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Transverse Vibration Test Rig for Threaded Fasteners

Darshan D. Deshpande, Krushna B. Handibag, Shubham D. Chavan, Prof. J. A. Kute
JSPM'S Rajarshi Shahu College of Engineering, Tathawade, Pune, India

Abstract: *The target of this review is to foster test rig for vibration releasing of bolts. Plan and advancement of framework parts for compatibility of various bolts. Test and Preliminary on test apparatus to foster the relaxing qualities of bolts what's more, determine the rot charts versus cycles. A catapulted joint including fasteners without washer just nut, screws with plain washer what's more, nut, fasteners with spring washer and nut, screws with Nyloc washer and nut was considered to complete the examinations. A similar examination was done with Relative investigation of rot qualities of individual bots with different end condition and to foresee vibration relaxing utilizing this review.*

Keywords*:* Transverse Vibration

I. INTRODUCTION

In every industrial application imaginable, one could not help but notice the numerous bolted joint connections that go into an equipment, component, or sub-assembly.

Bolted joints are one of the most commonly used methods to bring parts together and to secure them in place. However, there is an inherent weakness to this joining method by design. Without proper locking system in place there is a potential, under certain circumstances, where the bolted joints can experience self-loosening.

This is a phenomenon commonly observed under extreme vibrations and constant dynamic loadings where the joining parts begin to shift or slide. With many critical applications in the aviation industry that can experience both modes of external forces on a daily basis, it is imperative that the necessary preventative maintenance protocols are taken place. One of those protocols is to incorporate an effective locking system to the application's bolted joint. This will discuss the various locking methods and their effectiveness to prevent bolt self-loosening that are prevalent in this industry's environment.

Most bolted joints, especially the ones associated with machinery, are subject to significant vibration levels during their life span. Rotating or reciprocating machines, such as gas/steam turbines, electric motors and IC engines, are subject to vibration of relatively high frequency. Gyratory crushers, jack hammers and so forth are subject to medium frequency vibrations. Forging/stamping machines are subject to relatively low frequency, high amplitude vibrations. Certain dynamic structures (e.g., bridges and buildings subject to wind and cyclonic loads) also undergo dynamic load fluctuations. Cyclic temperature variations may also cause dynamic (very low frequency) load fluctuations in bolted joints.

II. LITERATURE REVIEW

The most influential paper on the self- loosening of threaded fasteners to-date was by Gerhard H. Junker, in "New criteria for self-loosening of fasteners under vibration", SAE Paper 690055, He studied a theory developed to predict self-loosening under vibratory loading occurs. He found that transverse dynamic loads generate a far more severe condition for self-loosening than dynamic axial loads. The reason for this is that radial movement under axial loading is significantly smaller than that which is sustained under transverse loading. [3]

According to Ravinder Kumar, research and development palwal, in "Causes and prevention of loosening in pre stressed bolts" he studied that vibration loosening of fasteners sets with different types of washers and Initial clamp load is highest in case of bolt with wave washer so due to this clamp load reduction is least. [19]

According to Umesh Dalal, Dr A.G.Thakur, in "Transverse Vibration Loosening Characteristics of Bolted Joints Using Multiple Jack Bolt Nut" they studied that significant difference in the loss of preload between fasteners with standard

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nut and with Multiple Jack Bolt Nut. With Multiple Jack Bolt Nut, loss in preload was only11.1% over 12000 cycles and bolt does not suffer any failure. While loss of preload in case of fastener assembled with standard nut was 42% for 8325 cycles and fatigue failure occurred due significant loss of preload. [6]

According to Haviland G S in "Unraveling the Myths of Fastener World" SAE paper 810509, he studied that the transverse joint movement and subsequent loosening can be arise from other mechanism besides direct shear loading. Differential thermal expansion due to temperature differences or dissimilar joint material can lead to joint slip. Bending of the joint can also lead to displacement and joint slip. [8]

According to Yamamoto A and S Kasai in "A Solution for self-loosening mechanism of threaded fasteners under transverse vibration", bull. Jpn. Soc. Precision Eng., they studied that a loosening is elastic torsion generated in the bolt shank during relative motion in the mating threads. The beginning of loosening process and one that leads to allowing transverse joint movement and significant fastener joint preload loss. [9]

III. PROBLEM STATEMENT

Fasteners coming loose are a common problem across many industries. We can complete an assessment of a fastener's self- loosening characteristics using a transverse vibration test machine. The fastener preload decay graphs produced can allow an assessment to be made of a fastener's resistance to self-loosening. Pre-loaded bolts (or nuts) rotate loose, as soon as relative motion between the male and female threads takes place. This motion cancels the friction grip and originates an off torque which is proportional to the thread pitch and to the preload. The off torque rotates the screw loose, if the friction under the nut or bolt head bearing surface is overcome, by this torque.

IV. PROPOSED METHODOLOGY

Designed model of pre-load developed for each predetermined pre-stress torque value, resultant clamp force generated.

Derivation of theoretical safe cycles for given system forces of vibration, amplitude and frequency.

Determination of forces and utilizing system of forces to determine the linkage dimensions of critical parts of drive.

3-D modeling of set-up will be done using Unigraphics Nx-8.0 and CAE of critical component and meshing using Ansys Work-bench 14.5.

The experimental validation part of the clamping force developed by individual bolt-washer-nut combination with those determined using Ansys software vibration", SAE Paper 690055, He studied a theory developed to predict self-loosening under vibratory loading occurs. He found that transverse dynamic loads generate a far more severe condition for self-loosening than dynamic axial loads. The reason for this is that radial movement under axial loading is significantly smaller than that which is sustained under transverse loading. [3]

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The experimental validation part of the clamping force developed by individual bolt-washer-nut combination with those determined using Ansys software

VII. EXPERIMENTAL SETUP

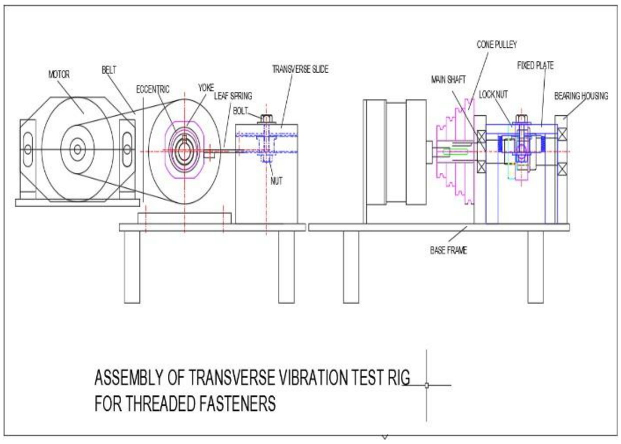

Fig. Schematic diagram for transverse vibration set up test rig

Motor Selection:
1- Phase induction motor Make: - godrej -boyce 230 volts, 50 hz,
Power = 0.25hp (0.185 kw) Speed = 1440 rpm (synchronous) Frame size =70
Current = 1.70 amp Torque= 0.17 kg. M
Torque at spindle is given by;

$$P = 2\pi N\ T/60$$

Where; T= Torque at spindle (Nm) P = POWER (Kw)
N = Speed (rpm)

$\Rightarrow T = 185 \times 60/\ 2\pi \times 1440$

$\Rightarrow T = 0.79$ N.m

Considering 100 % overload;

Tdesign = 2 T
= 1.56 N-m

$\Rightarrow$ Tdesign = 1.56 N.m

Maximum reduction ratio = 116/52 =2.23

Tdesign = 1.56x2.23 = 3.5 N-m

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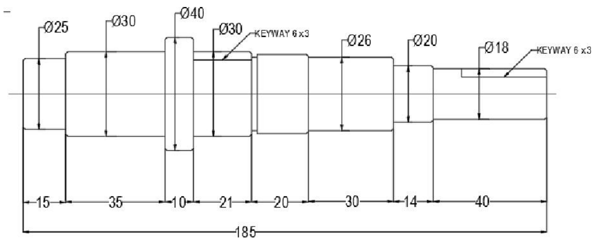
Design of drive shaft :



Material Selection ref :- psg (1.10 & 1.12) + (1.17)

Designation	Ultimate tensile Strength N/mm^2	Yeild strength N/mm^2
En24	800	680

⇒ fs allowable = 0.18 X 800=144 N/mm^2

⇒ T design = 3.5 Nm

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft Te = Π/16 fs d^3

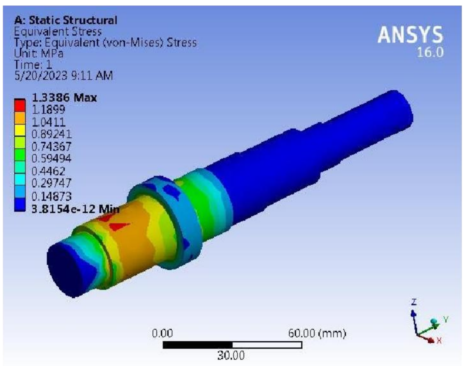
$$fs_{act} = \frac{16 \times 3500}{\Pi \times 16^3}$$

f act = 4.35 N/mm As fs act < fs all

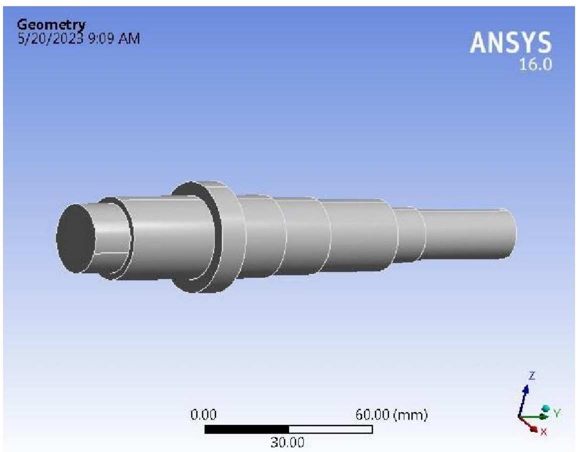
Drive shaft is safe under torsional load

Analysis of main shaft

Geometry

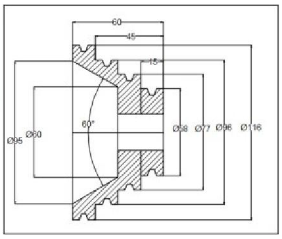


Maximum Stress



As the maximum stress induced is 1.3386 Mpa which is far below the allowable value the drive shaft is safe.

Design of step cone pulley:



Material selection : -Ref :- PSG (1.10 & 1.12) + (1.17)

⇒ $f_{s\,max}$ = UTS x 0.18 = 72 N/mm^2

DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
Aluminium	400	320

This is the allowable valve of shear stress that can be induced in the shaft material for safe operation. Assuming 100 % efficiency of transmission

⇒ T design = 3.5 N-m

Considering the torion failure of the hollow portion of the coupling shaft

$Td = \Pi/16 \times fs_{act} \times (D^4 - d^4)/D$

⇒ $fs_{act} = 16 \times Td / \Pi \times (D^4 - d^4)/D$

Diameter of smallest step on pulley = 58mm

Inside diameter of pulley =16mm

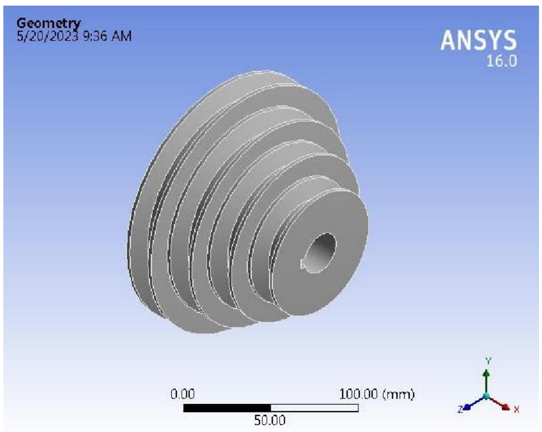
$= 16 \times 3500 \times 10^3 \times 58 / \Pi \times (58^4 - 16^4)$

⇒ $fs_{act} = 0.019\ N/mm^2$ As $fs_{act} < fs_{all}$

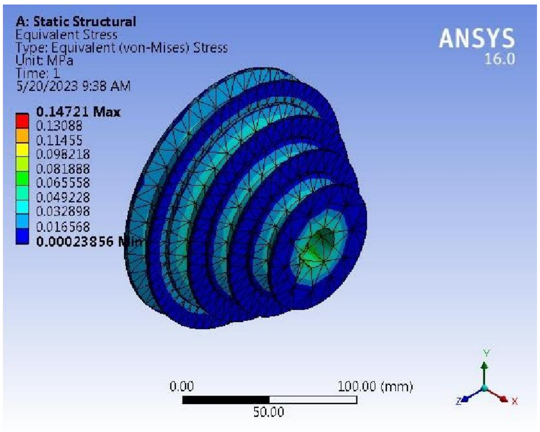
Step cone pulley is safe under torsional load.

Analysis of step cone pulley

Geometry

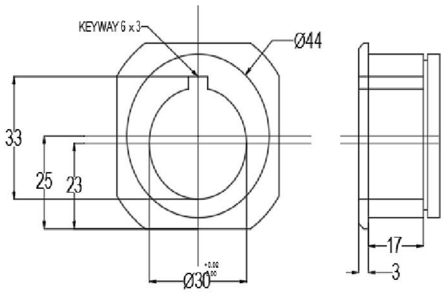


Maximum Stress



As the maximum stress induced is 0.14721 Mpa which is far below the allowable value the conepulley is safe.

Design of eccentric cam:



Material selection: -Ref :- PSG (1.10 & 1.12) + (1.17)

Designation	Ultimate tensile Strength N/mm^2	Yeild strength N/mm^2
EN24	800	680

⇒ fs allowable $= 0.18 \times 800 = 144\ N/mm^2$

⇒ T design = 3.5 Nm

Considering the torsion failure of the eccentric cam

$Td = \Pi/16 \times fs_{act} \times (D^4 - d^4)/D$

⇒ $fs_{act} = 16 \times Td/\Pi \times (D^4 - d^4)/D$

Diameter of eccentric cam = 44mm Inside diameter of cam =30mm

$= 16 \times 3500 \times 10^3 \times 44/\Pi \times (44^4 - 30^4)$

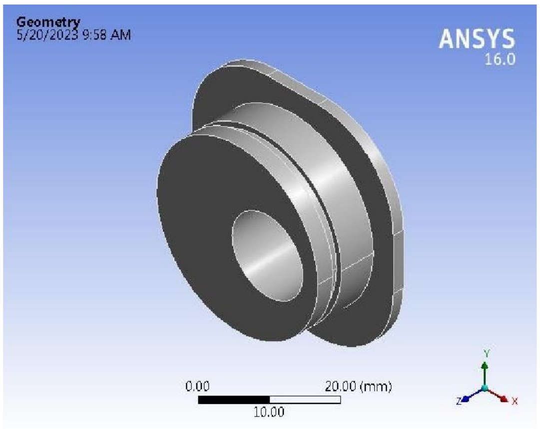
⇒ $fs_{act} = 0.266\ N/mm^2$

As $fs_{act} < fs_{all}$

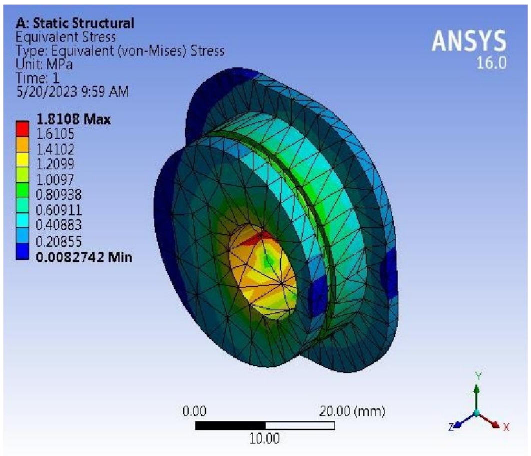
Eccentric cam is safe under torsional load.

Analysis of eccentric cam

Geometry

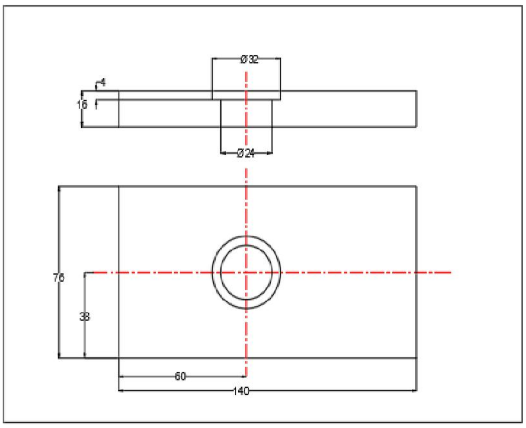


Maximum Stress



As the maximum stress induced is 1.81 Mpa which is far below the allowable value the eccentric cam is safe.

Design of moving plate:



Material selection: -Ref :- PSG (1.10 & 1.12) + (1.17)

Designation	Ultimate tensile Strength N/mm^2	Yeild strength N/mm^2
En9	600	480

⇒ fs allowable = 0.18 X 600=108 N/mm^2

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⇒ T design = 3.5 Nm

Force = T/ R = 3500/2.5 = 1400 N

Considering the shear failure of the eccentric cam

Stress = Force / Area

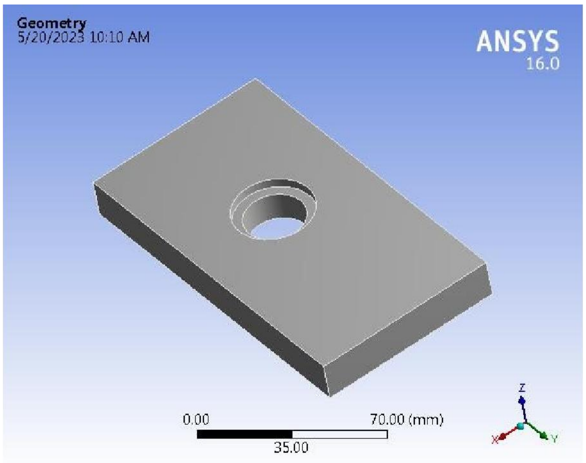
Stress = 1400 [(75x16)- (32 x4) +(26x12) Stress = 1400/760 = 1.842

⇒ fs_{act} = 1.842 N/mm^2 As $fs_{act} < fs_{all}$

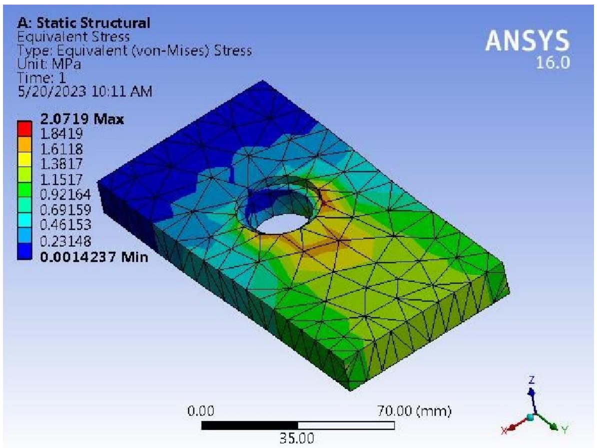
Moving plate is safe under torsional load.

Analysis of Moving plate

Geometry

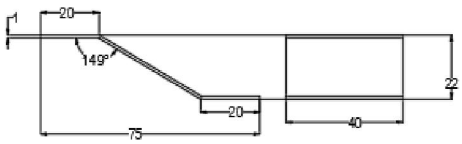


Maximum Stress



As the maximum stress induced is 2.0719 Mpa which is far below the allowable value the moving plate is safe.

Design of z-spring:



Material of Spring : EN42J/C80

Spring steel EN42J / C80 sheet plate is a kind of steel used to make cold-rolled steel

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It is sometimes referred to as spring steel, hot-rolled carbon steel, or C80 according to its chemical makeup. It possesses an excellent combination of toughness and elongation properties, a high elongation rate, and strong tensile strength.

Specifications: ASTM E45-05 / ASME E45-05

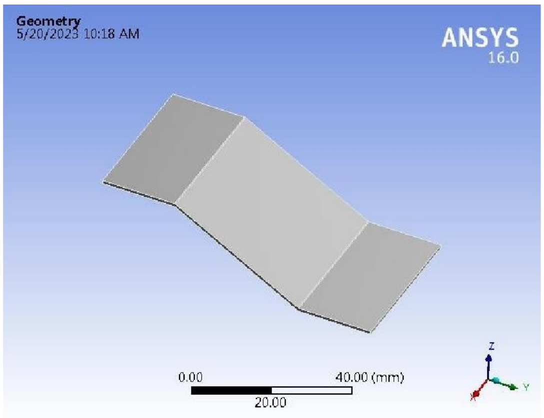
Dimensions: ASTM, ASME, and API Thickness Range**:** 0.25 MM up to 5.00 MM Width Range: 5.00 MM up to 400 MM Considering the shear failure of the eccentric cam

Stress = force / area Stress = 1400 [40x1] Stress = 1400/40 = 35

$\Rightarrow fs_{act} = 35\ n/mm^2$ as $fs_{act} < fs_{all}$

Spring is safe under shear load. Analysis of Spring

Geometry



Maximum Stress

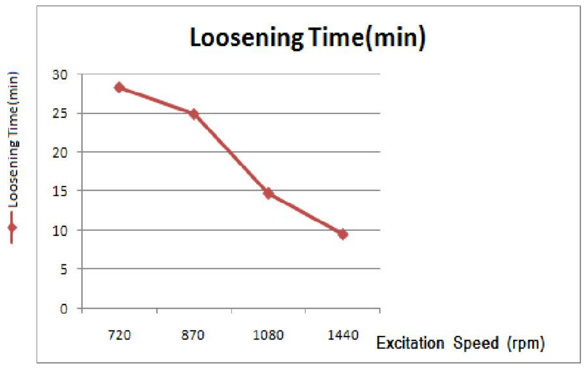
As the maximum stress induced is 64.769 Mpa which is far below the allowable value the spring is safe.

OBSERVATION

Observations M8 Bolt with Regular Washer

Sr . No.	Excitation Speed (rpm)	Time in min	No of cycles	Acceleration M2/sec
1	1400	10.3	14178	227
2.	1000	18.75	24690	252
3.	850	26.9	36160	323
4.	700	29.3	40980	357

Graph of Loosening time Vs Excitation Speed (Regular washer)



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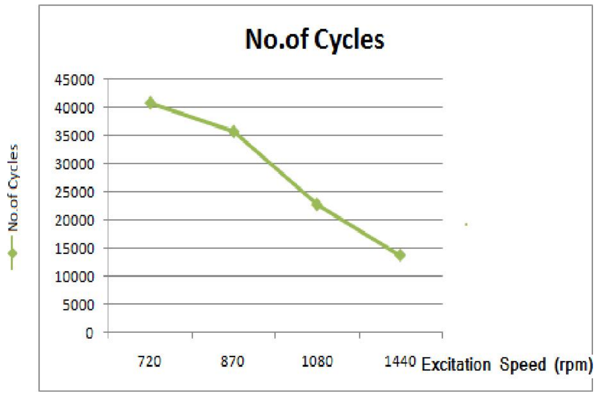
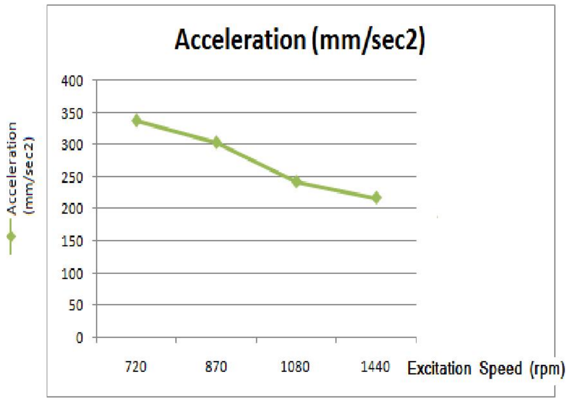
Graph of No of cycles Vs Excitation Speed (Plain washer)



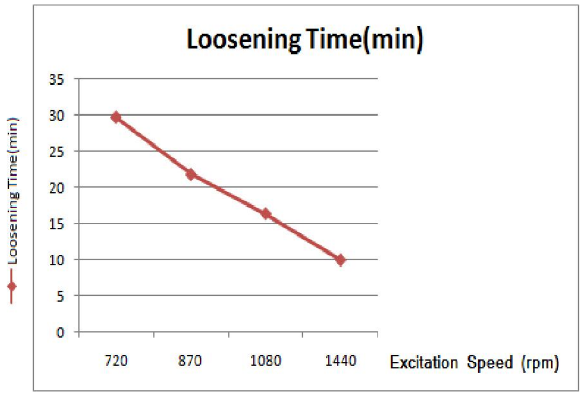
Graph of acceleration Vs Excitation Speed (Plain washer)



Observations M8 Bolt with Spring Lock Washer

SR . NO.	Excitation Speed (rpm)	TIME IN MIN	No of cycles	ACCELERATION m2/sec
1.	1400	12	14980	205
2.	1000	18.5	23800	266
3.	850	24.5	32360	298
4.	700	32	43760	337

Graph of Loosening time Vs Excitation Speed (spring washer)



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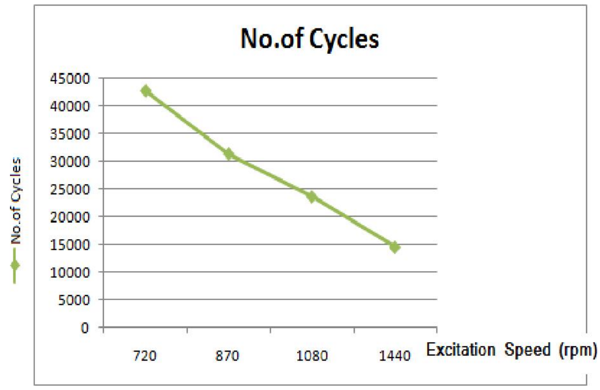
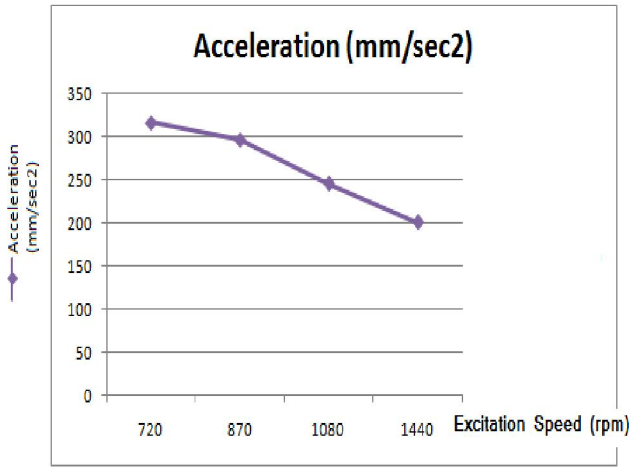
Graph of No of cycles Vs Excitation Speed (spring washer)



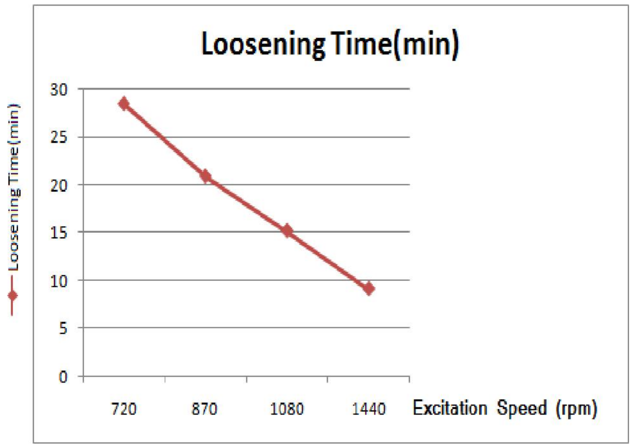
Graph of acceleration Vs Excitation Speed (Spring washer)



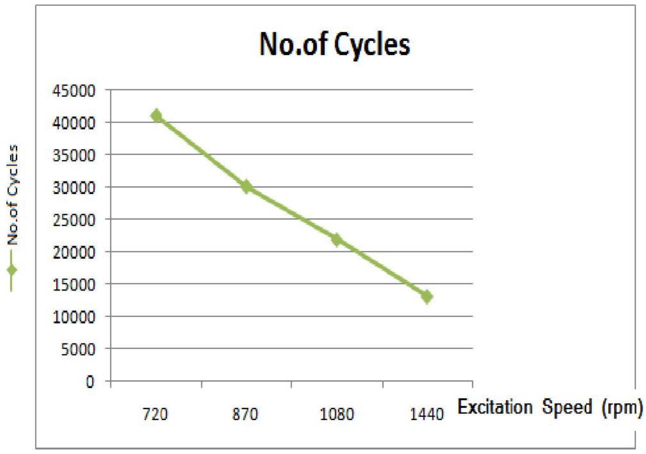
Observations M8 Bolt with external tooth Washer

SR NO.	Excitation Speed (rpm	TIME IN MIN	No of cycles	ACCELERATION m2/sec
1.	1400	11.2	13760	221
2.	1000	16.22	22410	266
3.	850	22.92	31320	317
4.	700	27.49	42030	357

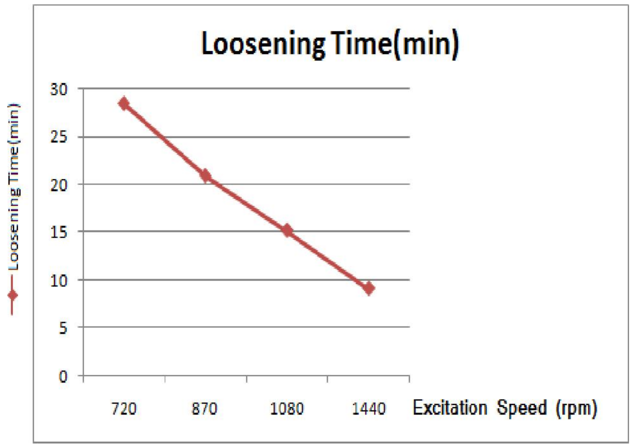
Graph of Loosening time Vs Excitation Speed (external tooth Washer



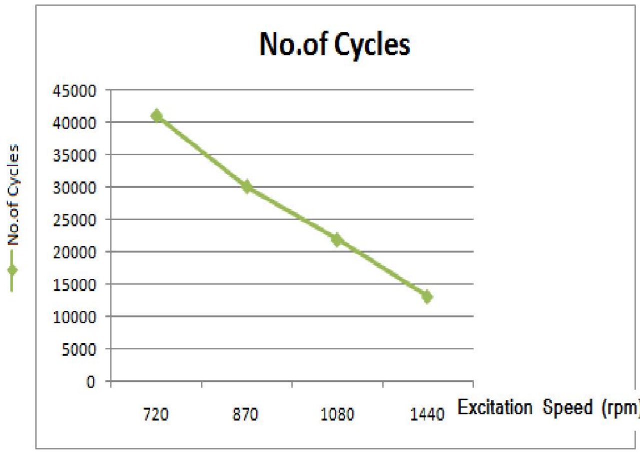
Graph of No of cycles Vs Excitation Speed (external tooth Washer**)**



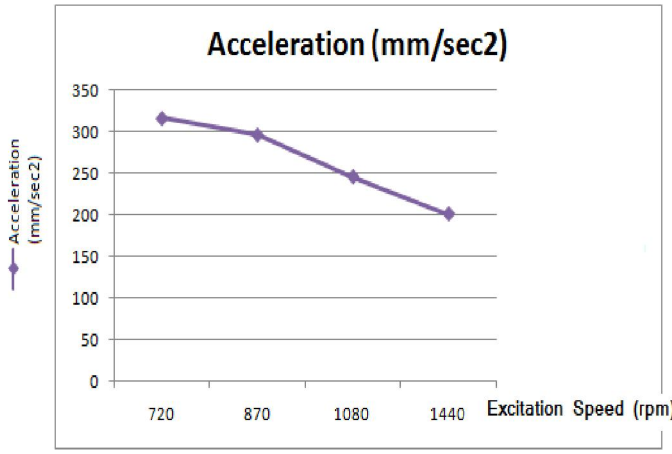
Graph of Loosening time Vs Excitation Speed (external tooth Washer



Graph of No of cycles Vs Excitation Speed (external tooth Washer)



Graph of acceleration Vs Excitation Speed (external tooth Washer)



VIII. CONCLUSION

The literature review revealed the necessity of the vibration loosening of the bolts with change in end conditions. The test rig was designed using theoretical method, the parts were modeled using unigraphix and analysis was carried out using Ansys Work bench. The analysis showed that the parts are well with safe limits. The testing was carried out and it was found that for all end conditions namely, plain washer, spring washer and external tooth washer. The loosening time drops with the excitation speed, the number of cycle before loosening drops with the excitation speed and the acceleration drops with excitation. The spring washer end condition exhibited the best result with 32 min loosening time, 43760 cycles before loosening and the minimum acceleration of 337 mm/sec2.

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