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An Experimental Investigation of Bolt Preload Decay in Socket Head Cap Screw by Transverse Vibration Test Rig

Mr.Jahid P.Pinjar[1], Prof. Abhay Kumar M.Kalje [2]

[1]*M.E. Mech-Design, Departement of Mechanical Engineering, N.B.Navale Sinhagad College of Engineerign, Kegaon, Solapur, Maharashtra.*

[2]*Associate Professor, Departement of Mechanical Engineering, N.B.Navale Sinhagad College of Engineerign, Kegaon, Solapur, Maharashtra.*

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Abstract - *A significant advantage of a bolted joint over other joint types, such as welded and riveted joints, is that they are capable of being dismantled. This feature however, can cause problems if it unintentionally occurs as a result of operational conditions. Such unintentional loosening, frequently called vibrational loosening. It is important for the Designer to be aware of the bolt loosening mechanisms which can operate in order to design reliable joints. This paper describes the dissertation work of design of vibration test rig which leads to the failure of fasteners by means of transverse vibration. The experimental work has also carried out to study the behavior of loosening mechanism considering the bolt preload decay with the help of digital strain indicator.*

***Key Words*: Fasteners, Preload,Vibration, Loosening, Design, Strain, Frequency.**

1.INTRODUCTION

A significant advantage of a bolted joint over other joint types, such as welded and riveted joints, is that they are capable of being dismantled. This feature however, can cause problems if it unintentionally occurs as a result of operational conditions. Such unintentional loosening, frequently called vibrational loosening in much of the published literature, is an important phenomenon and is widely misunderstood by Engineers. It is important for the Designer to be aware of the bolt loosening mechanisms which can operate in order to design reliable joints. The information presented below is key information for the Designer on the theory of vibration loosening of threaded fasteners and how such loosening can be prevented. **[1]**

It is widely believed that vibration causes bolt loosening. By far the most frequent cause of loosening is side sliding of the nut or bolt head relative to the joint, resulting in relative motion occurring in the threads. If this does not occur, then the bolts will not loosen, even if the joint is subjected to severe vibration. By a detailed analysis of the joint it is possible to determine the clamp force required to be provided by the bolts to prevent joint slip.

Often fatigue failure is a result of the bolt self-loosening which reduces the clamp force acting on the joint. Joint slip then occurs which leads the bolt being subjected to bending loads and subsequently failing by fatigue.

1.1 Problem Statement

Pre-loaded bolts (or nuts) rotate loose, as soon as relative motion between the male and female threads takes place. **[2]**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. There are three common causes of the relative motion occurring in the threads:

a. Bending of parts which results in forces being induced at the friction surface. If slip occurs, the head and threads will slip which can lead to loosening.
b. Differential thermal effects caused as a result of either differences in temperature or difference in clamp material.
c. Applied forces on the joint can lead to shifting of the joint surfaces leading to bolt loosening.

1.2 Objective

The objectives of dissertationis as follows,

a. Development of Test Rig for Vibration of Loosening of bolts with different variable speed.
b. Design and development of system of components for interchangeability of bolts and

nuts namely plain washer, spring washer and without washer.

c. Manufacturing and assembly of Test Rig.Test and trial for socket head cap screw namely M4, M5, M6 with predetermined loads from ANSYS and for unknown torque for different combinations.
d. Changing the end conditions of screws by changing the combination with plain washer and spring washer.

2. FINITE ELEMENT ANALYSIS OF BOLT PRELOAD

The Finite Element Model is developed in PRO-E software. This assembly model bolted joints consist of fixed plate and guide plate that is moving plate with different combinations like with no washer and with spring washer, plain washer.The model is for the following specifications of the socket head cap screws. The socket head cap screw threads: ANSI B1.13M, ISO261, ISO 262 (coarse series only) Property Class: 12.9-ISO 898/1 is selected for the model preparation. The table 1 shows the desired dimensions for modeling.

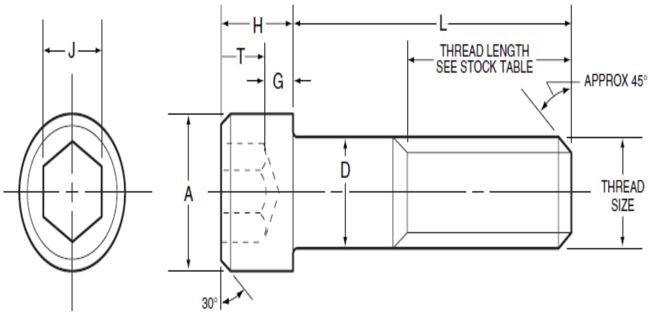


Fig.1 Dimensional features of socket head cap screw.

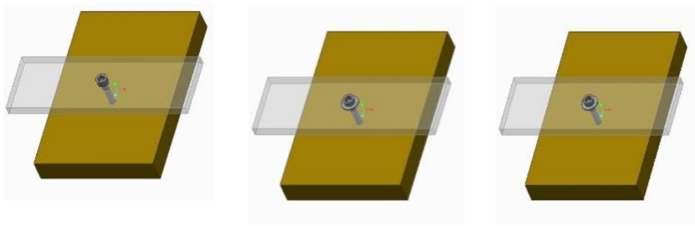
Table 1 Dimensions of Metric Socket Head Cap Screw.

Screw size	Pitch	A	D	H	J	L
M4	0.7	7.0	4.0	4.0	3.0	60
M5	0.8	8.5	5.0	5.0	4.0	60
M6	1	10	6	6	5.0	60

The mechanical property Ultimate tensile strength of these screws is 1300 MPa.

2.1 Modeling of screwed joint-

While modeling the two plates of having dimensions are 110 x 44 x 7 mm is placed at the top and after maintaining a clearance gap of 27 mm the guide plate of dimension of 175 x 75 x 20 mm is prepared. The model is kept as simple as simple for the analysis point because large parts and variation in geometry may consume the time for analysis. The following Fig.6.2 shows simple modeling of socket head cap screw of size M4.



(a) Without any washer (b) With Plain Washer (c) With Spring Washer.

Fig.2 Modeling of socket head cap screw.

The Similar models were prepared for M5 and M6 size of socket head cap Screws.

2.1.1 Defining Material Properties.

The CAD modeling of socket head cap screws with different combinations are imported into the ANSYS 14.0 for its analysis. The material property data needs to be assigned priory as a input for analysis work and expected approximate results. The following table shows the material properties for screw and Fixed plate for the stress analysis.

Material Properties for Guide plate and Fixed plate.

Material - Plain Carbon Steel.

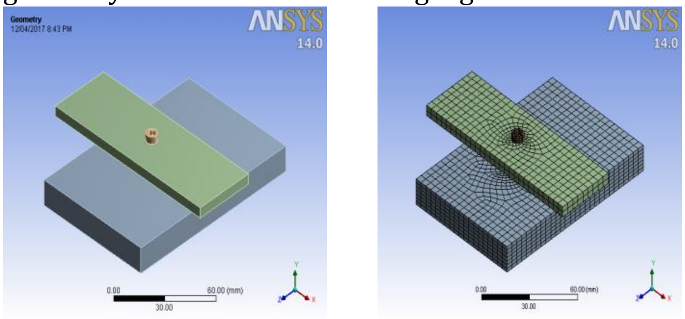
- Modulus of Elasticity- 202 MN/m^2.
- Modulus of rigidity- 78.5 MN/m^2.
- Poisson's Ratio -0.292.
- Density – 7820 Kg/m^3.

Material Properties for Socket Head Cap Screw.

- Material – Medium Carbon Steel (Annealed).
- Ultimate Strength- 460 MN/m^2 .
- Yield Point - 316 MN/m^2.
- Elastic Limit - 288 MN/m^2 .

2.1.2 Mesh Generation (Pre-processing)

The IGES file imported to ANSYS for the meshing purpose. The meshing is done in order to forming a good input for finite element modeling. Initially the surfaces are developed from the solid component after which solids are removed as the finite elements are generated out from the surface. The global tolerance value is of 0.44. For meshing Quadrilateral Element is used. It has two displacements per each node. For meshing edge length 2.5 mm is considered. The model and meshing geometry is as shown in following Fig 3.



(a) Model Geometry (b) Meshing of Geometry

Fig.3 Meshing of Assembly

After meshing the total number of elements are 10886 and number of nodes are 39235.

2.2 Finite Element Analysis of screwed joint assembly

The finite element analysis is carried out for different combinations of the screws. By considering a tightening torque i.e. angular rotation of screw for different end conditions for the combination of screws like with washer and without washer one has to analyze.**[4]**

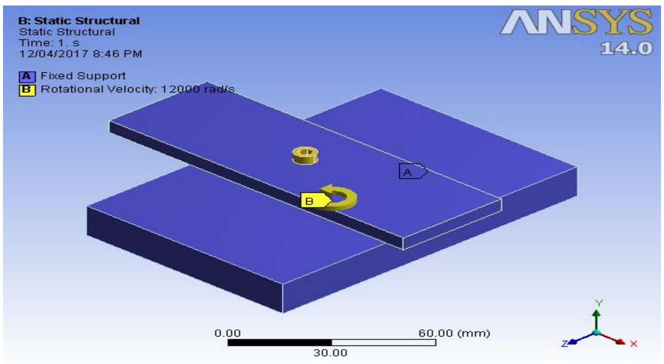


Fig 4. Static Structural Analysis of M4 Screw with no washer.

In this analysis the torque is applied and the stresses are produced on the fixed plate due to tightening load i.e. clamp load. As the torque only can apply in the static condition of tightening of fasteners here static structural analysis is carried. The following fig 4, 5 and 6 shows the analysis of bolt preload for M4 size socket head cap screw without washer. The rotational velocity of 12000 rad/sec is given i.e. solver for the analysis is used as a Mechanical APDL in ANSYS.

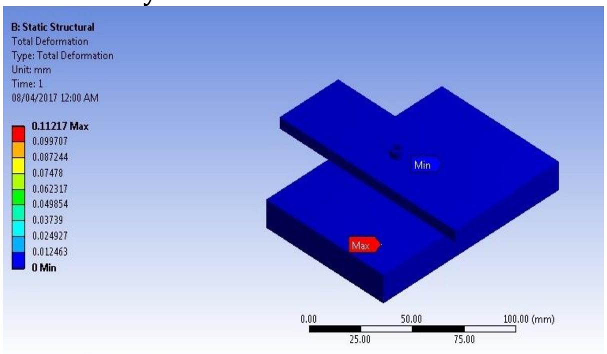


Fig .5 Static Structural Analysis showing deformation of M4 Screw with no washer.

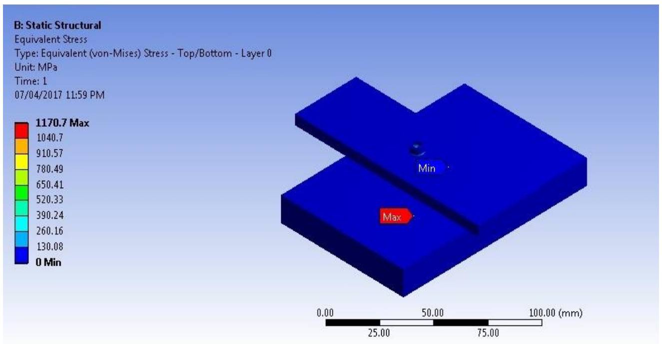


Fig .6 Static Structural Analysis showing stresses induced in M4 Screw with no washer.

From this analysis it is found that the preload value is in terms of stress as 1170.7 MPa with deformation 0.11217 mm.

Similar Methods were adopted for remaining conditions for M4 size screw with conditions of spring washer and plain washer.

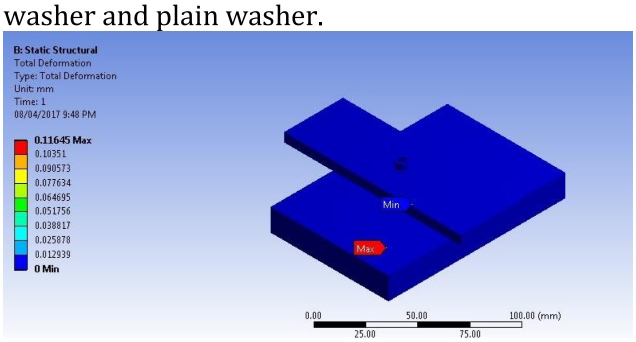


Fig.7 Analysis showing Deformation M4 Screw with plain washer.

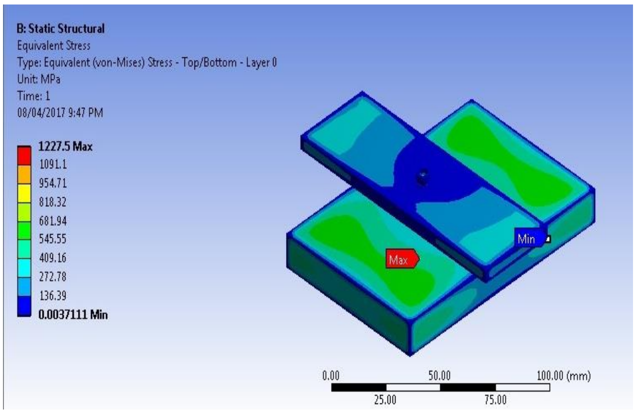


Fig. 8 Analysis showing stresses induced in M4 Screw with plain washer.

From this analysis it is found that the bolt preload value is in terms of stress as 1170.7 MPa with deformation 0.11217 mm.

After modeling and analyzing the results were found and they are tabulated as follows.

Table -2: Analysis Results for Stress Values in MPa

Size-	M4	M5	M6
Conditions	Stress in MPa		
With No washer	1170	1198	1198
With plain washer	1227	1247	1263
With Spring washer	1270	1280	1298

3. EXPERIMENTAL INVESTIGATION OF BOLT PRELOAD DECAY.

The transverse vibration test rig is used to predict the bolt preload decay. The image of test rig is as shown.

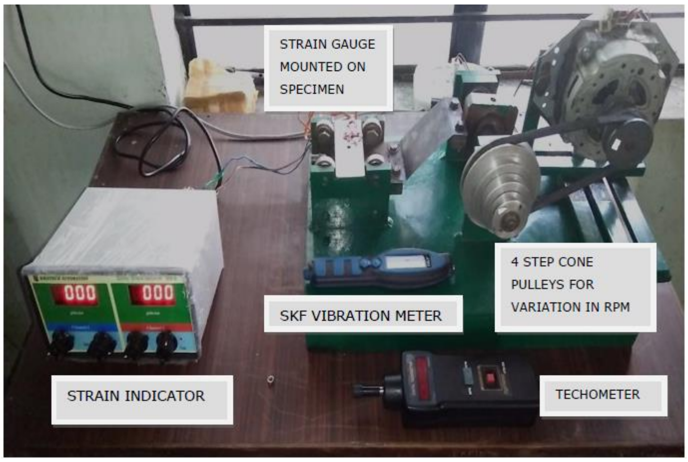


Fig. 9 Transverse Vibration Test Rig.

The experimental work is carried out at 840 rpm.

The procedure is as follows,

1. Tight the screw for desired combination.
2. Note down the strain indicator reading.
3. Switch ON the motor which is coupled with pulley and shaft. Measure its RPM
4. Observe the changes on strain indicator reading and note it for the change.
5. With the help of SKF vibration meter note the velocity of guide plate along with the frequency in Hz given by the vibration meter.
6. Run the trial for desired change in the strain indicator and for every change note the velocity and frequency.

The strain indicator should be set to zero before tightening the screw and same is followed to subsequent combination trials.

The trial readings were noted in the following table 3 shows observations for M4 size socket head cap screw.

Table-3: Decay in Preload of M4 Size screw

Size M4/ Reading-		1	2	3	4
With no washer	Pi	14.30	14.30	13.28	12.26
	F	0	12.90	13.48	14.36
Plain washer	Pi	15.32	15.32	14.28	13.28
	F	0	11.22	12.16	12.90
Spring washer	Pi	16.34	16.34	16.34	15.32
	F	0	8.62	11.22	12.48

*Pi= Preload in kN , *F= Frequency in Hz

Note-The '0' in preload readings noted at static condition of the system. Same method is followed for every combination. The changes in the reading are observed by noting the time. The displacement of a guide plate is a very minute in the range of 0.0001 to 0.00025 mm. Hence it is difficult to predict the displacement.

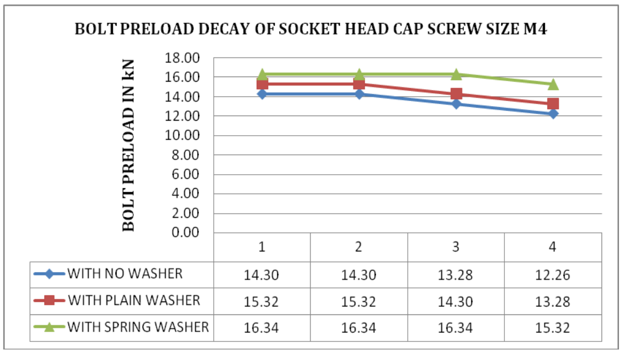


Chart -1: Preload decay of M4 Size.

Table-4: Decay in Preload of M5 Size screw

Size M5/ Reading-		1	2	3	4
With no washer	Pi	23.94	22.34	22.34	20.74
	F	0	11.48	12.46	13.59
Plain washer	Pi	25.53	25.53	23.94	22.34
	F	0	11.08	12.33	13.42
Spring washer	Pi	27.12	25.53	23.94	23.94
	F	0	10.05	11.30	12.42

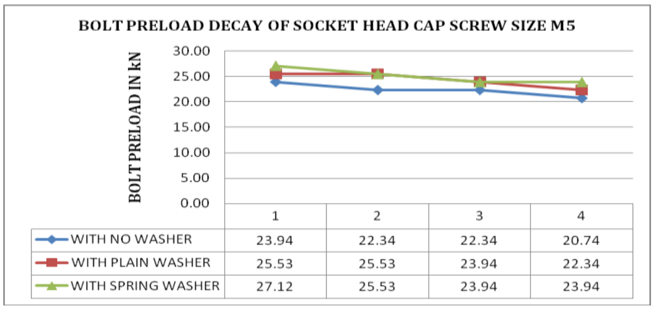


Chart -2: Preload decay of M5 Size.

Table-5: Decay in Preload of M6 Size screw

Size M5/ Reading-		1	2	3	4
With no washer	Pi	34.46	32.16	29.87	29.87
	F	0	11.48	12.16	13.19
Plain washer	Pi	36.76	36.76	34.46	32.16
	F	0	11.08	11.48	12.62
Spring washer	Pi	36.77	36.77	34.47	32.17
	F	0	9.45	10.90	12.10

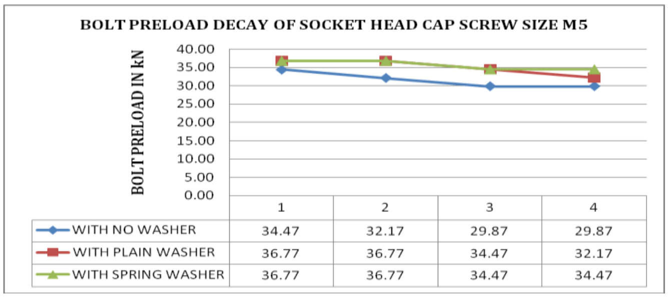


Chart -3: Preload decay of M6 Size.

3. CONCLUSIONS

From the experimental investigation the following conclusions are drawn.

- The bolt preload decay is non linear in nature as the frequency increases with the reduction in preload.
- The loosening prevention devices like spring washer has more preload values than the plain washer for every combination.
- With the reduction of preload the tightening torque to produce preload is also reduced. Hence monitoring of fastener system is necessary when the joints are subjected to vibrations.

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