DYNAMIC ANALYSIS OF BOLTED JOINTS UNDER AXIAL AND TRANSVERSE LOADS

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ABSTRACT

DYNAMIC ANALYSIS OF BOLTED JOINTS UNDER AXIAL AND TRANSVERSE LOADS

Throughout their lifetime, fasteners are exposed to a large number of repeated vibrations which cause self-loosening. This situation causes security problem for bolted joints. In this study, novel self-loosening formulations are derived for bolted joints. By using the solution of the differential equation of a tensioned cantilever beam under the lateral tip load and bending moment, transverse displacement distribution along the bolt axis due to the transverse excitation is obtained. Therefore, radial slip amplitudes of the all engaged threads are found by using transverse displacement distribution along the bolt axis. Finally, loosening of bolted joint due to progressive circumferential displacement is discussed.

Keywords: Bolted joint, self-loosening

ÖZET

EKSENEL VE YANAL YÜKLER ALTINDAKİ CIVATALI BAĞLANTILARIN DİNAMİK ANALİZİ

Bağlantı elemanları kullanım ömürleri boyunca çok sayıda tekrarlı titreşime maruza kalırlar. Bu durum bağlantı elemanlarında kendi kendine gevşeme durumuna sebep olur. Bu çalışmada, cıvatalı bağlantıların kendi kendine gevşeme problemi için özgün bağıntılar türetilmiştir. Ucunda yanal kuvvet ve eğilme momenti olan çeki altındaki ankastere kirişin diferansiyel denkleminin çözümü ile yanal yerdeğiştirmeden dolayı cıvata eksenindeki yanal yerdeğiştirme dağılımı elde edilmiştir. Böylece, cıvata eksenindeki yanal yerdeğiştirme dağılımı kullanılarak birbirine geçmiş tüm vida dişlerindeki radyal kaymalar bulunmuştur. Son olarak, kademeli çevresel yerdeğiştirmeden dolayı cıvatalı bağlantının gevşemesi tartışılmıştır.

Anahtar Kelimeler: Cıvatalı bağlantı, kendi kendine gevşeme

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LIST OF SYMBOLS

A_p	Effective cross-sectional area of the clamped parts
A_s	Cross-sectional area of shank of bolt
A_t	Cross-sectional area of threaded part of bolt
b	Thread thickness
c_i	Damping factor
C	Joint stiffness factor
d	Major diameter
d_h	Bolted joint hole diameter
d_m	Minor diameter
d_p	Pitch diameter
d_r	Root diameter
d_1	Bearing diameter of bolt or nut
d_2	Bottom diameter of frustum
E_b	Young Modulus of bolt material
E_p	Young Modulus of clamped material
f	Friction force
F, F_i	Preload
h	Distance
h	Depth of thread
k_b	Bolt stiffness
k_i	Stiffness
k_p	Stiffness of clamped material
L	Numbers of threads
L_b	Bolt length
L_c	Clamp length
L_{lp}	Length of lower plate
L_s	Shank length of bolt
L_t	Free thread length of bolt
L_{up}	Length of upper plate
m	Mass
m	Working thread length between nut and bolt

M	Moment
M_n	Bending moment acting on bolt-nut threads
n	Number of threads
N	Reaction Force
p	Pitch
P	Preload
r_n	Bearing radius of bolt
r_t	Effective thread contact radius between nut and bolt
s	Dynamic excitation
s	Transverse displacement
T	Torque which applied to bolted joint
T_u	Screw torque
T_p	Lowering torque
α	Thread angle
β	Half thread angle
λ	Lead angle
μ	Friction coefficient
μ_b	Friction coefficient under head of the bolt
μ_t	Friction coefficient between nut and bolt threads
θ	Angle

CHAPTER 1

INTRODUCTION

1.1. Self-Loosening Problem

Bolted joints are used in all kinds of constructions to fasten structural members to each other. If the structure is under dynamic loads, generally, bolted joints are subjected to axial, transverse or both types of repeated loadings. The characteristics of the bolted joints and the repeated loads play a critical role on self-loosening. This is known as self-loosening problem.

In order to avoid that undesired situation, the reasons of self-loosening of bolted joints are identified by considering all possible physical characteristics.

1.2. Literature Survey

Regarding the thesis subject, the most critical literatures are presented below:

Goodier and Sweeney (1945) derived the equations of external torques required to loosen a nut in both increasing and decreasing bolt loads, therefore their study is related with self loosening of threaded fasteners under axial loads. Their test apparatus is shown in Figure 1.1.

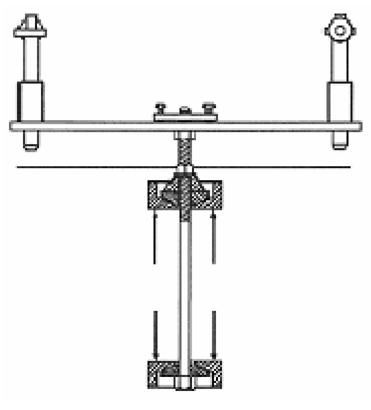

Figure 1.1 Goodier and Sweeney test machine
(Source: Bickford and Nassar, 1998)

Later Sauer et al (1950) also made experiments on the rotational loosening conditions based on the amplitude ratio defined as axial static load to axial dynamics load.

Junker (1969) demonstrated that loosening is very important when the joint is subjected to dynamic loads perpendicular to the thread axis. His test machine is shown in Figure 1.2.

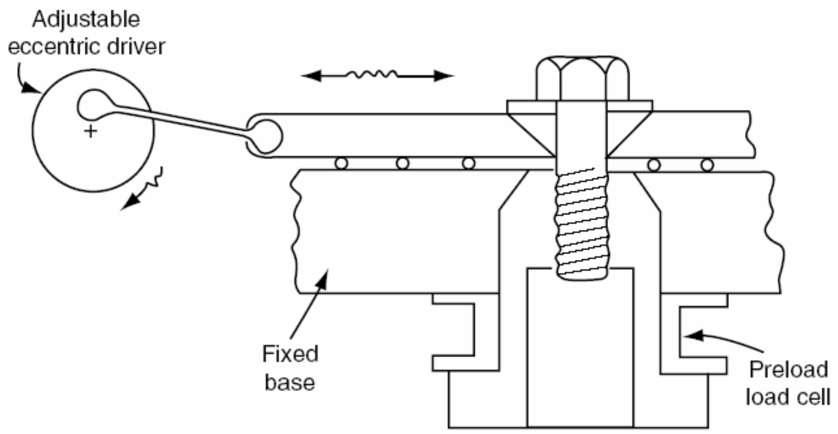

Figure 1.2 The Junker vibration test machine
(Source: Bickford, 2008)

After introducing the important studies above, it is better to present the other past studies by classifying them depending on the methods such as analytical or finite element methods.

Sakai (1978) analyzed theoretically self rotation, critical slippage between the clamped parts and loosening angle for the bolted joints under transverse load. He verified his theoretical studies by his experimental results.

Sakai (1979) found theoretically and experimentally that tightening of bolts over their yield points has an advantage over the tightening of bolts below their yield point for fatigue life.

Harnchoowong (1989) studied the effects of various parameters on nut loosening under axial loading conditions. He concluded that loosening resistance for bolted joints can be increased by decreasing the helix angle and/or increasing friction between the joint parts, bolt length, nut wall thickness, and nut height.

Ramey and Jenkins (1995) reported prediction of bolt loosening of bolted joints under dynamic tensile and shear forces. In their report, theory for bolt/nut interactions and vibrational loads on bolted joints are included. They used Taguchi methods to find the empirical equation for prediction of bolt loosening.

Hess and Sudhirkashyap (1996) examined the dynamics of threaded components under axial harmonic vibration. Their model is shown in Figures 1.3 and 1.4, respectively. It can be seen from Figure 1.3 that the real system with thread angle β is unwrapped. $\{k, c\}$, $\{k_2, c_2\}$, and $\{k_3, c_3\}$ are associated with thread interface, the dominant elastic mode of the screw, and the dominant elastic mode of the base, respectively. Also, m and m_2 are the masses of the screw and the effective mass of the base, respectively. The model predicts the intuitive twisting with load and the counter-intuitive twisting against load.

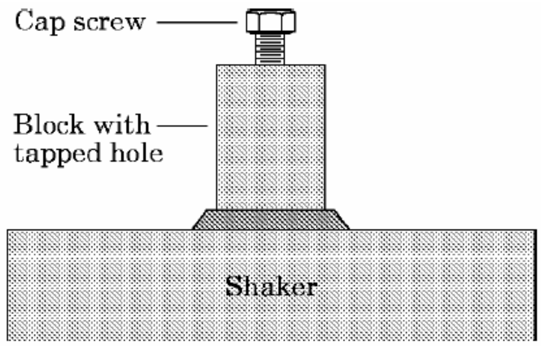

Figure 1.3 Cap screw in base with tapped hole apparatus
(Source: Hess and Sudhirkashyap, 1996)

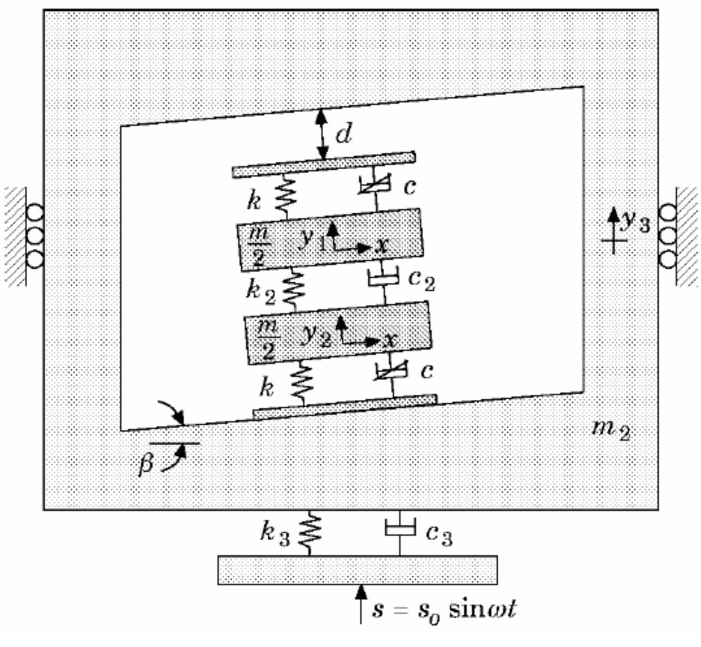

Figure 1.4 Dynamic model for axial loading
(Source: Hess and Sudhirkashyap, 1996)

Hess and Sudhirkashyap (1997) modified their previous analyses of threaded fasteners under vibration as the system shown in Figure 1.5. It can be seen from comparison of Figure 1.5 with Figure 1.3 that clamped mass and spring between clamped mass and base are added to the system. Elastic, damping and frictional characteristics in contact regions are shown in the model shown in Figure 1.6. They concluded that the system parameters can be adjusted in order to have no twist, loosening or tightening.

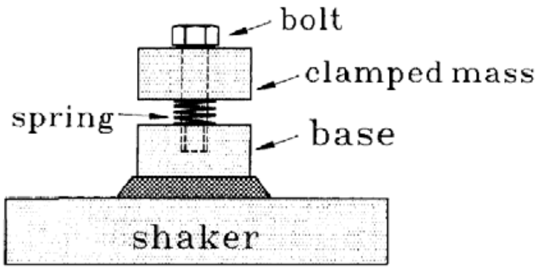

Figure 1.5 Single-bolt assembly apparatus
(Source: Hess and Sudhirkashyap, 1997)

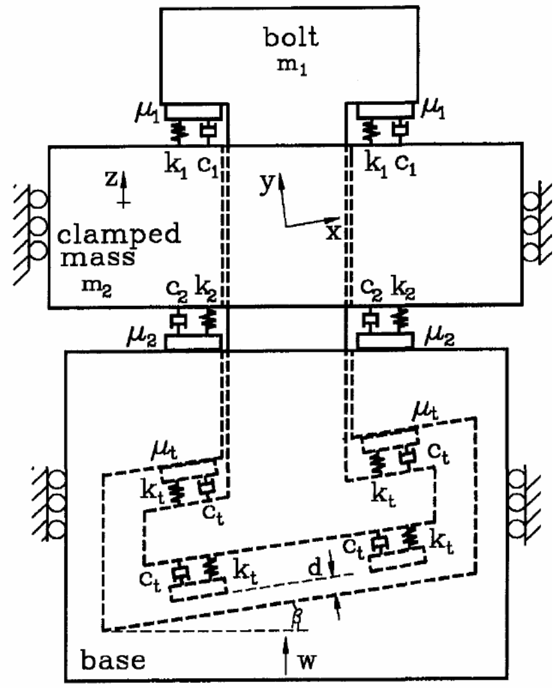

Figure 1.6. Dynamic model of single-bolt assembly
(Source: Hess and Sudhirkashyap, 1997)

Rashquinha and Hess (1997) extended the study of Hess and Sudhirkashyap (1997) to include the correct friction forces at the bolt and thread interfaces. Thus, they considered four cases:

1. both threads and head stick,
2. only threads stick,
3. only bolt head slicks,
4. both threads and head slip.

Due to the cases given above, the problem is highly nonlinear. They obtained that when the preload or the coefficient of friction increases, resistance of the joint to vibration induced loosening increases. Also, increasing the mass of the clamped component causes loosening and tightening at lower forcing levels.

Basava and Hess (1998) examined the effect of vibration level and initial pre-load on clamping force in a single-bolt assembly model under axial vibration by using the former studies of Hess and his co-authors. They found that changes in clamping force are generally transient and a steady value is reached over time.

Fernando (2005) utilized a bolt loosening mechanism similar to a mechanism of a vibratory conveyor transporting a particle. He presented effects of vibration excitation, thread and bearing frictions, thread pitch and thread flank angle on loosening.

Yokoyama (2009) theoretically formulated an analytical model for the mechanics of bolted joints subjected to transverse load. The following five factors are addressed:

1. bolt bending due to transverse force acting on the thread surface,
2. bolt bending due to thread surface reaction moment,
3. inclination of bolt head,
4. thread surface slip,
5. bearing surface slip.

He confirmed his formulations by Finite Element Method (FEM) results.

Due to the geometrical complexity of the thread part of the bolted joints, researchers mostly used finite element model to see the loosening conditions, thus, the following studies are mainly based on finite element models of bolted joints.

Pai and Hess (2002) studied on failure of threaded fasteners due to vibrational shear loads by using a three-dimensional finite element model shown in Figure 1.7. They made use of finite element model to find the slippage and prediction of different loosening mechanisms. They showed that loosening can occur at relatively low shear loads due to the process of localized slip.

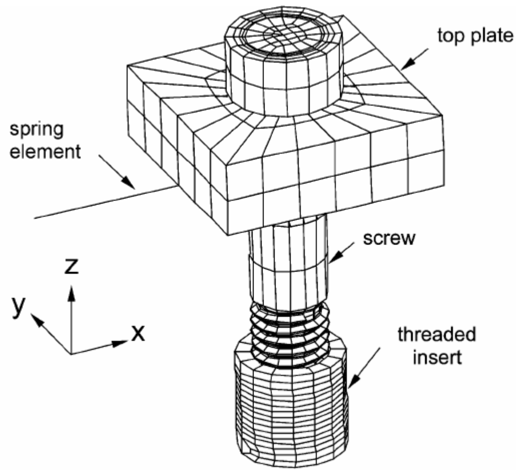

Figure 1.7. Joint model with typical finite element mesh
(Source: Pai and Hess, 2002)

Pai and Hess (2003) extended their past studies in order to see the influence of fastener placement on vibration-induced loosening by using the system shown in Figure 1.8. When the beam bends, the bolted joint is loaded by shear forces. They modeled the system by finite elements then estimated the slip according to distance *h*.

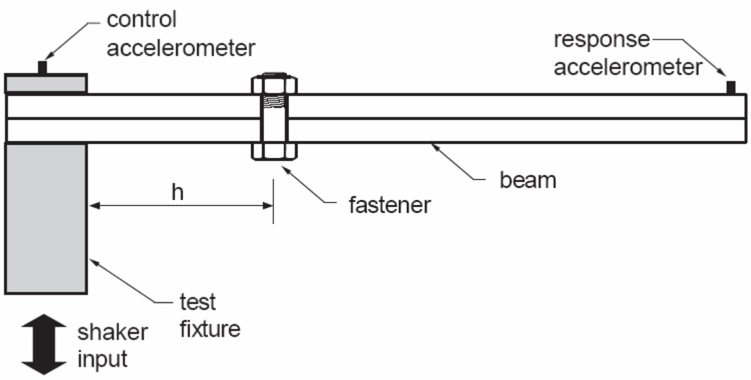

Figure 1.8. Test system to investigate the effect of fastener placement
(Source: Pai and Hess, 2002)

Jiang et al. (2003) investigated the early stage of self-loosening of bolted joints under transverse cyclic loading experimentally and numerically. They glued the nuts to bolts to test the benefit of this action. Their numerical model in ABAQUS® includes the plastic deformation near the roots of the engaged threads. They verified their finite element model experimentally.

Izumi et al. (2005) studied tightening and loosening processes of bolted joints by using the finite element model shown in Figure 1.9. They validated their model by doing some experiments. They found that loosening is initiated when complete thread slip occurrs prior to head slip.

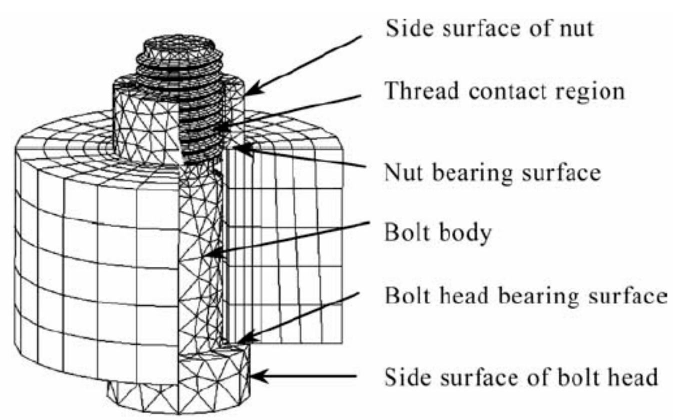

Figure 1.9. Finite element model
(Source: Izumi et al., 2005)

Nishimura et al. (2007) investigated critical relative slippage for bolted joints and performed experiments for this purpose. The relative slippages of bolted joint parts are illustrated in Figure 1.10.

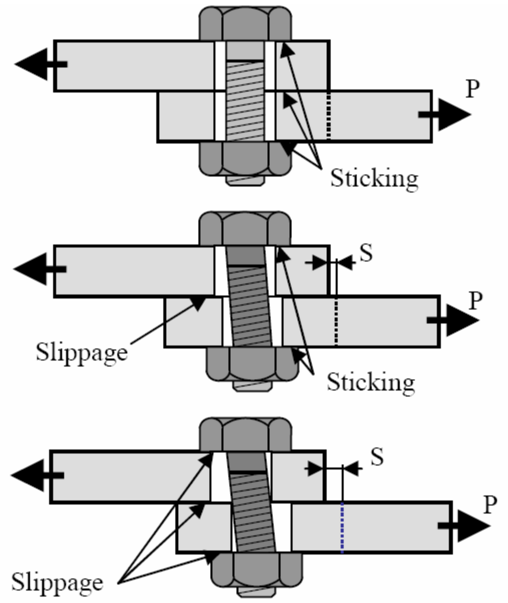

Figure 1.10. Sticking and slipping
(Source: Nishimura et al., 2007)

Zhang et al. (2007) simulated self-loosening of bolted joints by using a three-dimensional finite element model. They used ABAQUS® to model the mechanical surface interactions. They presented preload vs transverse displacements in a table.

Also, they presented simulations and experimental results for clamping force vs. number of cycles. Moreover, nut rotation vs. number of cycles is plotted for various transverse displacements. In addition to these, they simplified the two mating surfaces of the engaged threads to two inclined blocks representing bolt and nut.

Izumi et al. (2009) modeled double nut and spring washer usage in bolted joints in ANSYS® 10.0 as shown in Figure 1.11. They found that double nut is an effective method to inhibit self-loosening but spring washer has no positive effect to stop loosening.

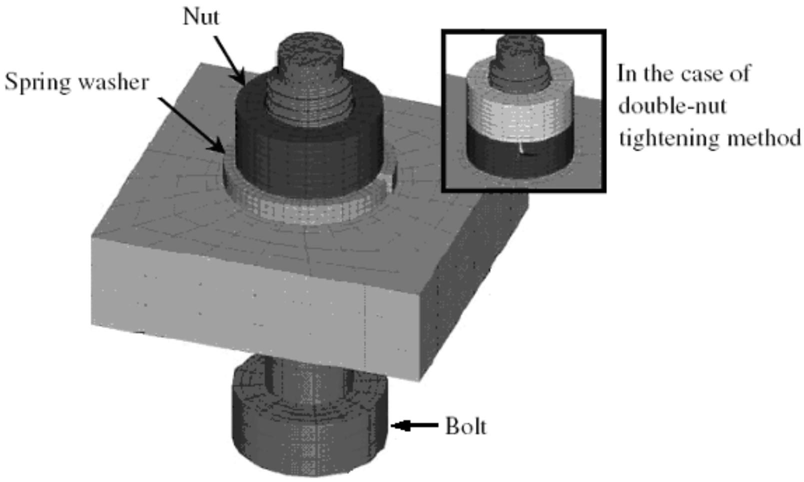

Figure 1.11. FEM model of double nut and spring washer
(Source: Izumi et al., 2009)

Hou and Liao (2014) studied the effect of increment step length, initial clamping force, amplitude of the shear load, thread tolerance, friction coefficients on the loosening process by FEM. They concluded that if the magnitude of dynamic load increases, the loosening will increase in the same way.

1.3. Scope of the Study

Considering the literature survey presented in the former section, it is observed that self-loosening problems of bolted joints are generally related with fluctuating external load, pre-load on the bolt and friction conditions of bolted joints. Therefore, the available formulations existing in the literature are very complex to understand and have various assumptions for self-loosening mechanisms. Also, they are not practical to use and apply easily.

In this thesis, by using the simple self-loosening mechanism rules and fundamental equations from mechanics of materials, new self-loosening formulations are derived for bolted joints.

CHAPTER 2

FUNDAMENTAL CONCEPTS

2.1. Definitions on Bolted Joints

Helical thread with single thread–right hand is illustrated in Figure 2.1 (left). Pitch p, lead L, and lead angle λ are shown in Figure 2.1. The relationship between lead and pitch is given by $L = np$, where n is the number of threads. It can be seen from Figure 2.1 that the left one is single thread–right hand, while the right one is double thread–left hand.

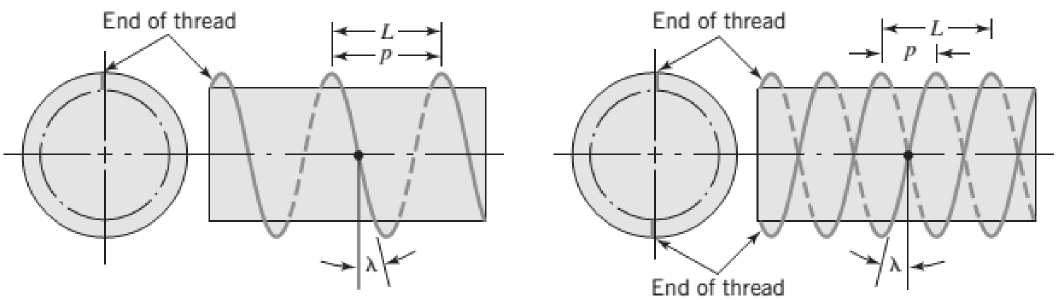

Figure 2.1. Helical thread of pitch p, lead L, and lead angle λ
(Source: Juvinall, 2011)

Unified and ISO thread geometry is shown in Figure 2.2. The major, minor/root and pitch diameters are denoted by d, d_r, and d_p, respectively. Also, h is the depth of thread and b is the thread thickness at the root. Half-thread angle is 30º.

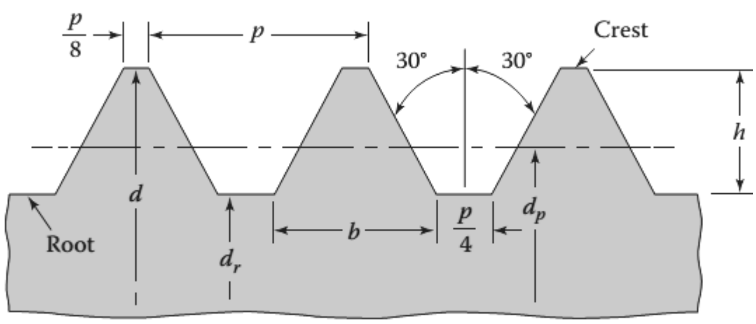

Figure 2.2. Unified and ISO thread geometry
(Source: Ugural, 2015)

Another example for thread form is square thread that is shown in Figure 2.3. Major diameter d, root diameter d_r and pitch diameter d_p are introduced in Figure 2.3.

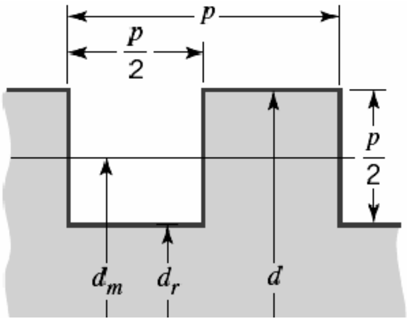

Figure 2.3. Square thread
(Source: Juvinall, 2011)

The important lengths of bolted joint are shown in Figure 2.4.

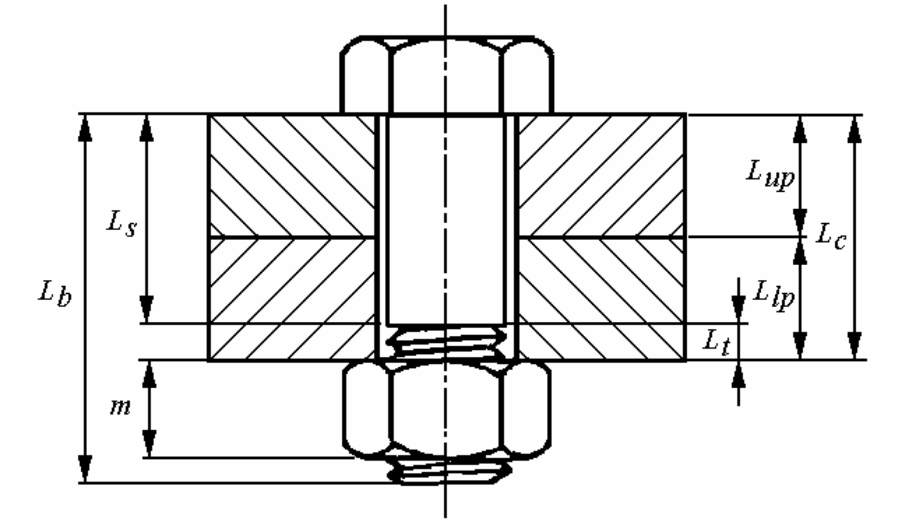

Figure 2.4. A simple bolted joint

It can be seen from Figure 2.4 that L_b, L_s, L_t, L_{up}, L_{lp} and L_c represent the length of bolt, shank of bolt, free thread length of bolt, length of upper plate, length of lower plate and the clamp length, respectively. Also, m is used to show the working thread length between nut and bolt.

Stiffness properties of the components shown in Figure 2.4 play a critical role. The most critical one is bolt stiffness k_b. It can be found by

$$\frac{1}{k_b} = \frac{L_s}{A_s E_b} + \frac{L_t}{A_t E_b} \tag{2.1}$$

where A_s, A_t, and E_b are cross-sectional area of shank of bolt, cross-sectional area of threaded part of bolt, and Young modulus of bolt material. Another stiffness property of the bolted joint is due to clamped materials. Referring to Figure 2.5, the effective cross-sectional area of the clamped parts, A_p, can be found as (Ugural, 2015)

$$A_p = \frac{\pi}{4}\left[\left(\frac{d_1 + d_2}{2}\right)^2 - d_h^2\right] \tag{2.2}$$

where $d_1 = 1.5d$ and $d_2 = d_1 + L_c \tan 30°$. Therefore, stiffness of the clamped parts can be found by using the following simple approach

$$k_p = A_p E_p / L_c \tag{2.3}$$

where, E_p is the Young modulus of the clamped part materials. On the other hand, for standard hexagon-headed bolts, stiffness of the clamped parts k_p is found by using the following expression (Ugural, 2015)

$$k_p = 0.29\pi E_p d / \ln\left(5\frac{0.58L_c + 0.5d}{0.58L_c + 2.5d}\right) \tag{2.4}$$

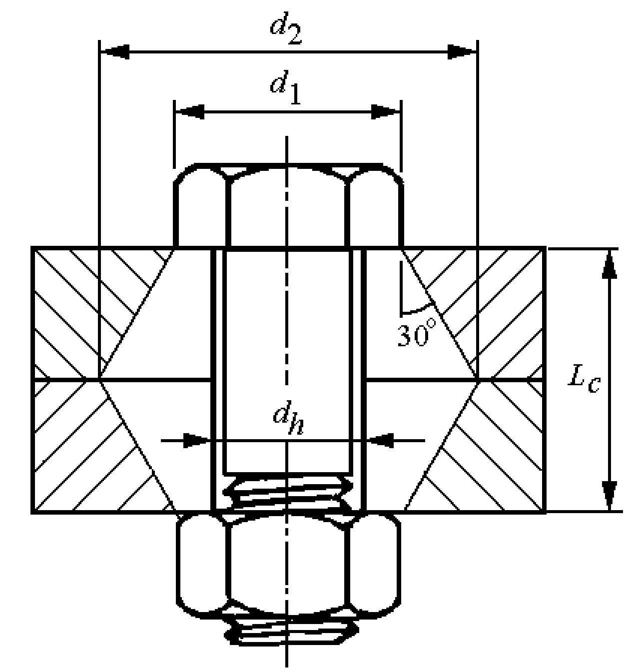

Figure 2.5. Conical effective clamped volume

2.2. Mechanics of Bolted Joints

A square-threaded bolt-nut with single thread is loaded by the axial compressive force P as shown in Figure 2.6.

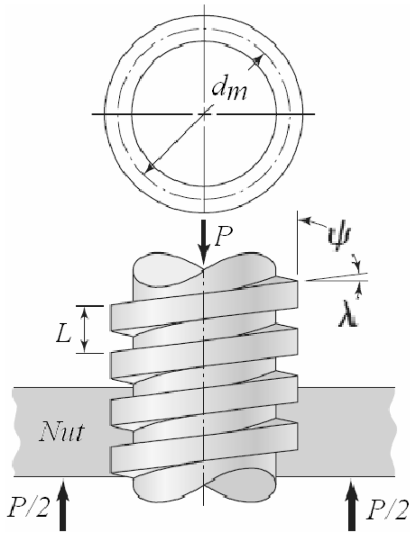

Figure 2.6. Axially loaded bolt-nut joint
(Source: Budynas and Nisbett, 2011)

If one revolution of helix is unwrapped, the inclined plane shown in Figure 2.7 is obtained. Nut is represented by a block on inclined plane in the figures. The lead angle λ plays a critical role in the equilibrium condition for the nut under the axial load. Two possible cases: lifting and lowering the axial load are considered to draw the free-body diagrams shown in Figure 2.7.

By using the equilibrium conditions, several equations available in machine elements textbook (Norton, 2006) can be derived by using Figure 2.7.

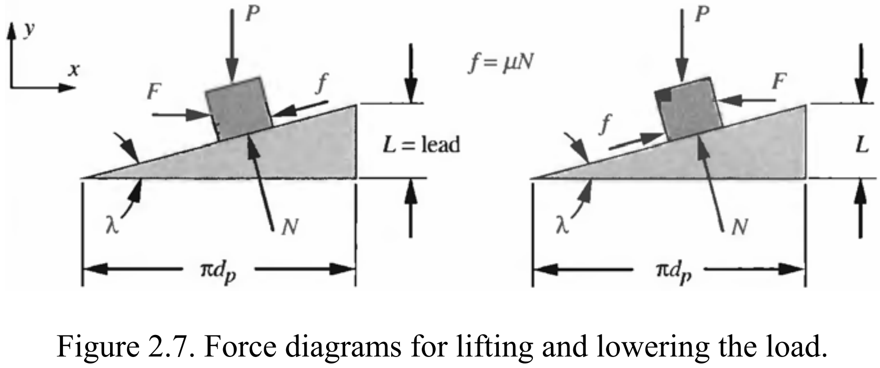

Figure 2.7. Force diagrams for lifting and lowering the load.
(Source: Norton, 2006)

The inclination angle of the plane, λ, in Figure 2.7 can be found as

$$\tan\lambda = L / \pi\, d_p \tag{2.5}$$

For the load lifting case shown in left side of the Figure 2.7, the following equations can be found from the equilibrium equations,

$$N = \frac{P}{(\cos\lambda - \mu\sin\lambda)} \tag{2.6}$$

$$F = P\frac{(\mu\cos\lambda + \sin\lambda)}{(\cos\lambda - \mu\sin\lambda)} \tag{2.7}$$

where μ is the friction coefficient on the contact surface. The screw torque required to lift the axial load P is expressed as

$$T_u = F(d_p / 2) \tag{2.8}$$

or by using Equation (2.7) in Equation (2.8)

$$T_u = \frac{Pd_p}{2}\frac{(\mu\cos\lambda + \sin\lambda)}{(\cos\lambda - \mu\sin\lambda)} \tag{2.9}$$

For the case of lowering the load shown in the right side of Figure 2.7, the screw torque required to lower the axial load P is found as

$$T_d = \frac{Pd_p}{2}\frac{(\mu\cos\lambda - \sin\lambda)}{(\cos\lambda + \mu\sin\lambda)} \tag{2.10}$$

Self-locking condition can be found by considering the lower limit of the lowering torque which is $T_d \geq 0$. Therefore, it the following relation can be written:

$$\mu \geq \tan\lambda \tag{2.11}$$

Motosh (1976) proposed an equation called torque T-tension P formula given by

$$T = P\left(\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos\beta} + \mu_n r_n\right) \tag{2.12}$$

where p, μ_t, μ_b, r_t and r_n are pitch, friction coefficient between nut and bolt threads, friction coefficient under head of the bolt, effective thread contact radius between nut and bolt, and bearing radius of bolt, respectively. β is the half-thread angle. In Equation (2.12), the first, second and third terms are due to bolt stretch component, frictional restraint between nut and bolt threads, and frictional restraint between the face of the nut and the washer or joint, respectively.

When applying input torque to the bolted joint, there are three reaction mechanisms. Approximately, torque input is consumed by nut friction torque or under head of the bolt friction torque, thread friction torque and part stretching torque as %50, %40, and %10, respectively (Bickford, 2015).

In order to find the internal forces of bolt and clamped material under the external tensional force along the bolt axis, the bolted joint shown in Figure 2.8 is considered.

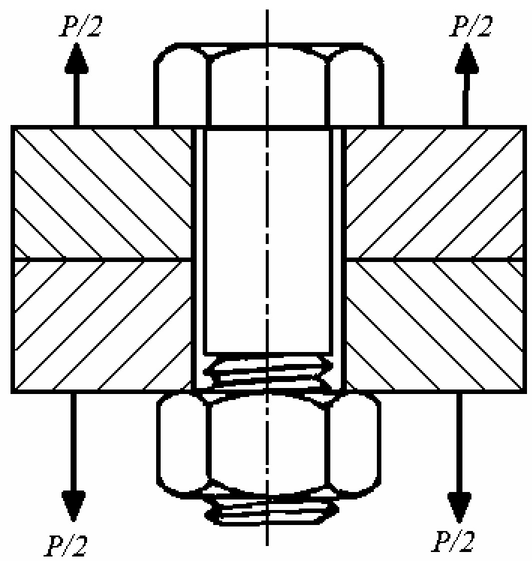

Figure 2.8. A bolted connection under axial force P

If F_b and F_p are used to represent the bolt and plate forces along the bolt axis due to external load P, respectively, force equilibrium along the bolt axis gives

$$P = F_b + F_p \tag{2.13}$$

Displacements of bolt and plate are written by using their stiffness properties as follows

$$\delta_b = F_b / k_b \tag{2.14}$$

$$\delta_p = F_p / k_p \tag{2.15}$$

On the other hand, compatibility condition requires the same displacement, i.e.

$$\delta_b = \delta_p \tag{2.16}$$

By using Equations (2.13) and (2.16) along with Equations (2.14) and (2.15), the bolt and plate forces along the bolt axis due to external load P can be found in the following forms:

$$F_b = CP \tag{2.17}$$

where C is known as joint stiffness factor and expressed as

$$C = \frac{k_b}{k_b + k_p} \tag{2.18}$$

and

$$F_p = (1 - C)P \tag{2.19}$$

If the bolt in Figure 2.8 has a pre-tension force F_i, Equation (2.17) and (2.19) become

$$F_b = CP + F_i \tag{2.20}$$

$$F_p = (1 - C)P - F_i \tag{2.21}$$

Graphical representation of Equations (2.20) and (2.21) are shown in Figure 2.9.

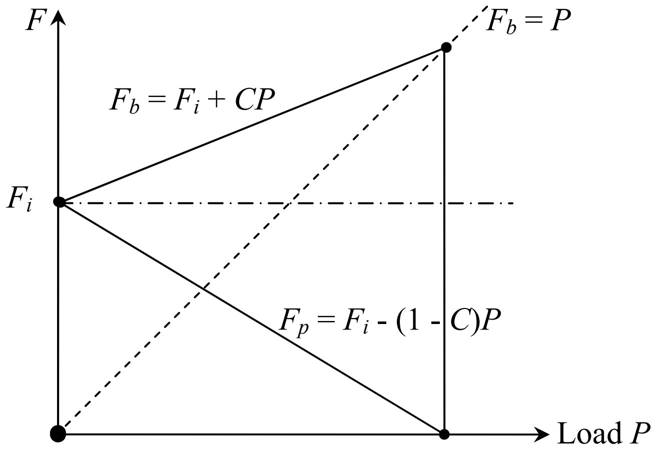

Figure 2.9. A bolted connection under axial force P
(Source: Ugural, 2015)

On the other hand, elastic deformations of bolt and clamped material due to the external force applied to bolted joint as shown in Figure 2.8 are needed. Assuming $k_p > k_b$ and using Equations (2.14) and (2.15), the graph shown in Figure 2.10 can be plotted.

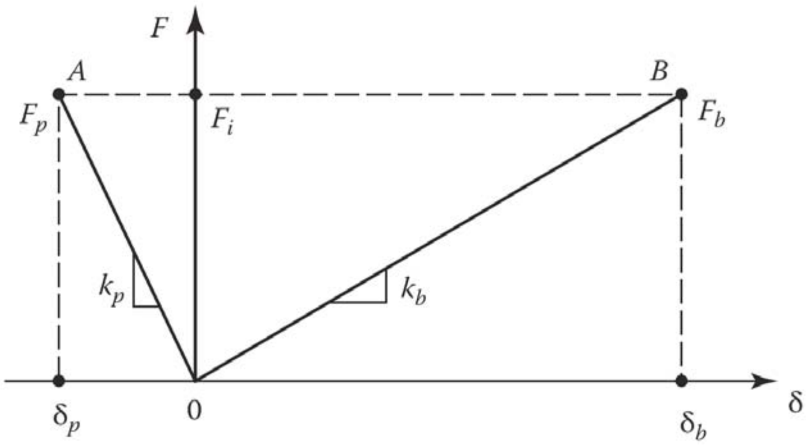

Figure 2.10. Displacements of bolt and clamped material due to pre-load F_i
(Source: Ugural, 2015)

Force distribution in threads of bolt-nut joints are critical concepts for determination of proper dimensions of the components. To illustrate this issue, Figure 2.8 is modified to show the bolt-nut threads in the region of nut, thus Figure 2.11 is obtained.

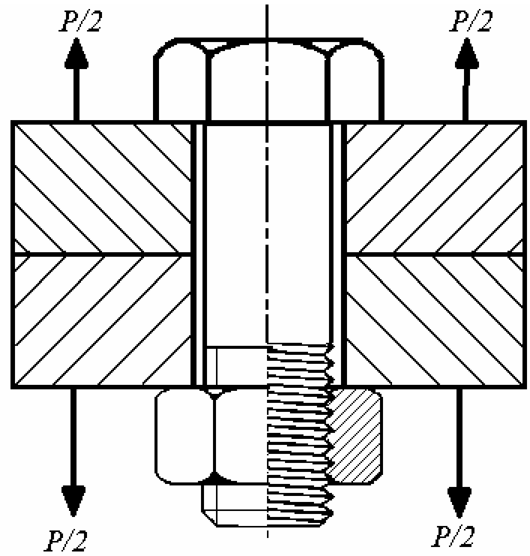

Figure 2.11. A bolted joint with local section under axial force *P*

Numerous studies are available for load distribution on bolt-nut threads. Kenny and Patterson (1989) presented a review on load and stress distribution in the threads of fasteners. They reported that the shape of the thread load distribution is generally accepted to be parabolic. They also, presented several studies based on analytical, experimental (for example by photoelastic material), and finite element methods. However, the most practical study is given by Trebuna et al. (2013). Their plot on this topic is presented in Figure 2.12. It can be seen from Figure 2.12 that load distribution on bolt-nut threads is in parabolic form. Moreover, first and second effective threads are loaded about 55% of the total load.

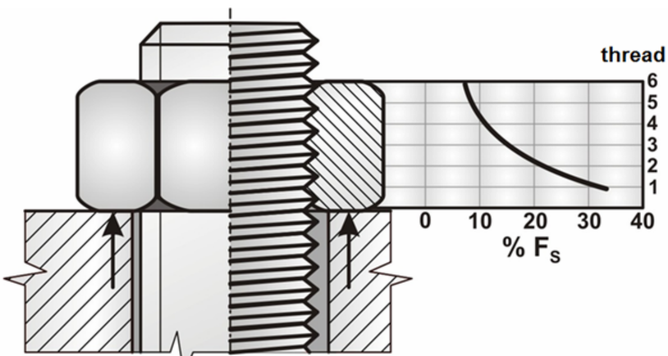

Figure 2.12. Force distribution in thread of nut
(Source: Trebuna et al., 2013)

Neglecting the nonlinear variation of load distribution on bolt-nut threads presented above, the reaction moment M_n acting on the thread in the bolted joint shown in Figure 2.13 is approximately derived by Yamamoto and Kasei (1977) as

$$M_n = \frac{m}{4} \frac{\mu_s F_i}{\cos^2 \beta} \tag{2.22}$$

where m is the nut height and μ_s is the friction coefficient of the thread surface.

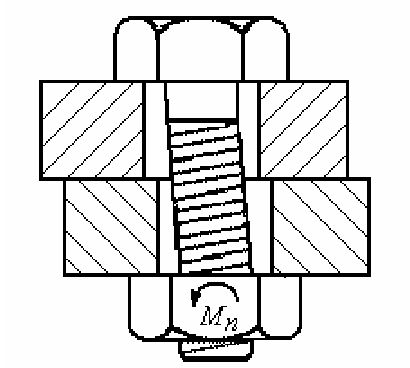

Figure 2.13. A nut under bending moment M_n

2.3. Loosening Mechanisms

Goodier and Sweeney (1945) investigated the loosening mechanism of a bolted joint under axial load. In this type of loading, the bolt is in tension and the nut is in compression. When a vibratory load is applied, the tension of bolt increases and decreases successively. During dynamic loading, the bolt tends to contract radially and on the other hand, nut tends to expand radially due to the Poisson effect. Therefore, the repeated actions cause the reduction of friction forces between bolt and nut contact areas.

Self loosening mechanism under transverse vibration effects is tested and discussed by Junker (1969, 1972). His theory is based on the effects of friction on two contacting solid bodies. He stated that "as soon as the friction force between two solid bodies is overcome by an external force working in one direction, an additional movement in any other direction can be caused by the action of forces that can be essentially smaller than the friction force." He proved his theory by a test involving a solid body on an inclined surface as shown in Figure 2.14. When the force is applied in *s* direction to the block resting on inclined plane, it moves not only in the direction of force but also downward direction although there is no external force in downward direction.

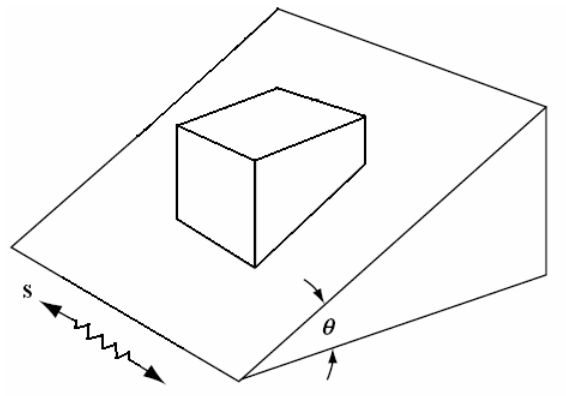

Figure 2.14. Junker's self-loosening test
(Source: Junker, 1969, 1972)

Zadoks and Yu (1997) discussed this subject considering the block with a mass *m*, shown in Figure 2.15, having a motion in direction 1 due to the force infinitesimally larger than the value of the frictional force. Next, a force in direction 2 is applied to the block, so the block moves with $a_2 = F_2/m$ in direction 2 addition to the direction 1, namely, it moves in direction 3. It should be stated that the force of friction is zero in the perpendicular direction to the direction of motion. Now, direction 4 is free from frictional force. This means that, if a force in direction 4 is applied, the block additionally accelerates in direction 4 with $a_4 = F_4/m$.

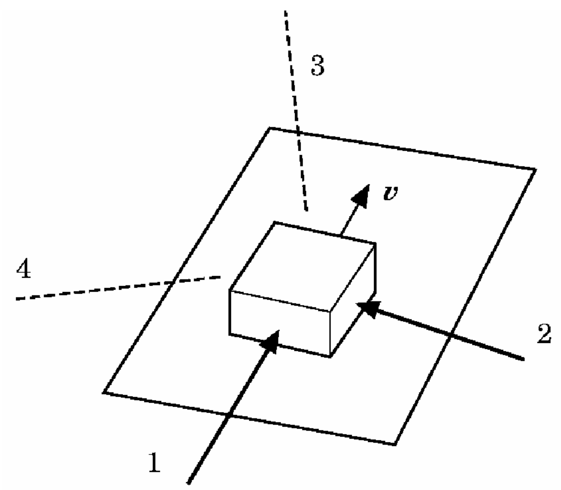

Figure 2.15. A sliding block on a plane
(Source: Zadoks and Yu, 1997)

Junker (1969) proposed the most well-known theory on self-loosening. The theory is confirmed by experimental studies by using “Junker Test Machine”.

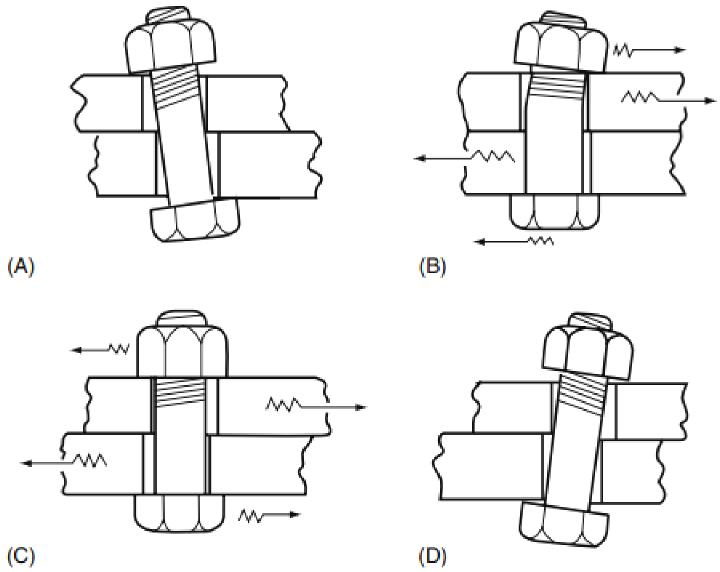

Figure 2.16: Self-Loosening process
(Source: Junker, 1969)

According to the theory, self-loosening in bolted joints occurs in four stages as shown in Figure 2.16:

A: Dynamic loads cause movement of threads and beginning of the separation between threads,

B: If the dynamic loads continue, the bolt will bend in reverse direction,

C: If loads still persist to proceed, nut moves in the reverse direction but loosening does not occur yet,

D: Last stage of preload loss and separation of threads, the process repeats over and over again until bolted joint is loosened.

Junker (1969) explains reasons to cause self-loosening assuming that there is no locking material and prevailing torque presence:

- Preload in the bolt causes to attempt nut’s rotation and this situation causes self-loosening.
- If dynamic loads are severe for bolted joints, thread mechanisms between the nut and the bolt and under head of the bolt begins to slip.
- If critical slip value is reached in the bolted joint, nut commence to rotate in the reverse direction of tightening.

- There is no self-loosening until friction mechanisms ceased between the nut and the bolt.
- The energy missing during each cycle also depends on the magnitude of the net ‘‘off-torque’’ on the nut during critical slip.
- The loosening will increase in each cycle before slip ends.

Ramey and Jenkins (1995) reported the following primary parameters for self-loosening mechanism:

1. Bolt size (diameter)
2. Lubrication on bolt
3. Hole tolerance
4. Initial preload
5. Locking device
6. Grip length
7. Thread pitch
8. Lubrication between mating materials
9. Class of fit
10. Joint configuration
11. Mass of configuration

Moreover, they pointed out that i) vibration direction and ii) magnitude of vibration also effect loosening.

Loosening sequences has two steps: Initially, a fastener subjected to excitation cycles loses the preload slowly. After reduction of sufficient preload, the nut loses rapidly as shown in Figure 2.17.

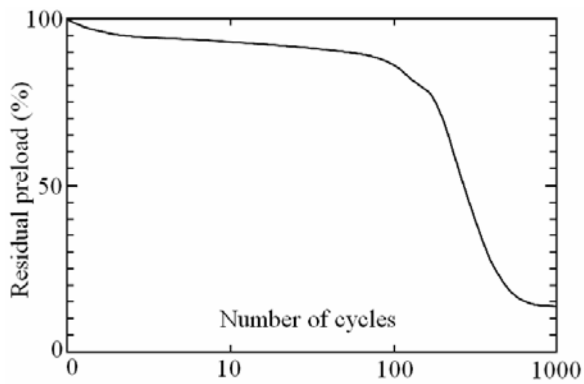

Figure 2.17: Loosening sequences
(Source: Junker, 1969)

2.4. Axially Loaded Bolted Joints

Let us consider a turnbuckle shown in Figure 2.18. The bolted joints in this system are loaded dynamically only in axial direction due to the usage of it.

Figure 2.18. A turnbuckle

2.5. Transversely Loaded Bolted Joints

Dynamic loading of bolted joints in transverse direction is possible in two different fashions. The bolted joint shown in Figure 2.19 illustrates the direct shear loading. On the other hand, Figure 2.20 illustrates the direct shear loading.

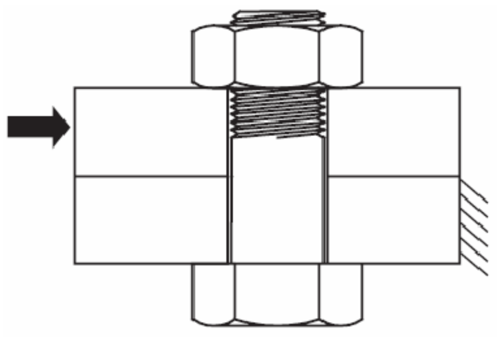

Figure 2.19. Direct shear loading
(Source: Pai and Hess, 2002)

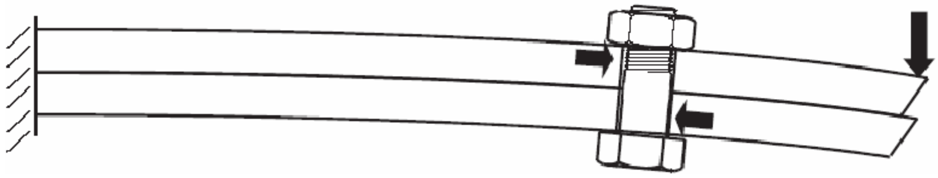

Figure 2.20. Bending induced shear loading
(Source: Pai and Hess, 2002)

2.6. Axially and Transversely Loaded Bolted Joints

There are numerous examples for dynamic loading of bolted joints under dynamic loading in both axial and transverse directions. The first example is bolted pressure vessel flange shown in Figure 2.21. In this example, bolts are under axial load under direct effect of inner pressure. Additionally, flange deflects in outward direction due to the inner pressure and then contracts in radial direction. Therefore, bolts undergo shear loadings. If the pressure is fluctuating, bolted joints are loaded dynamically.

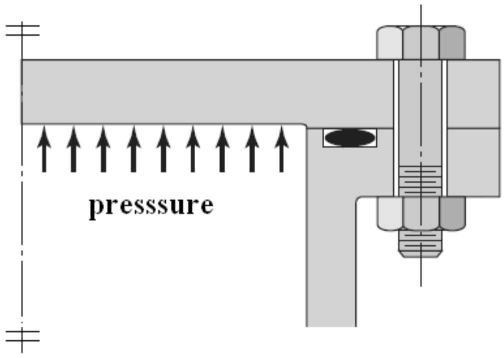

Figure 2.21. Bolted pressure vessel flange

Second example is beam to beam bolted connection shown in Figure 2.22. In this example, bolts on wide beam are subjected to shear and axial loadings due to the load acting on the narrow beam vertically.

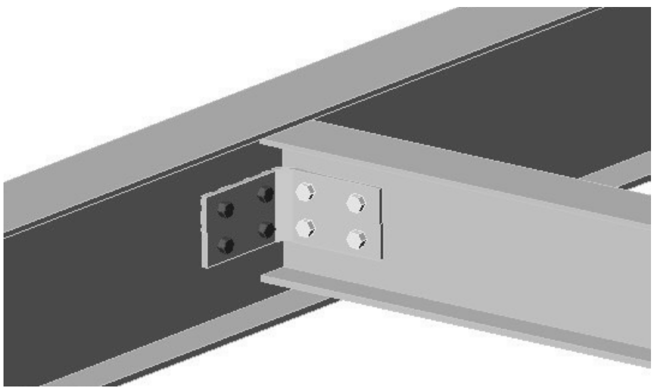

Figure 2.22. Beam to beam bolted connection

Last example is bolted connections in suspension system shown in Figure 2.23. Here, bolts are subjected to dynamic shear and axial loadings due to motion of the vehicle.

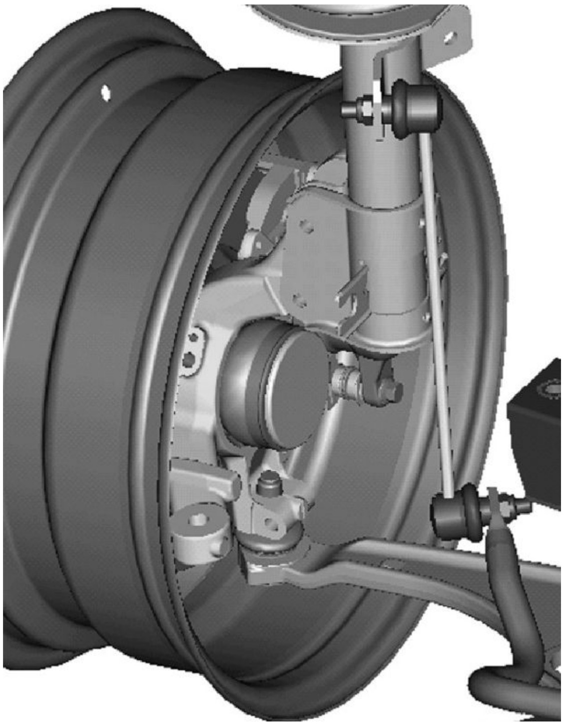

Figure 2.23. Bolt joints in suspension system
(Source: Genta and Morello, 2009)

CHAPTER 3

A NEW MODEL FOR SELF-LOOSENING

3.1. Introduction

Definitions and mechanics of bolted joints are presented in Chapter 2. In the context of mechanics of bolted joints, torque-axial load relationships, torque-tension equation, stiffness of bolted joint, load distribution on bolted joint, deformations of bolted joints, load distribution on bolt-nut threads, and finally moment acting on bolt-nut threads are given. Using the Junker theory, loosening mechanisms and sequences are explained by illustrations. Then, practical applications of axially, transversely, and axially-transversely loaded bolted joints are given.

In this chapter, the study on self-loosening mechanism by using finite element method and experiments presented by Zhang et al. (2007) are considered. The aim in this chapter is to reproduce their results with an analytical model, which is much simpler compared to the finite element model. Based on the geometrical model and the transverse excitation of the upper plate detailed by Zhang et al. (2007), the following analyses are studied analytically:

1. transverse displacement distribution along the bolt,
2. radial slip of thread regions of bolt,
3. nut rotation,
4. reduction of preload.

3.2. Geometrical Model of Bolted Joint and its Properties

In order to analyze and discuss the bolt-nut interaction, geometrical model used by Zhang et al. (2007) is redrawn and shown in Figure 3.1. All the geometrical and numerical data presented in this section are taken from Zhang et al. (2007). Top plate and upper cast iron plate are fixed to each other. Also, fixed plate is glued to lower cast iron plate. Therefore, just cast iron plate contact surfaces can move with respect to each other during the transverse excitation of the joint of bolt-nut.

Critical dimensions in this type of analysis are due to the bolt geometry, length of all clamped materials, length of the load cell, and the hole diameter of the plate.

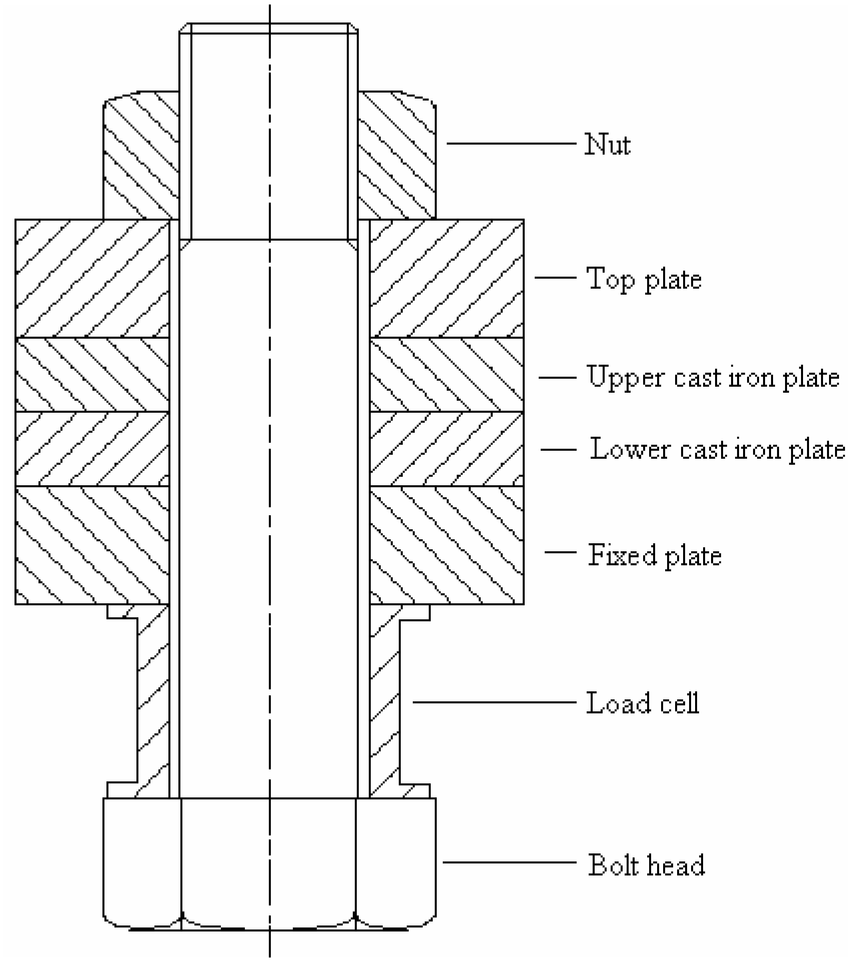

Figure 3.1. Bolted joint
(Source: Zhang et al., 2007)

As a case study, the ISO standard for M12x1.75 metric screw thread and nut with tolerance 6H/6g are used for the bolted joint. Hole diameter of the plates and load cell is 12.7 mm. The length of the load cell is 15 mm. The length of fixed plate with cast iron plate is 13 mm. Also, top plate with cast iron plate is 13 mm. Therefore effective clamp length is 41 mm. The elastic modulus for steels and cast iron are 206.8 GPa and 103.4 GPa, respectively. Poisson's ratio for both of them is taken as 0.28.

The coefficient of friction between the threads of the bolt and the nut is 0.09 and the nut-bearing friction coefficient is 0.12. The pre-load of the bolted joint is taken as 25 kN. The amplitude of transverse displacement is chosen as 0.45 mm.

3.3. Analytical Model

Figure 3.1 is modified to show the deformed configuration under transverse displacement δ. For this purpose, top plate with upper cast iron plate, the bolt and the nut is moved in transverse direction while the others are fixed as shown in Figure 3.2.

The following assumptions are made:

1. The slope of the deformed axis of the bolt head is not zero.
2. The nut does not have a relative displacement with respect to the upper plate.
3. The slope of the deformed axis of the bolt at bolt-nut region is limited due to the thread tolerances.

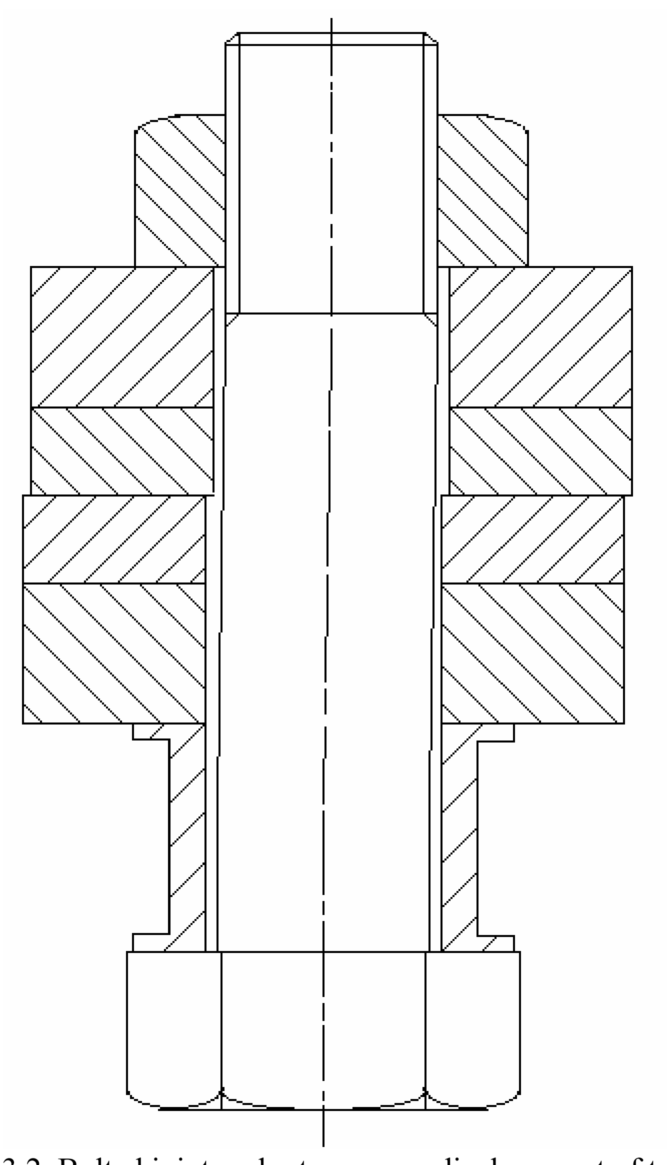

Figure 3.2. Bolted joint under transverse displacement of top plate

Transverse displacement distribution along the bolt axis $v(x)$ due to the excitation of the top plate (δ) can be found by using the differential equation given by Oden and Ripperger (1981) as

$$\frac{\partial^2 v(x)}{\partial x^2} - \frac{N}{EI(x)} v(x) = \frac{M_l(x)}{EI(x)} \tag{3.1}$$

where EI(x) is bending stiffness of the bolt, N is axial tension force and $M_l(x)$ is bending moment except the effect of axial force N at the distance x from the bolt head. For the cantilever beam of length L and loaded at the free end by transverse load P, tension load N, and bending moment M_n, internal bending moment is $M_l(x) = M_n + P\,(x - L)$. For this beam under tension, particular solution of Equation (3.1) can be found as

$$v_P(x) = \frac{P}{N}(L - x) - \frac{M_n}{N} \tag{3.2}$$

On the other hand, complementary solution can also be found as

$$v_c(x) = A\cosh(kx) + B\sinh(kx) \tag{3.3}$$

where k is defined by

$$k = \sqrt{\frac{N}{EI(x)}} \tag{3.4}$$

Thus, the general solution $v(x)$ is

$$v(x) = v_c(x) + v_p(x) \tag{3.5}$$

or

$$v(x) = A\cosh(kx) + B\sinh(kx) + \frac{P}{N}(L - x) - \frac{M_n}{N} \tag{3.6}$$

where A and B are constants depending on boundary conditions: $v(0) = 0$ and $v'(L_c) = \theta_c$. They are found as follows:

$$A = \frac{M_n - PL}{N} \tag{3.7}$$

$$B = \frac{\theta_c - A\,k\sinh(kL_c) + P/N}{k\cosh(kL_c)} \tag{3.8}$$

After subtituting A and B into Equation (3.6), it yields

$$\begin{aligned} v(x) = &\left(\frac{M_n - PL}{N}\right)\cosh(kx) + \left(\frac{P}{kN}\right)\sinh(kx) \\ &+ \frac{P}{N}(L - x) - \frac{M_n}{N} \end{aligned} \tag{3.9}$$

Radial slip of thread regions of the bolt can be obtained by using Equation (3.9) and the slope limitation of the deformed axis of the bolt at bolt-nut region due to the thread tolerances.

Analytical model of the microslip mechanism provided by Zhang et al. (2007) is shown in Figure 3.3.

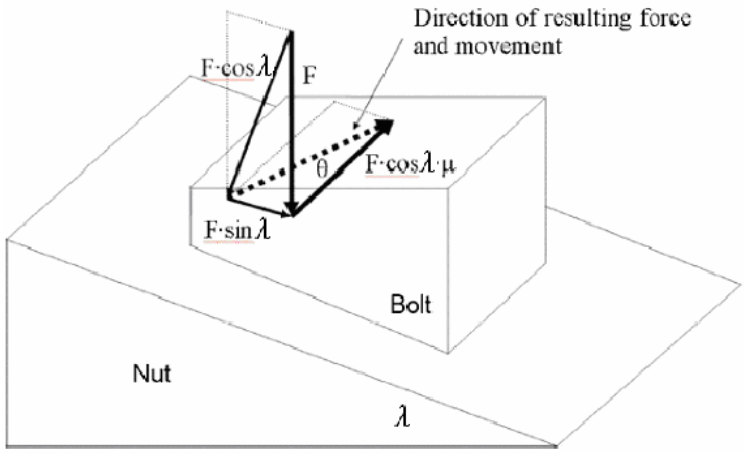

Figure 3.3. Analytical model for microslip mechanism
(Source: Zhang et al., 2007)

Microslip mechanism is explained as follows: Axial force F in bolt is resolved into two components as $F \cdot \sin\lambda$ which is parallel to inclined surface and $F \cdot \cos\lambda$ that is perpendicular to the inclined surface. λ is the inclination angle of the bolt-nut contact surface. Therefore, the friction force due to friction coefficient μ against to the action perpendicular to the inclined surface is $\mu \cdot F \cdot \cos\lambda$. Under the forces shown in bolt-nut contact surface, bolt moves in the resulting force shown by the dotted line. The angle θ is calculated as follows:

$$\tan\theta = \frac{F\sin\lambda}{\mu F\cos\lambda} = \frac{\tan\lambda}{\mu} \tag{3.10}$$

In similar fashion, when the displacement s is applied in the direction perpendicular to the ramp, the downward displacement q should be

$$q = s \cdot \tan\theta = s\frac{\tan\lambda}{\mu} \tag{3.11}$$

In order to imagine the radial and circumferential displacements in bolt-nut joint, Figure 3.4 can be considered. Comparing Figure 3.4 with Figure 3.3, it is seen that s is radial slip and q is helical displacement (Figure 3.5).

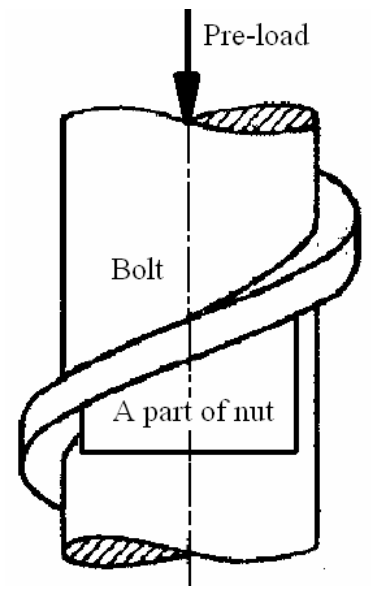

Figure 3.4. Bolt-nut interaction

Unwrapped bolt-nut contact surface for one pitch (p) is shown in Figure 3.5

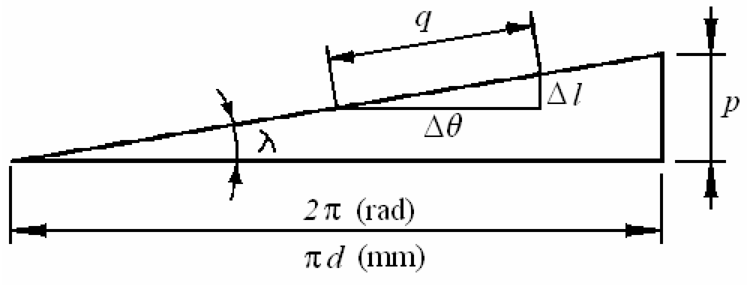

Figure 3.5. Unwrapped bolt-nut contact surface

If helical displacement q due to radial slip s is found, corresponding angle of rotation of bolt $\Delta\theta$ and relaxation Δl are calculated by using Figure 3.5 as

$$\Delta\theta = \frac{2\pi q}{\sqrt{p^2 + (\pi d)^2}} \tag{3.12}$$

and

$$\Delta l = \frac{p q}{\sqrt{p^2 + (\pi d)^2}} \tag{3.13}$$

Equation (3.12) can be reduced to approximate form by neglecting p, comparing with the πd, as

$$\Delta\theta \cong \frac{2q}{d} \tag{3.14}$$

Equation (3.13) gives the amount of axial loosening which reduces the clamping force due to the following equation

$$\Delta F = AE\frac{\Delta l}{L} \tag{3.15}$$

where L is the clamped length of the bolt.

CHAPTER 4

NUMERICAL STUDIES ON THE NEW MODEL

4.1. Numerical Results and Comparisons

Transverse displacement distribution along the bolt axis $v(x)$ due to the excitation of top plate δ = 0.45 mm is found by using analytical formulation given in Chapter 3 and the finite element model developed in ANSYS by using APDL (ANSYS Parametric Design Language). The results are shown in Figure 4.1.

Transverse displacements of the bolt part within the nut, namely engaged threads, are given in Table 4.1. Also, assuming the transverse displacement of the coordinate of the bolt axis at the upper surface of the nut is 0.45 mm, which is the transverse excitation δ, the possible radial slips are calculated and given in Table 4.1.

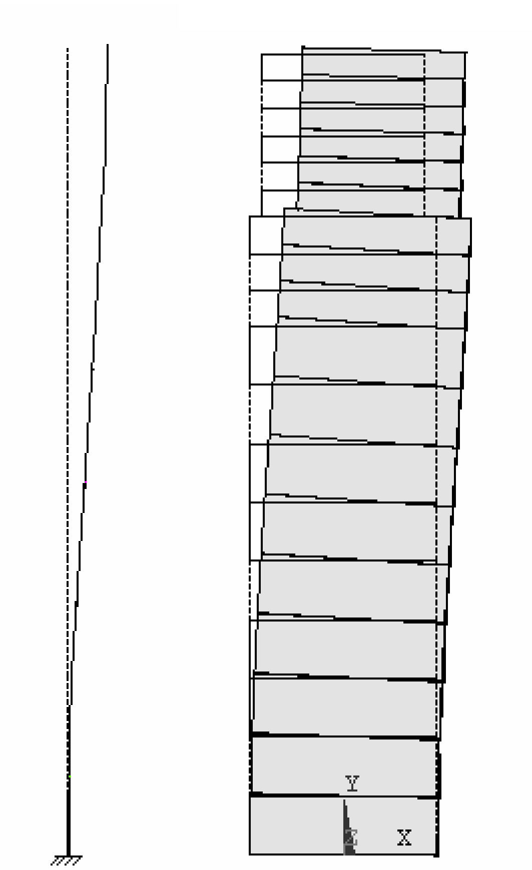

Figure 4.1. Transverse displacement of bolt axis

Table 4.1. Displacements of the bolt threads within the nut

Thread number from the bearing surface	Displacement (mm)		Radial slip (mm)	
	Analytical	ANSYS	Analytical	ANSYS
First thread	0.41243	0.41426	0.03757	0.03574
Second thread	0.42120	0.42302	0.02880	0.02698
Third thread	0.43009	0.43188	0.01991	0.01812
Fourth thread	0.43919	0.44090	0.01081	0.00910

Radial slip amplitudes of the all engaged threads given by Zhang et al. (2007) are illustrated in Figure 4.2.

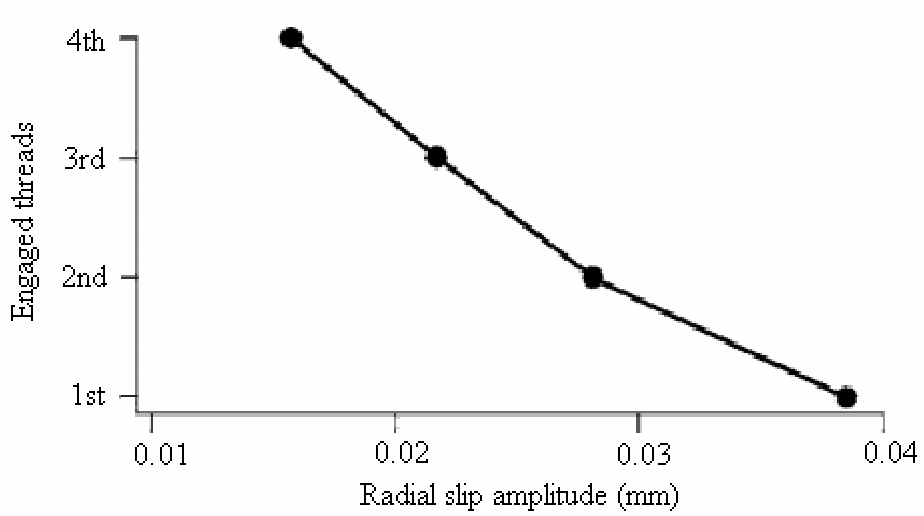

Figure 4.2. Radial slips of all engaged threads
(Source: Zhang et al., 2007)

Comparison of radial slip in Table 4.1 with the results of Zhang et al. (2007) given in Figure 4.2 gives the following conclusions:

- Radial slips in first and second threads are in excellent agreements,
- Radial slips in third thread are close to each others,
- Radial slips in fourth thread are not close to each others.

It is known that for the bolted joint configuration considered here, self loosening is mainly based on the radial slips in first and second threads. Therefore, the presented results and conclusion can be accepted positive.

As presented in Chapter 3; nut rotation, reduction of the clamping force, and progressive nut rotation etc due to transverse excitation are based on the radial slip in the first engaged thread. Therefore, the following results taken from Zhang et al. (2007) are obtained by using the formulation given in Chapter 3.

Considering friction coefficient in threads μ = 0.09, lead angle λ = 2.66º and radial slip s = 0.03757 mm in Equation in (3.11), circumferential displacement per step is

$$q = s\frac{\tan\lambda}{\mu} = 0.03757\frac{\tan 2.66^\circ}{0.09} = 0.01939 \text{ mm} \tag{4.1}$$

Since a loading cycle has four steps, $4q$ = 4·0.01939 = 0.07756 mm is the circumferential displacement per cycle. Figure 4.3 shows gradual circumferential displacement due to the periodic transverse displacements with amplitude δ = 0.45 mm.

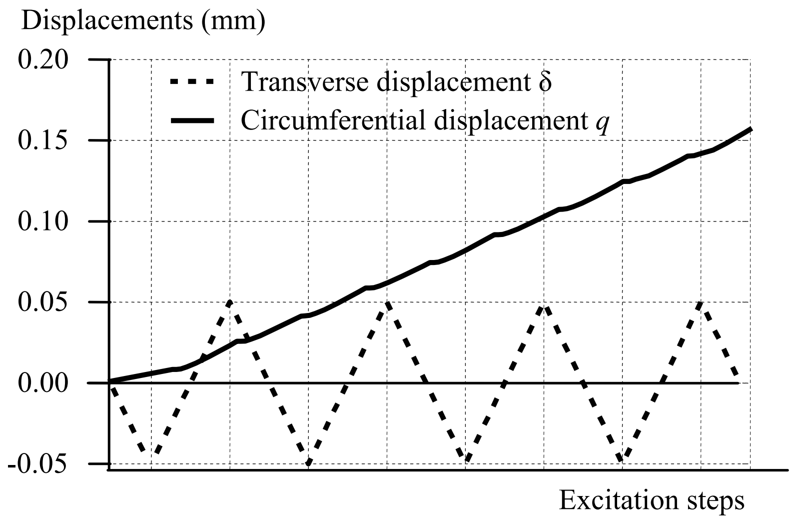

Figure 4.3. Progressive circumferential displacement due to transverse displacements

Nut rotation by using circumferential displacement is calculated approximately by Equation (3.14). For example, nut rotations per step and per cycle are calculated below, respectively.

$$\Delta\theta \cong \frac{2q}{d} = \frac{2\times 0.01939}{12} \tag{4.2.a}$$

$$\Delta\theta \cong 0.0032 \text{ rad/step} = 0.1833 \text{ deg/step} \tag{4.2.b}$$

and

$$\Delta\theta \cong 4\frac{2q}{d} = 4\frac{2\times 0.01939}{12} \tag{4.3.a}$$

$$\Delta\theta \cong 0.0128 \text{ rad/cycle} = 0.7334 \text{ deg/cycle} \tag{4.3.b}$$

Figure 4.4 shows gradual reduction of clamping force due to the progressive nut rotation. It can be seen from Figure 4.4 that the present results in Equation (4.3.b) are close the results in Figure 4.4. Also, by using Equation (3.15), percentage reduction of the clamping force P/P_o % can be found by considering P_o = 25 kN.

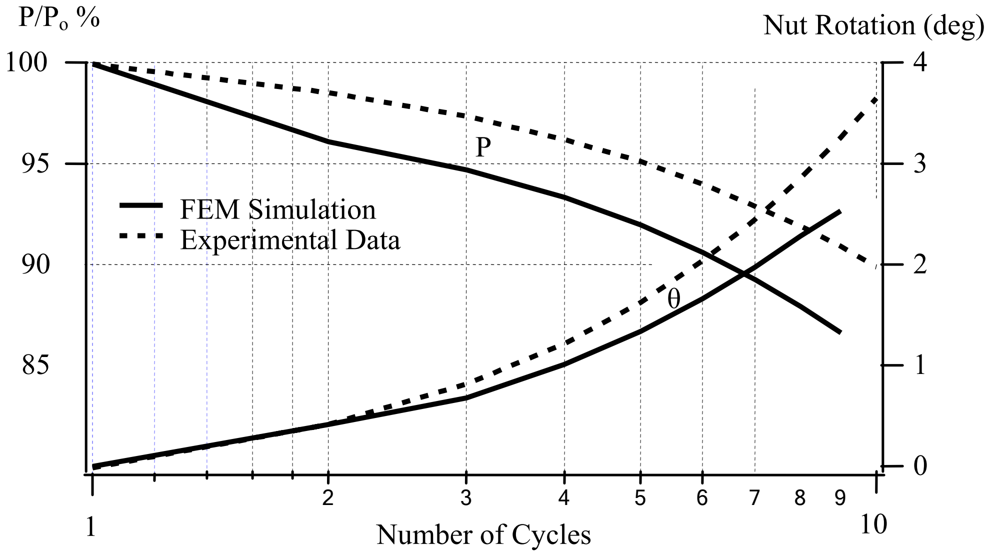

Figure 4.4. Reduction of clamping force due to the nut rotation
(Source: Zhang et al., 2007)

CHAPTER 5

CONCLUSIONS

In the available formulations existing in the literature for solutions of self-loosening problems of bolted joints are very complex to understand and have various assumptions. Also, they are not practical to use and apply easily.

Solutions of self-loosening problems of bolted joints are simply based on understanding the self-loosening mechanism rules. By using these rules and fundamental equations of solid mechanics, new, clear, and practical self-loosening formulations for bolted joints are derived.

Derivation is based on the solution of the differential equation of a tensioned cantilever beam under the lateral tip load and bending moment. The solution is transverse displacement distribution along the bolt axis due to the transverse excitation. By using this transverse displacement distribution along the bolt axis radial slip amplitudes of the all engaged threads are found. Then, loosening of bolted joint due to progressive circumferential displacement is discussed.

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