DESIGN AND PRODUCTION OF VIBRATION-RESISTANT BOLT PRODUCED BY COLD FORGING

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ABSTRACT

DESIGN AND PRODUCTION OF VIBRATION-RESISTANT BOLT PRODUCED BY COLD FORGING

Throughout their lifetime, fasteners are affected from vibrations. Thus, bolted joints have gradual loosening which causes security problems. In this thesis, improving the loosening situation of bolts under transverse vibration is studied by designing new bolt thread forms which have high vibration resistance. In the first step, the possible thread forms are drawn in 2-D and 3-D by considering the elastic contact. Secondly, static finite element simulations in contact regions are performed for different thread forms designed in the first step. After finding the thread forms within the gap tolerances between bolt and nut threads, thread rolling tools are designed and manufactured to verify them experimentally. Finally, numerous tests on transverse vibration and torque-tension are conducted to test the performance of the new thread forms.

ÖZET

SOĞUK DÖVME İLE ÜRETİLEBİLEN TİTREŞİM DİRENÇLİ CIVATA TASARIMI VE İMALATI

Bağlantı elemanları ömrü boyunca titreşimlerden etkilenirler. Bu nedenle, cıvatalı bağlantılar güvenlik problemine neden olan kademeli gevşemeye sahiptirler. Bu tezde, yüksek titreşim direncine sahip yeni cıvata diş formlarının tasarımı ile enine titreşim altındaki cıvataların gevşeme durumunun iyileştirilmesi incelenmiştir. İlk aşamada, elastik temas göz önüne alınarak olası diş şekilleri 2-D ve 3-D de çizilmiştir. İkinci olarak, ilk aşamada tasarlanan farklı cıvata diş formları için temas bölgelerindeki statik sonlu elemanlar simülasyonları gerçekleştirilmiştir. Cıvata ve somun dişleri arasıdanaki boşluk toleransları içinde vida diş formları bulunduktan sonra onları deneysel olarak doğrulamak için ovalama tarakları tasarlanmış ve imal edilmiştir. Son olarak, yeni vida diş formunun başarımını test etmek için çeşitli yanal titreşim ve tork-gerilme testleri yapılmıştır.

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LIST OF SYMBOLS

d	major diameter
d_p	pitch diameter
d_r	root diameter
f	friction coefficient
F_h	horizontal force acting on thread
F_p	preload
F_t	transverse force
F_v	vertical force acting on thread
h	depth of thread
h_s	height of thread bulge triangle
L_n	nut length
n	number of helix
N	reaction force
p	pitch
P_h	axial movement for one revolution of screw
Q	tangent of rotational force acting on thread
r_n	effective radius of contact between nut and bearing surface
r_t	effective contact radius of the threads
R_s	radius of thread tip
S	slip distance
T_{in}	tightening torque
T_l	torque required to decrease the load
T_{off}	disassembly torque
T_r	torque required to raise the load
W	axial load
α_n	thread angle
δ	distance between two helix function
Δs	unit slip distance
Δx	distance of fastener to sheaker
λ	helix or lead angle
μ_n	friction coefficient between face of nut and bearing surface of bolt

μ_t	friction coefficient between bolt and nut threads
θ	angle of thread bulge

CHAPTER 1

INTRODUCTION

1.1. Importance of Self Locking

Bolt or bolt and nut pair are used to fasten two or more objects either under static or dynamic load conditions. If the bolted joint is under dynamic loads in either axial or transverse directions, it may have loosing problem. To eliminate this problem, several applications such as use of washer, double nut, and chemical bonding are common in practice.

However, the problem of self-loosening of bolted joints is still an open area of research especially in automobile industry.

1.2. Past Studies

Goodier and Sweeney (1945) developed a test apparatus shown in Figure 1.1 to study the self loosening of threaded fasteners under axial loads. They derived the equations of external torques needed to loosen a nut under both increasing and decreasing bolt loads.

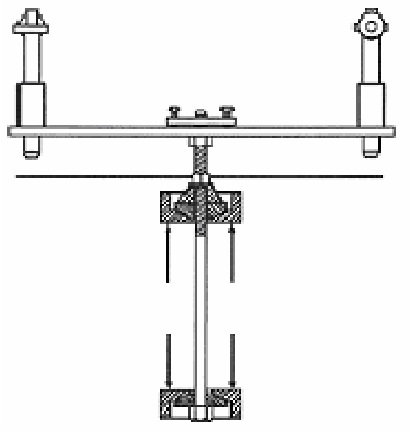

Figure 1.1. Goodier and Sweeney test machine
(Source: Bickford and Nassar, 1998)

Junker (1969) designed a test machine shown in Figure 1.2 to test the loosening of bolted joints under dynamic loads in perpendicular direction to the thread axis.

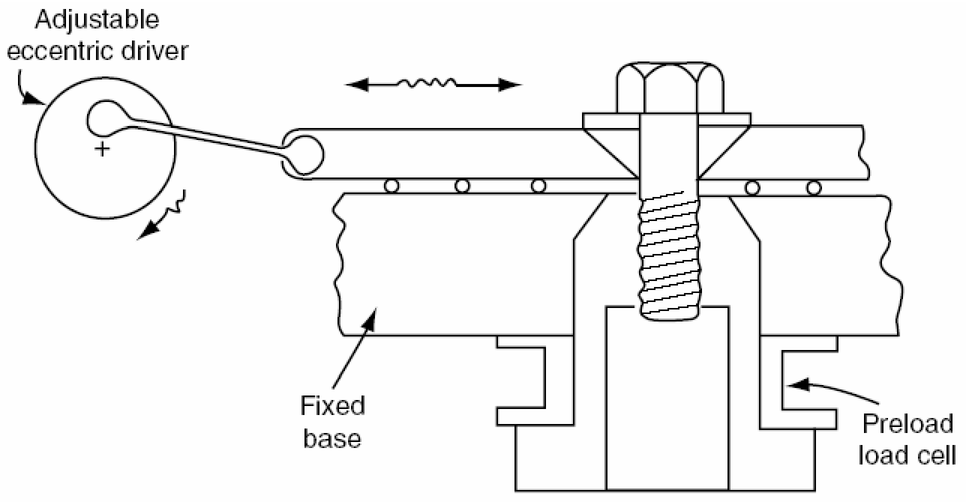

Figure 1.2. The Junker vibration test machine
(Source: Bickford, 2008)

After presenting the mostly cited studies above, the other past studies are summarized by ordering them according to the methods: analytical and numerical. Each group is given in the order of timeline.

Sakai (1978) studied on the bolted joints under transverse load to determine the self rotation for loosening, critical slippage of the clamped parts theoretically. He performed also experiments to verify his calculations. He (1979) also investigated the fatigue life to find the proper tightening of bolts considering yield points of the bolted joint materials.

Ramey and Jenkins (1995) reported the bolt loosening under dynamic tensile and shear forces. They found the empirical equation for prediction of bolt loosening by using Taguchi method.

Yokoyama (2009) expressed an analytical formulation for the bolted joints under transverse load. He reported the following five items:

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

The geometrical model of bolted joints includes complex geometrical shapes such as thread forms. The analytical models for analyzing the mechanics of the loosening of bolted joints under transverse loads are analytically very complicated. Therefore, finite element models of bolted joints are preferred by researchers. The following studies are based on finite element method.

Pai and Hess (2002) used the model shown in Figure 1.3 to find the failure of threaded fasteners due to vibrational shear loads. The benefits of their study are to find the slippage and prediction of different loosening mechanisms. They pointed out that the localized slip is effective on loosening at relatively low shear loads.

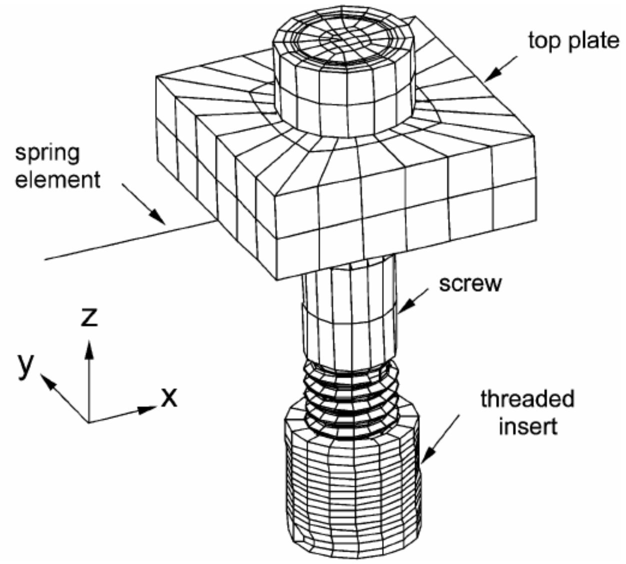

Figure 1.3. Joint model with typical finite element mesh
(Source: Pai and Hess, 2002)

Pai and Hess (2003) examined the effect of fastener placement, distance *Δx*, on loosening of fastener by using the system shown in Figure 1.4.

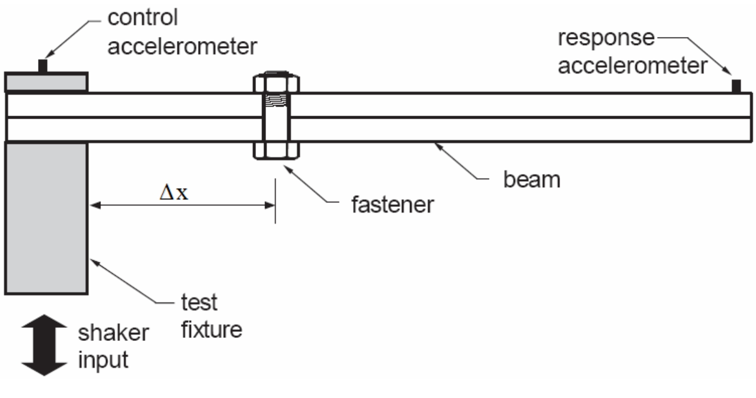

Figure 1.4. Test apparatus for the effect of fastener placement on loosening
(Source: Pai and Hess, 2003)

Jiang et al (2003) presented an experimental investigation on self-loosening of bolted joints. They also used numerical model in ABAQUS to see the plastic deformation near the roots of the engaged threads under the transverse cyclic loads.

Izumi et al (2005) studied on tightening and loosening mechanisms of threaded fastener by using the model shown in Figure 1.5. They concluded that loosening is initiated by thread slips, but not the head slip.

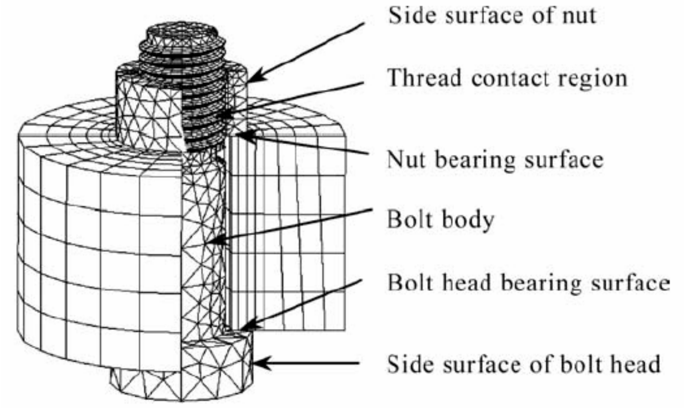

Figure 1.5. Finite element model
(Source: Izumi et al, 2005)

Nishimura et al (2007) showed numerically and experimentally that the relative slippage of bolted joints can be represented by steps shown in Figure 1.6.

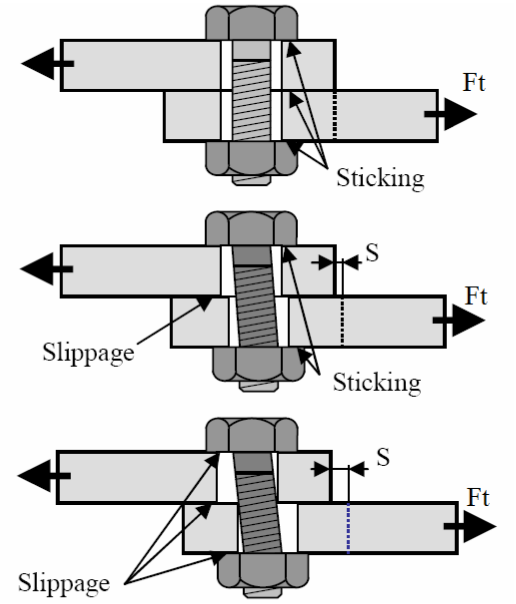

Figure 1.6. Sticking and slipping
(Source: Nishimura et al, 2007)

Hou and Liao (2014) studied the self-loosening of bolted joints to see the effect of time step, initial clamping force, amplitude of the shear load, thread tolerance, and friction coefficients.

The presented studies above are based on friction based self locking mechanisms. The other studies that related with the form of threads are cited in the next paragraphs.

Sase et al (1998) proposed an idea for anti-loosening screw fasteners named 'The Step-Lock Bolt' (SLB) shown in Figure 1.7.

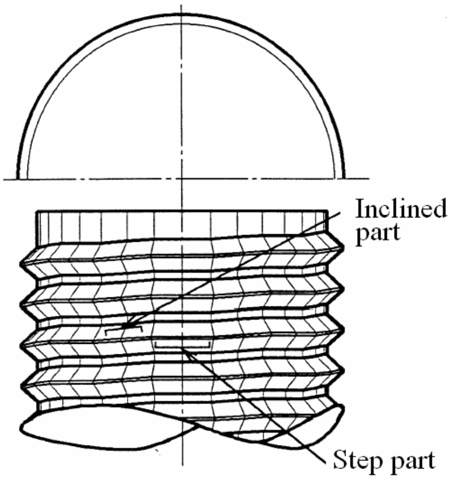

Figure 1.7. Outline of the Step-Lock Bolt
(Source: Sase et al, 1998)

Montminy (2007) conducted a new thread geometry based on without flank-to-flank thread contact. His new model is shown in Figure 1.8.

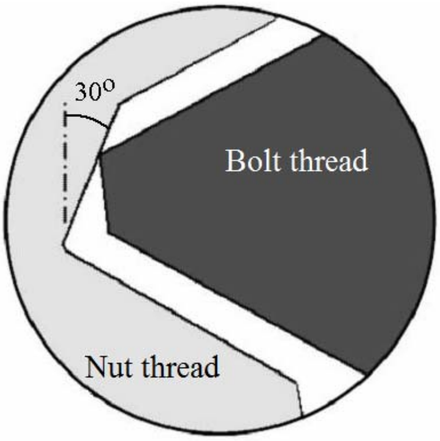

Figure 1.8. New model of Montminy
(Source: Montminy, 2007)

Ranjan et al (2013) used a cubic function represented in Figure 1.9 relating the axial and rotational motion in the bolt. It causes a non-helical curve of the thread in the bolt. However, they used a regular helical curve for the nut. Thus, additional torque is required for tightening due to elastic deformation occurred in the interference between the bolt and the nut.

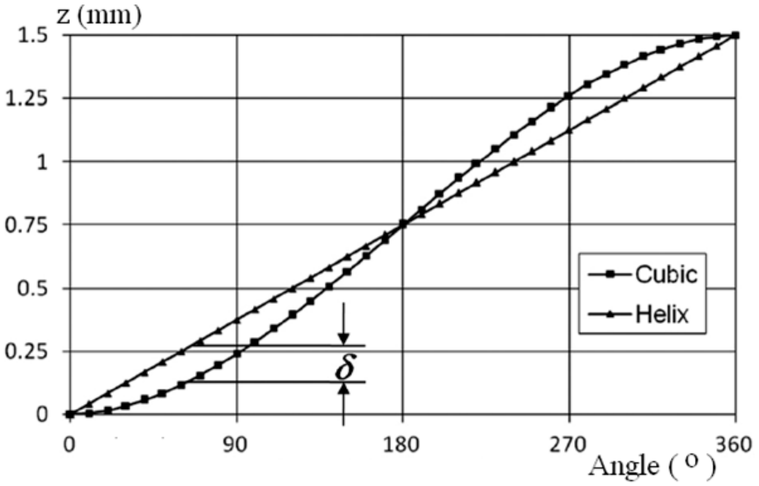

Figure 1.9. Comparison of the cubic and linear helical curve for M10
(Source: Ranjan et al, 2013)

1.3. Aim of the Thesis

Depending on the literature survey presented in last section, it is seen that self-loosening of bolted joints occurs due to the repeated loads applied to the joint in axial and/or transverse directions.

The aim of this thesis is to offer a new bolt thread form by modifying standard ISO metric form to be used in standard nuts. For this purpose, elastic deformations of the thread contacts in the limited range are examined via finite element simulations. Also, evaluating of the performance evaluation of the new bolt thread form are performed by using the transverse vibration and torque-tension tests

CHAPTER 2

THEORETICAL VIBRATION ANALYSIS

2.1. Main Definitions

The terminology for a hexagonal bolt and nut pair is illustrated in Figure 2.1.

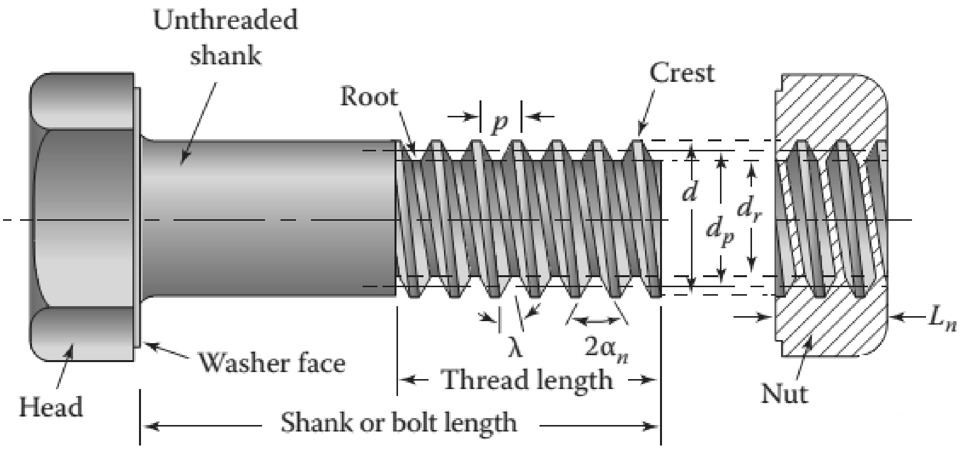

Figure 2.1. The terminology of hexagonal bolt and nut
(Source: Ugural, 2015)

The notations used in Figure 2.1 are explained below:

p: pitch,
λ: helix or lead angle,
α_n: thread angle,
d: major diameter,
d_p: pitch diameter,
d_r: root diameter,
L_n: nut length.

Pitch p is the axial distance between two successive threads. Lead P_h is the axial movement for one revolution of the screw. Helix angle λ may be either right handed (as in Figure 2.1) or left handed. Most of them are right handed. There is a relationship between pitch p and lead P_h as

$$P_h = n\,p \tag{2.1}$$

where n is the number of threads per lead. If n = 1, it is called as single-threaded screw, if n = 2, it is called double-threaded screw. Most bolts and screws have a single thread where $P_h = p$.

Unified and ISO thread form is shown in Figure 2.2. As seen from Figure 2.2, the major, root and pitch diameters are d, d_r, and d_p, respectively. Also, h is the depth of thread.

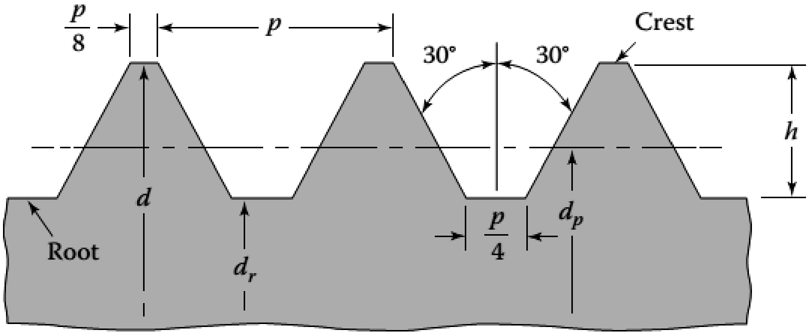

Figure 2.2. Unified and ISO thread geometry
(Source: Ugural, 2015)

2.2. Mechanics of Screws

In order to derive the torques required for increasing and decreasing of the axial load W, the bolt under axial load W as shown in Figure 2.3 can be considered.

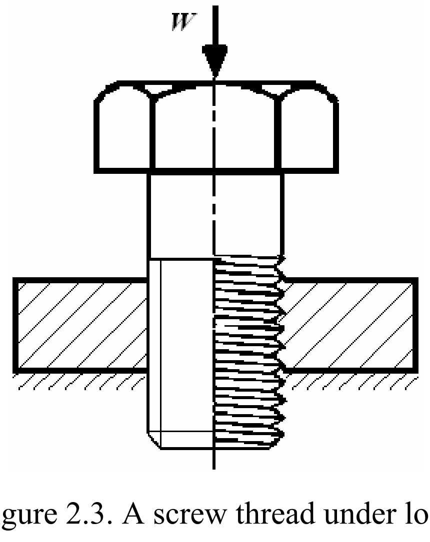

Figure 2.3. A screw thread under load W

It is noted from Figure 2.3 that the nut is assumed to be fixed. If one revolution of helix of the bolt-nut thread under the axial load *W* is unwrapped, Figure 2.4 is obtained. Free body diagram of small block which represents the bolt is shown in Figure 2.4.

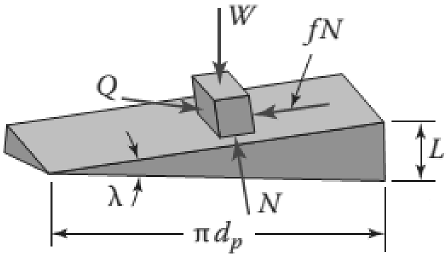

Figure 2.4. A developed screw thread
(Source: Ugural, 2015)

The following relationship is obtained on the triangle obtained by unwrapping the helix of the nut is written

$$\tan\lambda = \frac{L}{\pi d_p} \tag{2.2}$$

Two dimensional free body diagrams of bolt segment are shown in Figure 2.5. The view shown on the left side of Figure 2.5 is obtained by looking the nut thread from the inside region of the nut in radial direction. Therefore, the view of the section *A-B* shown on the right side of Figure 2.5 is perpendicular to helical path.

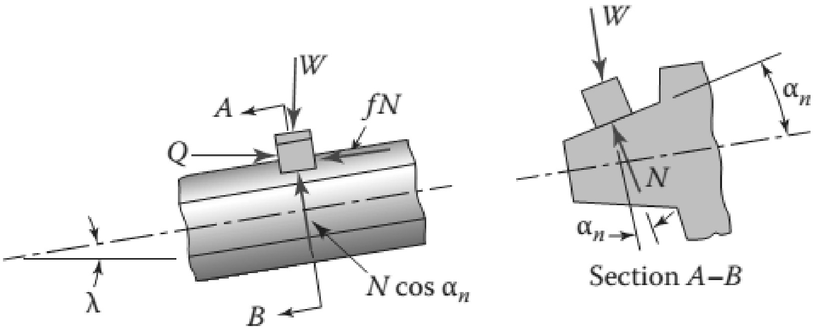

Figure 2.5. A segment of the thread
(Source: Ugural, 2015)

Torque T_r to raise the load W is derived by using Figure 2.5 as follows: The equilibrium conditions in the horizontal and vertical directions gives,

$$\sum F_h = 0: \; Q - N(f\cos\lambda + \cos\alpha_n \sin\lambda) = 0 \tag{2.3}$$

and

$$\sum F_v = 0: \; W + N(f\sin\lambda - \cos\alpha_n \cos\lambda) = 0 \tag{2.4}$$

where α_n is the normal thread angle and the other variables are defined in the Figure 2.5. Tangential force Q is found by eliminating force N from Equations (2.3) and (2.4) as

$$Q = W\frac{f\cos\lambda + \cos\alpha_n \sin\lambda}{\cos\alpha_n \cos\lambda - f\sin\lambda} \tag{2.5}$$

Therefore, the torque required to move the load up the inclined plane is

$$\begin{aligned} T_r &= \frac{d_p}{2}Q \\ T_r &= \frac{d_p}{2}W\frac{f + \cos\alpha_n \tan\lambda}{\cos\alpha_n - f\tan\lambda} \end{aligned} \tag{2.6}$$

Torque T_l required to decrease the load W is obtained by reversing the directions of Q and fN shown in Figure 2.5.

$$\begin{aligned} T_l &= \frac{d_p}{2}Q \\ T_l &= \frac{d_p}{2}W\frac{f - \cos\alpha_n \tan\lambda}{\cos\alpha_n + f\tan\lambda} \end{aligned} \tag{2.7}$$

The torques required for increasing and decreasing of the load W shown in Figure 2.3 can be treated as the tightening torque and disassembling torque for the bolted joints under tension W.

2.3. Classification of Loosening

Goodier and Sweeney (1945) found the reason for loosening of bolted joint under axial load. According to their study, Poisson's effect plays a critical role in the contacts of the bolt and nut threads. Due to the radial expansion of the nut and contraction of the bolt under the repeated axial load, loosening of bolted joint occurs.

Junker (1969) proposed the well-known theory on self-loosening. He confirmed his theory experimentally by using "Junker Test Machine". He tested self loosening mechanism under transverse vibration effects and 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 this statement by a solid body on an inclined surface as shown in Figure 2.6. If the force is applied to the block in *s* direction when it is stationary on the inclined plane, it slides not only in the direction of the force but also in downward direction although there is no external force in the downward direction. The self-loosening in bolted joints occurs in four stages as shown in Figure 2.7.

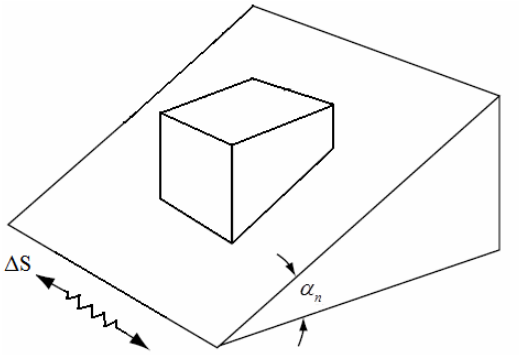

Figure 2.6. Junker's self-loosening test
(Source: Junker 1969, 1972)

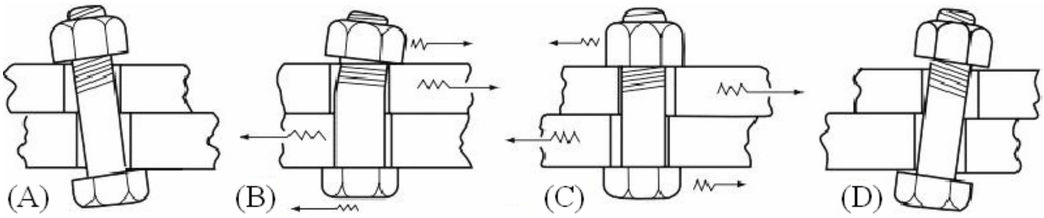

Figure 2.7. Self-Loosening process
(Source: Junker, 1969)

Junker (1969) explained the main reasons for self-loosening by as follows:

- If vibration is strong enough, transverse slip occurs between the joint surfaces,
- This slip momentarily overcomes all frictional restraint between parts, then self loosening starts,
- Depending on the thread-slip distance and thread clearance, elastic energy stored in the bolt is lost in each cycle,
- The energy lost during each cycle depends on the magnitude of the off-torque on the nut during slip. It is obtained by taking the coefficient of friction between nut and bolt threads μ_t and the coefficient of friction between the face of the nut and bearing surface μ_n equal to zero in the Motosh equation

$$T_{off} = \frac{F_P p}{2\pi} \tag{2.9}$$

which is originally

$$T_{in} = F_p \left(\frac{p}{2\pi} + \frac{\mu_t r_t}{\cos \alpha_n} + \mu_n r_n \right) \tag{2.8}$$

where T_{in}: torque applied to the fastener, F_p: preload created in the fastener, r_t: the effective contact radius of the threads, and r_n: the effective radius of contact between the nut and bearing surface.

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

1. Geometric: bolt diameter, thread pitch, grip length, class of fit, hole tolerance, and joint configuration,
2. Frictional: lubrication on bolt and lubrication between mating materials,
3. Kinetic: mass configuration and initial preload,
4. Additional: locking device.

Moreover, they pointed out that

i) vibration direction,

ii) magnitude of vibration,

also effect loosening.

2.4. Vibration Resistance

A bolted joint under axial and transverse excitation shown in Figure 2.8 can be vibration resistant joint by using the options listed below (Bickford, 2008):

1. Keep the friction forces in thread and joint surfaces more effective than the external forces applied to joint to loosen.

2. Do not allow slip between nut and bolt or nut and joint surfaces mechanically as shown in Figure 2.9.

3. Provide prevailing torque greater than the back-off torque. See Figure 2.10.

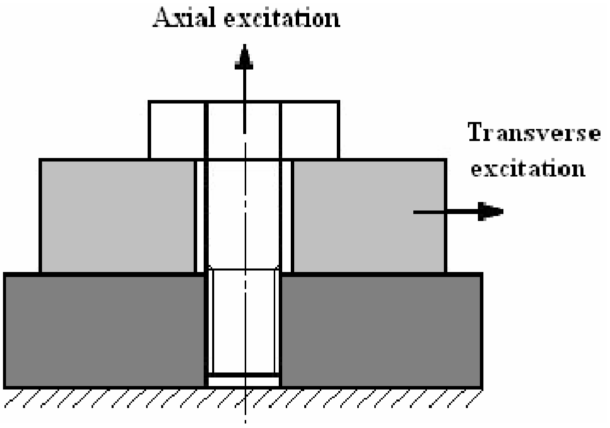

Figure 2.8. Axial and transverse excitation

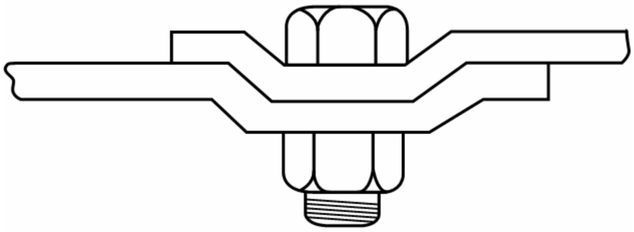

Figure 2.9. Bolted joint without slip mechanically
(Source: Bickford, 2008)

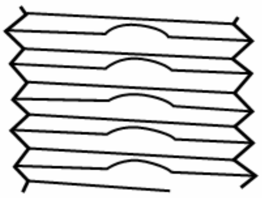

Figure 2.10. Interference fit threads (SPS)
(Source: Bickford, 2008)

2.5. Cold Deformation

Cold deformation is a metal forming process based on plastic deformation of materials using external force under recrystallization temperature of material. Cold forging process is preferable for mass production of metals because of its low cost comparing to machining process and rapid production abilities. Also products that have narrow tolerances can be manufactured by cold forging process easily. Bolts, nuts, and other small automotive parts are commonly produced by using cold forging process. Forming of product can be produced step by step in machine for obtaining final geometry of complex parts. The geometrical shapes of forming steps in production of bolt are illustrated on Figure 2.11.

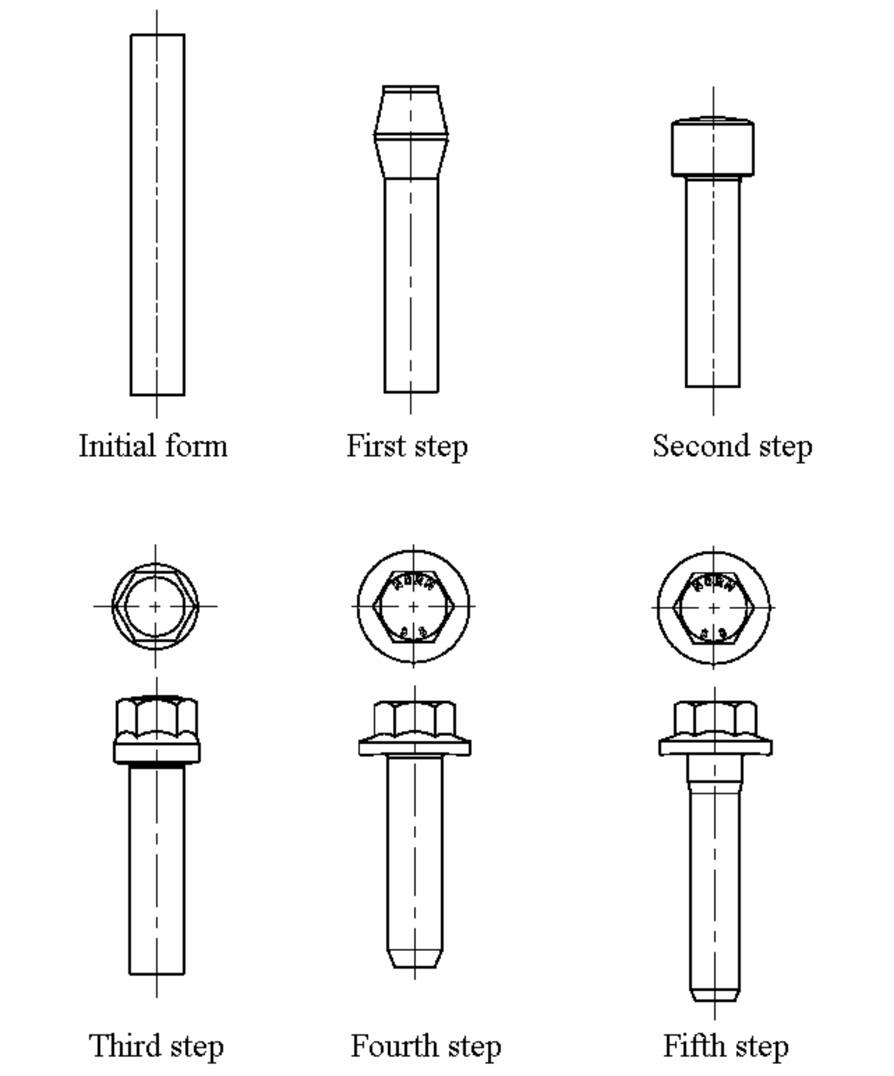

Figure 2.11. Cold deformation steps to produce bolt without thread

2.6. Vibration Test

The principle of Junker vibration test machine introduced in Figure 1.2 is given in Figure 2.12 in more detail.

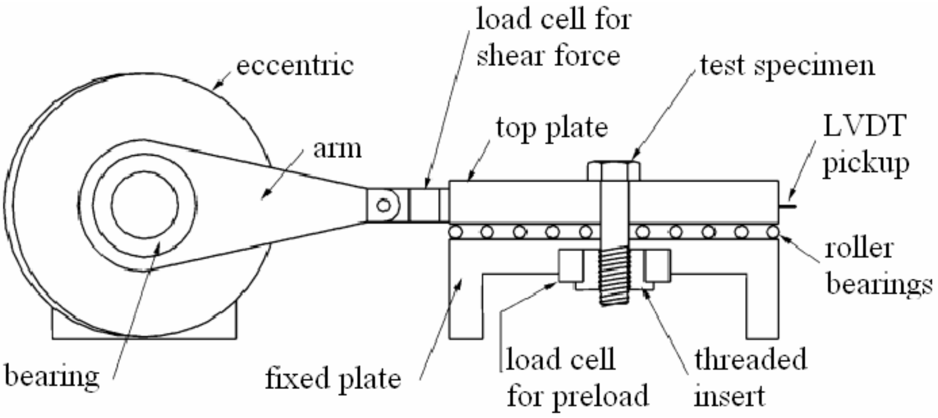

Figure 2.12. The Junker vibration test machine
(Source: Pai and Hess, 2002)

Adjustable eccentric driver shown in Figure 2.12 generates desired transverse displacement and frequency on the test specimen. Load cells are used to measure the preload acting on the joint as function of time, and then the control unit stores the experimental results. The test machine used in this thesis is shown in Figure 2.13.

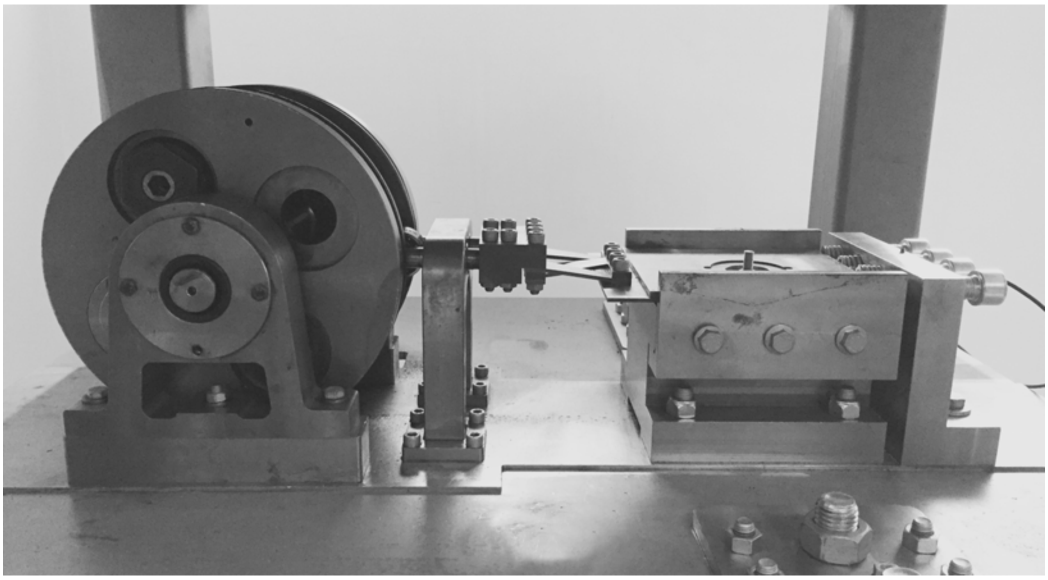

Figure 2.13. Junker vibration test machine in Norm Fastener Co.

2.7. Torque-Tension Test

This test is used to determine the relationship between torque, angle of rotation, and clamp load for bolted joints. Several graphs such as torque vs. clamp load, torque vs. angle, or torque and clamp load vs. angle can be plotted by using the data collected from the experiments. The components of the test system have the followings:

- data acquisition,
- rotary torque angle transducer,
- clamp force load cell,
- DC electric drive motor and controller,
- suitable fixture assembly for mounting test system components.

The test machine including torque preload transducer used in this thesis is shown in Figure 2.14 and 2.15, respectively. A detail on torque preload transducer is given in Appendix B.

Figure 2.14. Torque-tension test machine in Norm Fastener Co.

Figure 2.15. Torque preload transducer in Norm Fastener Co.

CHAPTER 3

NOVEL DESIGN AND NUMERICAL SIMULATIONS

3.1. Novel Design of Thread Form of Bolt

The ISO metric bolt thread form and novel thread form obtained by modification of the truncated cone of ISO metric bolt thread form are shown together in Figure 3.1. To reach the thread form shown in Figure 3.1, several concepts regarding bolt thread form are examined. Moreover, some of them in the models imagined and modeled are encountered in the patented forms.

It can be seen from Figure 3.1 that the modifications are based on the design parameters h_s, θ and R_s. It is possible to keep the deformations of the tip of the new bolt and ISO metric standard nut joints by selecting aforementioned design parameters in different combinations.

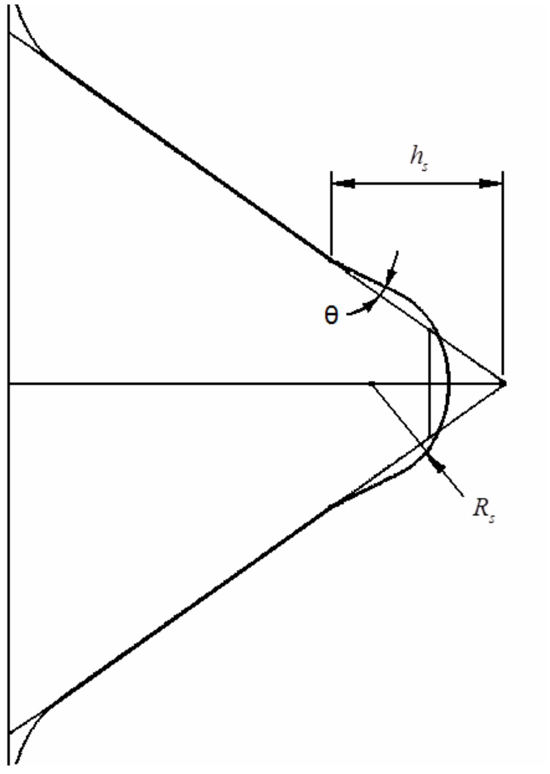

Figure 3.1. Novel thread form applied to ISO metric bolt thread

Some possible new thread forms of bolt are shown in Figure 3.2. In order to decide the thread form shape under the clamping force, static finite element simulations are performed to see the elastic deformations in bolt-nut interactions. Numerous finite element simulations are completed to decide the design parameters of new thread form under the desired application. Finally, the present one is determined. An example for finite element mesh of bolt-nut interaction and the static simulations under clamped force are shown in Figure 3.3. It is meshed by tetrahedral elements with element sizes 0.05-0.15 mm. The material is carbon steel 23MnB4. Other simulation parameters are listed below:

Bolt: ISO15071 modified M10x1.5x45, strength class: 8.8
Nut: DIN934 M10x1.5, strength class: 10
h_s = 0.375 mm
$\theta = 5^{\circ}$
R_s = 0.16 mm
Preload F_P =25.3 kN

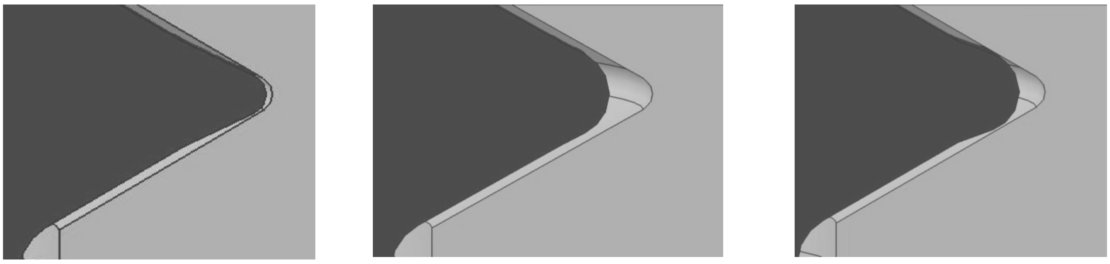

Figure 3.2. Novel thread form applied to ISO metric bolt thread

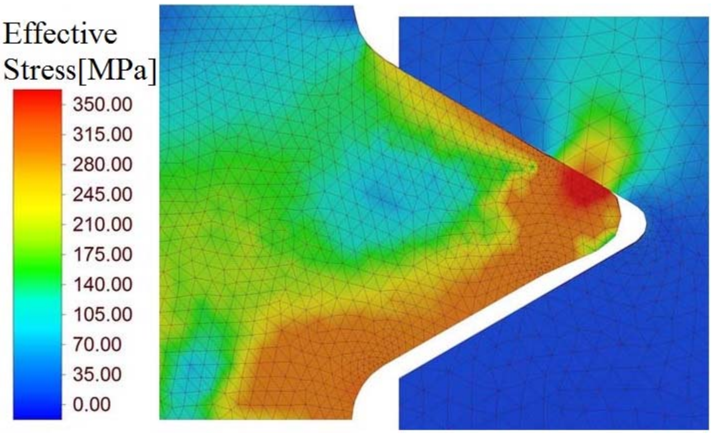

Figure 3.3. Effective stress distributions in bolt-nut interaction

3.2. Design and Production of Thread Rolling Tool

3D model of thread rolling tools with bolt before thread rolling are illustrated in Figure 3.4.

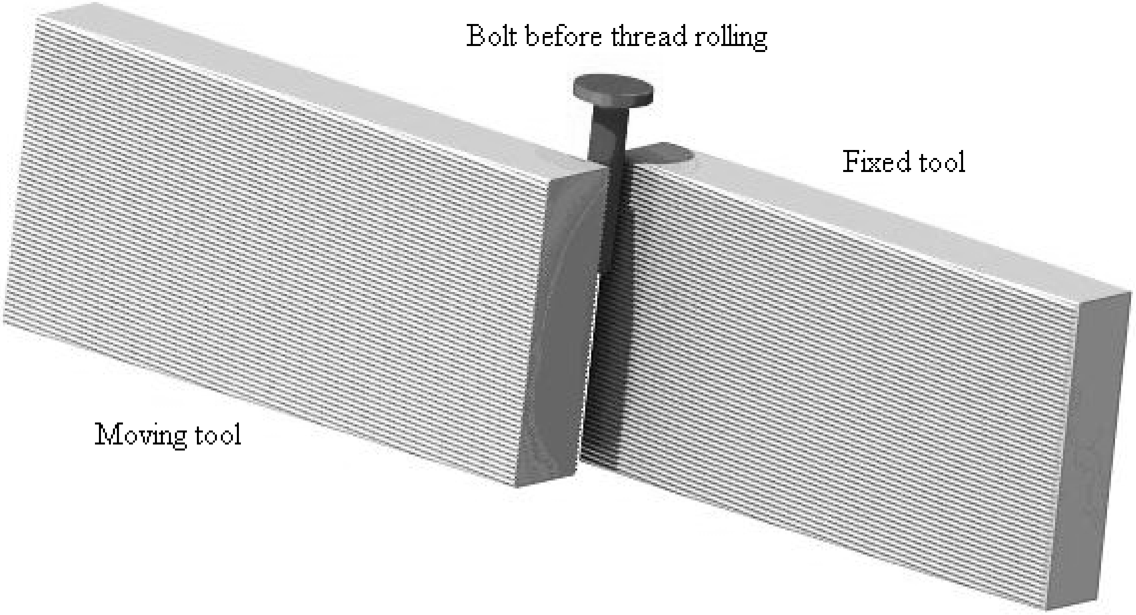

Figure 3.4. 3D model of thread rolling tools with bolt before thread rolling

The geometrical model of thread rolling tools are obtained in CATIA V5 R19, and then thread forming of bolt by using thread rolling tools are simulated in SIMUFACT 12. Some critical instants from simulation are shown in Figure 3.5. Additionally, thread forming views of bolt in 2D are given step by step in Figure 3.6.

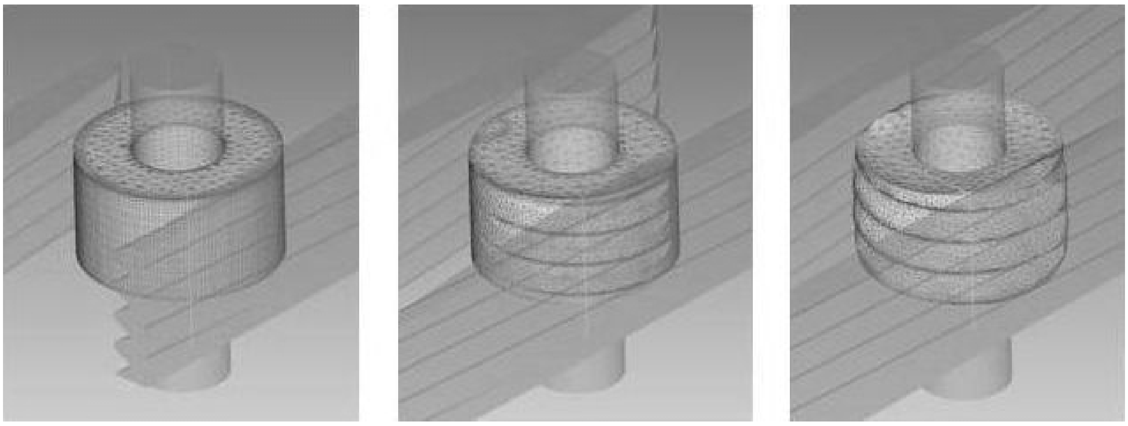

Figure 3.5. Thread forming steps view in 3D

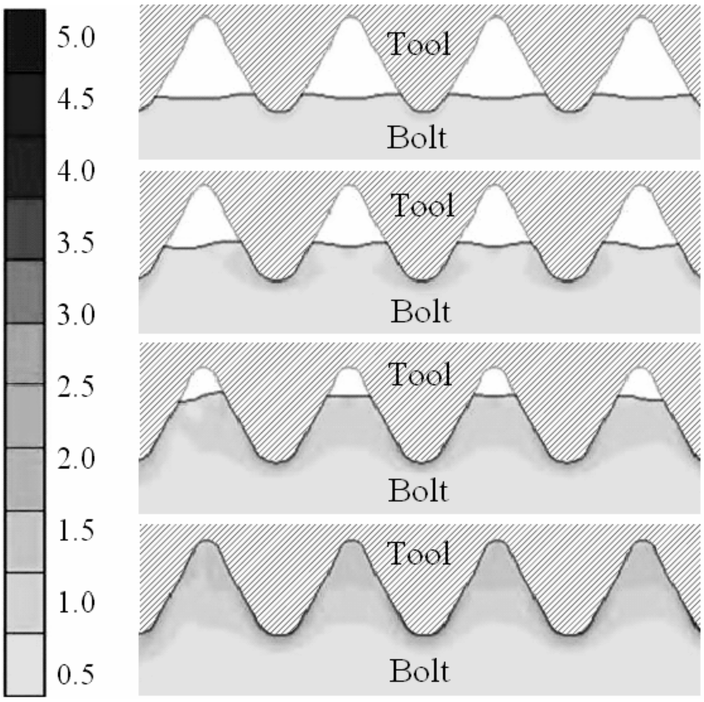

Figure 3.6. Thread forming steps view in 2D

After completing successfully the design and theoretical verification of thread rolling tools, they are produced in Taiwan due to the complex surface and desired accuracy. The photograph of the produced tools is given in Figure 3.7.

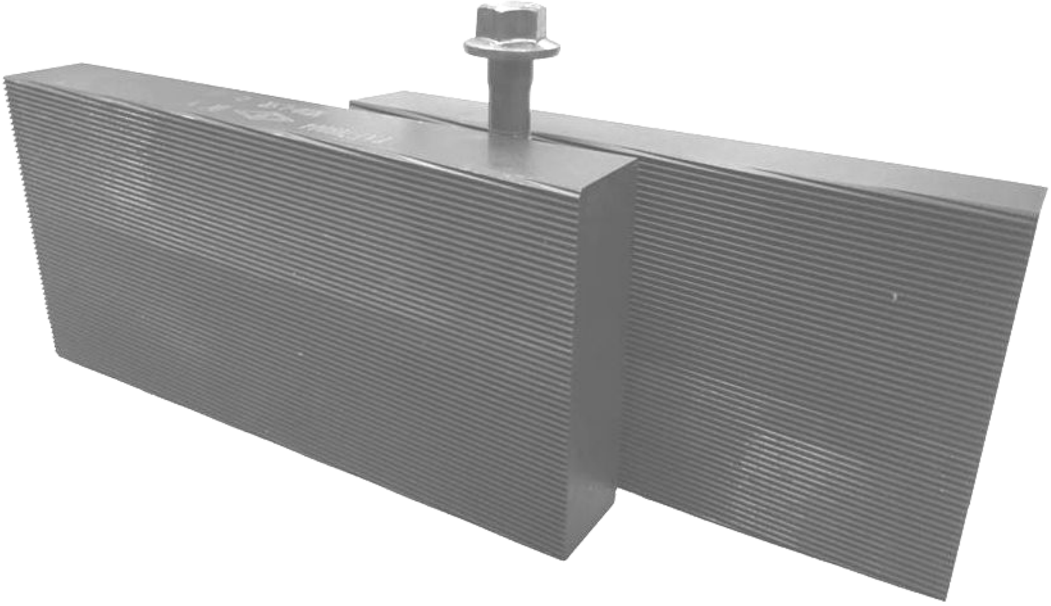

Figure 3.7. Thread rolling tools produced in Taiwan

3.3. Production of New Bolts

Production of new bolt having the profile shown in Figure 3.1 is accomplished by using the rolling thread tools shown in Figure 3.7. The bolt before rolling step is shown on the left of Figure 3.8. The photograph illustrated in the middle of Figure 3.8 shows the bolt just after rolling process. The bolt having surface treatment is shown on right side of Figure 3.8. For surface treatment zinc-flake coating is used. Bakalite images of the standard and new threads are shown in Figure 3.9.

Figure 3.8. New bolts produced by using new rolling threads

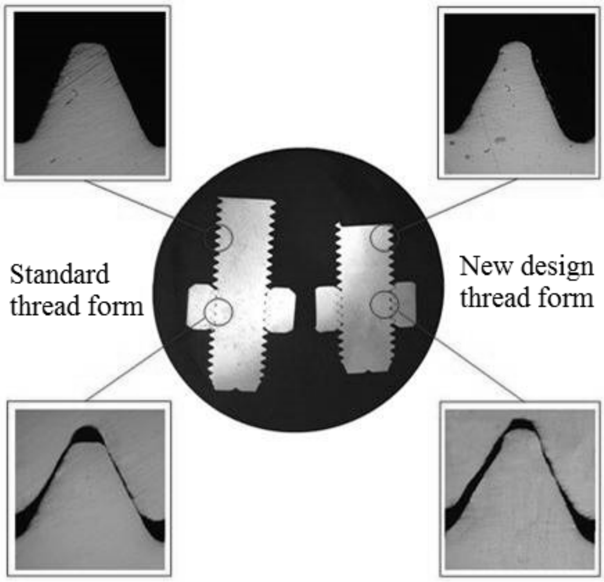

Figure 3.9. Thread forms of standard and new produced bolts, respectively

CHAPTER 4

EXPERIMENTAL RESULTS AND DISCUSSIONS

4.1. Torque-Tension Tests

Four samples of ISO-metric thread form and four samples of new thread form are chosen and tested. The results are presented in Figures 4.1 and 4.2.

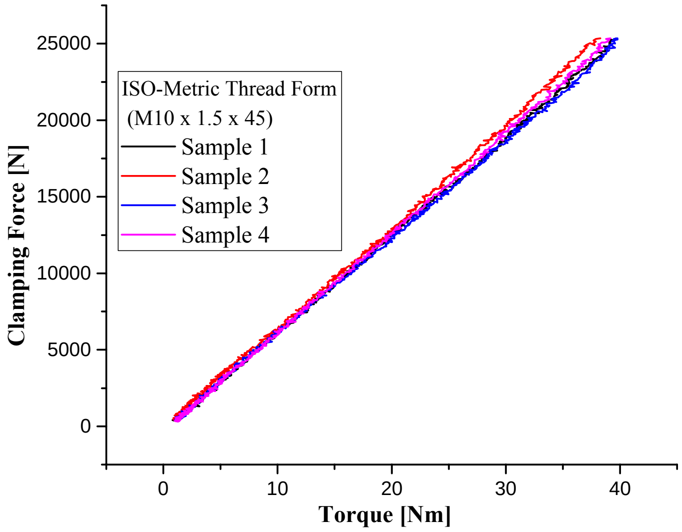

Figure 4.1. Results of ISO-metric thread form

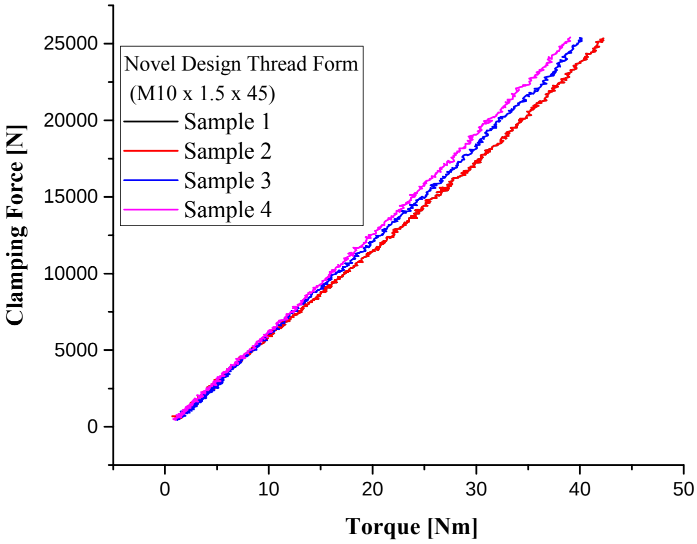

Figure 4.2. Results of new thread form

The average results of new and ISO-metric thread forms are found by using the results presented in Figures 4.1 and 4.2 and then given in Figure 4.3. It can be seen from Figure 4.3 that the torque increment for novel designed thread is about %5 with respect to ISO-Metric thread for the clamping force F_p = 25.3 kN. It should be mentioned that prevailing torque of novel design bolt is provided to overcome the back-off torque.

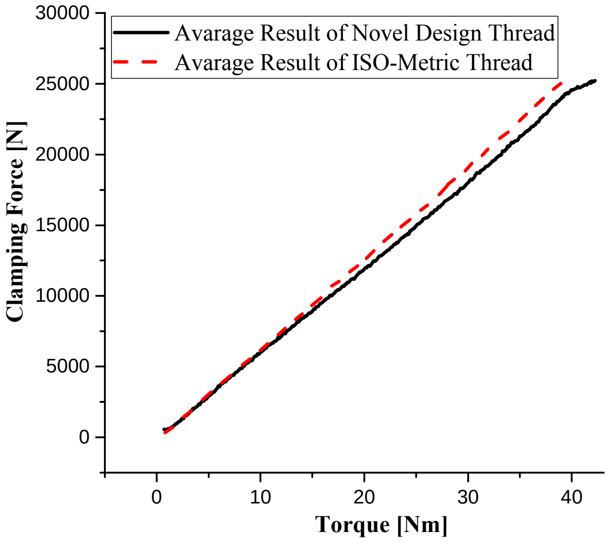

Figure 4.3. Average results of new and ISO-metric thread forms

4.2. Junker Vibration Tests

The test specimens for ISO-metric thread form with spring washer is shown on the left side of Figure 4.4 and new thread form without spring washer is shown on the right side of Figure 4.4.

Figure 4.4. Test specimens for ISO-metric thread form with spring washer and new thread form without spring washer, respectively.

The aim of the tests performed in this step is to determine the effects of spring washer and new thread form on vibration resistance.

Four samples of ISO-metric thread form with and without spring washer and four samples of new thread form are chosen and tested. The results are presented in Figures 4.5, 4.6 and 4.7.

Similar to former section, average values in the each plot are calculated and plotted together in Figure 4.8 to compare them among each others.

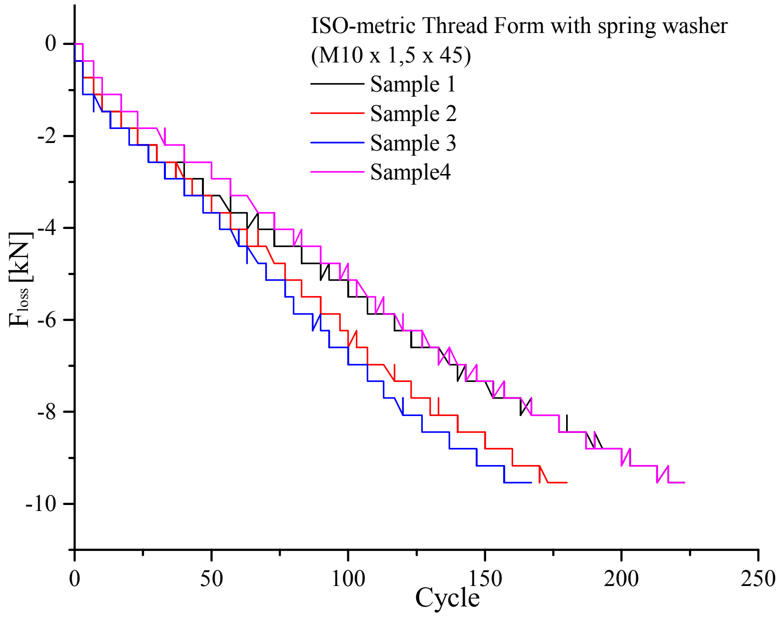

Figure 4.5. Results of ISO-metric thread form with spring washer

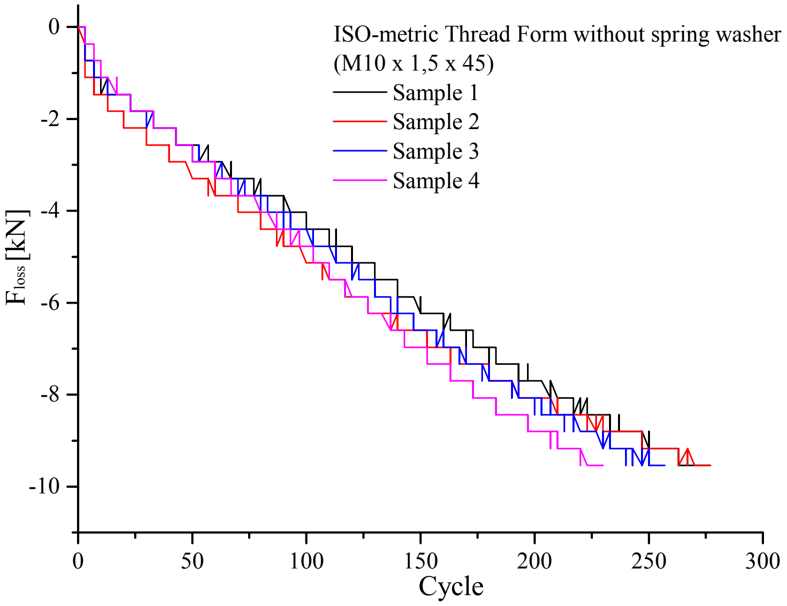

Figure 4.6. Results of ISO-metric thread form without spring washer

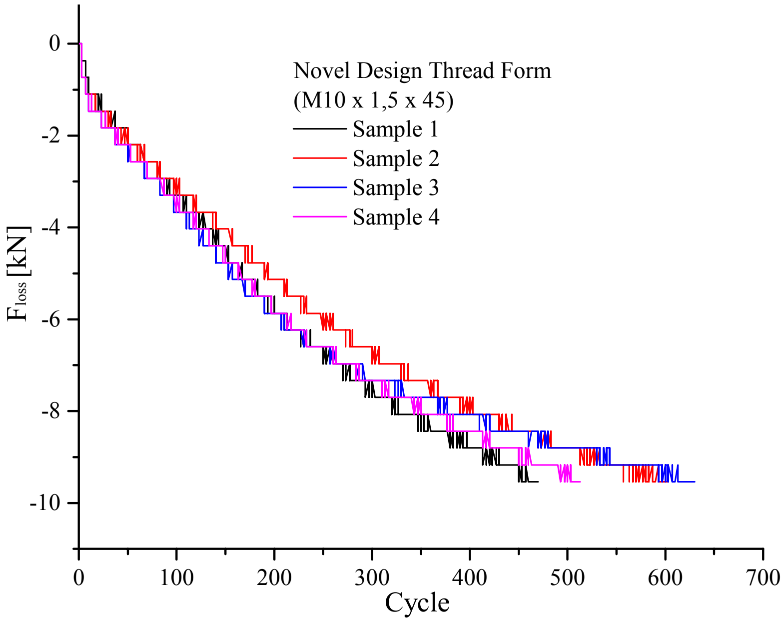

Figure 4.7. Results of new thread form

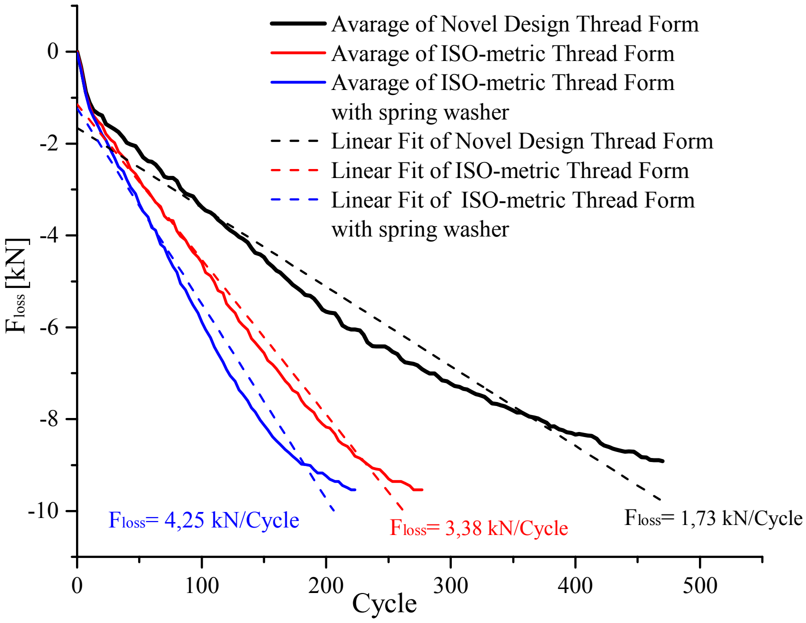

Figure 4.8. Average results of vibration tests

4.3. Discussion of Results

In this section, Figures 4.3 and 4.8 are considered to discuss on them. It can be seen from the aforementioned figures that new thread form has better performance than ISO-metric thread form with and without spring washer due to the less slope in the graphics.

CHAPTER 5

CONCLUSIONS

The improving the loosening of bolted joints under transverse vibration is studied by designing a novel bolt thread forms which has high vibration resistance. The main idea used for novel bolt thread forms is based on the elastic deformations of the thread contacts in the limited range. Within the scope of this thesis:

1. Finite element simulations are used for theoretical studies,
2. Production of the prototype of the novel bolts are accomplished,
3. Tests on transverse vibration and torque-tension are conducted.
4. The vibration resistance performance of new bolt is compared with the ISO metric bolts.
5. Finally, by using some modification on rolling diameter of bolt, the possible good results are obtained.

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APPENDIX A

SAN-TEZ PROJECT DETAILS

PROJECT CODE : 0653.STZ.2014

PROJECT NAME : Design and production of vibration-resistant bolt produced by cold forging

PROJECT DIRECTOR : Prof. Dr. Bülent YARDIMOĞLU

PROJECT STUDENT : Gökay YALDIZ

PROJECT INSTITUTION : İzmir Institute of Technology

PROJECT PARTNER : NORM Cıvata Sanayi ve Ticaret A.Ş.

PROJECT TIME : 12 months

APPENDIX B

TORQUE PRELOAD TRANSDUCER

The BLM μ tester is a complete device for measuring the friction coefficient. In addition to usage in the tool crib, it can be used in production line for fast and easy fastener evaluation. The advantages of using a bench are many: portable measuring equipment "all in one" complete unit, PC with touch screen, fixed transducer, adapter kit set included in the drawer, battery with an operation time > 16 hours and built-in battery charger. The μ Tester 25 and 200 can be supplied with additional transducers.

The model used in the test is "BLM TPT 200 / 100 Torque Preload Transducer 200 Nm, 100 kN ".