



MECHANISMS AND PREVENTION OF VIBRATION LOOSENING IN BOLTED JOINTS *

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SUMMARY: *Loosening of bolted joints under a vibratory environment has been an ongoing problem associated with many engineering applications. Total loss of the fastener or subsequent fatigue failure due to loss of bolt pre-tension are the predominant failure modes of vibration loosening.*

The present paper analyses two possible mechanisms of vibration loosening and identifies critical parameters in preventing loosening. The mathematical models developed shed light on the effect of various bolted-joint related parameters on vibration loosening and joint integrity. It also develops simple rules-of-thumb for the prevention of vibration loosening. Finally, based on the results of the above mathematical model the paper discusses the available locking devices and their correct usage and limitations.

NOMENCLATURE

D	nominal bolt diameter	m	mass of the particle on the conveyor
d_0	amplitude of rectilinear vibration	m_1	mass of the transported component (bolt or nut)
$d_0(t)$	cyclic rectilinear displacement applied on the conveyor	P	effective downward force
F	friction force on the particle	p	thread pitch
F_0	friction force with respect to conveyor at relative rest of the particle	R	reaction force on the particle
F_b	bolt tension	r_t	effective radius of the thread
F_i	pre-tension of the bolt	r_b	effective radius of the head/nut bearing surface
f	non-dimensional friction parameter (eqns (30, 58))	T	tightening torque
G	non-dimensional inertial parameter (eqns (29, 57))	T_i	installation torque (initial tightening torque)
g	gravitational acceleration	T_l	inertial torque
J	polar moment of inertia	T_p	Break-Loose torque
K	nut factor (eqns (6, 7))	x	displacement in x-direction
K_1	thread pitch contribution of nut factor (eqn (2))	Greek Symbols:	
K_2	thread friction contribution of nut factor (eqn (3))	α	thread flank angle
K_3	bearing friction contribution of nut factor (eqn (4))	β	inclined angle of the conveyor
		δ	transformed time variable (eqn (40))
		Γ	non-dimensional inertial parameter (eqn (31))
		γ	transformed time variable (eqn (39))
		λ	pre-load factor (eqn (48))
		μ	thread friction coefficient

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- μ_b bearing friction coefficient
- μ_s static friction coefficient of the conveyor surface
- μ dry friction coefficient of the conveyor surface
- θ angle of the applied rectilinear vibration to the conveyor surface
- τ non-dimensional time (eqn (27))
- τ_0 non-dimensional time when the particle at momentary rest
- τ_1 non-dimensional time when the particle at momentary rest
- ω angular velocity of rectilinear vibration
- ξ non-dimensional displacement (eqn (26))
- ψ angular rotation of the wheel

Axis Systems:

- OXY inertial co-ordinate system placed on the conveyor
- oxy relative co-ordinate system placed on the particle

Derivatives:

- first derivative with respect to time – velocity
- second derivative with respect to time – acceleration

Subscript:

- mean time averaged

1 INTRODUCTION

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. Gyrotory 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 (eg. 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. Although the frequencies of these fluctuations are spread over a wide spectrum, the general effects of dynamic loading on bolted joints are similar. The main effects are:

- a) loosening of the nut/bolt and
- b) fatigue.

In most situations, both of the above effects may occur in one joint.

It is common experience that vibration loosening of joints may occur when a bolted joint undergoes dynamic load fluctuations.¹ There are a large number of thread-locking devices available in the market in order to alleviate joint loosening problems.⁴ However, use of such devices needs to be done with care. It is very important to understand the root causes and mechanisms behind the phenomenon of vibration loosening in order to decide the optimum method of alleviating the problem. This paper discusses the mechanisms of loosening, mechanisms preventing loosening, and guidelines on using various thread-locking devices commonly available in the market.

2 BREAK-LOOSE TORQUE

There are several definitions related to loosening and tightening of fasteners that need to be established.

Prevailing torque is the torque necessary to rotate the fastener relative to its mating component with the torque being measured while the fastener is in motion and with zero axial load in the assembly. *Prevailing-on* torque is measured when the fastener is advanced towards its seated position. *Prevailing-off* torque is measured when the fastener is being removed.⁵

Break-loose torque is the torque, applied in a removal direction, necessary to start the fastener in motion from its fully preloaded installed position. For most locking fasteners, break-loose torque is higher than *initial tightening torque* (installation torque) T_i .⁵

The *tightening torque* is the torque applied on the fastener while generating axial-tension on the bolt. The tightening torque (T) of a bolted joint can be represented as:²

$$T = F_b \cdot D \cdot \left[\frac{p}{2\pi D} + \frac{r_t \mu_t}{D \cos \alpha} + \frac{r_b \mu_b}{D} \right] \tag{1}$$

$$K_1 = \frac{p}{2\pi D} \tag{2}$$

$$K_2 = \frac{r_t \mu_t}{D \cos \alpha} \tag{3}$$

$$K_3 = \frac{r_b \mu_b}{D} \tag{4}$$

$$T = F_b \cdot D \cdot (K_1 + K_2 + K_3) \tag{5}$$

$$K = K_1 + K_2 + K_3 \tag{6}$$

$$T = F_b \cdot D \cdot K \tag{7}$$

where the geometry of the joint is as shown in Figure 1.

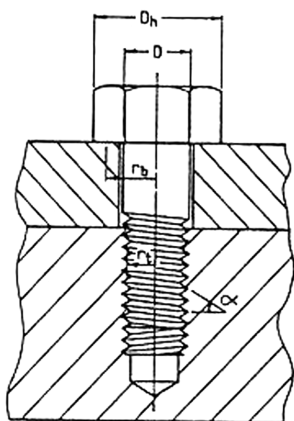


Figure 1: Geometry of a typical bolted joint (left hand thread).

When an *installation torque* (initial tightening torque) of T_i is applied on the bolted joint the resulting bolt pre-tension force F_i will produce a clamping force F_i between the joint members when the joint is free from external loads. For a non-locking fastener, this pre-tension force F_i will produce a break-loose torque T_p given by;

$$T_p = F_i \cdot D \cdot \left[\frac{-p}{2\pi D} + \frac{r_i \mu_i}{D \cos \alpha} + \frac{r_b \mu_b}{D} \right] \tag{8}$$

$$T_p = F_i \cdot D \cdot (-K_1 + K_2 + K_3) \tag{9}$$

This T_p will be the initial torque required to loosen the joint. The above break-loose torque (T_p) will be smaller than the installation torque (T_i) by amount:

$$T_i - T_p = F_i \cdot \left[\frac{p}{\pi} \right] \tag{10}$$

This indicates that the break-loose torque for a coarse threaded fastener will be smaller than that for a fine threaded fastener if everything else remains the same. In fact, the break-loose torque (T_p) will decrease linearly with the pitch (p) of the fastener. As is well known in the industry, fine threaded fasteners are more suitable for applications subject to high vibration.^{1,2}

As shown in the above eqn (8) the prevailing torque T_p is increasing with increasing pre-tension load (F_i) and the increasing friction coefficients and radii μ_i , μ_b and r_i , r_b . As r_i and r_b are proportional to the nominal diameter (D) of the fastener, both the tightening and loosening torque will increase for a given pre-tension load when the fastener size is increased. Therefore, as

the deficit between tightening and loosening torque is independent of the fastener diameter, it is better to use a larger diameter fastener for an application subject to significant vibration.

Similarly, increasing thread angle (α) will increase both break-loose and installation torque if all the other parameters are kept constant. Therefore a “buttress” thread ($\alpha=0^\circ$) will loosen easily compared to a metric thread ($\alpha=30^\circ$).

The break-loose torque (T_p) increases linearly with the pre-tension load at a rate lower than the increase of the tightening torque. Once the possible magnitude of the loosening torque anticipated during the life-span of the bolted joint is estimated, a minimum pre-tension load can be determined to prevent vibration loosening without using any additional locking devices.

The above analysis does not take into account any break-loose torque component due to thread damage or plastic/elastic deformation of thread. The analysis is only a first order (linear) analysis. When the fastener is tightened closer to the yield load, local plastic deformation occurring at the threads will incorporate further break-loose torque.

Now that we have some understanding of the break-loose torque or the torque preventing loosening, it is opportune to investigate the mechanisms of loosening.

3 LOOSENING MECHANISMS

There are several theories on the mechanisms of loosening in a vibratory environment.¹

3.1 Mechanism of a vibratory conveyor

One theory proposes that the bolt loosening mechanism is similar to the mechanism of a vibratory conveyor transporting a particle up a ramp. The static friction associated between the particle and the ramp surface and the inertia forces developed by particular vibration mechanism will drive the particle up the ramp. The vibration mechanism causes an overall upward (along the ramp) inertial force larger than the downward force on the particle. When the upward inertial force is larger than the resultant of gravitational force and the static friction the particle will travel upwards. When the resultant of inertial force, gravitational force and the static friction force is acting downwards (along the ramp) then the particle will travel downwards. Both of these actions will occur in a single cycle of vibration. When the resultant travel is upwards then the particle will continue to travel up the conveyor.

Consider a conveyor of inclination β as shown in Figure 2. A cyclic rectilinear displacement of $d_0(t)$ is



applied at an angle θ to the conveyor (axes system oxy) with respect to inertial axes system OXY. Consider a heavy material particle B of mass m having a motion of translation in the OXY plane. Its absolute coordinates X, Y are related to the relative coordinates x, y by the expressions;

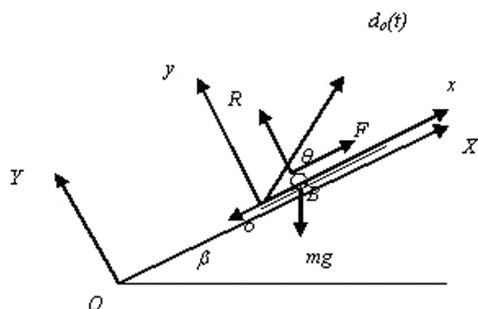


Figure 2: Forces on a particle on a conveyor.

$$X = x + d_o(t) \cos \theta \tag{11}$$

$$Y = y + d_o(t) \sin \theta \tag{12}$$

If the particle is not in contact with the conveyor it is acted upon only by gravitational force components $-mg \sin$ and $-mg \cos$ along OX and OY directions respectively.

If the static dry friction coefficient of the conveyor surface is μ , and the reaction force is R , then the friction force F :

$$F = -\mu R \operatorname{sgn} \dot{x} \tag{13}$$

The direction of this friction force will always be opposite to the direction the particle tends to move.

A simple harmonic displacement $d_o(t)$ with an amplitude d_o and angular velocity ω can be given as:

$$d_o(t) = d_o \cos \omega t \tag{14}$$

By applying Newton’s Second Law to the particle, the following equations of motion can be established.

$$m\ddot{X} = -mg \sin \beta + F \tag{15}$$

$$m\ddot{Y} = -mg \cos \beta + R \tag{16}$$

By substitution of (11), (12) and (14) in (15) and (16):

$$m\ddot{x} = md_o \omega^2 \cos \omega t \cos \theta - mg \sin \beta + F \tag{17}$$

$$m\ddot{y} = md_o \omega^2 \cos \omega t \sin \theta - mg \cos \beta + R \tag{18}$$

For the particle not to depart the surface $R > 0$ and

$y(t) = 0$. Hence the following inequality must be satisfied:

$$R = mg \cos \beta - md_o \omega^2 \cos \omega t \sin \theta > 0 \tag{19}$$

This will require;

$$\frac{g \cos \beta}{d_o \omega^2 \sin \theta} > 1 \tag{20}$$

Similarly, when the particle is at state of relative rest with respect to the conveyor: $\dot{x}(t) = \ddot{x}(t) = 0$, then,

$$md_o \omega^2 \cos \omega t \cos \theta - mg \sin \beta + F_0 = 0 \tag{21}$$

$$F_0 = mg \sin \beta - md_o \omega^2 \cos \omega t \cos \theta \tag{22}$$

$$|F_0| < \mu_1 R \tag{23}$$

$$\left| mg \sin \beta - md_o \omega^2 \cos \omega t \cos \theta \right| < \mu_1 (mg \cos \beta - md_o \omega^2 \cos \omega t \sin \theta) \tag{24}$$

where μ_1 is the static friction coefficient. If this condition is violated the particle starts to slip.

The equation of motion for the state of relative slipping can be derived by substituting (13) and (19) in (15):

$$\ddot{x} = d_o \omega^2 \cos \omega t \cos \theta - g \sin \beta - \mu (-d_o \omega^2 \cos \omega t \sin \theta + g \cos \beta) \operatorname{sgn} \dot{x} \tag{25}$$

In order to solve the above differential equation, introduce the following non dimensional variables:

$$\xi = \frac{x}{d_o \cos \theta} \tag{26}$$

$$\tau = \omega t \tag{27}$$

Now;

$$\ddot{\xi} = \cos \tau - \frac{g \sin \beta}{d_o \omega^2 \cos \theta} + \mu \tan \theta \left(\cos \tau - \frac{g \cos \beta}{d_o \omega^2 \sin \theta} \right) \operatorname{sgn} \dot{\xi} \tag{28}$$

by substituting:

$$G = \frac{g \sin \beta}{d_o \omega^2 \cos \theta} \tag{29}$$

$$f = \mu \tan \theta \tag{30}$$



$$\Gamma = \frac{g \cos \beta}{d_o \omega^2 \sin \theta} \quad (31)$$

in eqn (28);

$$\ddot{\xi} = \cos \tau - G + f(\cos \tau - \Gamma) \operatorname{sgn} \dot{\xi} \quad (32)$$

Depending on the values of above parameters G , f and Γ , various regular regimes of motion can be established. For the current analysis we only consider the regime with two momentary stops per period (2π) and no relative rest periods. The particle will slide in +ox direction for one part of the cycle and in the opposite direction during another. Lets consider that the particle moves forward in the time interval τ_0 to τ_1 and then backward to complete the cycle.

Now,

$$\begin{aligned} \dot{\xi} > 0; \text{ at } \tau_0 < \tau < \tau_1; \text{ and } \dot{\xi} < 0; \\ \text{at } \tau_1 < \tau < 2\pi + \tau_0; \\ \text{and } \dot{\xi} = 0 \text{ at times } \tau_0, \tau_1 \text{ and } 2\pi + \tau_0. \end{aligned}$$

The forward motion is now described by:

$$\ddot{\xi} = (1 + f) \cos \tau - (G + \Gamma f) \quad (33)$$

The solution to eqn (33) can be written as:

$$\begin{aligned} \dot{\xi} = \sin \tau - \sin \tau_0 + f \sin \tau - f \sin \tau_0 \\ - (G + f\Gamma)(\tau - \tau_0) \end{aligned} \quad (34)$$

This must become zero at $\tau = \tau_1$. Hence:

$$\begin{aligned} 0 = \sin \tau_1 - \sin \tau_0 + f \sin \tau_1 - f \sin \tau_0 \\ - (G + f\Gamma)(\tau_1 - \tau_0) \end{aligned} \quad (35)$$

Similarly, the following equations can be derived for the backward motion:

$$\ddot{\xi} = (1 - f) \cos \tau - (G - \Gamma f) \quad (36)$$

$$\begin{aligned} \dot{\xi} = \sin \tau - \sin \tau_1 - f \sin \tau + f \sin \tau_1 - (G - \Gamma f)(\tau - \tau_1) \end{aligned} \quad (37)$$

$$0 = \sin \tau_0 - \sin \tau_1 - f \sin \tau_0 + f \sin \tau_1 - (G - \Gamma f)(2\pi + \tau_0 - \tau_1) \quad (38)$$

Equations (35) and (38) can be solved for the two unknowns τ_0 and τ_1 . By substituting the simple transformations:

$$\frac{\tau_1 - \tau_0}{2} = \gamma \quad (39)$$

$$\frac{\tau_1 + \tau_0}{2} = \delta \quad (40)$$

we get;

$$\sin \gamma \cos \delta = G\gamma - \frac{\pi}{2}(G - \Gamma f) \quad (41)$$

$$f \sin \gamma \cos \delta = \Gamma f \gamma + \frac{\pi}{2}(G - \Gamma f) \quad (42)$$

By solving equations (41) and (42):

$$\gamma = \frac{\pi(1 + f)}{2f} \frac{\Gamma f - G}{\Gamma - G} \quad (43)$$

$$\delta = \cos^{-1} \left(\frac{\Gamma f \gamma + \frac{\pi}{2}(G - \Gamma f)}{f \sin \gamma} \right) \quad (44)$$

Now the mean velocity of the particle being conveyed in the forward direction is:

$$\xi_{mean} = \frac{1}{2\pi} \left(\int_{\tau_0}^{\tau_1} \dot{\xi} d\tau + \int_{\tau_1}^{2\pi + \tau_0} \dot{\xi} d\tau \right) \quad (45)$$

By substituting equations (34) and (37) with appropriate τ_0 and τ_1 derived from equations (43) and (44) we get the mean speed of conveying:

$$\xi_{mean} = -\cos \gamma \sin \delta - \frac{2f}{\pi} \left[\left(\gamma - \frac{\pi}{2} \right) \cos \gamma - \sin \gamma \right] \sin \delta \quad (46)$$

If the mean conveying speed is positive the particle will move upwards and if negative the particle will move downwards.

A complete analysis of the vibratory conveyor mechanism is beyond the scope of this paper and can be found in Bykhovsky 1972.³

3.2 Application of vibratory conveyor principle to vibration loosening of fasteners

By applying the above analogy to the threaded fastener; the thread essentially acts as the ramp with associated inclined plane and friction properties. The nut or the bolt – depending on which top surface of the thread the load is bearing on – represents the materials being transported. For example, if the nut/bolt assembly is in a vertical axis with the bolt head facing up, the load bearing will occur on the



top surface of the bolt thread, hence making the nut the material being transported. Similarly, if the bolt head is facing downwards then the bolt will become the material being transported. In either case negative mean conveying velocity will loosen the joint.

The pre-tension will provide an additional reaction force on the head/nut joint interface and on the thread (significantly larger than that caused by the mass of the nut), hence increasing the friction forces. In this case, the resulting inertial forces due to vibration should overcome the reaction force due to pre-tension before loosening can occur. It can be shown through an order of magnitude analysis that the gravity force becomes insignificant once a reasonable clamping force (pre-tension) is applied. If the top flank angle of the thread is α and the clamping force is F_i and the mass of the transported component is m_i , then the effective downward force (P) due to clamping force and mass is:

$$P = \frac{(F_i + m_i g)}{\cos \alpha} \quad (47)$$

for simplification, by substituting:

$$F_i = m_i \lambda \quad (48)$$

Now,

$$P = \frac{m_i (\lambda + g)}{\cos \alpha} \quad (49)$$

Once a pre-load is applied to a bolted joint the bearing friction, under nut or under head friction, will also help prevent it from loosening. The pre-load F_i will exert a frictional force of ${}_b F_i$ on the bearing surface. This can be approximated by two forces ${}_b F_i \tan$ in the direction of the pre-load and ${}_b F_i / \cos$ in the direction of friction force on the thread where $= \tan^{-1}(p/\pi D)$. Now the total effective downward force is:

$$P = \frac{m_i (\lambda + \mu_b \lambda \tan \beta \operatorname{sgn} \dot{x} + g)}{\cos \alpha} \quad (50)$$

For most cases β and μ_b are very small compared to the contribution from the pre-load λ and g . Therefore, for simplicity, we can omit the friction term from the above equation, hence making it same as equation (49).

Now the equation of motion (15), (16) can be re-written for this case as:

$$m_i \ddot{X} = -\frac{m_i (\lambda + g)}{\cos \alpha} \sin \beta + F - \frac{\mu_b F_i}{\cos \beta} \operatorname{sgn} \dot{x} \quad (51)$$

$$m_i \ddot{Y} = -\frac{m_i (\lambda + g)}{\cos \alpha} \cos \beta + R \quad (52)$$

then, equation (19) becomes:

$$R = \frac{m_i (\lambda + g)}{\cos \alpha} \cos \beta - m_i d_o \omega^2 \cos \omega t \sin \theta > 0 \quad (53)$$

This will require;

$$\frac{(\lambda + g) \cos \beta}{d_o \omega^2 \sin \theta \cos \alpha} > 1 \quad (54)$$

Now the equation of motion that will govern the state of relative slip can be re-written as:

$$\begin{aligned} \ddot{x} = & d_o \omega^2 \cos \omega t \cos \theta - \frac{(\lambda + g)}{\cos \alpha} \sin \beta \\ & - \mu_i \left(-d_o \omega^2 \cos \omega t \sin \theta + \frac{(\lambda + g)}{\cos \alpha} \cos \beta + \frac{\mu_b \lambda}{\mu_i \cos \beta} \right) \operatorname{sgn} \dot{x} \end{aligned} \quad (55)$$

By using the same transformations as in the previous case the corresponding equation of motion is:

$$\begin{aligned} \ddot{\xi} = & \cos \tau - \frac{(\lambda + g) \sin \beta}{d_o \omega^2 \cos \theta \cos \alpha} + \\ & \mu_i \tan \theta \left(\cos \tau - \frac{(\lambda + g) \cos \beta}{d_o \omega^2 \sin \theta \cos \alpha} - \frac{\mu_b \lambda}{\mu_i d_o \omega^2 \sin \theta \cos \beta} \right) \operatorname{sgn} \dot{\xi} \end{aligned} \quad (56)$$

by substituting:

$$G = \frac{(\lambda + g) \sin \beta}{d_o \omega^2 \cos \theta \cos \alpha} \quad (57)$$

$$f = \mu_i \tan \theta \quad (58)$$

$$\Gamma = \frac{1}{d_o \omega^2 \sin \theta} \left(\frac{(\lambda + g) \cos \beta}{\cos \alpha} + \frac{\mu_b \lambda}{\mu_i \cos \beta} \right) \quad (59)$$

get;

$$\ddot{\xi} = \cos \tau - G + f(\cos \tau - \Gamma) \operatorname{sgn} \dot{\xi} \quad (60)$$

Equation (60) is identical to the equation (32) for



the conveyor case but with different values for G , f and Γ .

Hence, the solution to this equation will also be of the same form as that for equation (32) with modified constants.

By using the above analysis, the trends of different variables of vibration loosening can be established. As the above analysis only considers the specific case with upward and downward slipping with momentary stops, the current trend analysis does not cover the full extent of the solution domain.

3.3 Inertial torque

Another mechanism of vibration loosening is the inertial torque. This is quite different to the mechanism of loosening discussed in the previous section. Treatment of this mechanism is relatively simple as the inertial torque can be readily approximated by using Newton's Second Law.

A wheel nut will be subject to rotational inertia due to rotation of the wheel. The equivalent inertial torque (T_i) on a nut with polar moment of inertia J and a rotational acceleration $\ddot{\psi}$ can be given as:

$$T_i = J\ddot{\psi} \tag{61}$$

For a right hand thread, an anti-clockwise acceleration of the wheel (stud/shaft) will cause the nut to tighten while clockwise acceleration of the shaft will cause loosening of the wheel nut if the above inertial torque T_i exceeds the break-loose torque T_p described in equation (8). A sudden anti-clockwise deceleration of the shaft could loosen a right hand wheel nut if adequate pre-load is not provided. Similarly, if the shaft is rotating clockwise, screwing of any thing to this shaft should be done with a left handed thread.

Depending on the arrangement of the rotating joint either the nut or the bolt has the potential for loosening. Typically this would be the moving or non-stationary component. If a bolt is used to join a wheel to a rotating shaft (fixed nut) the bolt will loosen. In this case the polar moment of inertia of the bolt should be taken as J . Typically, bolts have a larger polar moment of inertia compared to nuts. Hence as a rule-of-thumb it is advisable to use a nut with a fixed bolt instead of a bolt with a fixed nut. This is common in automobile wheel assemblies.

Rotational acceleration $\ddot{\psi}$ may have a harmonic form if the wheel is subject to harmonic motion. In this case, intermittent loosening may occur when the dynamic torque momentarily exceeds the break-loose torque. At each loosening the break-loose torque comes down due to relaxation of the pre-load hence loosening will continue at an exponential rate. In general, the loosening in this mode will start due

to the largest inertial acceleration / deceleration most likely caused by impact loads.

Once the polar moment of inertia of the nut/bolt and the possible maximum acceleration / deceleration levels are established for a particular joint, the minimum break-loose torque requirement can be calculated using eqn (61). Then, using eqn (8), the necessary minimum pre-load F_i to prevent this form of loosening can be calculated.

Now;

$$T_p = F_i \cdot D \cdot \left[\frac{-p}{2\pi D} + \frac{r_i \mu_i}{D \cos \alpha} + \frac{r_b \mu_b}{D} \right] > J\ddot{\psi} \tag{62}$$

$$F_i > \frac{J\ddot{\psi}}{\left[\frac{-p}{2\pi} + \frac{r_i \mu_i}{\cos \alpha} + r_b \mu_b \right]} \tag{63}$$

Please note that if the joint is subject to impact loads the resulting accelerations $\ddot{\psi}$ could be very large and appropriate approximations should be used when using eqn (63).

4 TREND ANALYSIS OF VIBRATION LOOSENING USING CONVEYOR THEORY

4.1 Effect of vibration excitation

In the present analysis, vibration is characterised by the amplitude (d_0), frequency (ω) and the direction of vibration (θ). Vibration loosening is identified by a negative mean non-dimensional velocity ξ_{mean} .

The actual mean relative velocity (-ve indicates loosening) is:

$$\dot{x} = \xi d_0 \cos \theta \tag{64}$$

In the present analysis, rectilinear vibration has been considered. This assumes the in-phase vibration in any two mutually perpendicular directions. For simplicity, two directions of excitation (θ), namely, lateral ($\theta = -\beta$) and longitudinal ($\theta = 90-\beta$) to the fastener axis have been considered. These two cases provide uniform effects along the full extent of the thread. Rectilinear vibration applied at any other direction will have different effects on diametrically opposite sides of the thread. These cases are not covered under current analysis.

For a M12 coarse threaded fastener with under head / nut and thread friction coefficient of 0.15 subjected to a vibration amplitude of 1mm lateral to the fastener axis, the effect of frequency of vibration at different pre-tension levels on vibration loosening is shown



in Figure 3. As shown in the figure, the required frequency of vibration for loosening is increasing with increasing pre-tension. Increase in negative mean velocity indicates faster loosening. For a given pre-load, with increasing frequency, the loosening velocity will reach a negative maximum value. When there is no pre-tension vibration loosening will start at a very low frequency.

loosening.

Figure 4 shows the effect of amplitude of vibration on mean velocity (loosening is -ve) while maintaining the lateral vibration frequency at 750Hz with all other parameters similar to the previous case.

As can be expected, this also shows that the increasing pre-load will increase the minimum amplitude

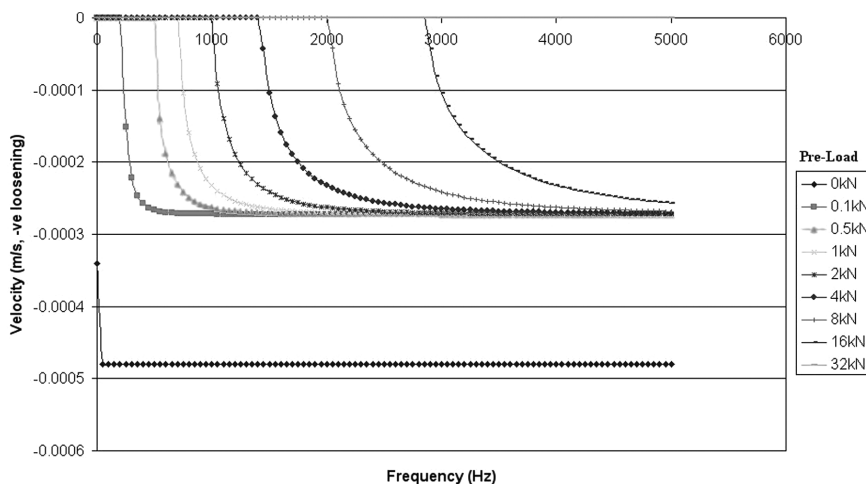


Figure 3: Effect of Frequency (M12, p=1.7mm, $\mu_t=\mu_b=1.05$, $d_0=1$ mm, Lateral)

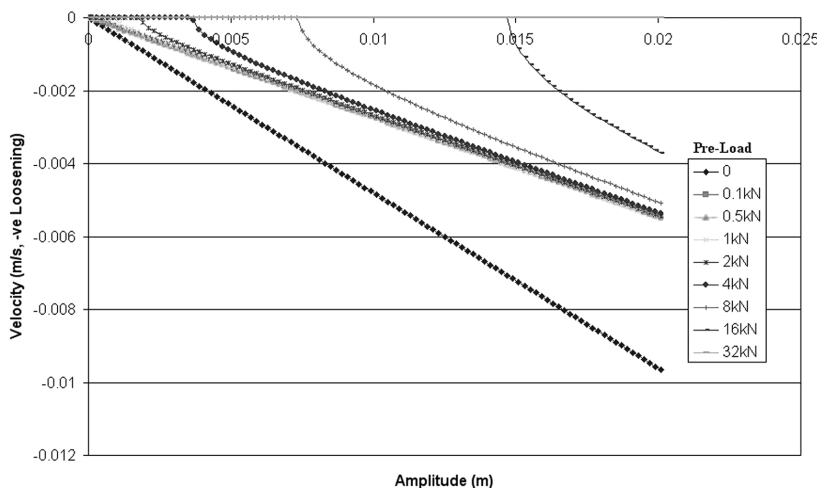


Figure 4: Effect of Amplitude (M12-p=1.75mm, freq=750Hz, $\mu_t=\mu_b=1.05$, $d_0=1$ mm, Lateral)

When interpreting the results we should keep in mind that once the loosening starts at a given pre-load the pre-load will progressively reduce. Once the pre-load is reduced the rate of loosening will increase until the failure occurs. Therefore, it is prudent in the design to eliminate the possibility of start of any

required to start loosening. Once loosening starts the pre-load will be lost and speed of loosening further increases. A screw with no pre-load will start to loosen at very small amplitude at the frequency of 750Hz.



Figures 5 and 6 show the corresponding cases for longitudinal vibration. In comparison, it is evident that the impact of longitudinal vibration in loosening is significantly less than that of lateral vibration. If the vibration is occurring in the longitudinal direction a relatively small pre-load will assure that no vibration loosening will take place.

fastener at a vibration amplitude of 1mm at a frequency of 750Hz for lateral and longitudinal vibration conditions respectively. The bearing friction coefficient is assumed to be 0.15. The variation of pre-load is also shown. As seen in Figure 7, the friction coefficient required to avoid loosening will come down with increasing pre-load. At zero pre-load,

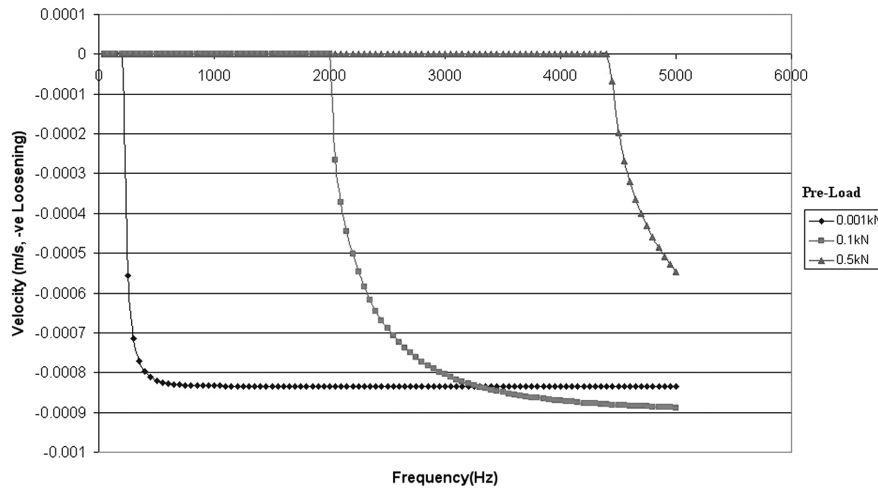


Figure 5: Effect of Frequency (M12-p=1.75mm, $\mu_t=\mu_b=0.15$, $d_0=1$ mm, Longitudinal).

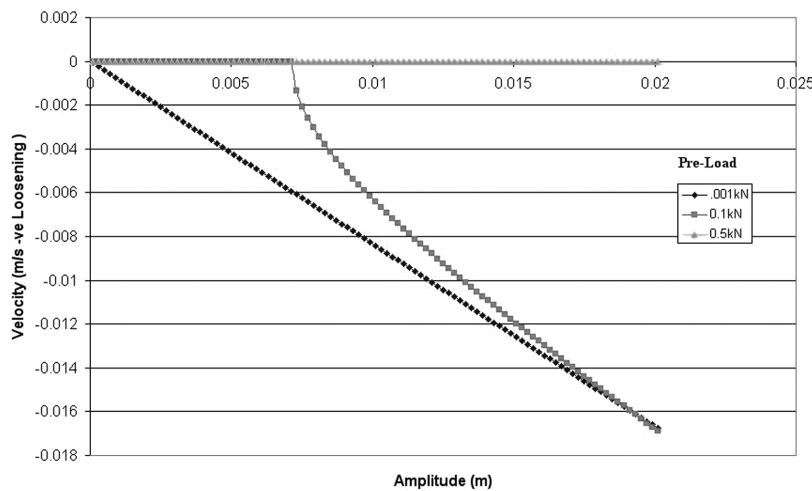


Figure 6: Effect of Amplitude (M12-p=1.75mm, $\mu_t=\mu_b=0.15$, $freq=750$ Hz, Longitudinal).

4.2 Effect of thread friction

Figures 7 and 8 show the effect of thread friction on vibration loosening of an M12 coarse thread

vibration tightening may occur at very small thread friction coefficients (≈ 0.02). At pre-loads higher than ≈ 2 kN, no loosening will occur at 750Hz and 1mm amplitude vibration condition.

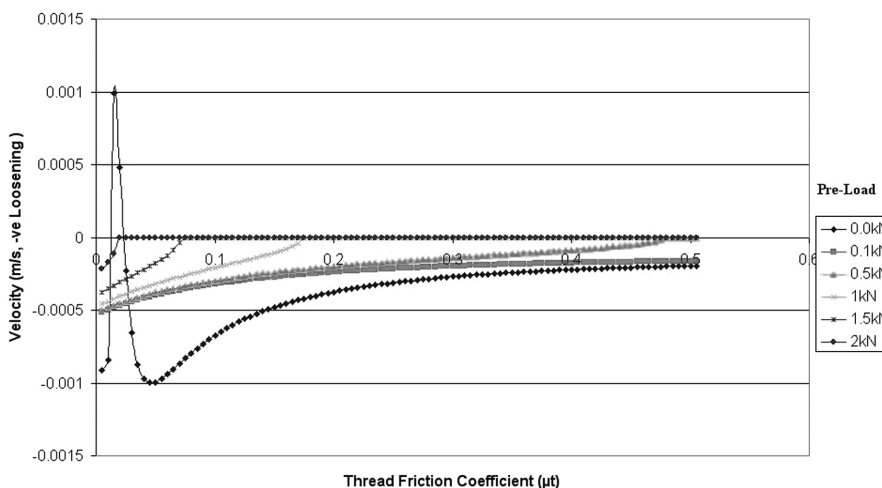


Figure 7: Effect of thread friction (M12-p=1.75mm, freq=750Hz, $\mu_b=0.15$, $d_0=1.0$ mm, Lateral)

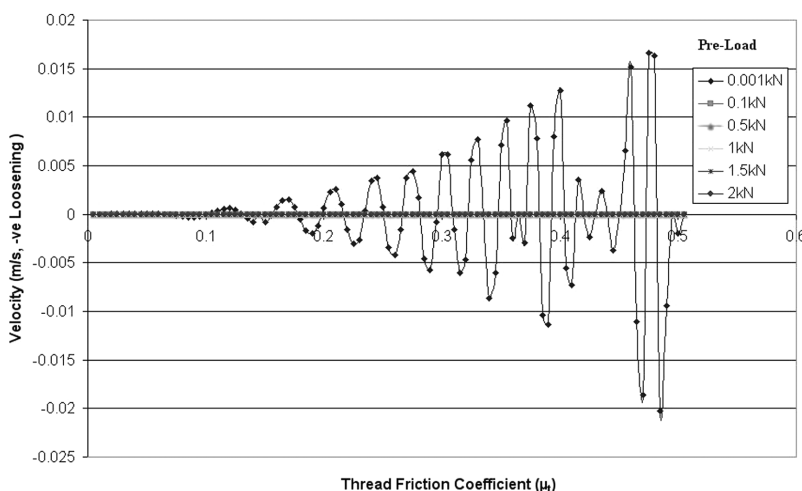


Figure 8: Effect of thread friction (M12-p=1.75mm, $\mu_b=0.15$, $d_0=1$ mm, freq=750Hz, Longitudinal)

Figure 8 shows no potential for loosening even at 0.1kN pre-load; however, it shows some strange behaviour at no pre-load. This could be due to a numerical instability in the solution at zero pre-load and may be ignored. Further refinement in simplifying assumptions may be required to avoid this instability.

4.3 Effect of bearing friction

Figures 9 and 10 show the effect of bearing friction on vibration loosening of an M12 coarse thread fastener at a vibration amplitude of 1mm at a frequency of 750Hz for lateral and longitudinal vibration conditions respectively. The thread friction coefficient is assumed to be 0.15. The effect of variation of clamp load is also shown.

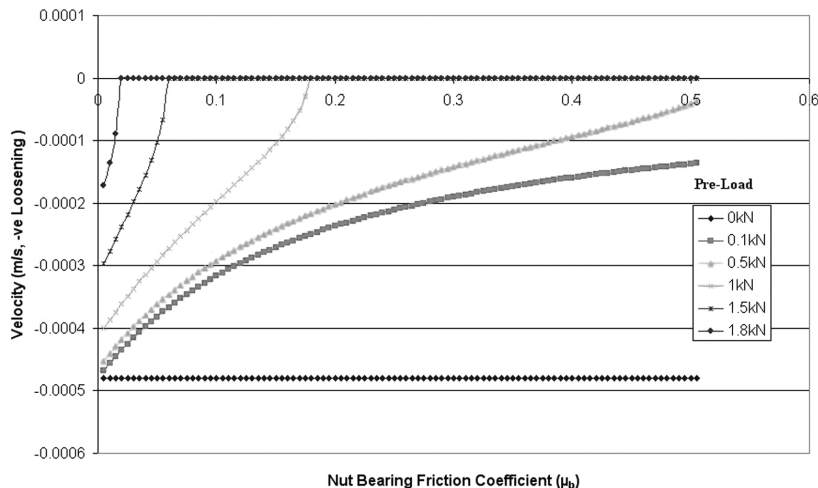


Figure 9: Effect of bearing friction (M12- $p=1.75\text{mm}$, $\mu_t=0.15$, $d_0=1\text{mm}$, $\text{freq}=750\text{Hz}$, Lateral)

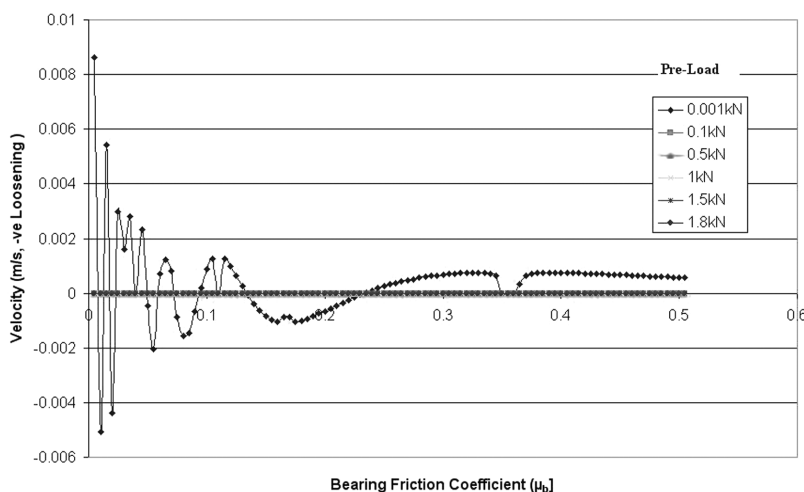


Figure 10: Effect of bearing friction (M12- $p=1.75\text{mm}$, $\mu_t=0.15$, $d_0=1\text{mm}$, $\text{freq}=750\text{Hz}$, Longitudinal)

As can be expected, and as seen in Figure 9, at zero pre-load the bearing friction coefficient has no effect on loosening. This is due to no under head/nut contact at zero pre-load. With increasing pre-load the minimum friction requirement goes down showing that a pre-load of 2kN will not loosen the screw at any friction coefficient for the given vibration condition.

The case for longitudinal vibration is shown in Figure 10. This shows a similar behaviour to the effect of thread friction. At non-zero pre-load, loosening will not occur, and at zero pre-load the results shown may

not present a valid case as zero pre-load nullifies the effect of bearing friction.

4.4 Effect of thread pitch

The effect of thread pitch on vibration loosening under different levels of vibration (amplitude 1mm, increasing frequency) with thread and bearing friction coefficients of 0.15 and a pre-load of 1kN for lateral and longitudinal vibration conditions is shown in Figures 11 and 12 respectively.

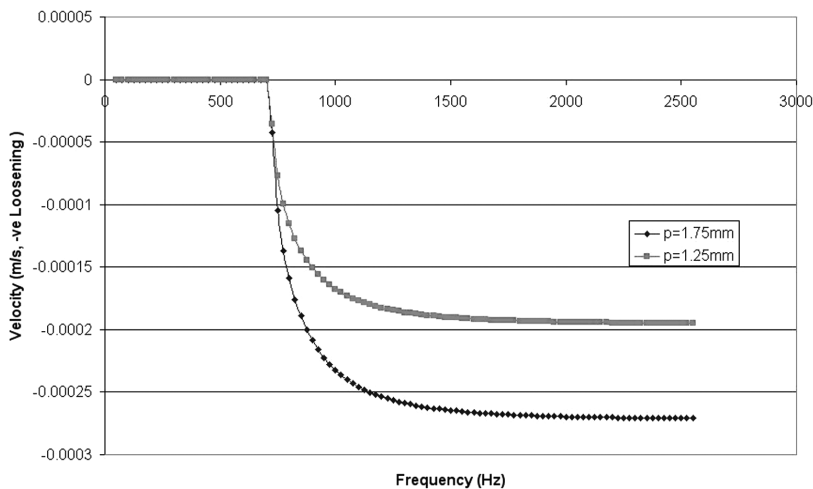


Figure 11: Effect of thread pitch (M12, $\mu_t = \mu_b = 0.15$, $d_0 = 1.0\text{mm}$, $F_i = 1.0\text{kN}$, Lateral)

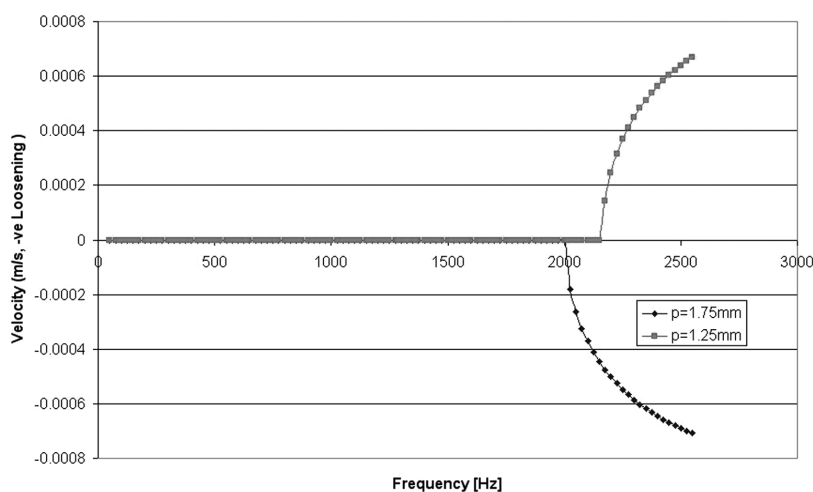


Figure 12: Effect of thread pitch (M12, $\mu_t = \mu_b = 0.15$, $d_0 = 1.0\text{mm}$, $F_i = 1.0\text{kN}$, Longitudinal)

As seen in Figure 11, loosening will start at the same frequency but the coarse thread will loosen significantly faster than the fine thread under lateral vibration.

Figure 12 shows that a fine thread ($p=1.25\text{mm}$) will cause the screw to tighten while a coarse thread ($p=1.75\text{mm}$) will cause it to loosen under longitudinal vibrations. Theoretically there should be a critical pitch at which the behavior would change from loosening to tightening. This is beyond the current scope of this paper. However, this opens up a new possibility that vibration tightening, through design, could be applied to machinery subject to severe

vibrations. Further research into this window of conditions may be of great importance for future developments.

4.5 Effect of thread flank angle

The effect of thread flank angle on vibration loosening under different pre-loads at a level of vibration amplitude 1mm, and frequency 750Hz with thread and bearing friction coefficients of 0.15 for lateral and longitudinal vibration conditions is shown in Figures 13 and 14 respectively.

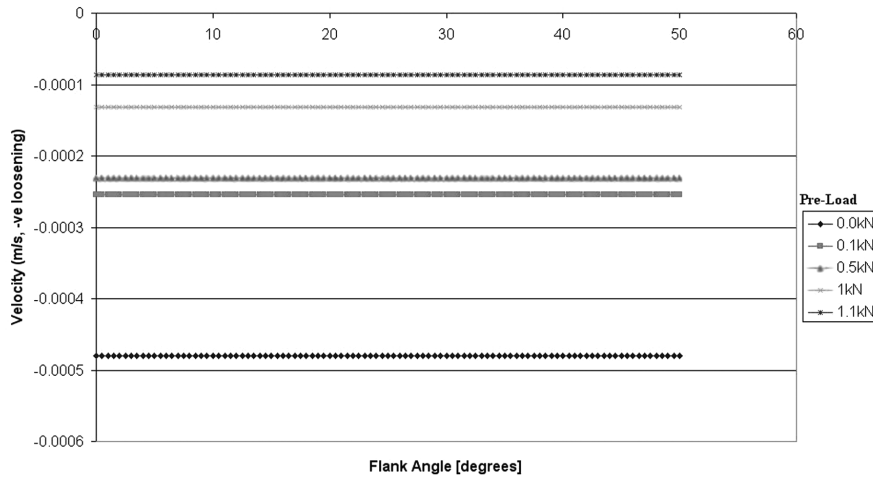


Figure 13: Effect of thread flank angle (M12- $p=1.75\text{mm}$, $\mu_t=\mu_b=0.15$, $d_0=1.0\text{mm}$, $\text{freq}=750\text{Hz}$, Lateral)

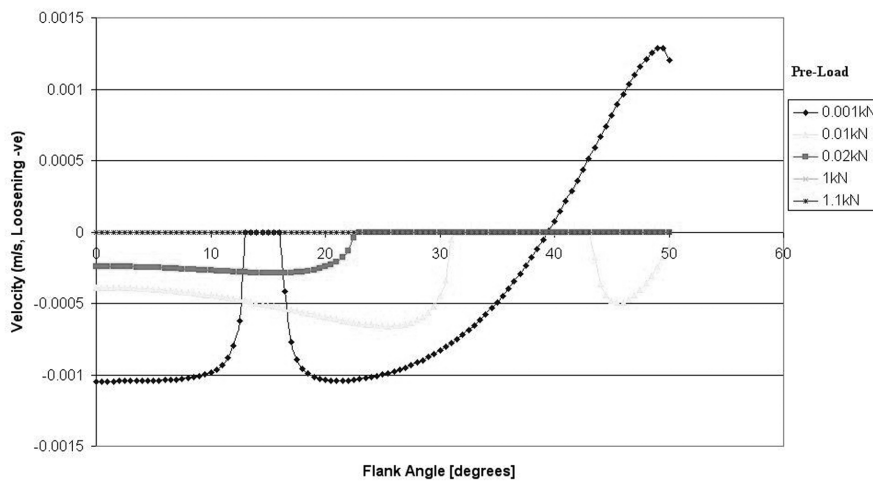


Figure 14: Effect of thread flank angle (M12- $p=1.75\text{mm}$, $\mu_t=\mu_b=0.15$, $d_0=1.0\text{mm}$, $\text{freq}=750\text{Hz}$, Longitudinal)

From Figure 13 it is evident that the flank angle does not affect loosening under lateral vibration.

As discussed before, pre-load will have a significant effect on vibration loosening. In Figure 14, at very low pre-loads lower flank angles are susceptible to

vibration loosening. Figures 15 and 16 below show the effect of flank angle under different levels of vibration (amplitude 1mm, increasing frequency) with thread and bearing friction coefficients of 0.15 and a pre-load of 1kN for lateral and longitudinal vibration conditions respectively.

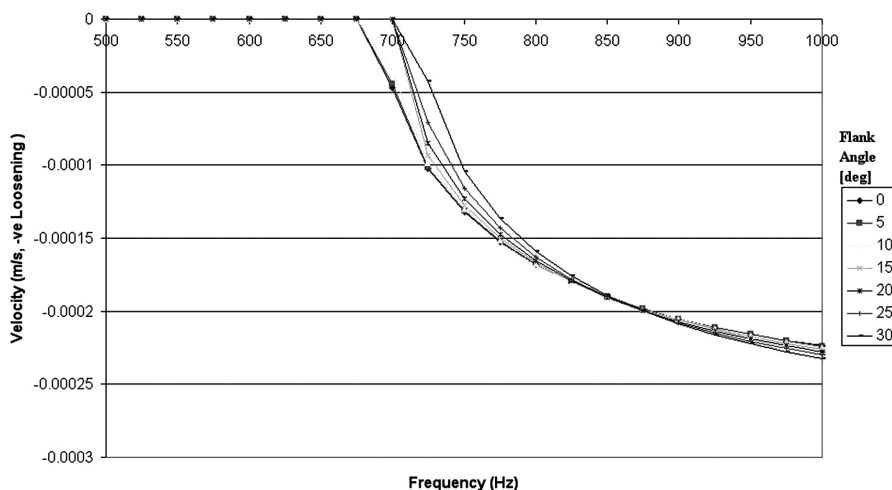


Figure 15: Effect of thread flank angle (M12, $p=1.75\text{mm}$, $\mu_t=\mu_b=0.15$, $d_0=1.0\text{mm}$, $F_i=1.0\text{kN}$, Lateral)

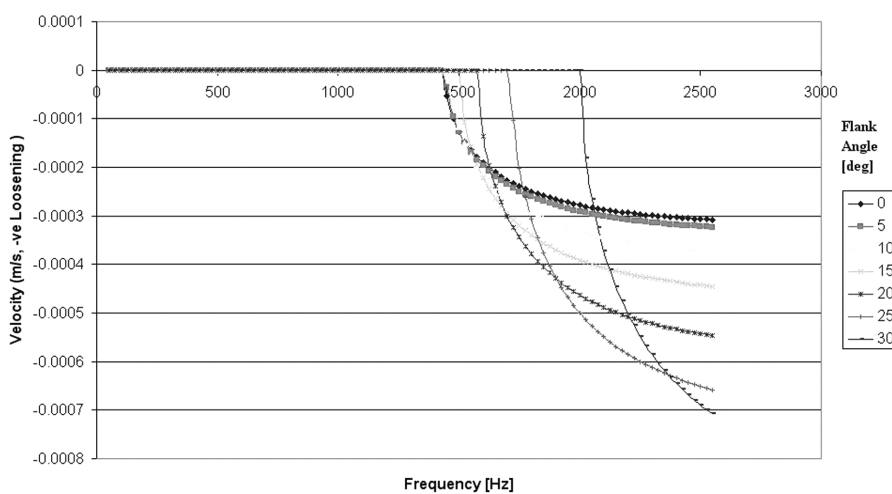


Figure 16: Effect of thread flank angle (M12, $p=1.75\text{mm}$, $\mu_t=\mu_b=0.15$, $d_0=1.0\text{mm}$, $F_i=1.0\text{kN}$, Lateral, Longitudinal)

As evident in Figures 15 and 16, lower flank angles again will start loosening at lower vibration levels under both lateral and longitudinal vibration.

This is one of the reasons why "buttress" threads are not recommended for applications subject to vibration excitation.

As discussed, all of the above considered parameters will have an impact on vibration loosening. In general, as shown, adequate pre-load will solve almost all vibration loosening concerns. Designers should consider these effects when designing machines subject to significant dynamic loads and vibration conditions.

5 PREVENTION OF VIBRATION LOOSENING

Now that we have established the possible mechanisms of vibration loosening, a method of estimating the break-loose torque and effect of various joint parameters on the potential for loosening it is advantageous now to evaluate methods of preventing vibration loosening. There are many devices available in the market that are designed for this purpose. However, there are no specific guidelines provided for the proper selection and use of these devices and their limitations. This section focuses on addressing this issue with the preceding analysis as a baseline.

5.1 Pre-load

Depending on the application, there are various methods of avoiding vibration loosening. As discussed in the previous sections, the most reliable and economical method of preventing vibration loosening is applying adequate pre-load. This will not only stop the chance of vibration loosening but also provide a sound fatigue-tolerant joint.⁶ As shown in Figure 3 a 16kN pre-load on an M12 bolt requires a vibration amplitude of 1mm applied at approximately 2800Hz to start vibration loosening. Similarly as shown in Figure 4 for a frequency of vibration of 750Hz an amplitude of 14.5mm is required to loosen a M12 bolt tightened to a load of 16kN.

For a Class 4.6 M12 bolt, 16kN is approximately 80% of the yield load. Same figures (3 & 4) also show (by interpolation) that a pre-load of 13kN (65% yield load) would still require severe vibration conditions to start loosening. This confirms our experience that a pre-load higher than approximately 65% of the yield load of the fastener is adequate to prevent loosening at vibration levels commonly experienced in the field. This is more so for larger diameter fasteners. Using equations 1 and 8 the tightening and prevailing torque correspond to 65% yield load for this fastener is approximately 40Nm and 32Nm respectively. For this case, no additional devices to prevent loosening are required.

As established in the previous sections by using as large as possible fastener, with metric (flank angle 30°) fine thread having a friction coefficient approximately larger than 0.15 tightened to at least 65% of the proof load will provide an effective vibration resistance joint for many common applications.

If the joint is subject to severe vibrations then a device that will increase the break-loose torque may be required. On the other hand if the design of the joint does not allow pre-tensioning of the bolt to at least 65% of the proof load then a locking device must be used if the joint is subject to vibration.

5.2 Lock nut or jam nut

As discussed earlier, use of any additional anti-loosening device is only necessary if either the joint cannot handle the required minimum pre-tension load or the tightening method does not assure the correct pre-load.

There are situations that the joint may not function if a large pre-load is applied. If this joint is subject to vibration an additional device to prevent loosening is required. There are a large number of anti-loosening devices available.⁴

Lock-nuts (use of full nut and a half nut; one on top of the other) are commonly used as a device for

preventing loosening. It is our experience that most of the practitioners incorrectly use the lock nuts. For example, it is common to use two identical nuts for this purpose. This method will hardly ever work as the both nuts will rotate together in a vibration environment hence loosening the joint. In this method, the interface between the two nuts will experience the pre-load and hence be safe from loosening one nut with respect to the other; however, as the clamp interface will not see a significant pre-load the joint could still loosen by turning both nuts together.

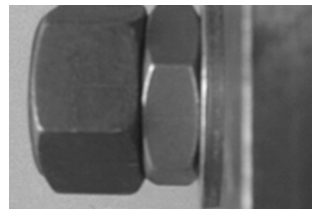


Figure 17: Correct use of lock nut.

Ideally two different nuts – one full nut and one half nut (Lock/Jam nut) – must be used in order to achieve a sound locking mechanism to prevent vibration loosening. The Lock/Jam nut must have a loose fitting thread and be made of a lower grade material. Here again we have seen that some practitioners use the full nut first with the half nut on top, tighten the first nut to achieve the desired tension (typically small) and then tighten the half nut to a greater tightness. This procedure is not correct and will not create a sufficient locking mechanism. In this case again, as the half nut cannot provide enough tensile force to create a jam, both nuts will rotate together if the tension in the joint is lost.

The correct method (Figure 17) is to use the half nut first and snug tighten it (typically small tension). At this point the bolt thread will be pulling the lock nut down and be bearing on the upper flank of the nut thread. The full nut is installed next. This nut will always bear on its upper thread flank while applying tension on the bolt. As this nut is further tightened, the downward force applied on the lock nut by the full nut will push it down, imparting more clamp load on the joint. This should be continued until the lower flank of the half nut bears on the bolt thread. It is important that the half nut is held while the full nut is tightened.

In this case, the two adjacent nuts are bearing on opposite thread flanks and the threads are deformed in opposite directions. Therefore, it will take a much larger torque to rotate them both together, hence providing a non-loosening nut even if the tension in the joint is lost. This device essentially achieves a very large effective friction coefficient between the bolt and the nut threads. Furthermore, the friction force does not totally depend on the pre-load on the joint. Instead, it depends on the reaction force

between the two nuts. The half nut being made of lower grade material with only few threads allows the threads to plastically deform hence substantially increasing the break-loose torque as well as prevailing torque. When loosening the joint, the half nut should be held and loosen the full nut first. Then unwind the nuts separately. Half nut may be re-used if permanent deformation has not occurred on the threads.

There are, however, a number of issues to keep in mind regarding this tightening method. Firstly, if the half nut is over tightened at the start there is a great possibility that a larger than desired pre-tension may end up in the bolted joint. Secondly, the standard torque tension relationships do not apply for this tightening process. In general, if you have to use a lock nut, it implies that the tension in the joint is not critical. If it is a critical joint a direct tension sensing bolt such as SMARTBOLT™ may be used.

5.3 Nyloc® nuts

These nuts have a captive polymer insert at the end of the thread (Figure 18). This insert is unthreaded and will be threaded during installation. Due to the elastic/plastic nature of the insert, additional frictional load is applied to the thread. This will increase the break-loose torque and therefore reduce the potential of loosening. These nuts provide a large prevailing torque. Hence are suitable for applications that do not require a large pre-load. These nuts may only be used once or twice in some applications as the prevailing torque of the polymer insert may be reduced below the design level once a thread is formed.



Figure 18: Nyloc® nut

These nuts are not suitable for joints that are subject to high temperatures. Typical torque tension relationships are not applicable to joints which use this type of nuts.

5.4 Nylon pellet inserts

A nylon or plastic insert is installed in a longitudinal slot along the bolt or nut thread. This insert will protrude out of the thread root diameter, hence interfering with the thread fit. Apparent increase in the thread friction will increase the break-loose and prevailing torques. This method is again not suitable for high temperature applications and typically good

for once only use. Typical torque tension relationships are not applicable to this type of bolted joints due to inherent variability in friction characteristics.

5.5 Deformed nuts

One end of the nut is deformed to form a non-circular hole (Figure 19). In the tightening process, the circular bolt shank has to work against the non-circular hole, hence storing additional elastic energy in the nut/bolt (thread) interface. This energy will act as a barrier for loosening. This is another way of increasing apparent thread friction. With multiple loosening and tightening the effectiveness of this feature will be reduced. Typical torque tension relationships are not applicable to this type of bolted joints.

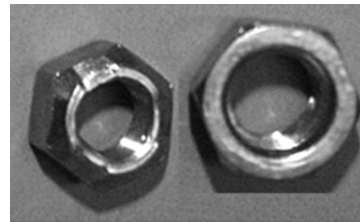


Figure 19: Deformed nuts

5.6 Deformed threads

By applying a secondary deformation to a part of the nut or bolt threads, apparent thread friction can be increased. This will in turn increase the break-loose torque and the prevailing torque. Typical torque tension relationships are not applicable to this type of bolts/nuts.

5.7 Interfering thread

By using a dissimilar thread profiles (typically with increased root radius) for the nut and the bolt an interfering thread can be achieved. Binding of the dissimilar threads prevents the nut from loosening. As common with other increased effective thread friction devices this type of nuts and bolts will not follow a regular torque tension relationship.

5.8 Tapered thread

This is a variation of interfering thread. In this case, the minor diameter of the nut is tapered towards the last few threads so that an interfering fit between the bolt and the nut is achieved. This will act as a thread locking device as well as a seal. BSPT is an example of this thread.

5.9 Thread locking adhesives

It is common practice to apply thread locking adhesives on the threads in order to prevent loosening (Figure 20). These adhesives will again increase the effective thread friction either forming mechanical interlocks or chemical bonding. In the case of chemical bonding the break-loose torque will be substantially increased however the prevailing torque may not increase as much. Typical torque tension relationships are not applicable to this type of bolts/nuts.

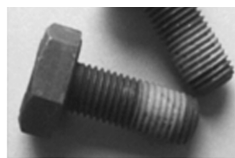


Figure 20: Thread locking adhesive

5.10 Locking washers

The function of this type of washer is to promote the retention of fasteners such as bolts, nuts and screws. This is typically achieved by increased friction between the fastener and the mating material through mechanical interlocking or interference. Increased bearing friction will provide characteristics similar to those shown in figures 9 and 10. The pre-tension is somewhat important in using this type of devices as they do not increase the prevailing torque. These washers also provide some spring take-up similar to spring washers but at a much lower magnitude. Locking washers are not suitable for joint which does not require pre-tension. Typical torque tension relationships are not applicable to this type of bolts/nuts as the under head/nut bearing friction is affected.

5.10.1 Tooth lock washers

Tooth lock washers may have internal teeth (Figure 21), external teeth (Figure 22) or both. These washers are primarily of two constructions: teeth twisted out of plane (type A) or edges of the teeth folded in opposite directions (type B).

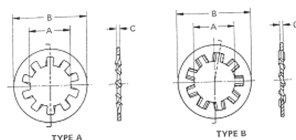


Figure 21: Tooth lock washer – internal

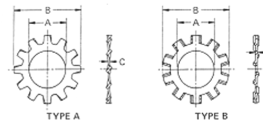


Figure 22: Tooth lock washer – external

These washers are not suitable for joints that are subject to severe vibration conditions.

5.10.2 Serrated washers and flanged nuts/bolts

Serration is made on washer, under head or under nut bearing surfaces (Figure 23). At installation, these serration will embed into the joint material substantially increasing the under head/nut friction hence increasing the break-loose torque. Damage tolerance of the joint surface is a consideration for the application of this method. Furthermore, these devices need a minimum remaining pre-tension in the joint to be effective. They do not provide any additional prevailing torque. Typical torque tension relationships are not applicable to this type of bolts/nuts.

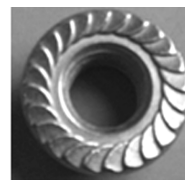


Figure 23: Serrated flanged nut

5.10.3 Ridge lock washers

A number of ramp segments are formed on the mating sides of the two washers. The ramp angle is larger than the helix angle of the thread. If the bolt/nut has to rotate in the loosening direction, the difference between the ramp angle and the helix angle will cause the joint to tighten, hence preventing further loosening. This device is only effective when there is pre-tension in the joint. They do not increase prevailing torque. As this device does not alter the friction characteristics of the joint (plain surfaces on washer/nut/head and washer/joint interfaces) torque tension relationships are valid when using this device.

A variation of ridge lock washers have ridges on either side of the mating washers (similar to serrated washers) so that they positively embed to the nut/head and the joint surface and prevent slipping. Damage tolerance of the joint surface is a consideration for the application of this method. Typical torque tension relationships are not applicable to this type of bolts/nuts.



5.11 Split-beam lock nut

This nut has usually six slots cut in one end where the thread diameter is slightly less than the standard thread (Figure 24). As the bolt enters this area the slotted portion of the nut act as beams deflecting away from the bolt applying additional frictional forces on the threads. If these deformations are limited to elastic range, this type of nut is useful for multiple tightening and loosening, however, thread wear may reduce the effectiveness of the locking mechanism. These devices provide increased prevailing torque as well as break-loose torque after tensioning. Typical torque tension relationships are not reliably applicable to this type of nuts.

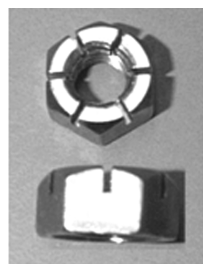


Figure 24: Split-beam lock nut

5.12 Castellated nuts with cotter pin

A castle nut typically has six diametrically aligned slots at the outer end of the nut (Figure 25). Once the correct tightening is achieved the nut is further rotated to align a slot with the single hole drilled through the thread of the bolt. A cotter pin is then inserted and bent at the end to prevent accidental removal. Mechanical interlock provided by the cotter pin will prevent the nut from rotating. This method is particularly good for applications where the tension required on the bolt is small. Torque tension relationship of the joint may not be affected with the use of this device. It may be necessary to slightly over tighten the joint in order to align a slot with the hole in the bolt when used in a tensioned joint. This device provides positive anti loosening properties and are suitable for critical applications.



Figure 25: Castellated nut

5.13 Nut cap and cotter pin

This is a variation of the castellated nut. Instead of cutting slots on the nut, a stamped sheet metal cap with slotted tabs will fit on the nut after tightening. A cotter pin is then inserted through the tabs and the hole in the bolt as in the previous case. This device is cheaper than the above device.

5.14 Lock wiring

This method is commonly applied when a group of bolts is fastened in to tapped holes. Once adequate tightening is achieved, a continuous wire is passed through holes drilled through the heads of the bolts. The principal is to arrange the lock wire in such a way that a loosening of one bolt will tighten the adjacent bolts. Once fed through all the bolts the wire will be crimped at either end. If this method is applied to nuts, corner drilling instead of through drilling should be used in order to prevent loss of strength. This is a more expensive method of avoiding vibration loosening.

As discussed, there are a large number of devices available for preventing vibration loosening. Typically, these devices are necessary only if the joint is not capable of carrying enough pre-load to prevent vibration loosening. If a minimum pre-load of $\approx 65\%$ Proof Load of the fastener can be assured it is highly unlikely that anti-loosening devices will be needed.

6 CONCLUSION

Based on the above analysis and discussion the following generalized rules of thumb may be established:

6.1 First order torque-tension analysis

- Both tightening torque and prevailing torque increase with increasing pre-load
- Keeping everything else the same, a smaller pitch fastener (fine threaded) will have a larger prevailing torque than a larger pitch fastener, hence performing better in a vibration environment
- To resist a given loosening torque, a larger diameter fastener is better
- A “buttress” thread will loosen easily compared to a metric thread
- If the magnitude of the loosening torque can be established for an application, it is possible to



calculate the minimum pre-load requirement to prevent vibration loosening

- When the pre-load is closer to the proof load of the fastener, associated local plastic deformations in the threads will dramatically increase break-loose torque.

6.2 Loosening mechanisms and analysis

- The mechanism of a vibratory conveyor resembles the mechanism of vibration loosening of a bolted joint
- Lateral vibration has a more prominent effect than the longitudinal vibration in loosening
- Loosening will increase with increasing frequency
- Loosening will increase with increasing amplitude
- The minimum amplitude and frequency required to start loosening will increase significantly with increasing pre-load
- Under very specific conditions, vibration tightening may occur
- Increasing thread friction will also increase the minimum vibration level required to loosen the joint
- Effect of pre-load is further accentuated with increasing thread friction
- Bearing friction is only effective when there is a pre-load. Increasing pre-load will accentuate the effect of bearing friction in preventing vibration loosening
- Finer threads will perform better in a vibration environment
- Fine threads have a high possibility of vibration tightening under certain conditions
- At lower pre-loads smaller thread flank angles are more susceptible to vibration loosening. This is why the "buttress" thread is not recommended for applications subject to vibration
- Pre-load is the most economical way of preventing vibration loosening
- A minimum pre-load of 65% of the proof load of the fastener should prevent vibration loosening under most common vibration environments
- When the inertial torque of the nut/bolt exceeds the prevailing torque loosening may occur

- For a right hand thread, an anti-clockwise acceleration of the shaft will cause the nut to tighten while a clockwise acceleration will cause the nut to loosen if adequate pre-load is not provided.
- To minimize vibration loosening of rotating equipment; for fastening to a clockwise rotating shaft a left handed screw thread and for fastening to an anti-clockwise rotating shaft a right-hand screw thread must be used.

6.3 Locking devices

- Locking device need to be used only if the joint cannot carry the required minimum pre-load to prevent vibration loosening.
- Using two similar nuts in a locking arrangement will not provide sufficient locking. Half nut (Lock/Jam nut) must be used to achieve a proper locking mechanism. The correct way is to install the half nut first to snug the joint and then tighten the joint using the full nut on top of the half nut. Load bearing on opposite flanks of the threads of the full nut and the half nut will ensure jamming of the thread.
- If a change in effective thread/bearing friction is used as the mechanism of thread locking, typical torque tension relationships applicable to standard bolts/nuts are no longer applicable. This is due to random nature of such locking mechanisms. Furthermore, the devices rely on increased bearing friction need remaining pre-tension to assure their effectiveness.
- Some locking methods are only suitable for once only tightening. In general, most locking devices reduce their effectiveness in the subsequent applications.
- Locking mechanisms with nylon or plastic inserts are not suitable for elevated temperature applications.
- Temperature conditions, tolerance to joint surface damage, cost, simplicity and reliability of installation, servicing requirements, etc, need to be considered when designing an appropriate thread locking mechanism.

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Saman joined Ajax Fasteners in 1997 as a pioneering member of the Ajax Technology Centre Pty Ltd a \$20M R&D Syndication on Advanced Fastening Systems Research. In 2004, in a management buy in (MBI), he bought into Global Engineered Fasteners (parent of Ajax Engineered Fasteners) business with four other executive members as the executive in charge of engineering and technology.

Saman became an internationally recognized fastener expert and innovator through his extensive research and publications in fastener and manufacturing engineering and by authoring twelve patented products and processes out of which four are (ONESHOT™, SMARTBOLT™, SMARTFIX™ and ONESIDE™) currently on the world market.

Saman is established as a Commercial R & D, Product Development and Commercialization Guru, due to his innovative idea to market process of focused research in bridging leading edge technologies to meet market expectations. He has extensive experience and success in collaborative research, team building, team leading, directing and training to achieve challenging goals both technically and financially successful.

Before joining Ajax he was the Manager and Principal engineer of the Building and Environment Technology Division of VIPAC Engineers & Scientists Ltd. During 8 years (1989-1997) at VIPAC Saman became an expert in Wind Engineering, Industrial Aerodynamics and Industrial Thermodynamics.

At UBC as an Assistant Professor (1987-89) Saman has conducted contract research for NASA, Lockheed Martin, Boeing and US Air-force.

His contributions in STOL wing configurations, side force alleviation devices for FA-11 aircraft; drag reduction devices for trucks & buses; natural ventilation systems, wind irrigation systems; artificial heart valves; sloshing dampers for slender and outer space structures; technologies for wind tunnel testing; solar impact analysis, environmental testing, energy labeling, bolting technology, manufacturing technology, forging simulations and mathematical modeling are being used world wide.