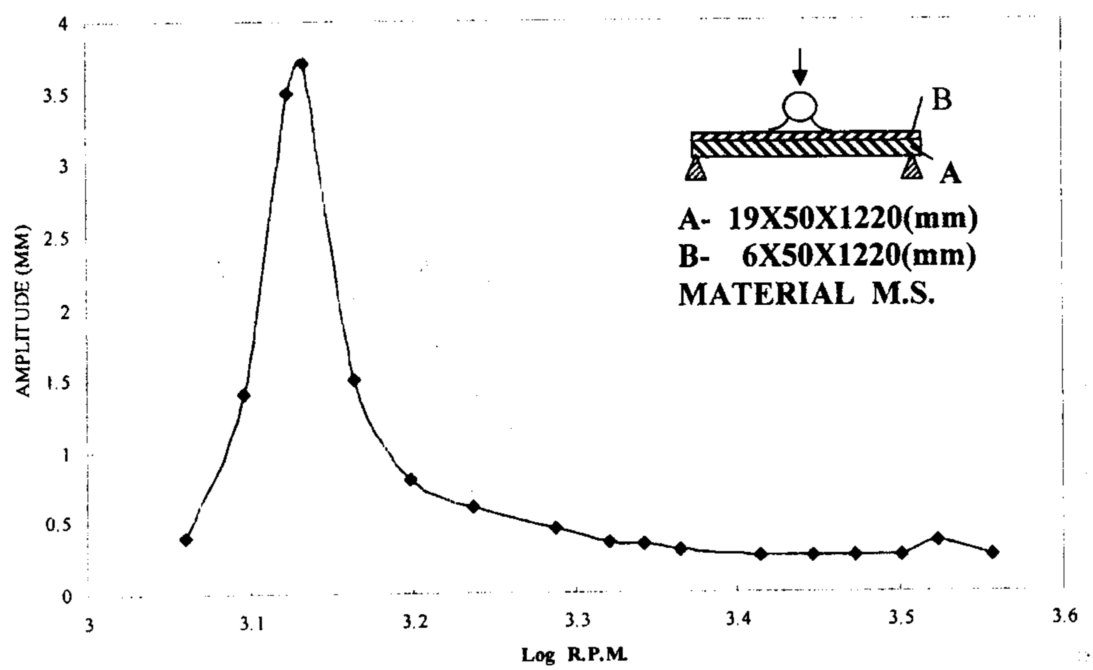

Fig. (5) Frequency response of a simply supported double beams.

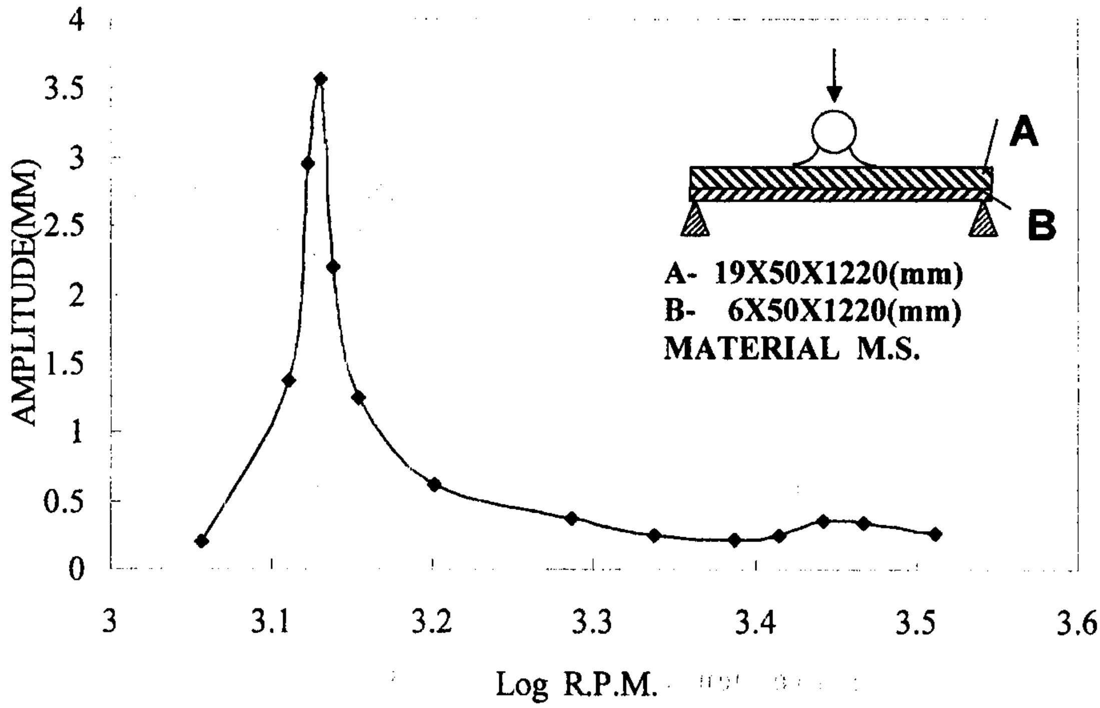

Fig. (6) Frequency response of a simply supported double beams.

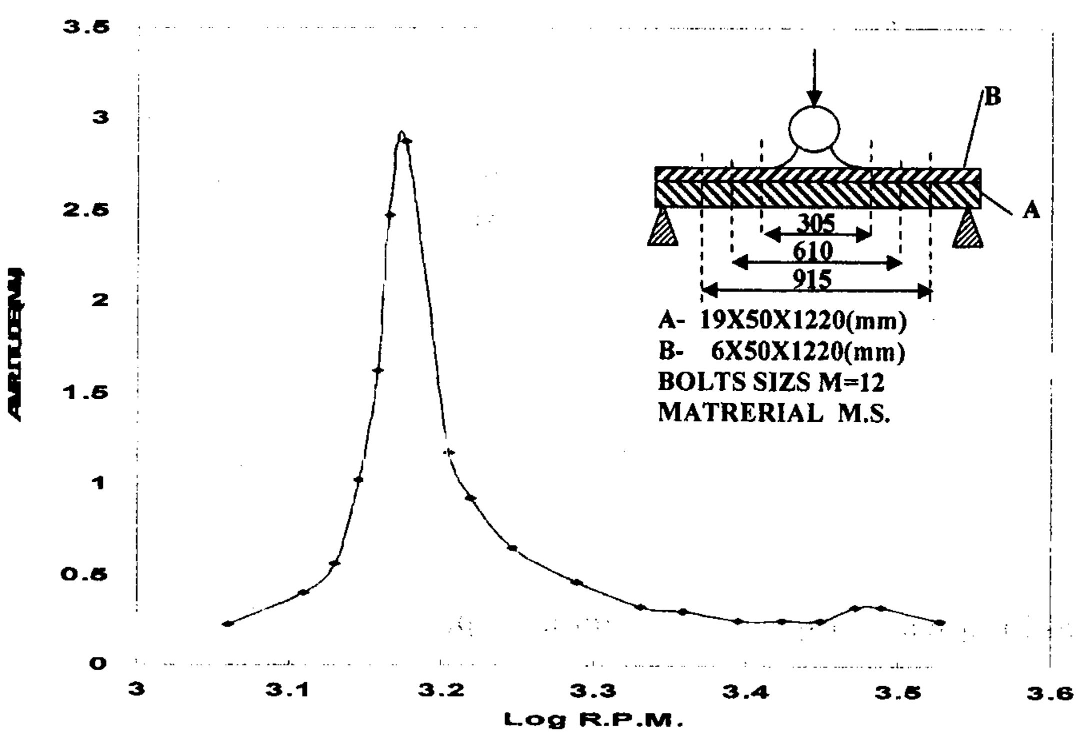

Fig. (7) Frequency response of a simply supported double beams fixed with six bolts. APPLIED TORQUE = 2 m.kg.

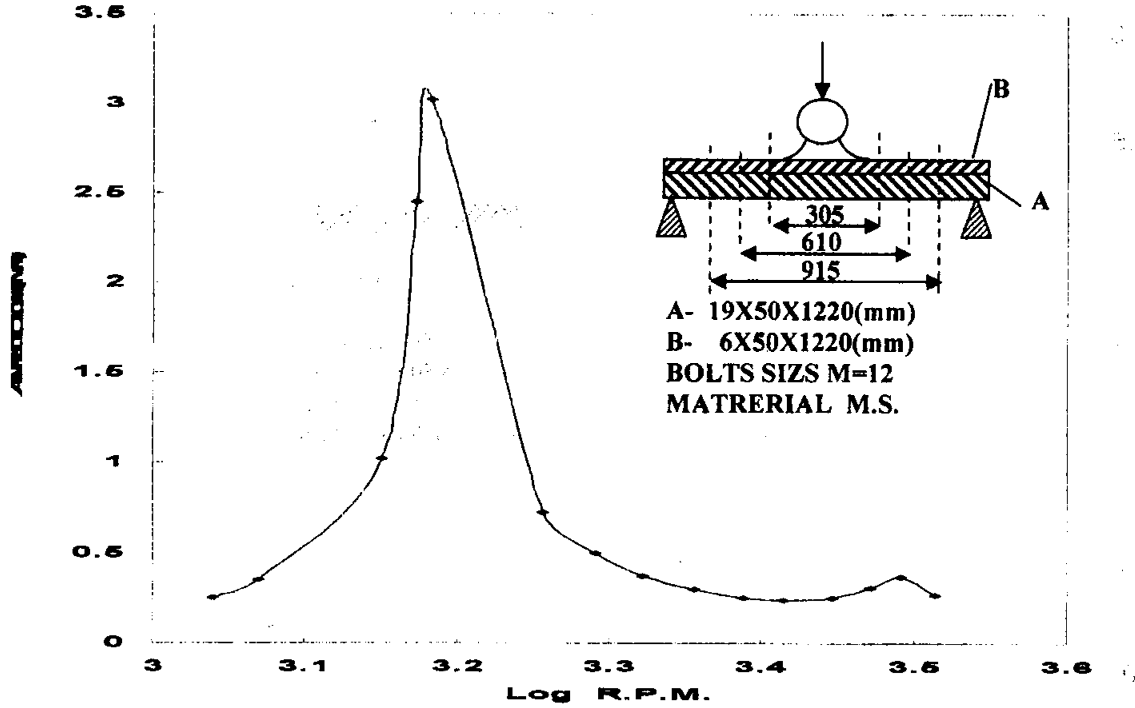

Fig. (8) Frequency response of a simply supported double beams fixed with six bolts. APPLIED TORQUE = 3 m.kg.

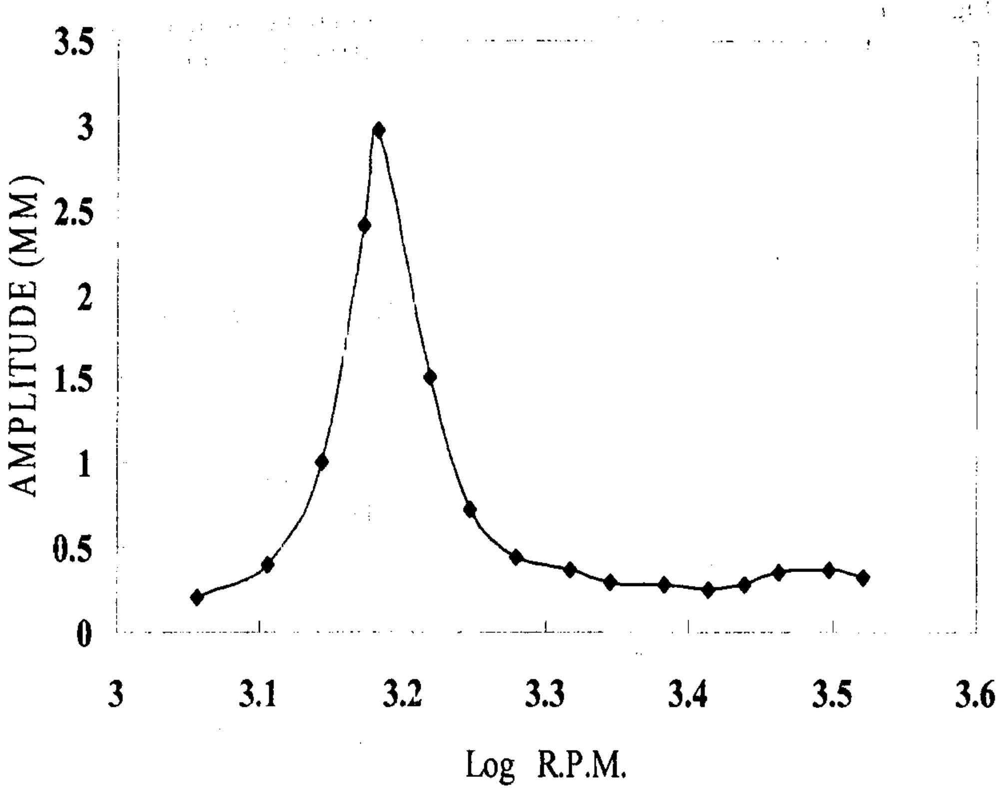

Fig. (9) Frequency response of a simply supported double beams fixed with six bolts APPLIED TORQUE = 5 m.k

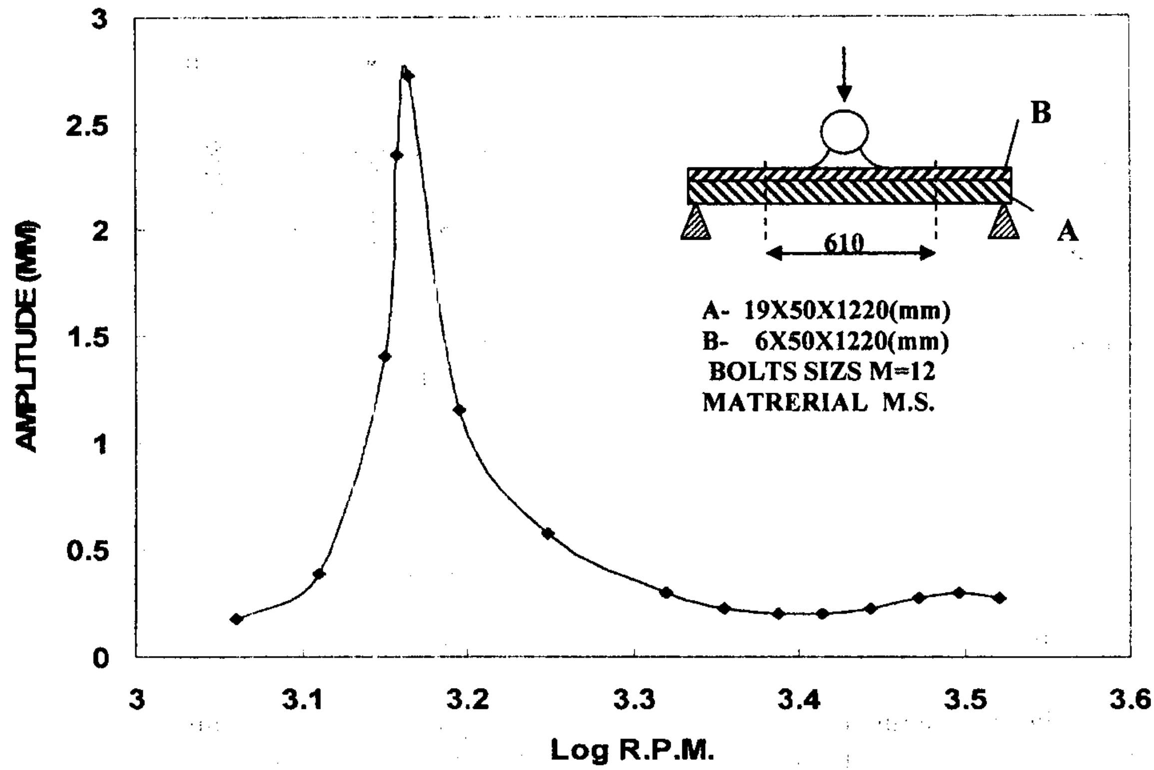

Fig. (10) Frequency response of a simply supported double beams fixed with two bolts. APPLIED TORQUE = 2 m.kg

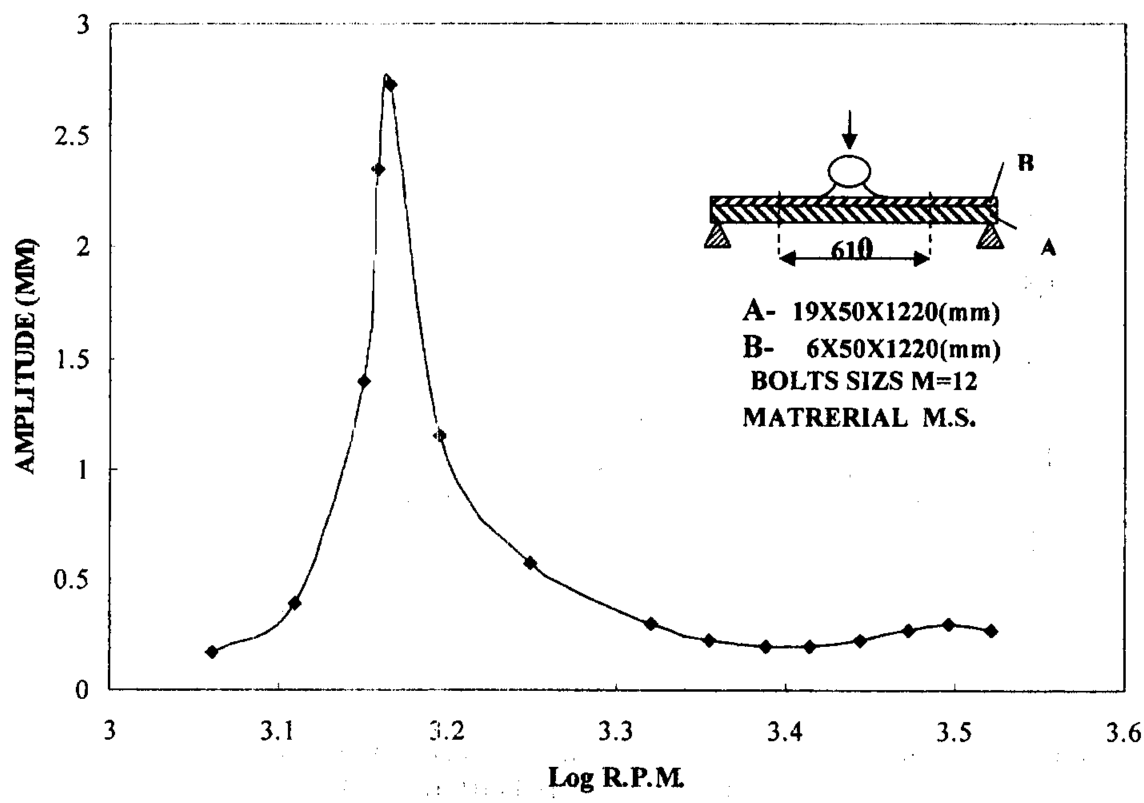

Fig. (11) Frequency response of a simply supported double beam fixed with two bolts. APPLIED TORQUE = 3 m.kg

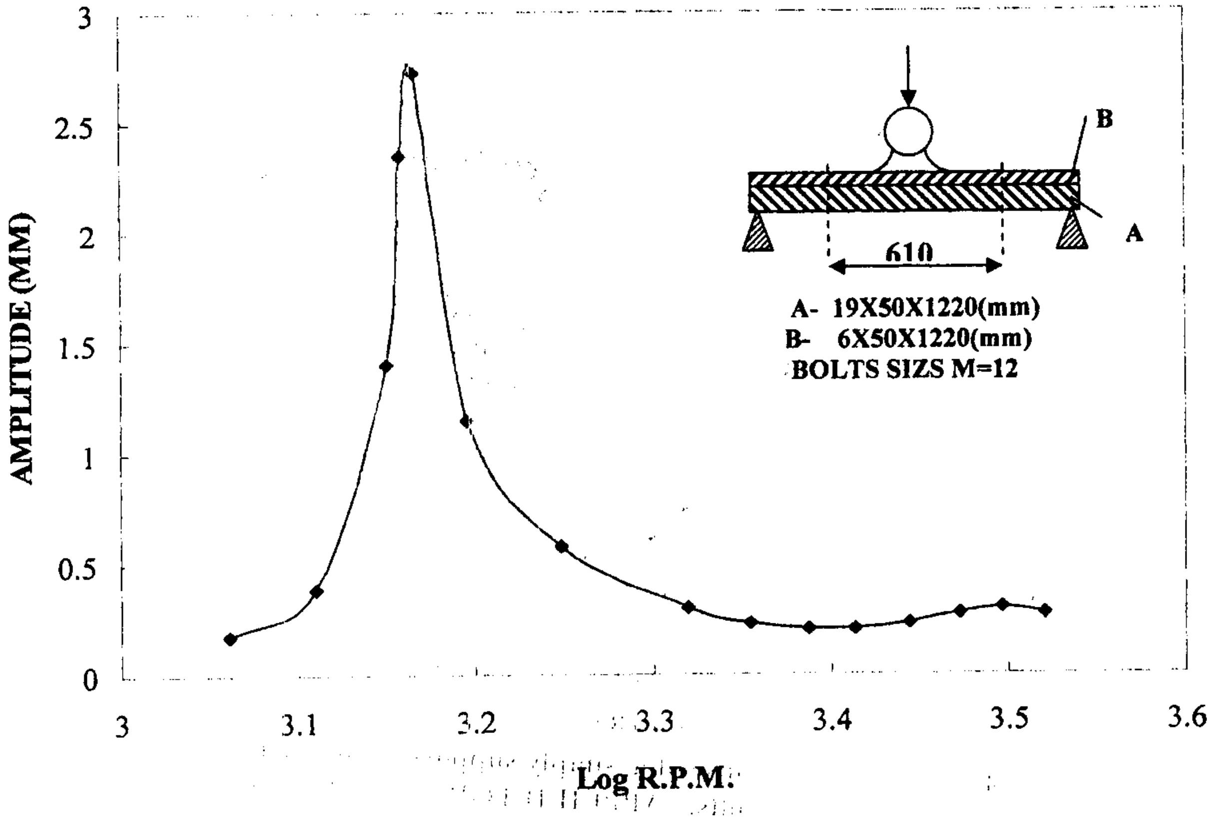

Fig. (12) Frequency response of a simply supported double beams fixed with two bolts. APPLIED TOROUE = 5 m.kg

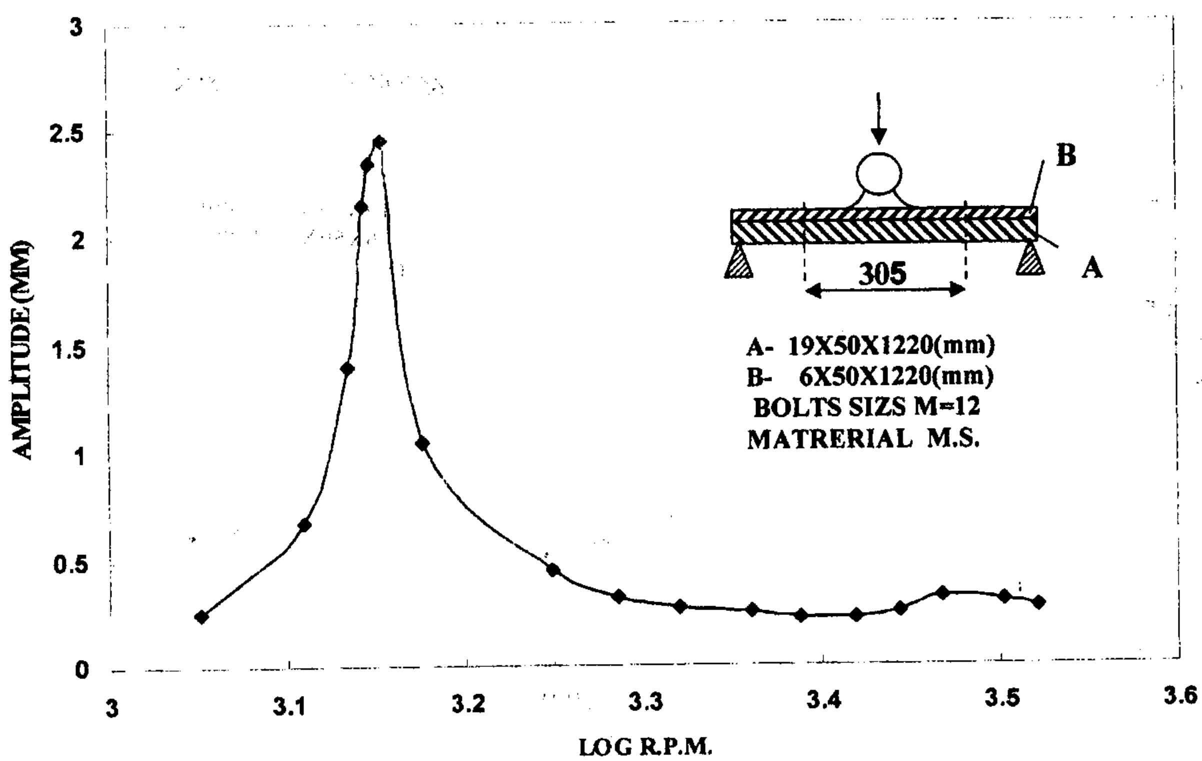

Fig. (13) Frequency response of a simply supported double beams fixed with two bolts. APPLIED TORQUE = 2 m.kg

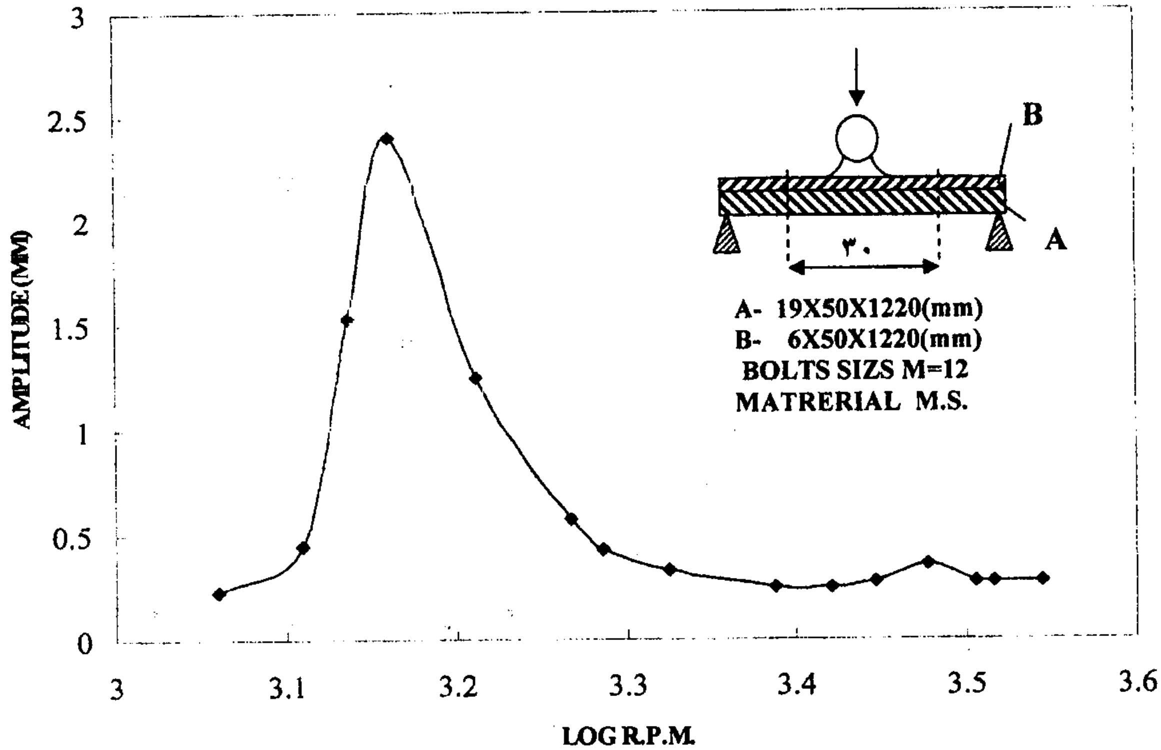

Fig. (14) Frequency response of a simply supported double beams fixed with two bolts. APPLIED TORQUE = 3 m.kg

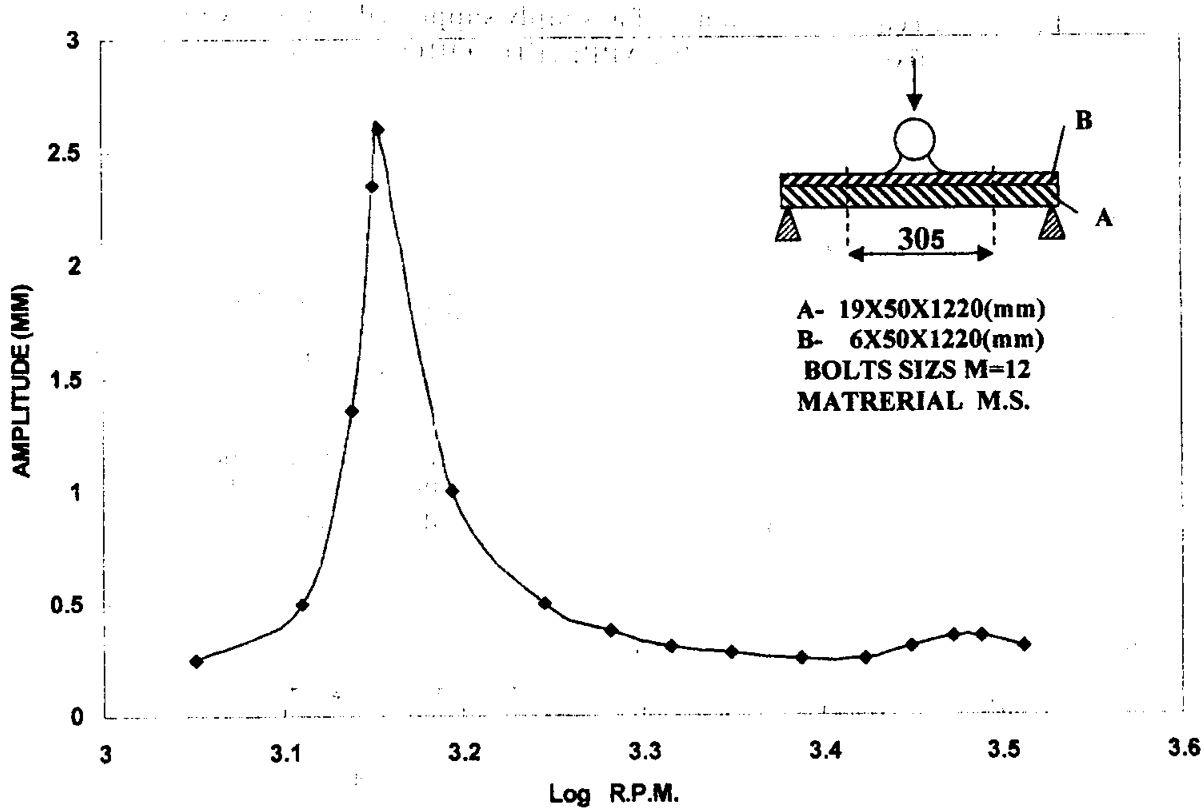

Fig. (15) Frequency response of a simply supported double beams fixed with two bolts. APPLIED TORQUE =5 m.kg

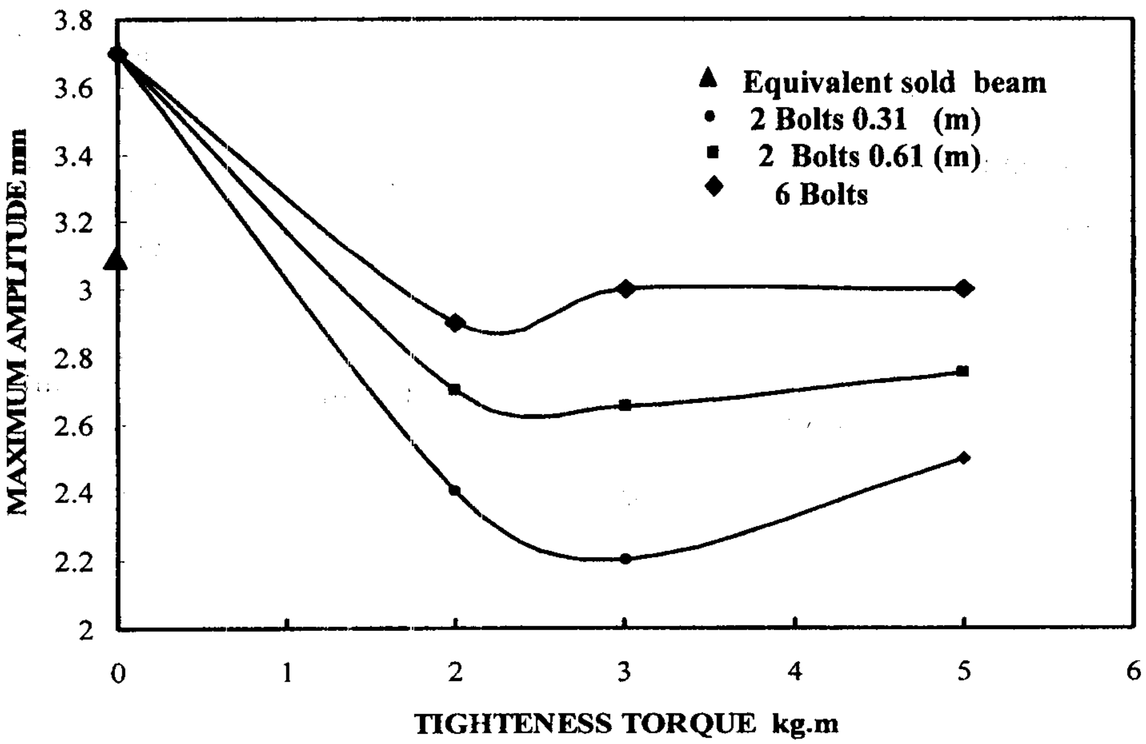

Fig. (16) Effect of bolt tightens on the maximum amplitude

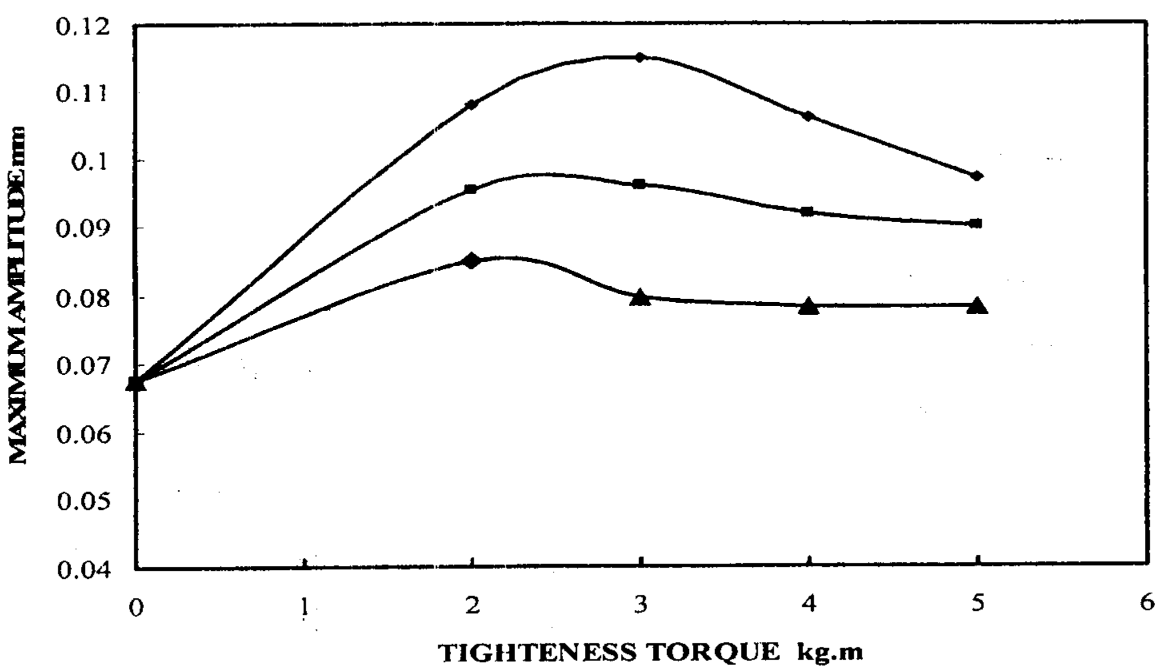

Fig. (17) Effect of bolt tightens torque on the structural ratio γ

Reference

1- William T. Thomason," Theory of vibration with Applications" ", 3rd Edition Butter Worth- Heinemann, (1988).
2- H.E. Barnacle and G.E. walker, "Mechanics of Machines ", Vol 1., (1986).
3- J.P. Denhartog, "Mechanical Vibration", Published by Mc Graw- Hill, (1987).
4- Paul H. black and O. Eugene Adams, JR, "Machine Design", ٤rd Edition John Wiley and Sons, New York, (1982).

تأثير الاحتكاك على المضائلة الذاتية للقضبان

الخلاصة

تم في هذا البحث دراسة تأثير الاحتكاك على المضائلة للقضبان الحديدية ذات المقاطع ألمستطيله المثبتة تثبيتا بسيطا من الجانبين والمثقلة في وسطها بمحرك كهربائي يحتوي على قرص غير متزن لتوليد الاهتزازات ألازمة.

لغرض زيادة الإحتكاك بين القضبان فقد تم ربطها بالبراغي بمسافات مختلفة من مركز القضبان. حيث تم دراسة تأثير عدد البراغي المربوطة والمسافات فيما بين هذه البراغي إضافة إلى تأثير قوة الشد المستخدمة في الربط على المضائلة للقضبان.

لقد تم التوصل في هذا البحث إلى إمكانية استخدام قوة الاحتكاك لتحسين الموصفات الاهتزازية للمنظومة الميكانيكية من خلال الاختيار الأمثل لمواقع الربط إضافة إلى اختيار العزم المستخدم في ربط هذه البراغي. حيث تم التوصل إلى أن استخدام عزم كبير في ربط البراغي لا يعني بالضرورة تحسين المواصفات الاهتزازية للمنظومة الميكانيكية.