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"Experimental Study of Preload Decay in Threaded Fasteners Application"

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ABSTRACT: The Nut and bolt assembly used in various mechanical applications are to be studied to find out decay happening due to vibrations applied to it. In this project I aim to study various combinations of threaded fasteners used in particular application such that decay due to vibrations could be explored. 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 research describes the dissertation work of studying the behavior of loosening mechanism considering the bolt preload decay. After review of literature it was concluded in this review that the most occurring root cause of loosening is side sliding of the nut or bolt head relative to the fastened joint, resulting in the relative motion occurring within the threads. If this does not occur or can be prevented, then the loosening of bolts can be reduced.

Keywords: Bolt, Bolted Joints, Vibration, Loosening of bolts, Junker test, Fasteners.

I. 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.

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.

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.

There are three common causes of the relative motion occurring in the threads:

I. 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.
II. Differential thermal effects caused as a result of either differences in temperature or difference in clamp material.
III. Applied forces on the joint can lead to shifting of the joint surfaces leading to bolt loosening.

Work completed during the 1960's in Germany indicated that transversely applied alternating forces generate the most severe

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conditions for self-loosening. The result of these studies led to the design of a testing machine which allowed quantitative information to be obtained on the locking performance of self-locking fasteners. Such machines, often called Junkers machines in the literature - after it's inventor, have been used over the last twenty years by the major automotive and aerospace manufacturers to assess the performance of proprietary self-locking fasteners. As a result, a rationalization of the variety of locking devices used by such major companies has occurred. For example, conventional spring lock washers are no longer specified, because it has been shown that they actually aid self-loosening rather than prevent it. There are a multitude of thread locking devices available.

A Junker test is a mechanical test to determine the point at which a bolted joint loses its preload when subjected to shear loading caused by transverse vibration. Design engineers apply the Junker Test to determine the point at which fastener securing elements – such as lock nuts, wedges and lock washers – fail when subjected to vibration. The data collected by the test enables design engineers to specify fasteners that will perform under a wide range of conditions without loosening. Research into the causes of vibration induced self-loosening of threaded fasteners spans six decades and the causes of self-loosening are now well understood. It was pioneering experimental research into the behavior of bolted joints under transverse loads, conducted by German engineer Gerhard Junker in the late 1960s which underpins modern theories on self-loosening behavior.

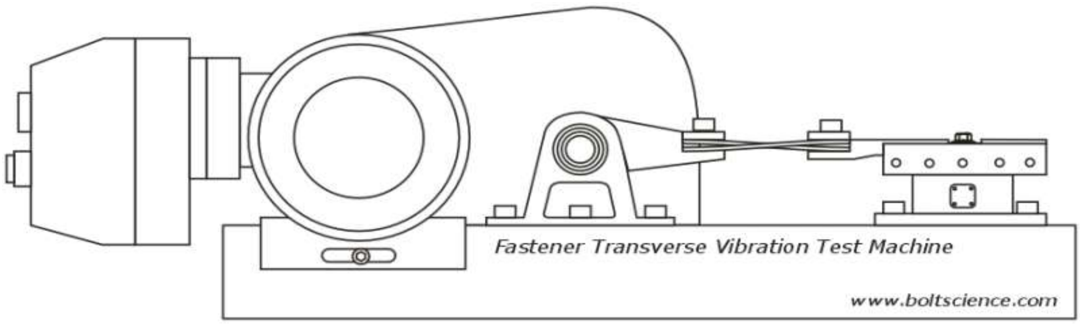

Fig. 1.1 Junker's Transverse Vibration Test Machine.

II. LITERATURE REVIEW:

N. G. Pai and D. P. Hess [1] presents results of a study on failure of threaded fasteners by vibration induced loosening caused due to dynamic shear loads. A three-dimensional finite element (FE) model is used to study details of four different loosening processes that are characterized by either complete or localized slip at the head and thread contacts.The results show that loosening can occur at relatively low shear loads due to the process of localized slip.

N. G. Pai and D. P. Hess [2] presents the experimental method in which they have developed the cantilever beam and nut bolt arrangement as a model to study the behavior of the loosening effect under the action of the vibration by studying the effect of loosening in connection with the position of the fastener placement.

C. A. Cheatham, C. F. Acosta, D. P. Hess [3] presents results from an experimental investigation of loosening of threaded inserts. The tests are performed on a transverse test machine which provides transverse shear.Both secondary locking features are found to provide improved resistance to loosening. The improvement with inserts with prevailing torque locking feature is found to be minimal at first, then increase with decreasing preload and eventually level out with a fraction of initial preload retained.

R. I. Zadoks and D. P. R. Kokatam [4] shows the axial stiffness of a bolt plays a critical role in the prediction of the self-loosening process of threaded connections subject to oscillatory excitation.The model is loaded by pulling the nodes around the outside of the bolt head in the axial direction while holding the bottom of the plate fixed. Several analyses are performed to investigate the axial stiffness of the bolt.

N. G. Pai and D. P. Hess [5] present a study on loosening of threaded fasteners subjected to dynamic shear loads. A fundamental analysis of loosening reveals that a fastener can loosen at lower loads than previously expected due to localized slip at the contact surfaces. Four different loosening processes of a screw under different conditions of slip at the head and thread contact

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regions are identified. Experimental results illustrating these loosening processes are presented. In addition, the minimum dynamics hear force required to initiate loosening is determined experimentally.

F. M. Leon, N. G. Pai, D. P. Hess [6] presents the results from tests that investigate the effect of thread dimensional conformance of fasteners on yield and tensile strength.Variations in bolt pitch diameter were found to affect the yield and tensile strength by about an order of magnitude more than variations in bolt major diameter or nut pitch and minor diameters. The mean tensile strength for conforming product was found to be as much as 20% greater than the tensile strength for nonconforming product.

Ingrid A. Rashquinha and Daniel P. Hess [7] shows the dynamics of threaded fastened assemblies represent a highly nonlinear constrained vibration problem that is nontrivial, but of considerable practical importance. In this paper, a dynamic model of a fastened assembly is developed which incorporates a dynamic fastener model with dynamic structural component models.

Satoshi Izumi, Takashi Yokoyama, Atsushi Iwasaki, Shinsuke Sakai [8] investigated the mechanisms of the tightening process and the loosening process due to shear loading using the framework of the three-dimensional finite element method (FEM). Results are compared with those of conventional theories based on material mechanics and with experimental results. We found some new aspects of threaded fastener theory. Previous theory overestimates the tightening torque in the relation between preload and tightening torque.

Abhay Kakirde, Dr. Shriram Dravid [9] “A review on loosening of bolted joints” Authors have reviewed the various studies carried out by previous researchers in this area of research. Main reasons of loosening are cyclic loading and unloading of threaded components. Many researchers have developed machines for their experimental work. Loosening i.e. unlocking rotation of threaded assembly is caused by the restoring action of an elastic torsion of a bolt shank because of the relative motion at mating surface on threads.

Bikash Panja and Santanu Das [10] “Development of anti loosening fasteners and comparing its performance with different other threaded fasteners” In the present experimental investigation, anti-loosening ability of various fastening elements, such as conventional nut, nylock nut, flat washer, spring washer, inside and outside serrated washer, is tested with a conventional M16 high-tension steel bolt. All these fasteners are tested in terms of their loosening characteristics. Accelerated vibrating conditions are used for the test on an indigenously made test rig.

III. MATHEMATICAL MODELLING OF BOLT PRE-LOAD

A bolted assembly is tightened by applying force to the wrench handle and rotating the hexagonal nut. The bolts are tightened with a specific magnitude of preload P_i. It is necessary to determine the magnitude of torque that will induce this pretension. The torque required to tighten the bolt consist of the following two factors;

I. Torque required overcoming threaded friction and inducing the pre-load i.e. $(M_t)_t$.
II. Torque required to overcome collar friction between the nut and the washer $(M_t)_c$.

3.1 Torque Requirement for Bolt Pre-Load

The equations derived for trapezoidal threads are suitably modified for ISO metric screw threads. The torque required to overcome the friction is given by,

$$M_t = \frac{P_i\, d_m}{2} \frac{\mu \sec\theta + \tan\alpha}{1 - \mu \sec\theta + \tan\alpha}$$

(3.1)

For ISO metric screw threads, $\theta=30^0$, $\alpha=2.5^0$ $d_m=0.9d$ where d is nominal or major diameter. The coefficient of friction varies from 0.12 to 0.20 depending upon the surface finished accuracy of the thread profile and lubrication assuming. i.e. $\mu=0.15$ and substituting the above values in equation (3.1) we get,

$M_t=0.098P_id$

(3.2)

According to uniform wear theory, the collar friction torque $(M_t)_C$ is given by,

$$(M_t)_C = \left(\frac{\mu P_i}{2}\right)\left(\frac{D_0+D_i}{2}\right)$$

(3.3)

In the above equation D_0 is the diameter of an imaginary circle across the flats of the hexagonal nut and D_i is the diameter of the hole in the washer for ISO metric screw threads.

$\left(\frac{D_0+D_i}{2}\right) = 1.4d$ & $\mu=0.15$

Substituting these values in equation (3.3) we get

$$(M_t)_C = \left(\frac{0.15\, P_i}{2}\right)(1.4d)$$

(3.4)

$$(M_t)_C = (0.105P_i)d$$
(3.5)

Adding equation (3.2) and (3.4) the total torque, $(M_t)_t$ is required to tighten the bolt is given by,

$$(M_t)_t = M_t + (M_t)_C = (0.098 + 0.105)P_i d$$
$$(M_t)_t = M_t + (M_t)_C = 0.2P_i d$$
(3.6)

The above equation gives a simple explanation to determine the wrench torque $(M_t)_t$ required to create the required preload P_i.

IV. EXPERIMENTAL PROCEDURE.

1. The strain indicator should be set to zero before tightening the screw. (same is followed to subsequent combination trials.)
2. Tight the screw for desired combination.
3. Note down the strain indicator reading. This reading is at static condition and gives the bolt preload value in terms of strain.
4. Switch ON the motor which is coupled with pulley and shaft. Measure its RPM with the help of tachometer.
5. Observe the changes on strain indicator and note it. Also note the time with the help of stopwatch.
6. With the help of SKF vibration meter note the velocity of guide plate along with the frequency in Hz given by the vibration meter for corresponding change observed on strain indicator.

The experiment conducted for different combinations of socket head cap screw for a selelcted RPM as the pulley is provided for the change in RPM speed.For indivisual size of head cap screw there is a set of three combination namely with no washer, with plain washer and spring washer.

V. 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 the bolt preload in terms of the deformation of the contacting bodies. 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 5.1, 5.2 and 5.3 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. the tightening torque value is given to the screw. The solver for the analysis is used as a Mechanical APDL in ANSYS.

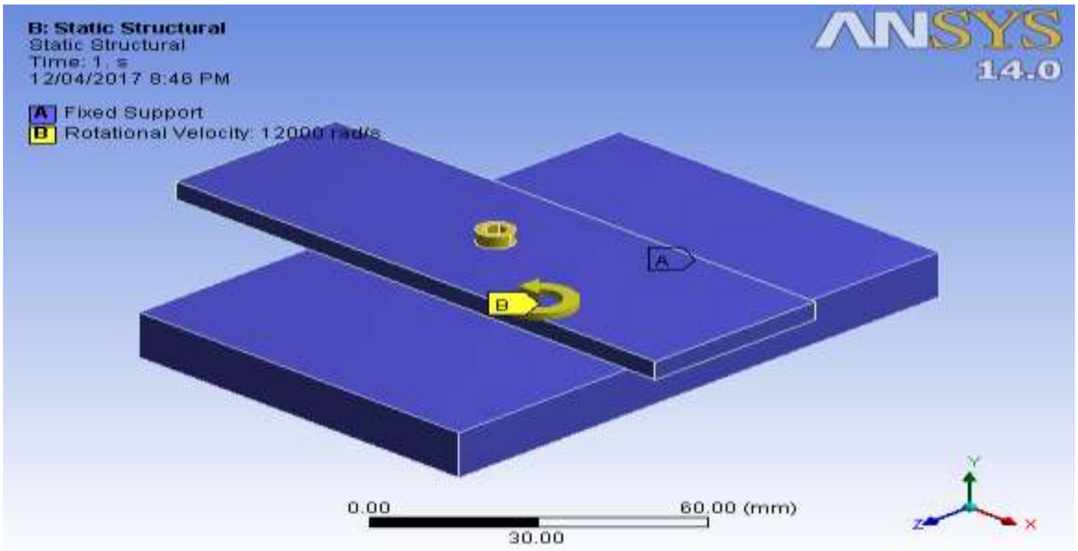

Fig 5.1 Static Structural Analysis of M4 Screw with no washer.

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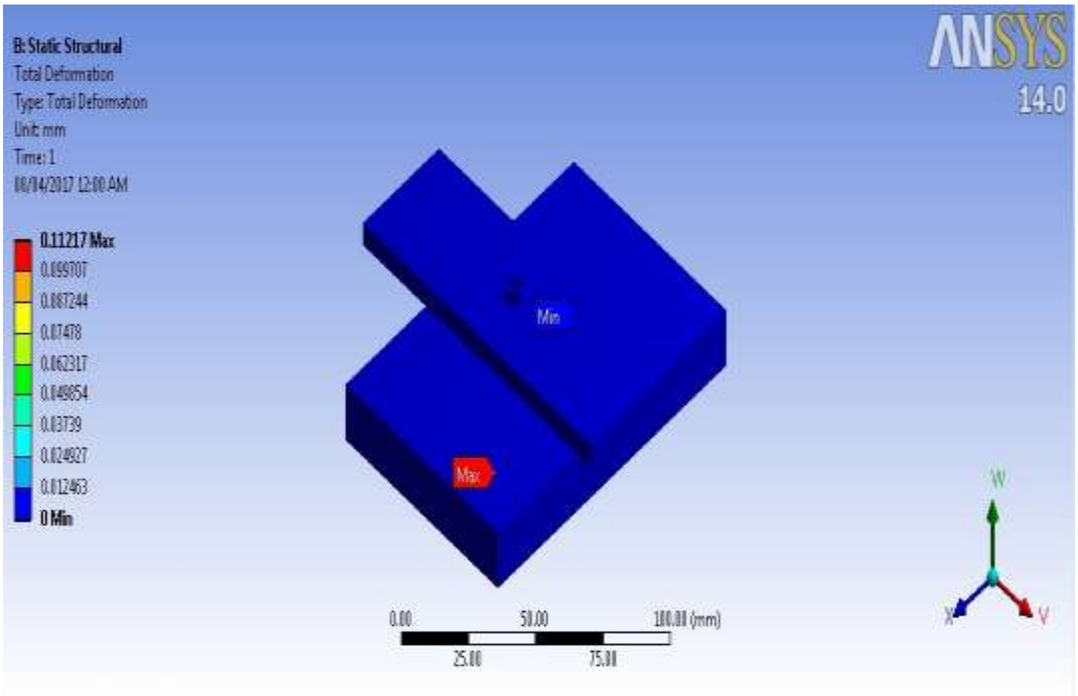

Fig 5.2 Static Structural Analysis showing deformation of M4 Screw with no washer.

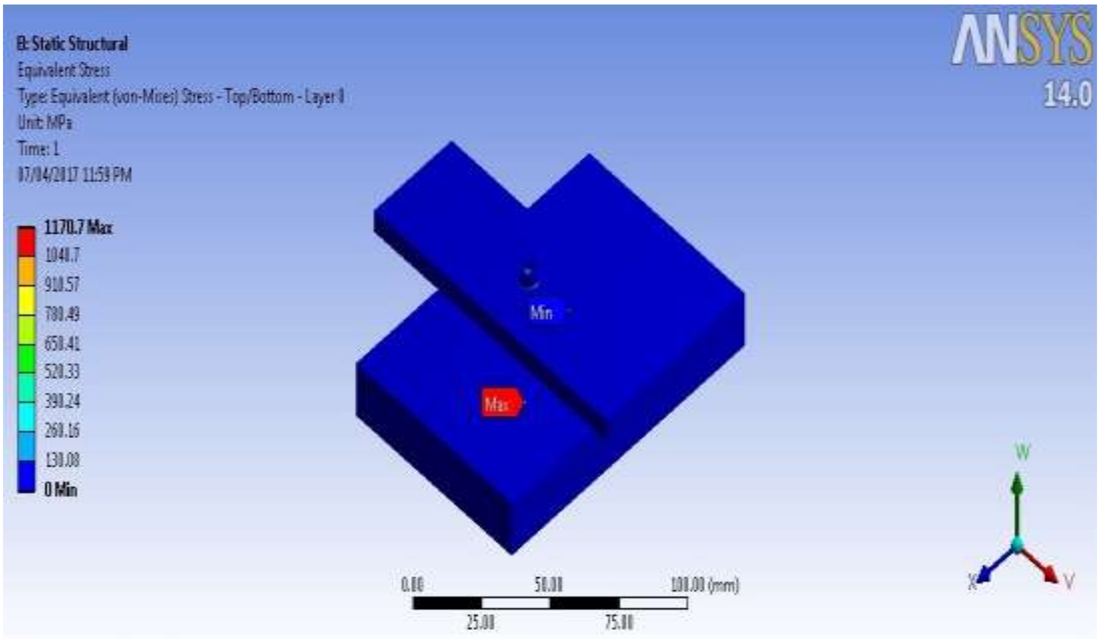

Fig 5.3 Static structural analysis showing of M4 screw with no washer

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.

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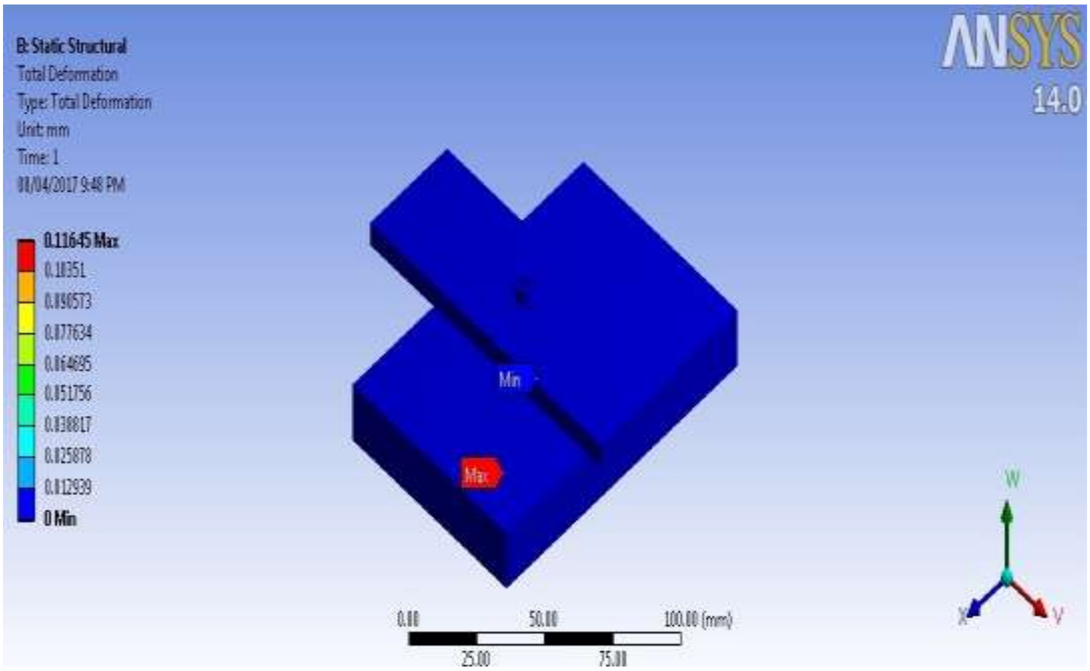

Fig 5.4 Analysis showing deformation M4 screw with plain washer

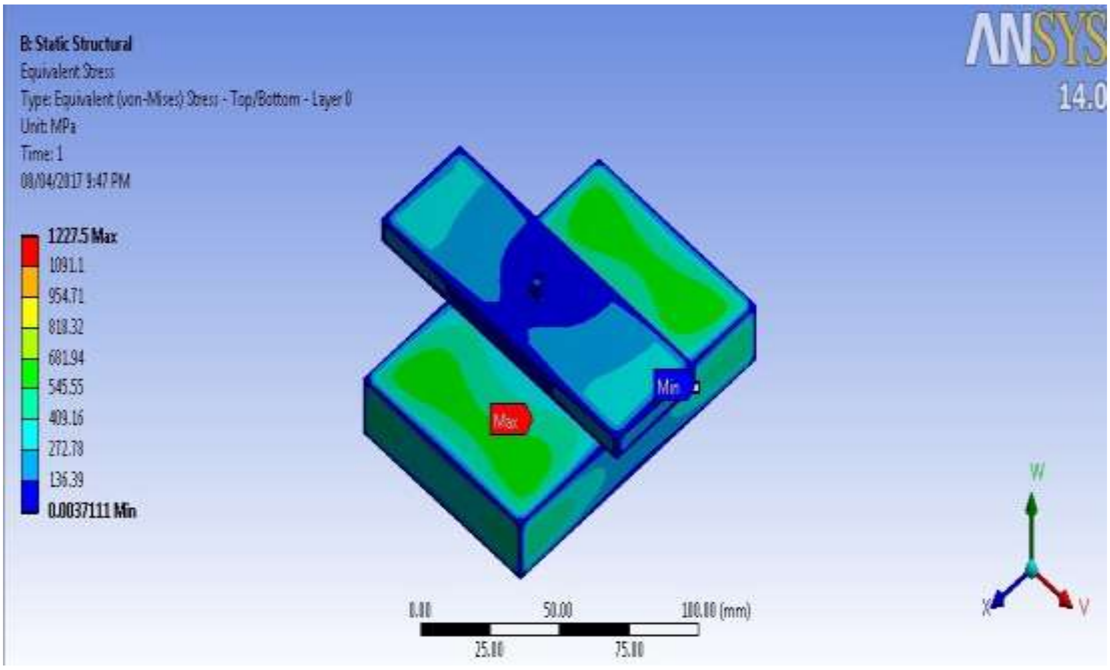

Fig 5.5 Analysis showing stresses induced in M4 Screw with plain washer

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.

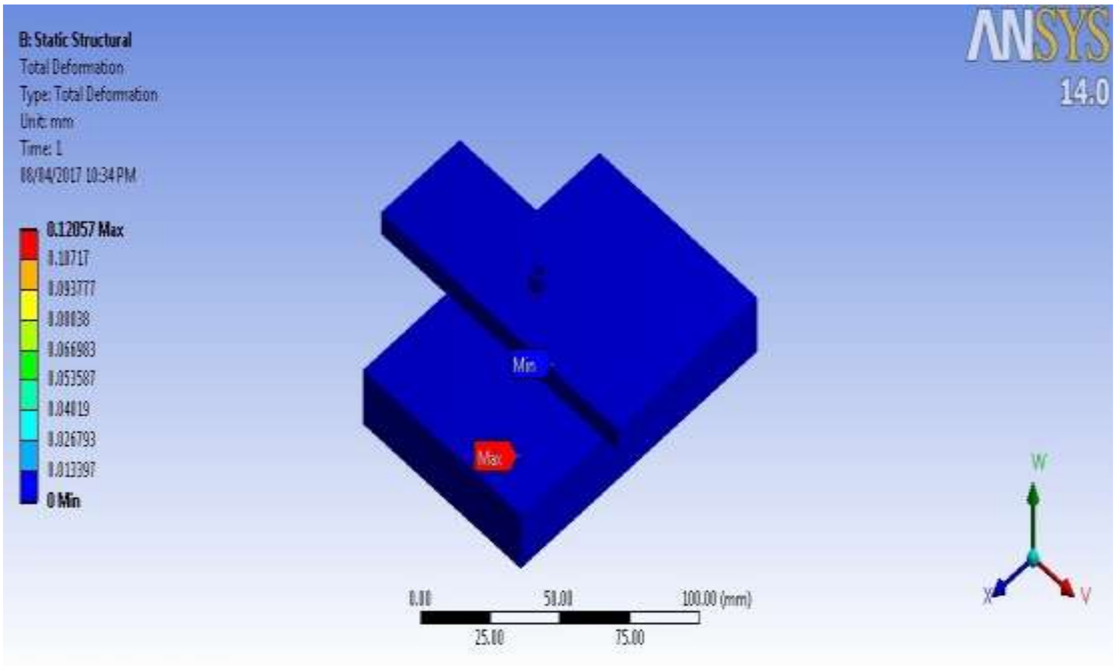

Fig 5.6 Analysis showing deformation in M4 screw with spring washer

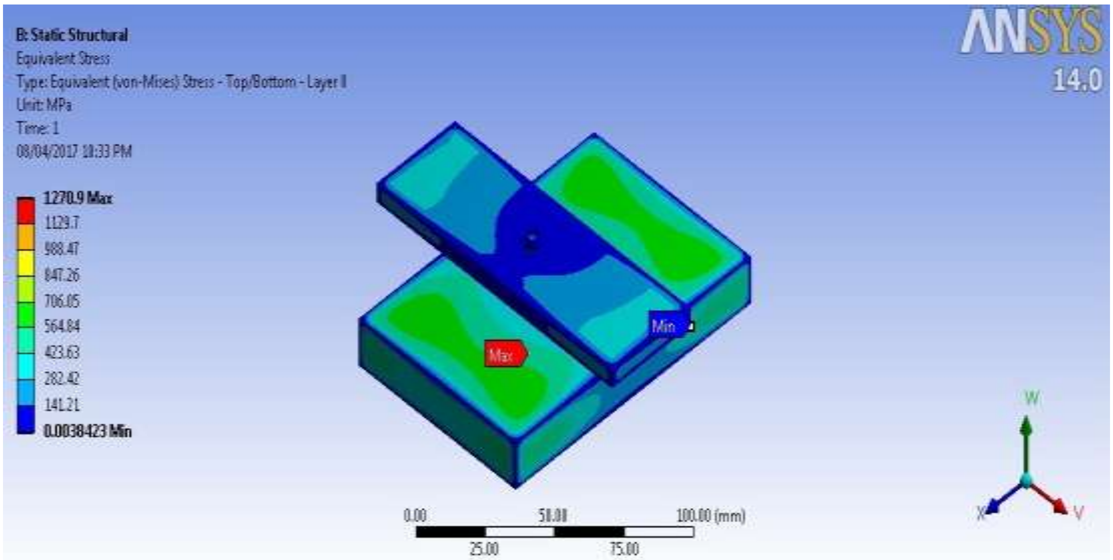

Fig 5.7 Analysis showing stresses induced in M4 Screw with spring washer.

The same analysis procedure was followed for the given conditions of M5 and M6 socket head cap screw with their different combinations.

The following table shows the FE analysis of the overall analysis which were carried out for different combinations.

Table 5.1 FEM Results of Stresses and Deformation

Sr. No.	Size	M4	
	End Conditions	Stress in MPa	Deformation in mm
1	Without Washer	1170.7	0.11645
2	Plain Washer	1227.5	0.1217
3	Spring Washer	1270.9	0.12057

VI. TRIALS AND OBSERVATIONS.

By following the above procedure the trial were conducted and the observations were tabulated. The following are the experimental parameters remains same for all the trials. The details are as follows.

- Speed in RPM - 840
- Pulley Diameter- 44 mm.
- Poissions ratio for material, $\mu = 0.292$
- Modulus of Elasticity E= 210 N/mm^2
- 1 mm per sec= 206264.806 mm
- Area(a) for M4 size screw = 12.5664 mm^2
- Area(a) for M5 size screw = 19.635 mm^2
- Area(a) for M6 size screw = 28.2744 mm^2

The trial observations are as tabulated as follows

Table 6.1 :- Trial 1 A – M4 screw with nut only.

Reading No's	1	2	3	4	5
Time in minutes	0	1	3	8	10
Strain 1(E_a)	40	39	37	35	34
Strain 2 (E_b)	26	25	24	23	23
Shear stress (τ)	1137.77	**1137.77**	1056.50	975.23	893.96
Pre-Load in N	14297.68	**14297.68**	13276.42	12255.16	11233.89
Pre-Load in kN	14.29768	14.29768	13.27642	12.25516	11.23389
Velocity in mm/sec	0	**45.2**	47.2	50.3	56.7
Displacement in mm	0	**0.00022**	0.00023	0.00024	0.00027
Velocity in rad/sec	0	2.0545	2.1455	2.2864	2.5773
Frequency in Hz	0	12.90912	13.48032	14.36568	16.19352
Tightening Torque N-m	**11.438147**	11.438147	10.621137	9.804126	8.987116

6.1 Sample Calculations for observation table

6.1.1 Calculation of Preload.

As the starin guage arrangement is of rectangular rosette we have the equation for calculating shear stress,

$$\tau = \frac{E}{2(1+\mu)}(E_a - E_b)$$

(6.1)

$$\therefore \ \tau = \frac{210}{2(1+0.292)}(40-26)$$

$$\therefore \ \tau = 1137.77 \text{ N/mm}^2$$

$$P_i = \tau \times a = 1137.77 \times 12.5664 = 14297.68 \text{ N}$$

Hence the experimental value of Pre-load (Pi) is 14297.68 N. This value is calculated for static condition. The remaining values are calculated in similar format for dynamic condition by noting the values from strain indicator channel.

6.1.2 Calculation of Displacement of Guide plate.

The displacement for the guide plate is started after the satrting of motor. The value of velocity is given by vibration meter and it is noted. Here the conversion of velocity into the displacement of guide plate is calculated as follows,

Velocity =45.2 mm/sec.

We have,

1 mm per sec= 206264.806 mm

$$\therefore \text{Displacement} = \frac{\text{Velocity in mm/sec}}{206264.806}$$

(6.2)

$$\therefore \text{Displacement} = \frac{45.2}{206264.806} = 0.00022 \text{ mm}$$

This value is calculated for Reading Number 2 condition. The remaining values are calculated in similar format.

6.1.3 Calculation of Frequency.

calculating frequency we need to convert the linear velocity (v) into angular velocity (ω).

We have,

$$v = r\omega$$

$$\omega = \frac{v}{r} = \frac{45.2}{22} = 2.0545 \text{ rad/sec}$$

We have,

$$1 \text{ rad/sec} = \frac{1}{2\pi} \text{ Hz} = 0.1591549 \text{ Hz}$$

$$\therefore \text{Hz} = \frac{1 \text{ rad/sec}}{\frac{1}{2\pi}}$$

$$\therefore \text{Hz} = \frac{2.0545}{\frac{1}{2\pi}} = 12.9091$$

This value is calculated for Reading Number 2 condition. The remaining values are calculated in similar format.

The same calculations method is used for remaining readings and trials.

6.1.4 Calculation of Tightening torque.

$$(M_t)_t = M_t + (M_t)_c = (0.098 + 0.105)P_i d = 0.2P_i d$$

(6.3)

$$(M_t)_t = 0.2P_i d = 0.2 \times 14297.68 \times 4/1000$$

$$(M_t)_t = 11.438144 \ Nm$$

Where d is the mean diameter of screw in meter.

The calculations for remaining trials and observation tables were prepared in MS-Excel sheet and tabulated as follows.

Table 6.2 :- Trial 1 B – M4 screw with plain washer.

Reading No's	1	2	3	4	5
Time in minutes	0	1	5	15	20
Strain 1(E_a)	45	44	43	42	41
Strain 2 (E_b)	30	29	29	29	29
Shear stress (τ)	1219.0402	1219.04	1137.770	1056.501	975.232
Pre-Load in N	15318.947	15318.94	14297.684	13276.421	12255.15
Pre-Load in kN	15.3189	15.3189	14.2976	13.2764	12.25
Velocity in mm/sec	0	39.3	42.6	45.2	48.7

Displacement in mm	0	0.00019	0.00021	0.00022	0.00024
Velocity in rad/sec	0	1.78636	1.93636	2.05455	2.213
Frequency in Hz	0	11.22408	12.16656	12.90912	13.908
Tightening Torque N-m	12.2551	12.255	11.438	10.6211	9.8041

Table 6.3 :- Trial 1 C – M4 screw with spring washer.

Reading No's	1	2	3	4	5
Time in minutes	0	3	10	18	30
Strain 1(E_a)	48	47	46	45	43
Strain 2 (E_b)	32	31	30	30	29
Shear stress (τ)	1300.30	1300.30	1300.30	1219.04	1137.77
Pre-Load in N	16340.21	16340.21	16340.21	15318.94	14297.68
Pre-Load in kN	16.34021	16.34021	16.34021	15.31894	14.29768
Velocity in mm/sec	0	30.2	39.3	43.7	46.3
Displacement in mm	0	0.00015	0.00019	0.00021	0.00022
Velocity in rad/sec	0	1.3727	1.7864	1.9864	2.1045
Frequency in Hz	0	8.62512	11.22408	12.48072	13.223
Tightening Torque N-m	13.072	13.0721	13.0721	12.255	11.438

VII. RESULTS AND DISCUSSION:

In this chapter the results obtained from the experiment and their significance with the bolt preload decay is discussed. The results are founded with the help of FEM method and Experimental method is compared. Also the discussion is carried out in consideration of bolt preload decay with the effect of vibration, also in tightening torque and how the loosening mechanism are correlated with the experimental value.

7.1 Comparison of Bolt Preload in Static Condition

Here the comparison of bolt preload with the ANSYS and with the Strain indicator reading is compared and errors in results are found. The comparison is tabulated as follows,

Table 7.1 Comparison of FEM and Experimental values of stress due to Preload

Screw Size	End Conditions	FEM Values of Stress due to Preload in N/mm^2	Experimental values of Stress due to Preload in N/mm^2	Percentage Error
M4	With nut only	1170.7	1137.77	2.81
	With plain washer	1227.5	1219.04	0.68

	with spring washer	1270	1300.30	2.38
M5	With nut only	1189	1219.04	2.52
	With plain washer	1247.1	1300.310	4.26
	with spring washer	1280	1300.30	1.58
M6	With nut only	1219	1219.04	0.00328
	With plain washer	1262.9	1300.31	2.96
	with spring washer	1298.9	1300.31	0.10

As the FEM values are approximate values but they are found close to the experimental values as the error in the results are very less. Hence the reliability of FEM method is good in giving the results approximately equal to the experimental or real value problems with the predefined boundary conditions.
Also it is validate that the experimental design of Test rig is achieved its desired objective to give the bolt preload value in static condition.

7.2 Results of Bolt Preload Decay in Consideration of Tightening Torque and Frequency for M4 Screw.

Here the experimental results were discussed in order to find how effect of the vibration leads to the loosening of the fastener system and it is also tries to establish the relation between tightening torque and preload values and the frequency at which the loosening occurs.

Table 7.2 Reduction in Bolt Preload for M4 size screw with end conditions.

End Conditions	Bolt Preload in kN	Rduction in Percentage	Average	Tightening Torque in kN-m	Reduction in Percentage	Average
With NutOnly	14.29768	100	91.426	11.43815	100	91.426
	14.29768	100		11.43815	100	
	13.27642	92.85		10.62114	92.85	
	12.25516	85.71		9.80413	85.71	
	11.23389	78.57		8.98712	7.57	
With plain washer	15.31895	100	91.92	12.2516	100	92.014
	15.31895	100		12.2516	100	
	14.29768	93.33		11.43815	93.36	
	13.27642	86.66		10.62114	86.69	
	12.25516	80		9.80413	80.02	
With spring washer	16.34021	100	96.24	13.07217	100	97.17
	16.34021	100		13.07217	100	
	16.34021	100		13.07217	100	
	15.31895	93.75		12.2516	93.72	
	14.29768	87.49		12.48072	92.17	

From the above table it is clear that the reduction in preload is not a immediate task and hence the loosening not occurs immediately. The reduction in preload is more when the screw is with nut only. The highest value of preload is obtained with end condition when screw is with spring washer.

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The reduction percentage of preload is in the range of 91 to 97.The lowest values i.e. the chances of loosening is more when washer is not used.

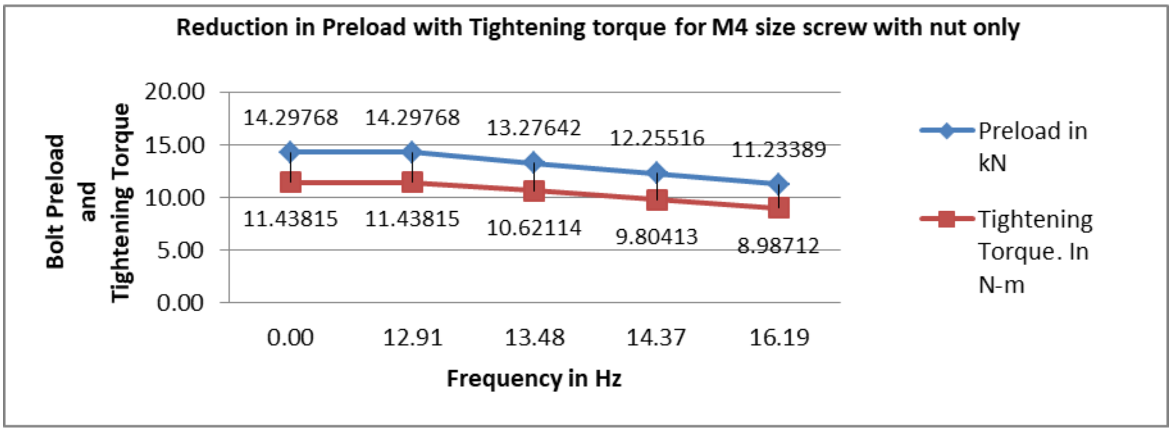

Fig.7.1 Graph of Frequency vs. Preload and Tightening Torque M4 size screw with nut.

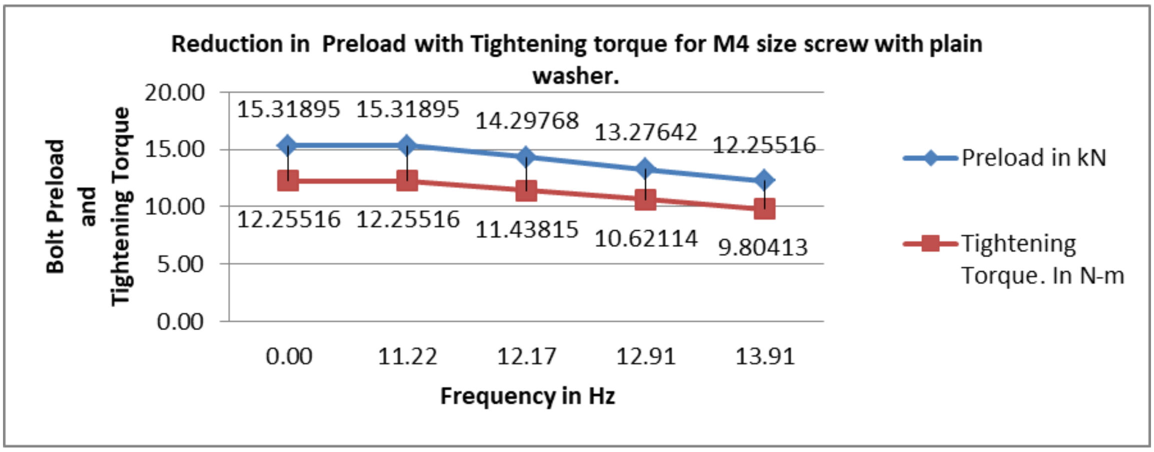

Fig.7.2 Graph of Frequency vs. Preload and Tightening Torque M4 size screw with plain washer

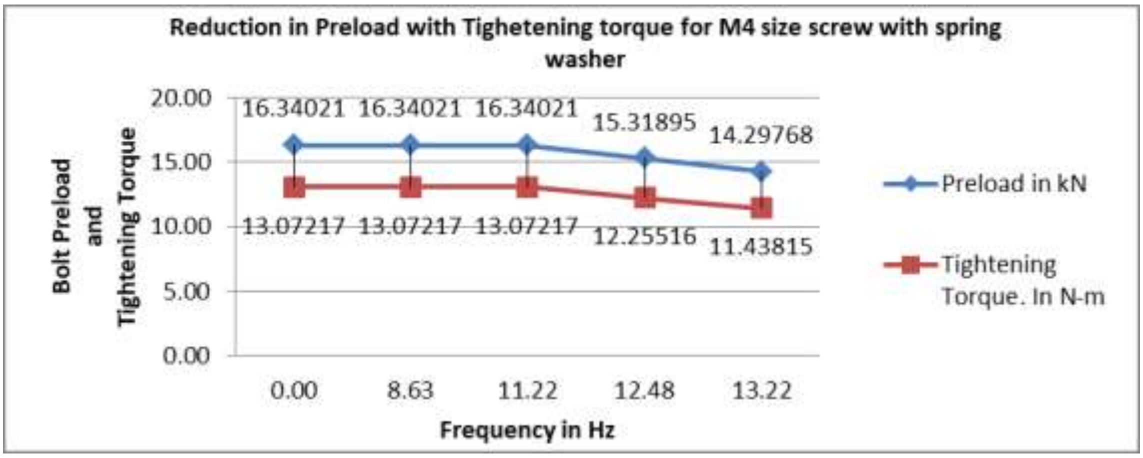

Fig.7.3 Graph of Frequency vs. Preload and Tightening Torque M4 size screw with spring washer.

From the graphical results it is found that the torque is also reduces with the reduction in preload with increase in frequency and it is nonlinear in nature. Hence the loosening characteristics of screw are nonlinear one.

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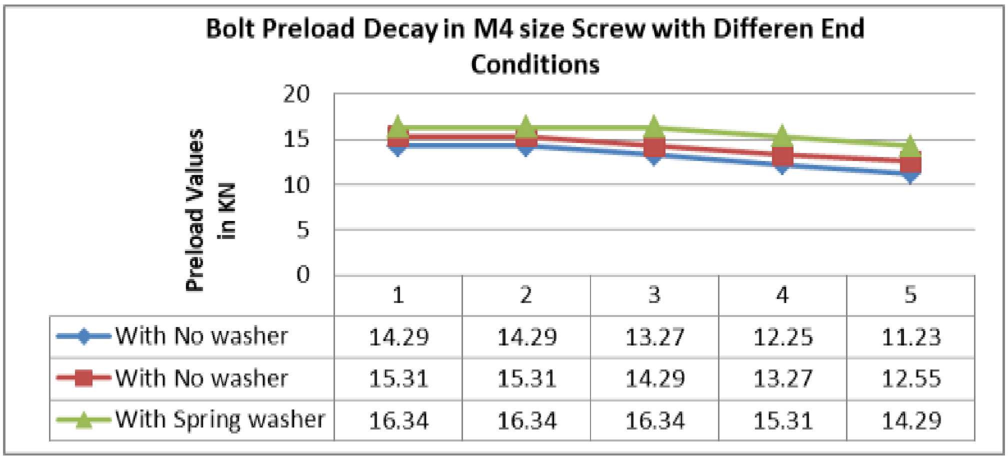
VIII. CONCLUSION:

The work carried out in this dissertation is mainly concerned with the analyzing the loosening characteristics i.e. bolt preload decay. The study is carried out to find out the significant effect of vibration on the fastener system. The experimental study shows that in dynamic condition fasteners loosening effect caused due to the increase in frequency which results in the reduction in the tightening torque as the tightening torque is directly proportional to the bolt preload. The stability of fastener system under vibration is totally dependent upon the tightening torque. Hence it is necessary to design the fastener system when it is subjected to transverse vibration. The concluding remarks of the experimental study are as follows.

- The loosening effect is not immediately occurs. It is a cyclic process for a fastener system which is subjected to the vibration.
- The loosening effect is more in the transverse vibration than longitudinal vibration.
- Loosening effect and bolt preload decay is a non-linear one as it does not occurs immediately.
- The experimental study shows that the loosening phenomenon is not a subsequent characteristic.
- The summery of nonlinear nature of bolt preload decay is shown by the following graphs.



	1	2	3	4	5
With No washer	14.29	14.29	13.27	12.25	11.23
With No washer	15.31	15.31	14.29	13.27	12.55
With Spring washer	16.34	16.34	16.34	15.31	14.29

Fig 8.1 Graph showing nonlinear nature of bolt preload decay with different end condition of M4 size screw.

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