

## Design and Optimization of Screwed Fasteners to Reduce Stress Concentration Factor

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### Abstract

The bolt and nuts are used for temporary fastening in the machines and structural assembly; they play an important role to restrict the movement of individual parts in the assembly. Around 10 to 20% of fasteners are used in various designs, sizes, strength levels and materials in complex mechanical systems for diverse applications. The unequal load distribution in the threaded fasteners causes the fatigue failure due to stress concentration

In this paper stress distribution in Buttress and ACME threads, for various symmetrical models such as groove added to bolt and nut, a step is added to nut, with or without groove on the bolt, and to the reduced diameter, a taper is added to the nut and a groove on the bolt are designed in 3D and analyzed and validated from 2D models developed to reduce stress concentration.

The objective of this study is to show the quantitative comparison between the Buttress and ACME bolt and nut design modifications to improve the design methods efficiently to reduce the stress concentration. In this study various geometrical designs are proposing to reduce the stress concentration factor.

**Keywords:** Fasteners; Threads; Design modification; Stress concentration

### Introduction

#### Fasteners

Machines are assembled from individual parts. The parts may need to be joined sometimes in permanent or temporary connections. The temporary connections or joints are normally made by using screws bolts or nuts and bolts. The temporary joints render the convenience of disconnection as frequently as required for such purposes as inspection, repairs, adjustment or replacement. The most important feature of the screws or bolt is that they are standardized and are available as standard part ready to use. The threaded fasteners are used to hold two parts. The fastener consists of two parts. An externally threaded member whose length is normally greater than the diameter carries threads over entire or part length from end. The other end of this member is larger in area (you can assume that the end is upset). This part is inserted in coaxial holes in two parts taking care that the "upset" or larger end is greater in dimension than the holes so that it does not pass through the hole. The end that comes out of the holes carries thread upon which another internally threaded part is wound. Fasteners can also be used to close a container such as a bag, a box, or an envelope; or they may involve keeping together the sides of an opening of flexible material, attaching a lid to a container, etc. Fasteners used in these manners are often temporary, in that they may be fastened and unfastened repeatedly. Bolt and nut are one among the different fasteners.

Bolted joints are most common elements of structures, machines in pressure vessels, automobiles, machine tools, home appliances etc. Hundreds of different bolt designs with various sizes, strengths and materials are used in the assembly of aircrafts. On an Averages of 2.4 million fasteners are used to assemble a Boeing 747 aircraft. In bolt nut fasteners the load distribution is very unequal with high stress concentration at the thread roots. This stress concentration can cause fatigue failure in the bolt-nut fasteners. For complex structures the reliability (trust or faith in) of the bolt-nut fasteners is the most important. So, study of bolt-nut fasteners is important.

#### Stress

The concept of stress, introduced by Cauchy around 1822, is a measure of the average amount of force exerted per unit area of the surface on which internal forces act within a deformable body. In other words, it is a measure of the intensity, or internal distribution of the total internal forces acting within a deformable body across imaginary surfaces. These internal forces are produced between the particles in the body as a reaction to external forces applied on the body. External forces are either surface forces or body forces.

The SI unit for stress is the Pascal (symbol Pa), which is equivalent to one newton (force) per square meter (unit area). The unit for stress is the same as that of pressure, which is also a measure of force per unit area. Engineering quantities are usually measured in Mega Pascal's (MPa) or Giga Pascal's (GPa). In imperial units, stress is expressed in pounds-force per square inch (psi).

For the simple case of a body axially loaded, e.g. a prismatic bar subjected to tension or compression by a force passing through its centroid, the stress  $\sigma$ , or intensity of the distribution of internal forces, can be obtained by dividing the total tensile or compressive force by the cross-sectional area where it is acting upon (Figure 1). Thus, we have

$$\sigma = (F/A)$$

Where  $\sigma$  = Nominal Stress; F = Total tensile or compressive force; A = Cross-sectional area.

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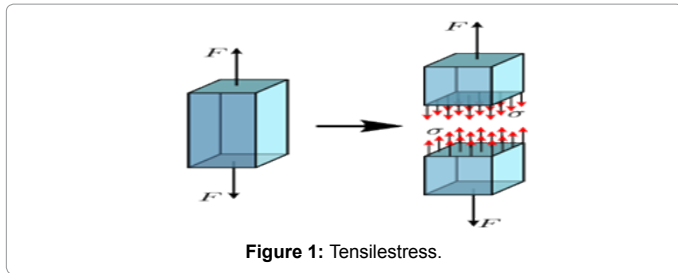


Figure 1: Tensile stress.

Experimental determination of stresses can be carried out using the photo-elastic method or FEM.

### Stress concentration

A stress concentration (often called stress raisers or stress risers) is a location in an object where stress is concentrated. An object is strongest when force is evenly distributed over its area, so a reduction in area, e.g. caused by a crack, results in a localized increase in stress. A material can fail, via a propagating crack, when a concentrated stress exceeds the material's theoretical cohesive strength. The real fracture strength of a material is always lower than the theoretical value because most materials contain small cracks that concentrate stress. Fatigue cracks always start at stress raisers, so removing such defects increases the fatigue strength.

**Causes:** Geometric discontinuities cause an object to experience a local increase in the intensity of a stress field. The examples of shapes that cause these concentrations are: Cracks, sharp corners, holes and, changes in the cross-sectional area of the object. High local stresses can cause the object to fail more quickly than if it wasn't there. Good design of the geometry minimizes stress concentration.

### Stress concentration factor

The ratio of the maximum stress and the nominal applied tensile stress is denoted as the stress concentration factor,  $K_t$ .  $K_t = \sigma_m / \sigma_o$

**Acme threads form:** There is a special type of trapezoidal thread called ACME thread. These two threads are identical in all respects except the thread angle. In ACME thread, the thread angle is  $29^\circ$  instead of  $30^\circ$  these threads has more thickness at core diameter than that of square threads. Therefore screw with ACME threads stronger than equivalent screw with square threads. Such a screw has a large load carrying capacity. From the Figure 2, If  $p$  = pitch,  $d$  = depth of thread, and  $n$  = number of threads per inch.

**Buttress threads form:** Buttress threads are used where heavy axial force acts along the screw axis in one direction only. It has higher efficiency compared with ACME threads. It can economically manufactured on thread milling machine. The axial wear at the thread surface can be compensated by means of split-nut. It can transmit power and motion in one direction only. The basic profile is the theoretical profile of the thread (Figure 3). An essential principle is that the actual profiles of both the nut and bolt threads must never cross or transgress the theoretical profile. So bolt threads will always be equal to, or smaller than, the dimensions of the basic profile. Nut threads will always be equal to, or greater than, the basic profile. To ensure this in practice, tolerances and allowances are applied to the basic profile.

### Applications of the screwed fasteners

1. Connecting threaded pipes and hoses to each other and to caps of fixtures.

2. Gear reduction via worm drives.
3. Moving objects connecting rotary to linear motion (lead screw of a jack).
4. Measuring by correlation linear motion to rotary motion in Micrometer.
5. Lead screw of a lathe.

### Advantages of the screwed fasteners

1. Screwed joints are highly reliable in service
2. Screwed joints are easy to assemble and disassemble.
3. A wide range of screwed joints is available to adopt under various working conditions.
4. Screws are relatively cheap.
5. Due to Standardization, cheap manufacturing processes can be adopted.
6. Screws can be used to transmit power such as lead screws.

### Limitations of the screwed fasteners

1. The stress concentration in the threaded portions is variable under variable conditions.
2. Screwed joints become loose due to machine vibrations.
3. The strength of the screwed joints is not comparable with Welded or riveted joints.

### Literature Review

Bolt-nut connectors are one of the basic types of fasteners used in machines and structures. They play an important role in the safety and reliability of structural systems. The load distribution in typical bolt-nut connectors is very unequal, with a high stress concentration at the thread roots. This stress concentration can cause fatigue failure

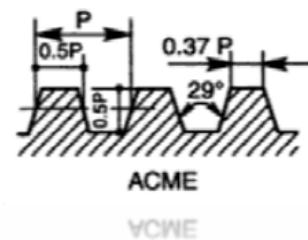


Figure 2: ACME thread form.

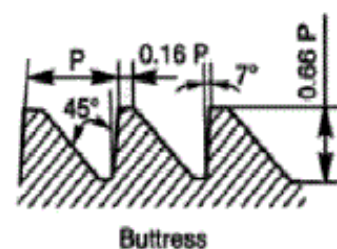


Figure 3: Buttress thread form.

in the bolt-nut connectors. For example, hundreds of different bolt designs with various sizes, strength levels, and materials are used in the assembly of an aircraft. On the average, 2.4 million fasteners are used to assemble a Boeing 747 aircraft. Of this total, 22% are structural bolts [1]. The importance of the reliability of bolt-nut connectors cannot be overemphasized in such applications. Hence, it is of considerable interest to study the stress concentration in bolt-nut connectors. In the past, several researchers have studied the stress distribution in bolt-nut connectors using computational and experimental methods. Hetenyi [2], Seika et al. [3], Patterson and Kenny [4,5] and Kenny and Patterson [6] have used experimental methods like photo elasticity to study the stress distribution in bolt-nut joints.

In present study an attempt made to understand the behavior of the bolted joints and the stress concentration factor when loaded statically with uni-axial external loads uses FEA [1]. Some journals like An innovative polariscope for photo elastic stress analysis [2], which uses photo-elasticity and Venkatesan and Kinzel [7], has studied the stress concentration in threads of bolt-nut joint using ax symmetry finite element model.

In this paper, the stress distribution in bolt-nut connectors is studied using an ax symmetric finite element model. Various geometric designs proposed in the literature were studied to determine the extent to which they reduce stress concentrations. Some well-known modifications do significantly reduce the stress concentration factor (up to 85%) while other changes produce much more modest changes. The design modifications include things such as grooves and steps on the bolt and nut, and reducing the shank diameter of the bolt. All of the changes also result in a reduction in weight.

In present study an attempt made to understand the behavior of the bolted joints and the stress concentration factor when loaded statically with uni-axial external loads [1]. Also an attempt has been made to minimize the error between experimental results and FEM simulated results by selecting the exact number of element and selecting suitable yield criteria. Linear finite element analysis method is used to determine the stress concentration factor of the threads in bolted connection.

In the current study the load transfer (tension and bending) as well as stress intensity factor solutions for various crack locations and sizes have been derived by advanced 3D FE analyses including threaded bolt and nut [8]. In addition, a large number of fatigue crack tests with Titanium and Steel fasteners have been performed to Determine realistic defect shapes for rolled- as well as machined threads, and the evolution of the shape as the defect propagate, Verify stress intensity factor and to determine the amount of bending inherent in the fastener due to the load introduction through the nut and due to the presence of a crack.

In this work the analysis of stresses and deformations arising in the bolt shank on various laws of the load distributions on the threads is considered [9]. The finite element method (FEM) [10-13] and ANSYS application on solution of this problem are used as well. Having solved the problem of finding the most loaded cross section of the bolt one can more precise to estimate the design reliability.

In the present work, joint materials are assumed to be isotropic and homogeneous, and linear elastic ax symmetric finite element analysis was performed to evaluate the member stiffness [14]. Uniform displacement and uniform pressure assumptions are employed in idealizing the boundary conditions. Wide ranges of bolt sizes, joint thicknesses, and material properties are considered in the analysis to evaluate characteristic behavior of member stiffness. Empirical formulas for the member stiffness evaluation are proposed using

dimensionless parameters. The results obtained are compared with the results available in the literature.

In this paper Finite Element Analysis (FEA) with extremely fine mesh in the vicinity of the blades of Steam Turbine Rotor is applied to determine stress concentration factors [15]. A model of Steam Turbine Rotor is shown.

In this paper, the details of an experimental method to measure the clamping force value at bolted connections due to application of wrenching torque to tighten the nut have been presented. A simplified bolted joint including a holed plate with a single bolt was considered to carry out the experiments. This method was designed based on Hooke's law by measuring compressive axial strain of a steel bush placed between the nut and the plate. In the experimental procedure, the values of clamping force were calculated for seven different levels of applied torque, and this process was repeated three times for each level of the torque. Moreover, the effect of lubrication of threads on the clamping value was studied using the same method. In both conditions (dry and lubricated threads), relation between the torque and the clamping force have been displayed in graph. The current thesis extends the work of Venkatesan and Kinzel [7], in which design modifications for 1-8 UNC thread forms were considered with more of emphasis on finding the stress concentration for different thread forms. The objective of this study is to show quantitative comparison of effect of various proposed design modifications in the literature [16] on the stress concentration factor. In this thesis Buttress thread and ACME thread forms with the same design modifications proposed in the literature were considered and analyzed using ANSYS, a popular finite element analysis package.

## Experimental Procedure

### Bolt Design:

$$D_{\text{minor}} = 0.8797 \text{ in}$$

$$D_{\text{major}} = 1 \text{ in}$$

$$\text{Pitch (p)} = 3.175 \text{ mm}$$

$$D_{\text{min}/2} = 11.17219 \text{ mm}$$

$$H_b = 0.866 P$$

$$= 2.74963 \text{ mm}$$

$$R_b = H/6$$

$$= .458271 \text{ mm}$$

Where,  $H_b$  = Height of the thread,  $W$  = Width of Nut

$R_b$  = Root radius of the bolt,

$H_n$  = Height of the nut

$R_n$  = Radius of the nut

### Nut Design:

$$W = 0.3125 \text{ in} = 007.9248 \text{ mm}$$

$$H_n = 0.690 \text{ in} = 017.526 \text{ mm}$$

$$R_n = H/1 = 0.2291355 \text{ mm}$$

$$D_{\text{max}/2} = 12.7 \text{ mm}$$

## ACME threaded model results

**Model 1:** The basic model in this study used is a 23.4 mm diameter plan. Bolt with corresponding hexagonal nut with a pitch of 3.175 mm ACME threads. The bolt and nut are made of high carbon steel and it has a poisons ratio of 0.29 with an angle of 29°. The contact between the bolt and nut is analyzed by finite element methods. The load is equal to stress 10 psi is applied on the bolt shank to produce the stress concentration (Table 1 and Figure 4).

$$\text{Stress concentration factor (K)}_t = \sigma_m / \sigma_o$$

$$K_t = 25.132/5.7489$$

$$K_t = 4.3716$$

Models	1-8 UNC Thread Published values of *Kt (2D)	Whitworth thread form of *Kt (3D)	ISO-Metric of *Kt	ACME Thread Form of *Kt(3D)	Buttress thread form*Kt (3D)
MODEL 1	7.63	7.015	6.641	4.371619	3.334031
MODEL 2	5.81	5.773	5.98	4.568464	2.700146
MODEL 3	5.71	5.67	5.68	2.516703	2.241218
MODEL 4	5.4	5.363	5.499	2.575923	2.015132
MODEL 5	4.61	4.481	4.704	2.182126	2.058545
MODEL 6	5.32	5.413	5.728	2.23223	2.303686
MODEL 7	3.54	3.558	3.927	2.067838	2.7955

Table 1: Buttress threaded model results.

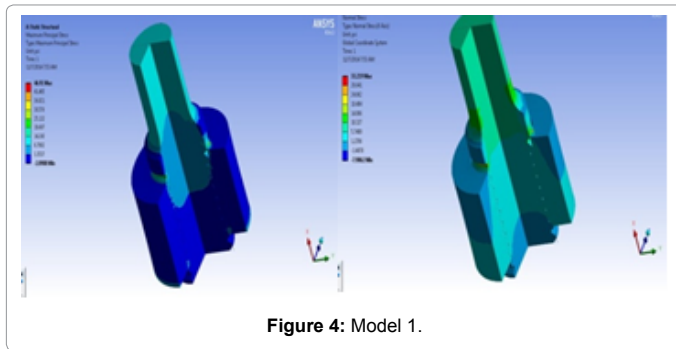


Figure 4: Model 1.

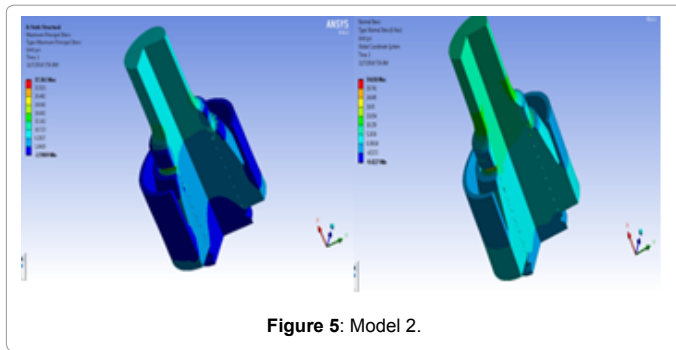


Figure 5: Model 2.

**Model 2:** The plane basic bolt and nut model is modified to find the stress concentration factor by adding groove to the face of the nut closes to the head of the bolt (Figure 5).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 24.042 / 5.2636$$

$$K_t = 4.5684$$

**Model 3:** The groove is added to the lower end of the bolt in addition to the groove of the bolt (Figure 6).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 24.334 / 9.669$$

$$K_t = 2.516$$

**Model 4:** A step is added to the nut with no grooves on the bolt and nut (Figure 7).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$= 18.83 / 7.31$$

$$K_t = 2.5759$$

**Model 5:** A step is added to the nut and a groove is added to the lower end of the bolt (Figure 8).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 19.743 / 9.0476$$

$$K_t = 2.1821$$

**Model 6:** A taper is added to the nut and a groove is added to the lower end of the bolt (Figure 9).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 20.272 / 9.0815$$

$$K_t = 2.232$$

**Model 7:** A step is added to the nut and a groove is added to the lower end of the bolt. The shank diameter of the bolt is reduced (Figure 10).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 15.997 / 7.7361$$

$$K_t = 2.0678$$

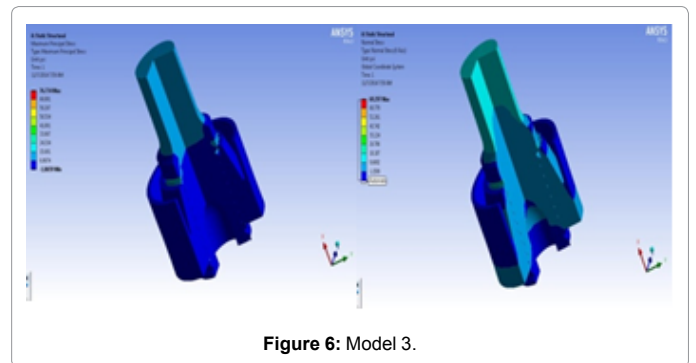


Figure 6: Model 3.

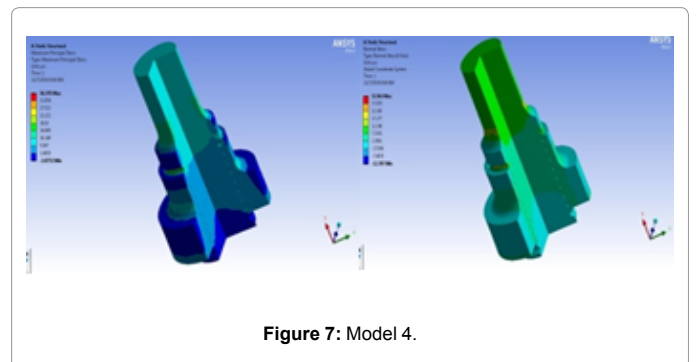


Figure 7: Model 4.

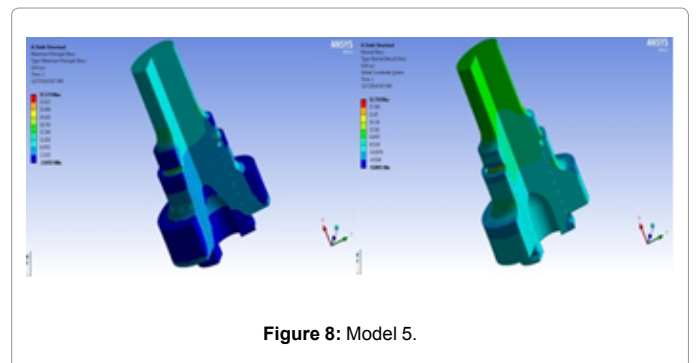


Figure 8: Model 5.



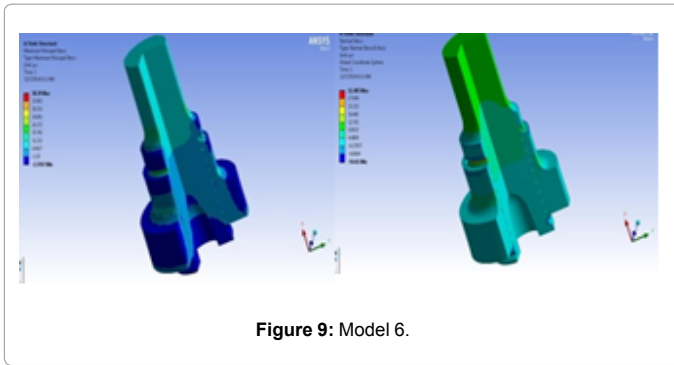


Figure 9: Model 6.

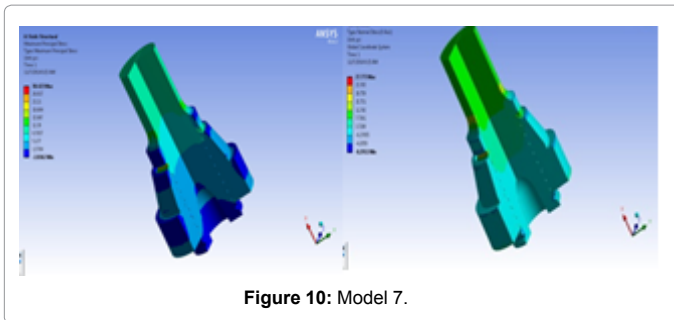


Figure 10: Model 7.

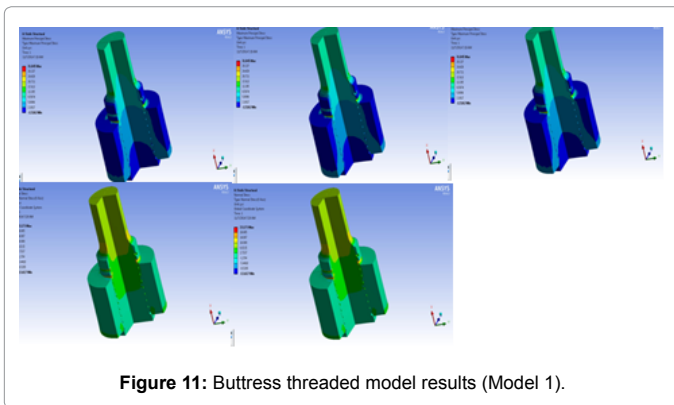


Figure 11: Butress threaded model results (Model 1).

### Butress threaded model results

**Model 1:** The basic model in this study used is a 23.4 mm diameter plan. Bolt with corresponding hexagonal nut with a pitch of 3.175 mm butress threads. The bolt and nut are made of high carbon steel and it has a Poisson's ratio of 0.29 with an angle of 45°. The load is equal to stress 10 psi is applied on the bolt shank to produce the stress concentration (Figure 11).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 20.721 / 6.215$$

$$K_t = 3.334$$

**Model 2:** The plane basic bolt and nut model is modified to find the stress concentration factor by adding groove to the face of the nut close to the head of the bolt (Figure 12).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 18.55 / 6.87$$

$$K_t = 2.70$$

**Model 3:** The groove is added to the lower end of the bolt in addition to the groove of the bolt (Figure 13).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 16.269 / 7.259$$

$$K_t = 2.241$$

**Model 4:** A step is added to the nut with no grooves on the bolt and nut (Figure 14).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 17.126 / 8.4987$$

$$K_t = 2.015$$

**Model 5:** A step is added to the nut and a groove is added to the lower end of the bolt (Figure 15).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 17.412 / 8.4584$$

$$K_t = 2.05$$

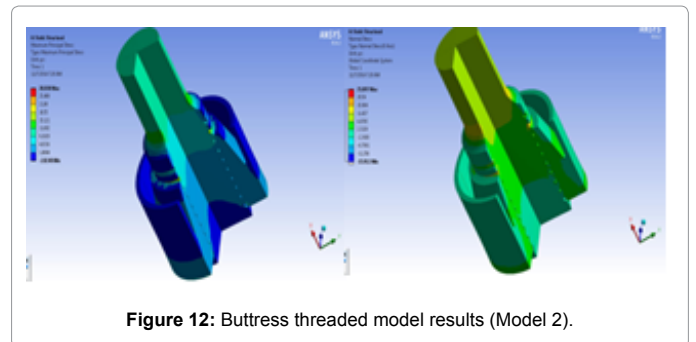


Figure 12: Butress threaded model results (Model 2).

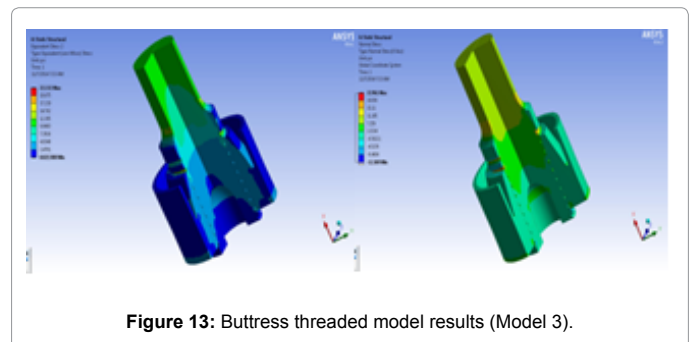


Figure 13: Butress threaded model results (Model 3).

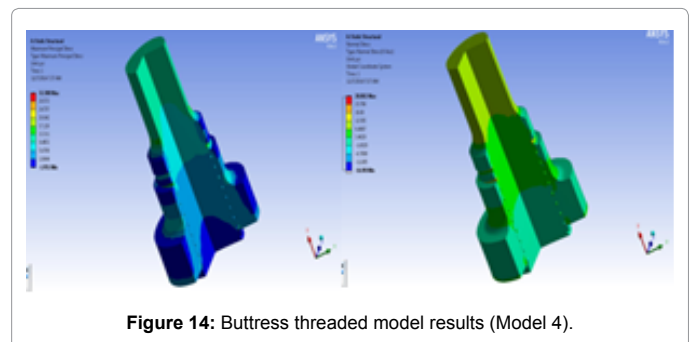


Figure 14: Butress threaded model results (Model 4).

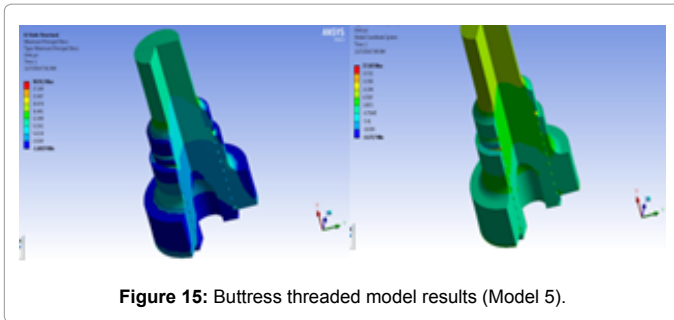


Figure 15: Butress threaded model results (Model 5).

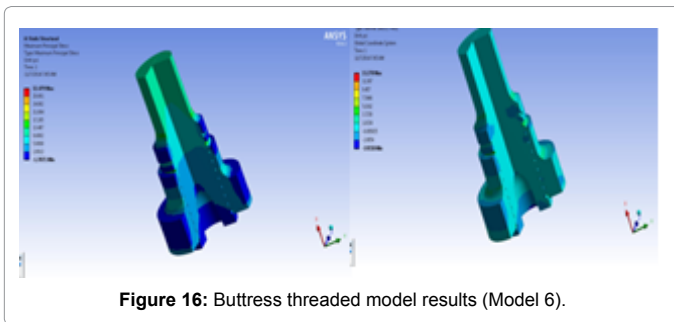


Figure 16: Butress threaded model results (Model 6).

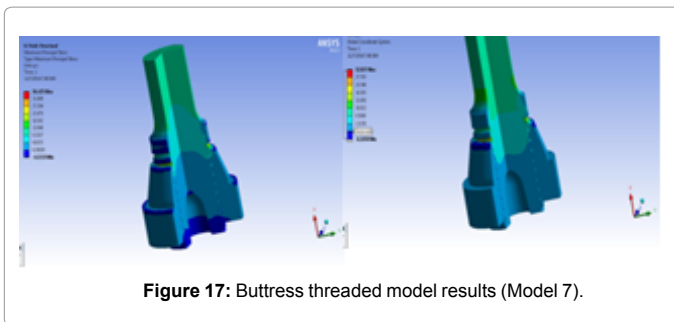


Figure 17: Butress threaded model results (Model 7).

**Model 6:** A taper is added to the nut and a groove is added to the lower end of the bolt (Figure 16).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 17.385 / 7.5466$$

$$K_t = 2.30$$

**Model 7:** A step is added to the nut and a groove is added to the lower end of the bolt. The shank diameter of the bolt is reduced (Figure 17).

$$\text{Stress concentration factor } (K_t) = \sigma_m / \sigma_o$$

$$K_t = 18.3636 / 5.686$$

$$K_t = 2.7955$$

## Results and Discussion

The main goal of this work was to investigate the effect of relative changes in the geometries of the nut and bolt on the stress concentration for the assembly [11]. Any changes in the geometries will increase the cost of the fastener system; the specific values for the stress concentration factors will change as the size of the nut and bolt changes [12-15].

Simulation work has been done to analyze the stress concentration

on various models using 3-D analysis. For this “ANSYS” has been used as simulation tool which works on finite element method [14]. The main aim of this work is to find the stress concentration factor values ( $K_t$ ) for nut and bolt connectors of 1-8 UNC thread form by using 3-D modeling and compare the relative  $K_t$  values with the standard values from 2-D modeling and to find the optimum nut and bolt connector model among the various models of 1-8 UNC thread form. This material is a low-alloy, high-carbon steel commonly used in aerospace fasteners and has an elastic modulus of 205 GPa and Poisson’s ratio of 0.29. The load applied is sufficient to produce a nominal stress of 10 psi in the shank of the bolt [15-18].

This study is to show quantitative comparison of effect of various proposed design modifications in the literature [3] on the stress concentration factor published previously to the work which has been done with ANSYS. Validation was also done for the same thread forms proposed in the literature (Figure 18).

### Model graph for different thread forms

Machines are assembled from individual parts. The parts may need to be joined sometimes in permanent or temporary connections. The temporary connections or joints are normally made by using screws bolts or nuts and bolts (Table 2). The temporary joints render the convenience of disconnection as frequently as required for such purposes as inspection, repairs, adjustment or replacement. The most important feature of the screws or bolt is that they are standardized and are available as standard part ready to use. The threaded fasteners are used to hold two parts. The fastener consists of two parts. An externally threaded member whose length is normally greater than the diameter carries threads over entire or part length from end. The other end of this member is larger in area (you can assume that the end is upset). This part is inserted in coaxial holes in two parts taking care that the “upset” or larger end is greater in dimension than the holes so that it does not pass through the hole. The end that comes out of the holes carries thread upon which another internally threaded part is wound (Figure 19).

The main goal of this work was:

- To find the effect of relative changes in the geometries of the nut and bolt on the stress concentration for the assembly on different thread (Butress Thread Form) after validating the results of published journal where he used 1-8 UNC Thread Form.
- To find the optimum design for bolt and nut connector.

With this in mind, the following conclusions can be drawn from the study

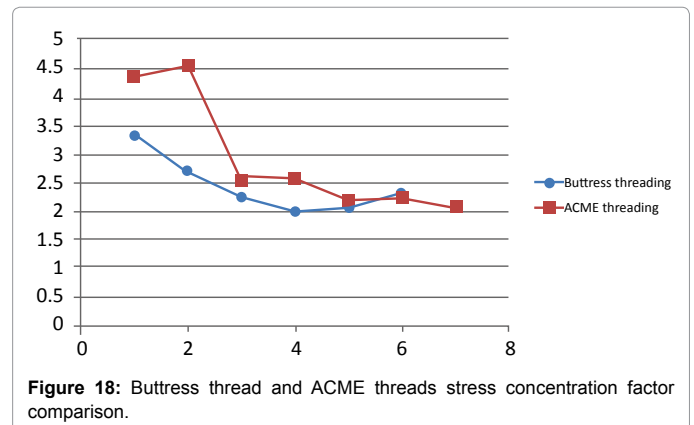


Figure 18: Butress thread and ACME threads stress concentration factor comparison.

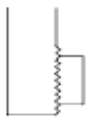
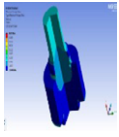
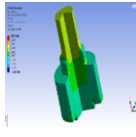

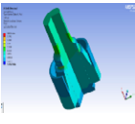
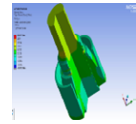
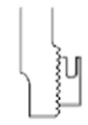
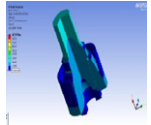
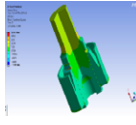
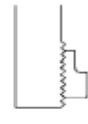
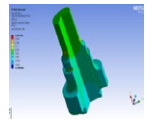
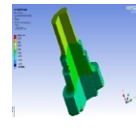
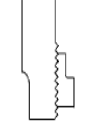
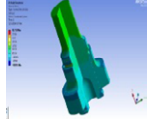
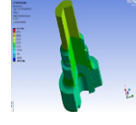

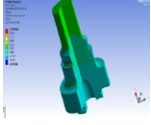
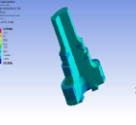

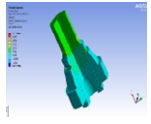
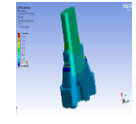
Design Description	Sketch 2D Model	3D Model For ACME Thread	Stress concentration factor	3D Model For Buttress Thread	Stress concentration factor
<b>Model-1</b> This is the base model which features a 1 – inch diameter bolt with corresponding nut.			$K_t=4.3716$		$K_t=3.334$
<b>Model-2</b> A groove is added to the face of the nut closer to the head of the bolt.			$K_t=4.5684$		$K_t=2.70$
<b>Model-3</b> A groove is added to the lower end of the bolt, in addition to the groove on the nut.			$K_t=2.516$		$K_t=2.241$
<b>Model-4</b> A step is added to the nut with no grooves on the bolt or the nut.			$K_t=2.5759$		$K_t=2.015$
<b>Model-5</b> A step added to the nut and a groove is added to the lower end of the bolt.			$K_t=2.1821$		$K_t=2.05$
<b>Model-6</b> A step is added to the nut and a groove is added to the lower end of the bolt. the shank diameter of the bolt is reduced.			$K_t=2.232$		$K_t=2.30$
<b>Model-7</b> A taper is added to the nut and a groove is added to the lower end of the bolt.			$K_t=2.0678$		$K_t=2.7955$

Table 2: 2D and 3D results and comparison; Results from various thread forms compare with validation.

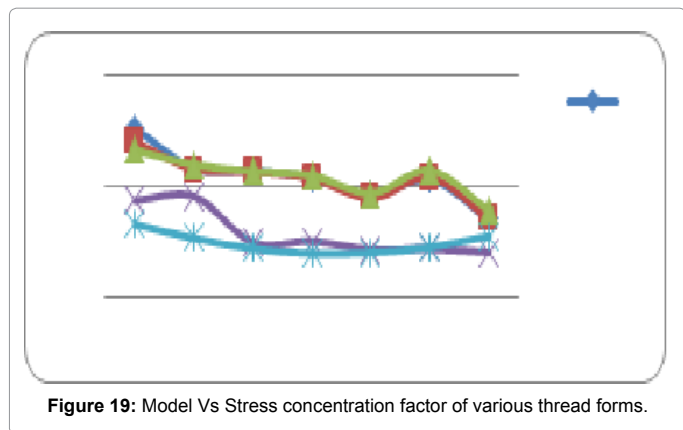


Figure 19: Model Vs Stress concentration factor of various thread forms.

1. Stress concentration factor is maximum at first thread and minimum at fourth thread. It is observed that in an individual thread

the highest stress occurred below the deepest point of the thread root.

2. The validated results are found to be reasonably accurate when compared with the published values. The minor error is due change in software used. Here we used Ansys whereas published Journal used Algor for FEM.

3. The stress concentration Factor has been reduced for different models of designed Buttress thread form with respect to base model. So, buttress Thread is effective when compared to 1-8 UNC Thread and it can as well use for structural works.

4. Among the three models done for buttress thread- model in which, a step is added to the nut with no grooves on the bolt and nut has less stress concentration factor. So, Model 4 is best design.

5. Essentially all of the stress reduction methods based on a modification of the threaded section and nut remove material. Therefore, even though the stress concentration changes due to changes in the nut and thread end appear to be less dramatic than those resulting

from reducing the shank diameter, the nut and thread changes may be more efficient from a material utilization and cost standpoint.

## Conclusion

3-D analysis gives exact values than 2-D analysis, we can consider the stress concentration factor ( $K_t$ ) values obtained in this study as more exact and standard values for 1-8 UNC, ACME and Buttress threads in nominal dimensions. And these  $K_t$  values for nominal dimensions can be used in design modifications.

The stress concentration factor is less for Buttress thread form, compared to the other forms of threads. So, Buttress thread form is effective when compared to 1-8 UNC thread form and ACME thread form and it can be used for structural works. Model 4 is the optimum model for nominal dimensions. This will help in reducing the material required for bolt in the design and cost consideration.

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