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DESIGN AND FINITE ELEMENT INVESTIGATION OF BOLT AND NUT FOR FASTENING STRESS REDUCTION

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Abstract: The study, design and finite element investigation of bolt and nut for fastening stress reduction were successfully investigated. researchers created a bolt of M6× H27 with a hexagonal head of 12mm, and a hexagonal nut of M12× H10 both having a right hand ANSI metric thread profile, with assigned material being Steel, High Strength, Low Alloy to reduce wear and tear using Auto Desk inventor. The created model was analyzed using FEA software. The bolt was first subjected to fastening moment of 240 N mm, with the nut being subjected to fixed constraints. Thereafter, the nut was subjected to fastening moment of 240 N mm with the bolt subjected to fixed constraints. Results showed that the ultimate tensile strength of assigned material reduced from 540 MPa to 448 MPa when fastening moment was applied at bolt head and Nut side respectively. This suggested that failure due to tensile stress would be predominant when fastening at the Nut side. Also, maximum fastening stresses were observed to be 7.109 MPa and 6.66822 MPa when fastening moment was applied at bolt head and Nut side respectively. This suggested that fastening at the Nut side would reduce fastening stress or thread wear. In addition, the maximum displacements were observed to be 0.00190463 mm and 0.00204884 mm when fastening moment was applied at bolt head and Nut side respectively. This indicated that fastening at the Nut side will give lesser revolution to achieve a tightened bolt and nut joint. Furthermore, the maximum contact pressures were observed to be 10.7241 MPa and 10.9924 MPa when fastening moment was applied at bolt head and Nut side respectively. This indicated that slippage is minimized at nut side fastening with high induce stress at the bolt shank, as the study revealed. The researchers made the following recommendations: Fastening at nut side should be adopted to reduce stress and maximize displacement/ advancement, bolt and nut design materials must have higher tensile strength rather than compressive strength or yield strength, since failure due to tensile stress is predominant, etc.

Index Terms---- Stress, Bolt and nut, Fastening, Finite element analysis, Turning moment, Constraints

INTRODUCTION

Bolt is a mechanical fastener consisting of a head and a cylindrical body with screw threads along a portion of its length. Nut is the female member of the pair, with internal threads to match those of the bolt. Washers are often used to spread stress and prevent crushing. According to Teltumade and Yenarkar (2013) bolted joint is a very popular method of fastening components together. The prime reason for selecting bolts as opposed to welding or rivets are that the connection can be easily released allowing disassembly, maintenance and inspection. It has a various application for mechanical joint as can be seen in pressure vessels, flanged pipes, spacecraft, ship, internal combustion engine, automobile, or oilrig, etc and for civil structure.

Finite element analysis here involves the use of simulation to predict and understand how bolt and nut thread wear stress can be reduced under severe physical applications of turning moment and constraints. Finite element uses finite element method, which is a numerical technique that cuts the structure of the bolt and nut into several elements and then reconnects the elements at point called nodes.

Kevin et al., (2008) stated that the critical component of designing bolted joints does not only involve the number of bolts, the size , and the placement of them but also covers the appropriate wear stress for the bolt and the torque that must be applied to achieve the desired wear stress. Fastening stress represents the stress that gradually drives the bolt and nut and could remove or wear the threads when excessive. Hence, the paper aimed at studying design and finite element investigation of bolt and nut for fastening stress reduction.

METHODOLOGY

The researchers created a bolt of M6× H27 with a hexagonal head of 12mm, and a hexagonal nut of M12× H10 both having a right hand ANSI metric thread profile, with assigned material being Steel, High Strength, Low Alloy to reduce wear and tear as shown in **Figure 1.0.** The model was prepared with the aid of inventor software and imported to; Finite Element Analysis software where wear stress was predicted. The bolt was first subjected to fastening moment of 240 N mm, with the nut being subjected to fixed constraints. Similarly, the nut was subjected to fastening moment of 240 N mm with the bolt being subjected to fixed constraints. Results were generated and compared to determine the best condition for the two situations.

MESHING

Meshing was used to divide the bolt and nut model into section with nodes of 9305 and elements of 5675. Increasing the number of elements, means more computations and more mathematical formula for the element. Hence, the more precise the results would be. Mesh settings used is shown below in table **1.0.**

TABLES AND FIGURES

Table 1.0: Mesh settings and general objective and settings

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	8/31/2023, 5:35 AM
Detect and Eliminate Rigid Body Modes	No
Avg. Element Size (fraction of model diameter)	0.1
Min. Element Size (fraction of avg. size)	0.2
Grading Factor	1.5
Max. Turn Angle	60 deg
Create Curved Mesh Elements	Yes
Part Number	NUT AND BOLT
Designer	EWURUM TENNISON
Cost	$1.00
Date Created	8/31/2023

Table 2.0: Physical and Material Properties

Material	Steel, High Strength, Low Alloy	
Density	7.85 g/cm^3	
Mass	0.0165836 kg	
Area	1637.12 mm^2	
Volume	2112.57 mm^3	
Center of Gravity	x=9.90606 mm y=0 mm z=0 mm	
Name	Stainless Steel	
General	Mass Density	8 g/cm^3
	Yield Strength	250 MPa
	Ultimate Tensile Strength	540 MPa
Stress	Young's Modulus	193 GPa
	Poisson's Ratio	0.3 ul
	Shear Modulus	74.2308 GPa
Part Name(s)	NUT AND BOLT	

Table 3.0: Model Operating Conditions

Load Type	Moment
Magnitude	240.000 N mm
Vector X	-240.000 N mm
Vector Y	0.000 N mm
Vector Z	0.000 N mm

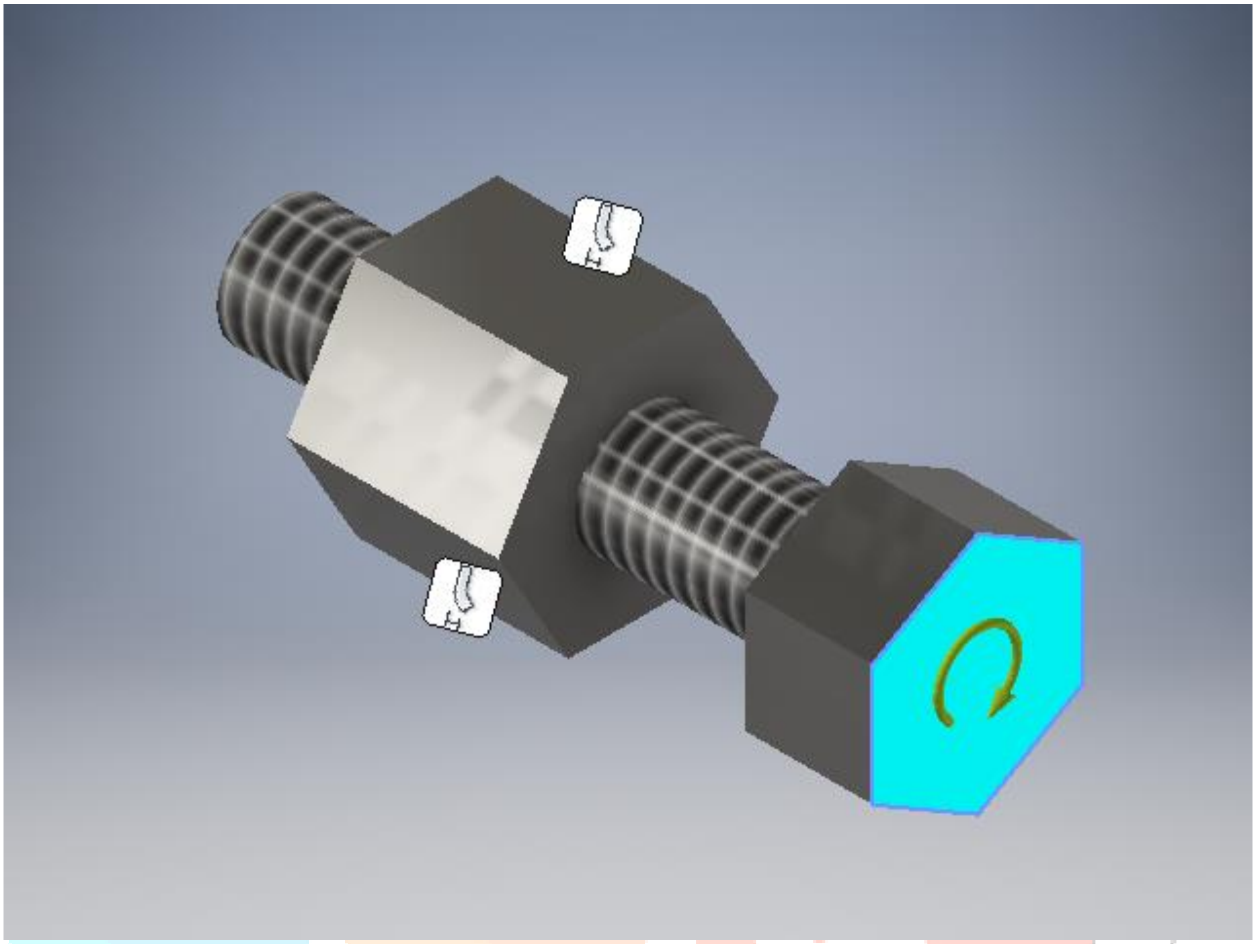

Fig 1.0(a): Bolt and Nut Assembly with Constraints and Fastening Moment

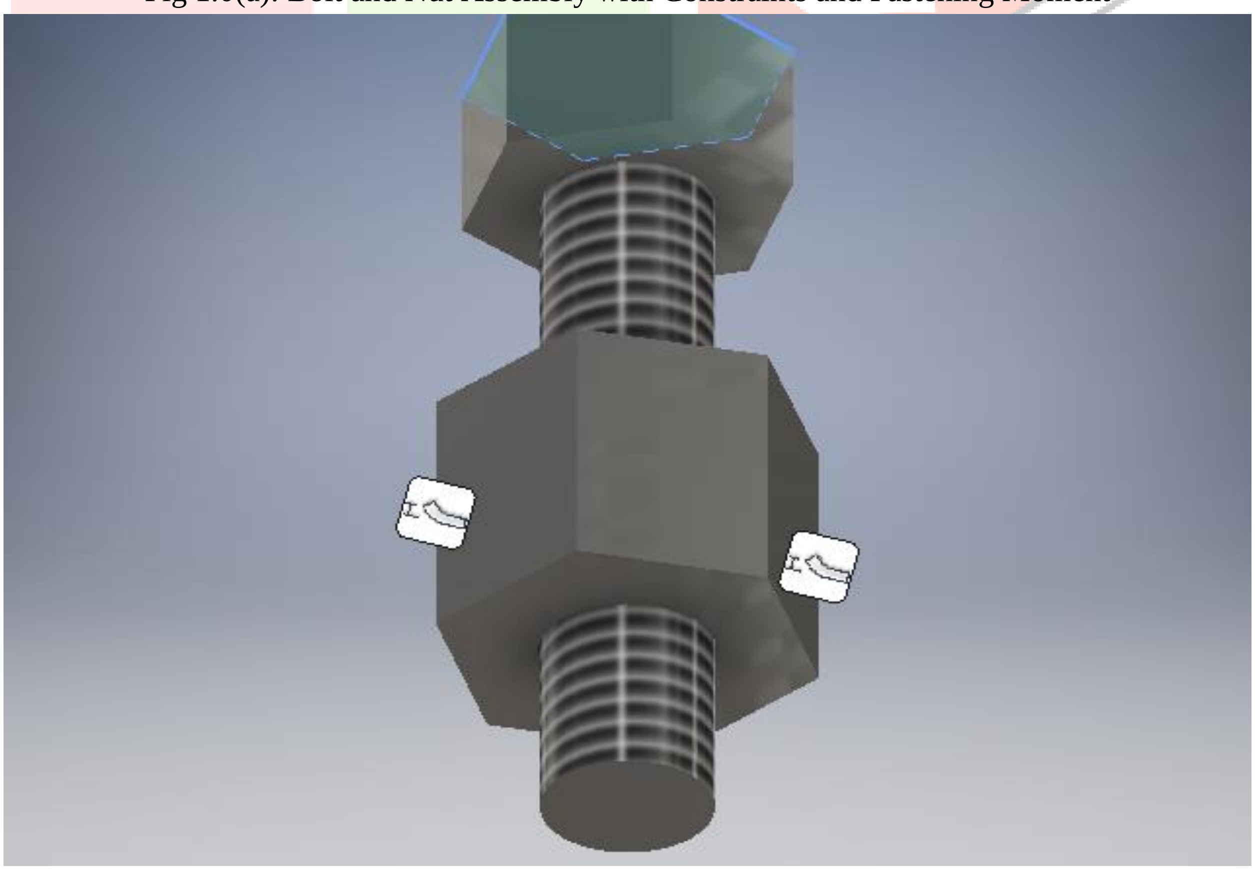

Fig 1.0(b): Bolt and Nut Assembly with Constraints and Fastening Moment

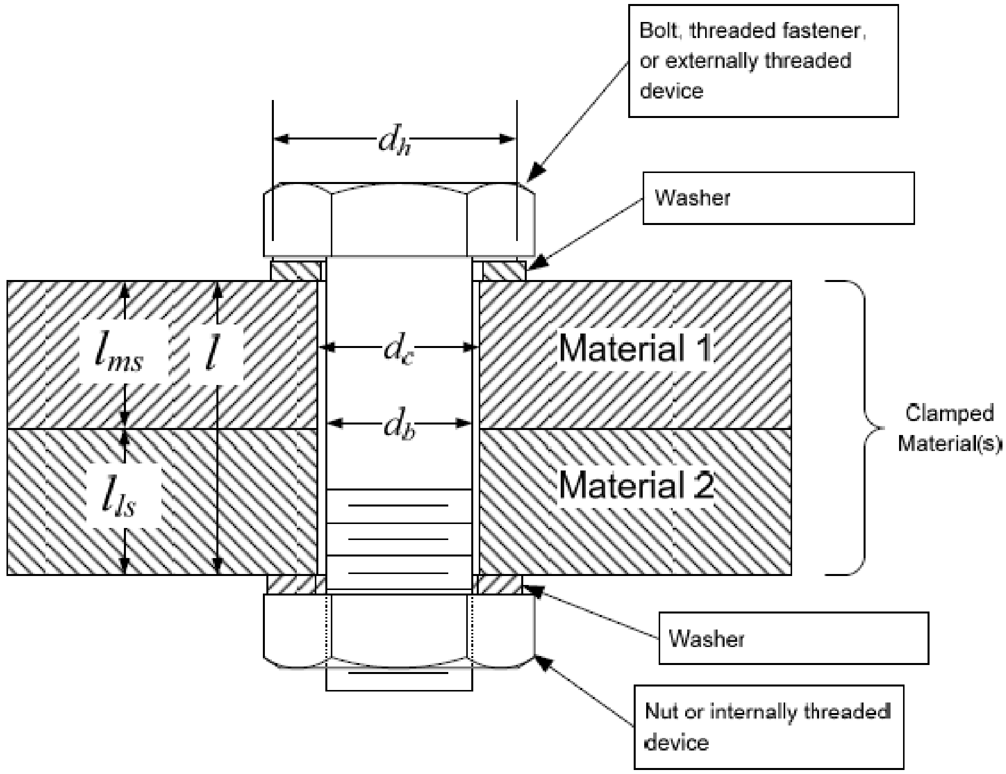

Fig 2.0: Bolt and Nut Assembly (source: Kevin et al., 2008)

EQUATIONS

The stress components in an element are given as below.

$(\sigma_x)_n = \frac{E}{(1+v)(1-2v)}[(1-v)a_n + ve_n]$ ..... (1.0) (Onyenobi et al., 2022)

$$(\sigma_y)_n = \frac{E}{(1+v)(1-2v)}[va_n + (1-v)e_n] \;..... (1.1)$$

$$(\tau_{xy})_n = \frac{E}{2(1+v)}(b_n + d_n) \;..... (1.2)$$

$$v = Poisson's\ ratio, E = modulus\ of\ elasticity$$

The displacement field is shown below.

$$a_n = \frac{\partial u_n}{\partial x} \;... (1.3)$$

$$e_n = \frac{\partial v_n}{\partial y} \;... (1.4)$$

$$b_n + d_n = \frac{\partial u_n}{\partial y} + \frac{\partial v_n}{\partial x} \;.... (1.5)$$

$$v\ and\ u\ are\ velocity\ components\ of\ x\ and\ y$$

The principal strains are given below

$e_x = \frac{1}{E}\left[\sigma_x - \frac{1}{m}(\sigma_y + \sigma_z)\right] \ldots..(1.5.1)$ (Rajput, 2008).

$$e_y = \frac{1}{E}\left[\sigma_y - \frac{1}{m}(\sigma_x + \sigma_z)\right] \ldots..(1.5.2)$$

$$e_z = \frac{1}{E}\left[\sigma_z - \frac{1}{m}(\sigma_x + \sigma_y)\right] \ldots..(1.5.3)$$

Von Mises Stress can be given as below.

$$\text{Von} - \text{mises stress} = \sqrt{\sigma_x{}^2 - \sigma_x\sigma_y + \sigma_y{}^2} \ldots.(1.5.4)$$

RESULTS

The following results were gotten when the fastening moment was applied at the bolt head.

Table 4.0: Reaction Force and Moment on Constraints

Constraint Name	**Reaction Force**		**Reaction Moment**	
	Magnitude	**Component (X,Y,Z)**	**Magnitude**	**Component (X,Y,Z)**
Fixed Constraint:1	0 N	0 N	0.244125 N m	0.244125 N m
		0 N		0 N m
		0 N		0 N m

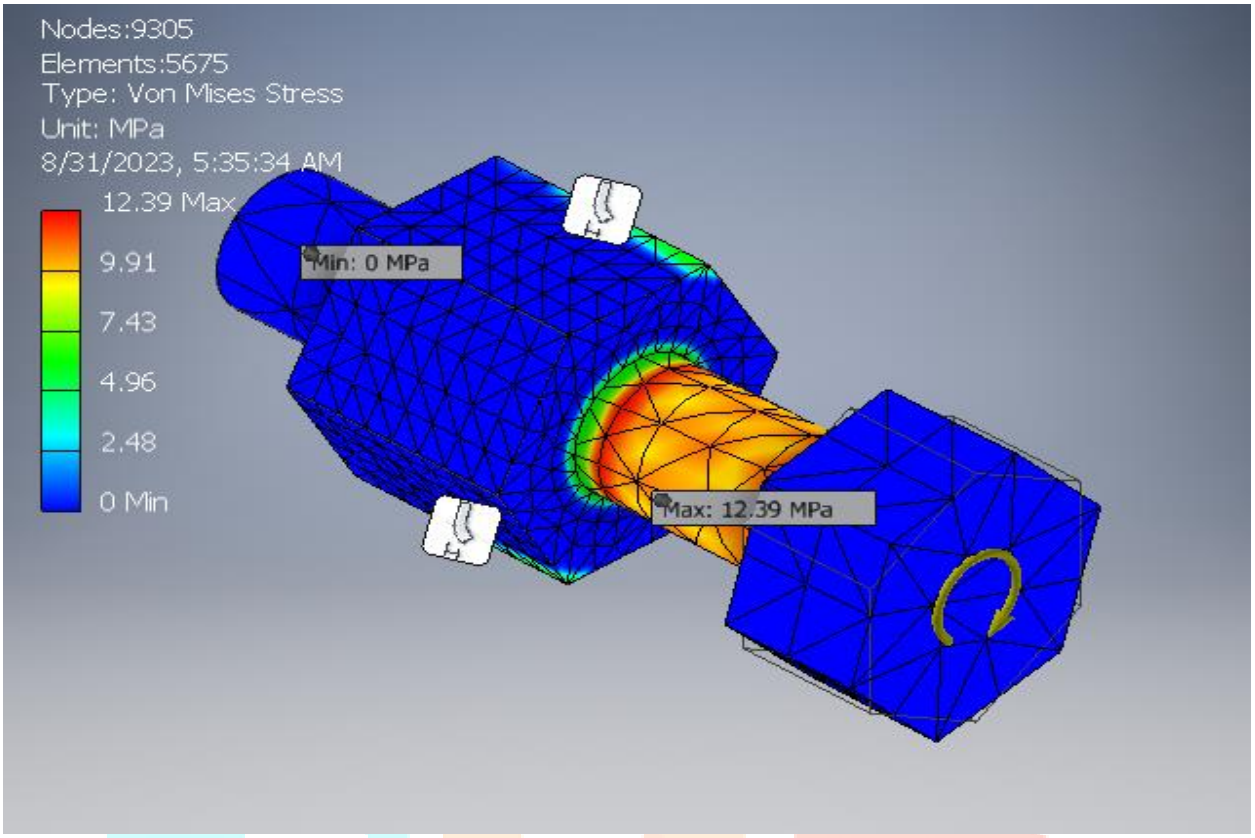

Fig3.0. Von Mises Stress

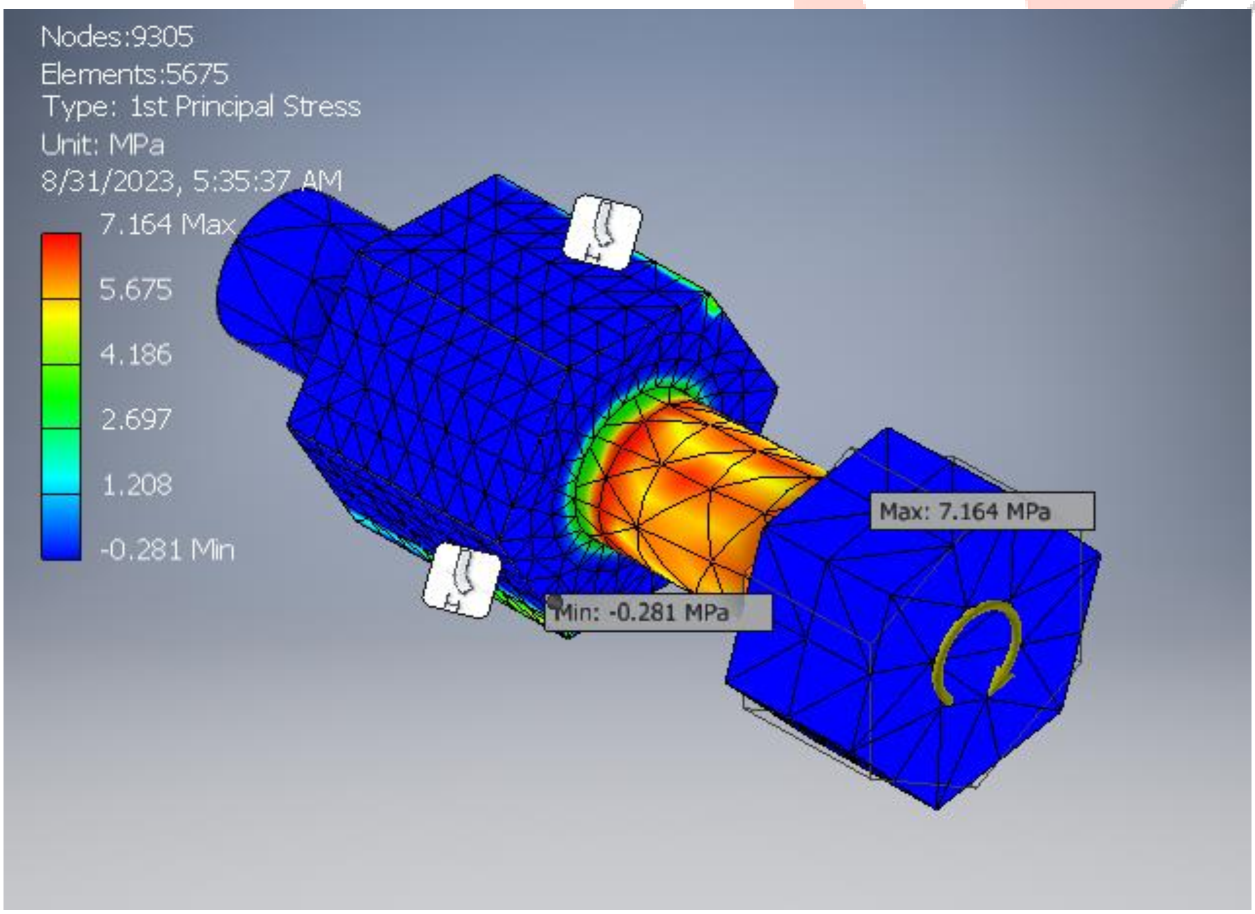

Fig4.0. 1st Principal Stress

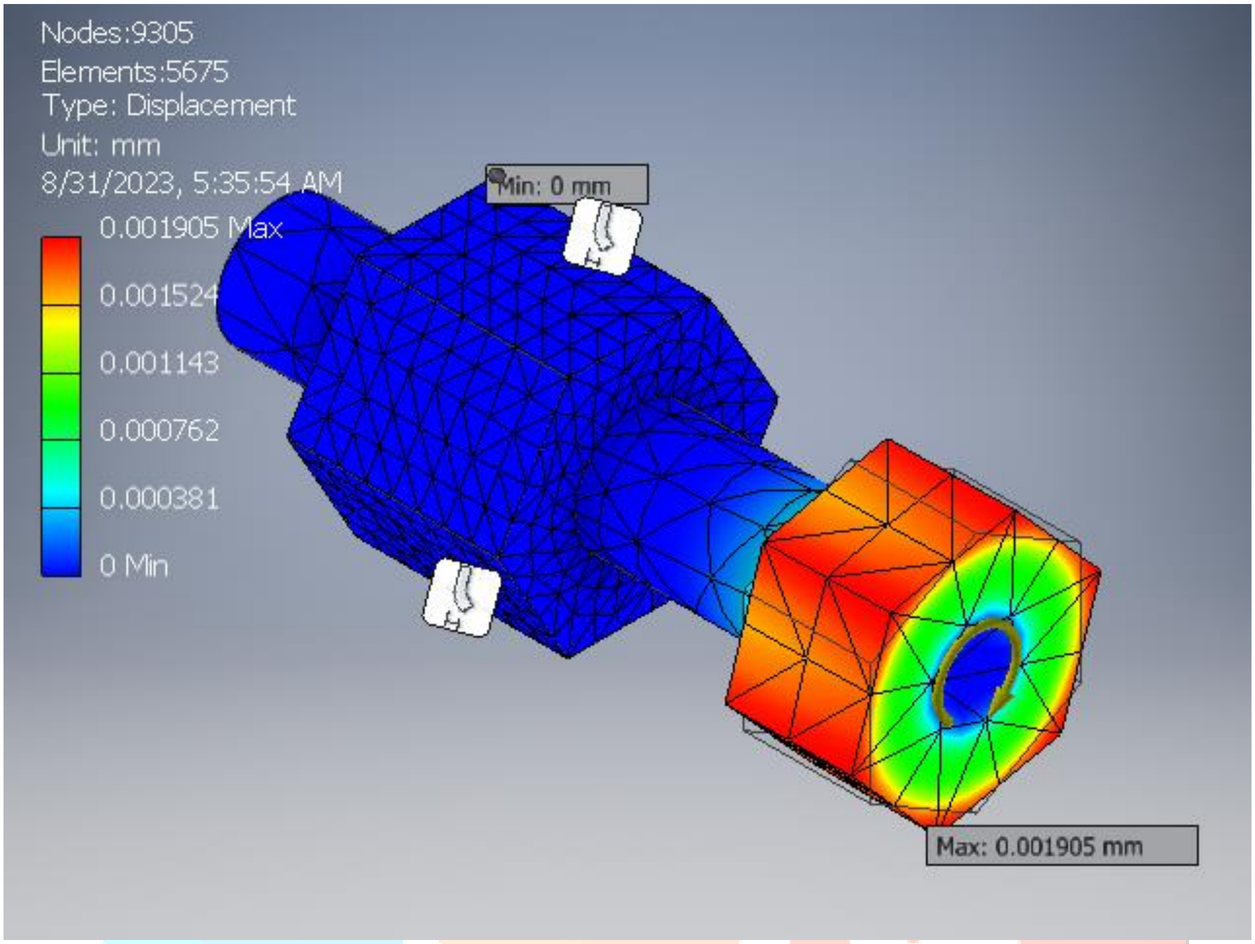

Fig5.0. Displacement

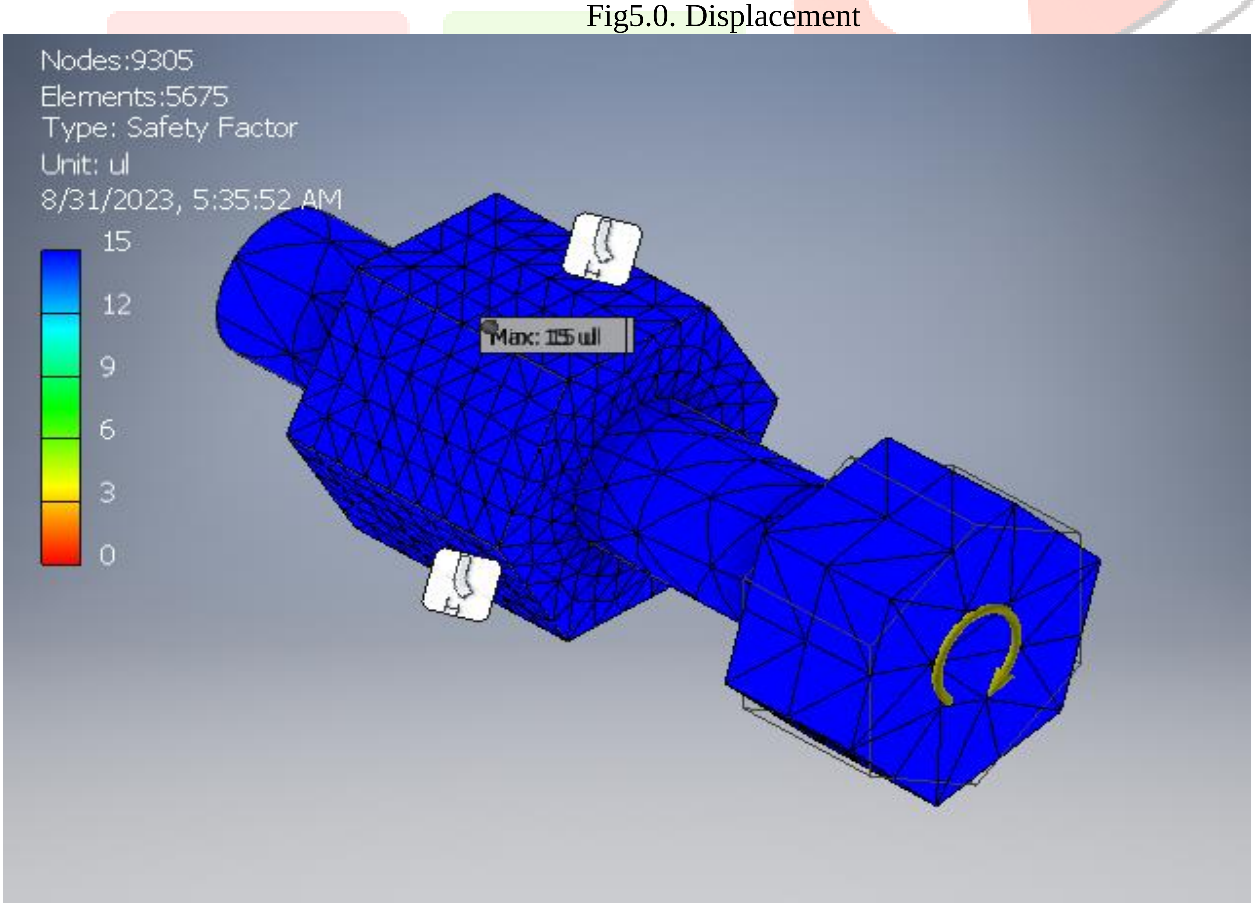

Fig5.0. Safety Factor

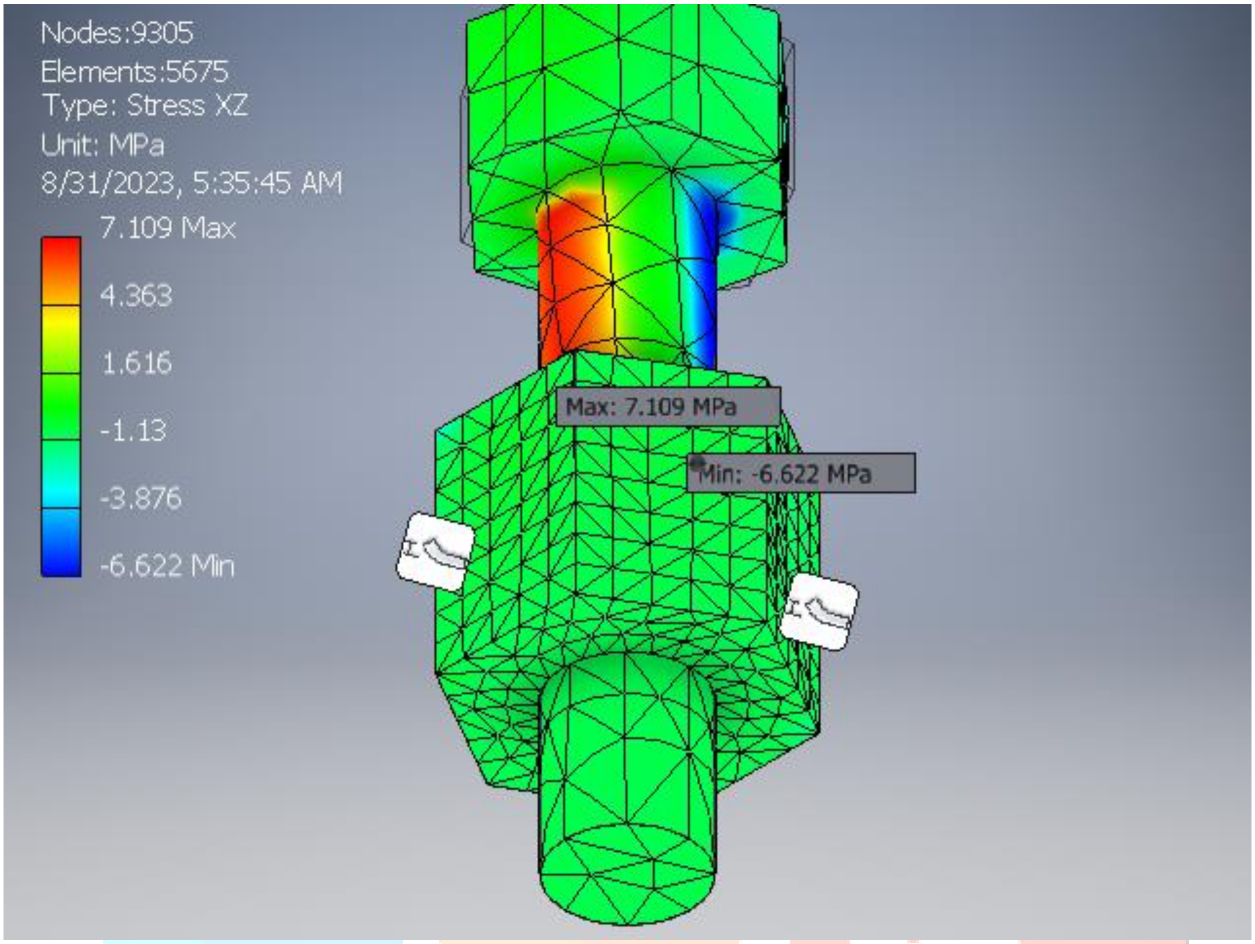

Fig6.0. Maximum Fastening Stress

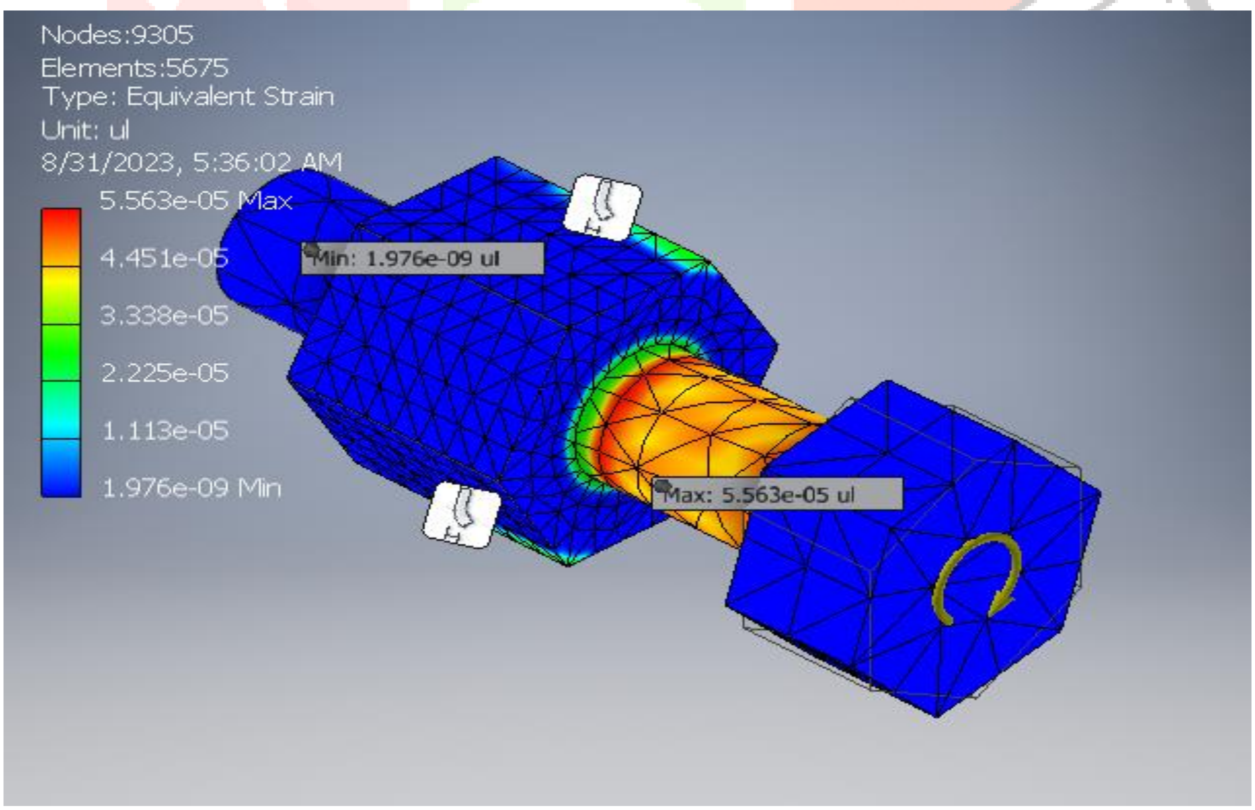

Fig7.0. Maximum Fastening Strain

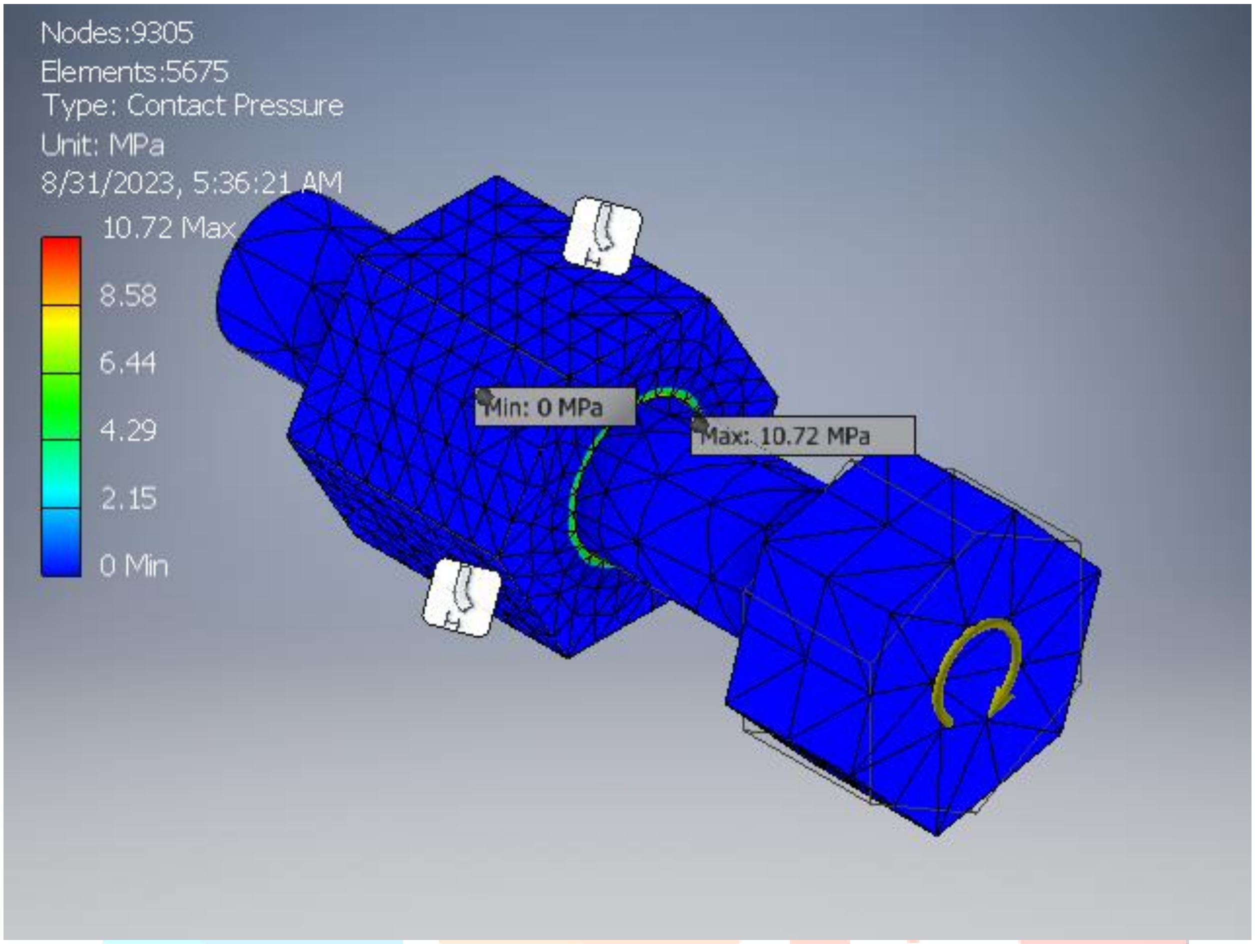

Fig8.0. Maximum Contact Pressure

Table 5.0: Output Summary

Name	Minimum	Maximum
Volume	2112.57 mm^3	
Mass	0.0169005 kg	
Von Mises Stress	0.000403007 MPa	12.3887 MPa
1st Principal Stress	-0.281274 MPa	7.16351 MPa
3rd Principal Stress	-7.20451 MPa	0.399612 MPa
Displacement	0 mm	0.00190463 mm
Safety Factor	15 ul	15 ul
Stress XX	-1.6188 MPa	1.52825 MPa
Stress XY	-6.80131 MPa	6.81892 MPa
Stress XZ	-6.6221 MPa	7.10874 MPa
Stress YY	-2.23439 MPa	5.34702 MPa
Stress YZ	-1.62168 MPa	2.43659 MPa
Stress ZZ	-5.95173 MPa	1.88875 MPa
X Displacement	-0.00000776802 mm	0.00000720736 mm
Y Displacement	-0.00164821 mm	0.00164913 mm
Z Displacement	-0.0019027 mm	0.00190463 mm

Equivalent Strain	0.00000000197618 ul	0.0000556317 ul
1st Principal Strain	0.00000000119594 ul	0.0000478141 ul
3rd Principal Strain	-0.0000485345 ul	-0.000000000631817 ul
Strain XX	-0.00000526186 ul	0.00000415731 ul
Strain XY	-0.0000458119 ul	0.0000459306 ul
Strain XZ	-0.0000446048 ul	0.0000478827 ul
Strain YY	-0.0000145925 ul	0.0000267183 ul
Strain YZ	-0.0000109233 ul	0.0000164123 ul
Strain ZZ	-0.0000295334 ul	0.0000131799 ul
Contact Pressure	0 MPa	10.7241 MPa
Contact Pressure X	-0.469201 MPa	0.337988 MPa
Contact Pressure Y	-10.2553 MPa	10.3689 MPa
Contact Pressure Z	-9.3786 MPa	10.0843 MPa

THE FOLLOWING RESULTS WERE GOTTEN WHEN THE FASTENING MOMENT WAS APPLIED AT THE NUT SIDE

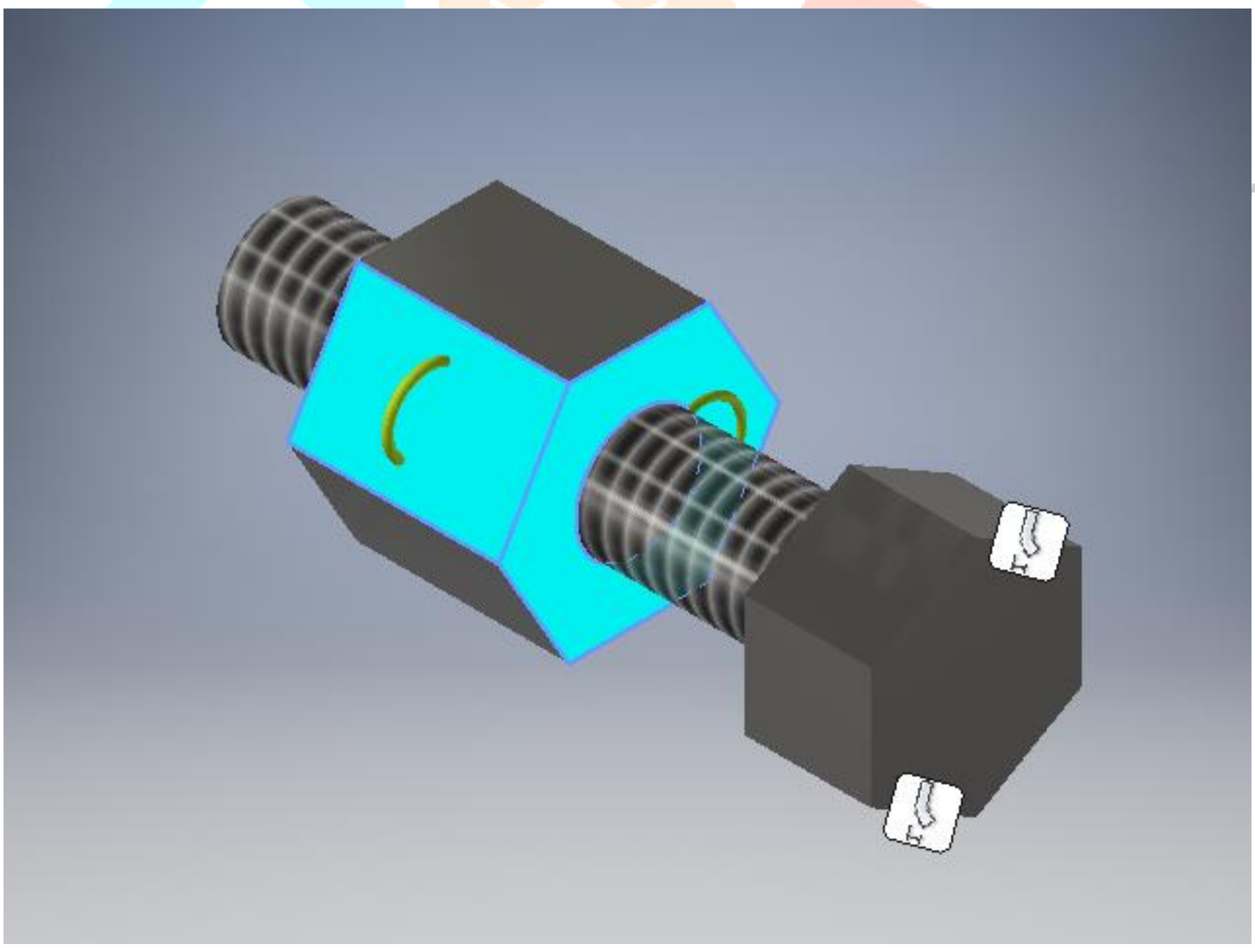

Fig9.0. Bolt and Nut Assembly with Constraints and Fastening Moment on Nut

Table 6.0: Reaction Force and Moment on Constraints

Constraint Name	Reaction Force		Reaction Moment	
	Magnitude	Component (X,Y,Z)	Magnitude	Component (X,Y,Z)
Fixed Constraint:1	0 N	0 N	0.234807 N m	0.234807 N m
		0 N		0 N m
		0 N		0.000193402 N m

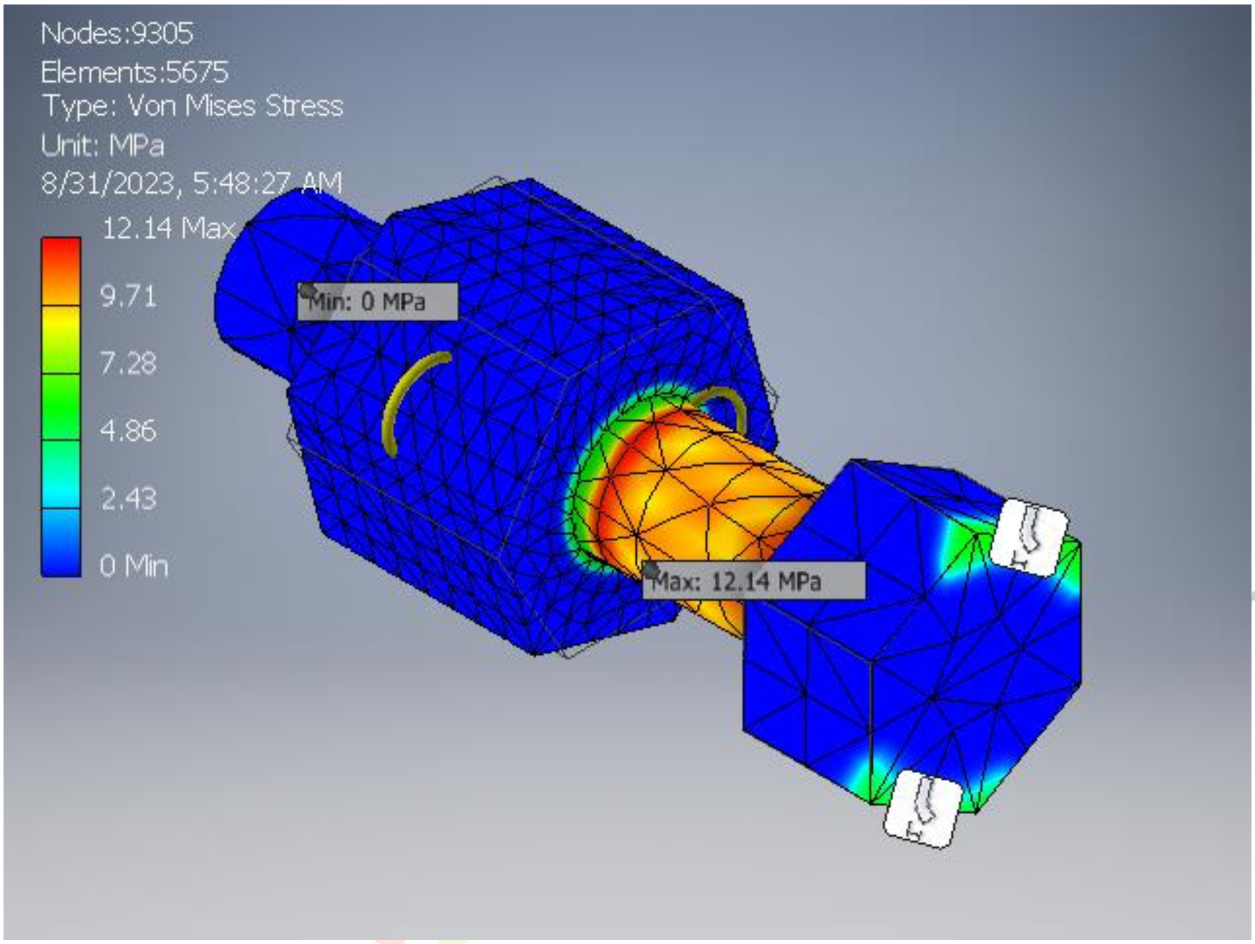

Fig10.0. Von Mises Stress

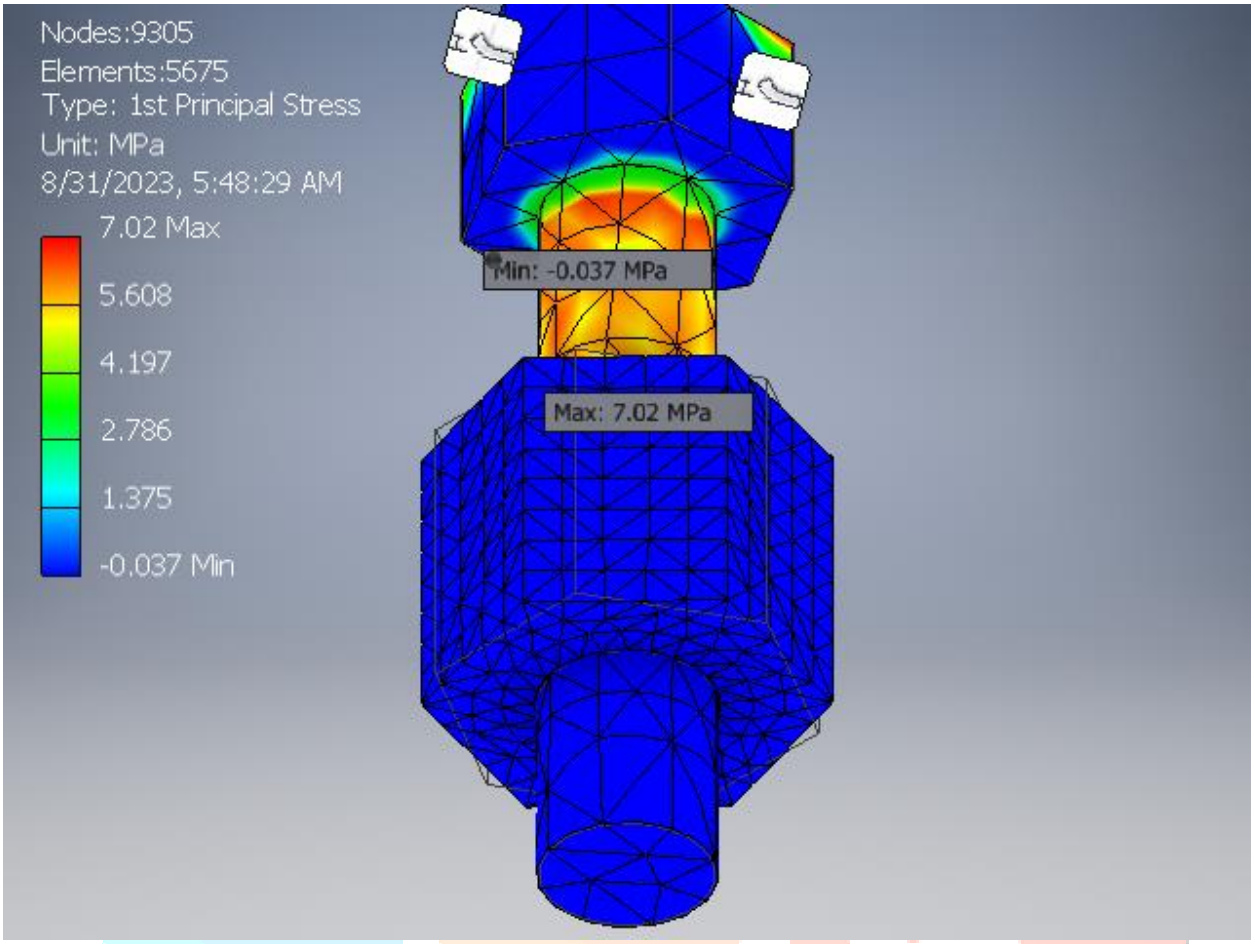

Fig11.0. 1ST Principal Stress

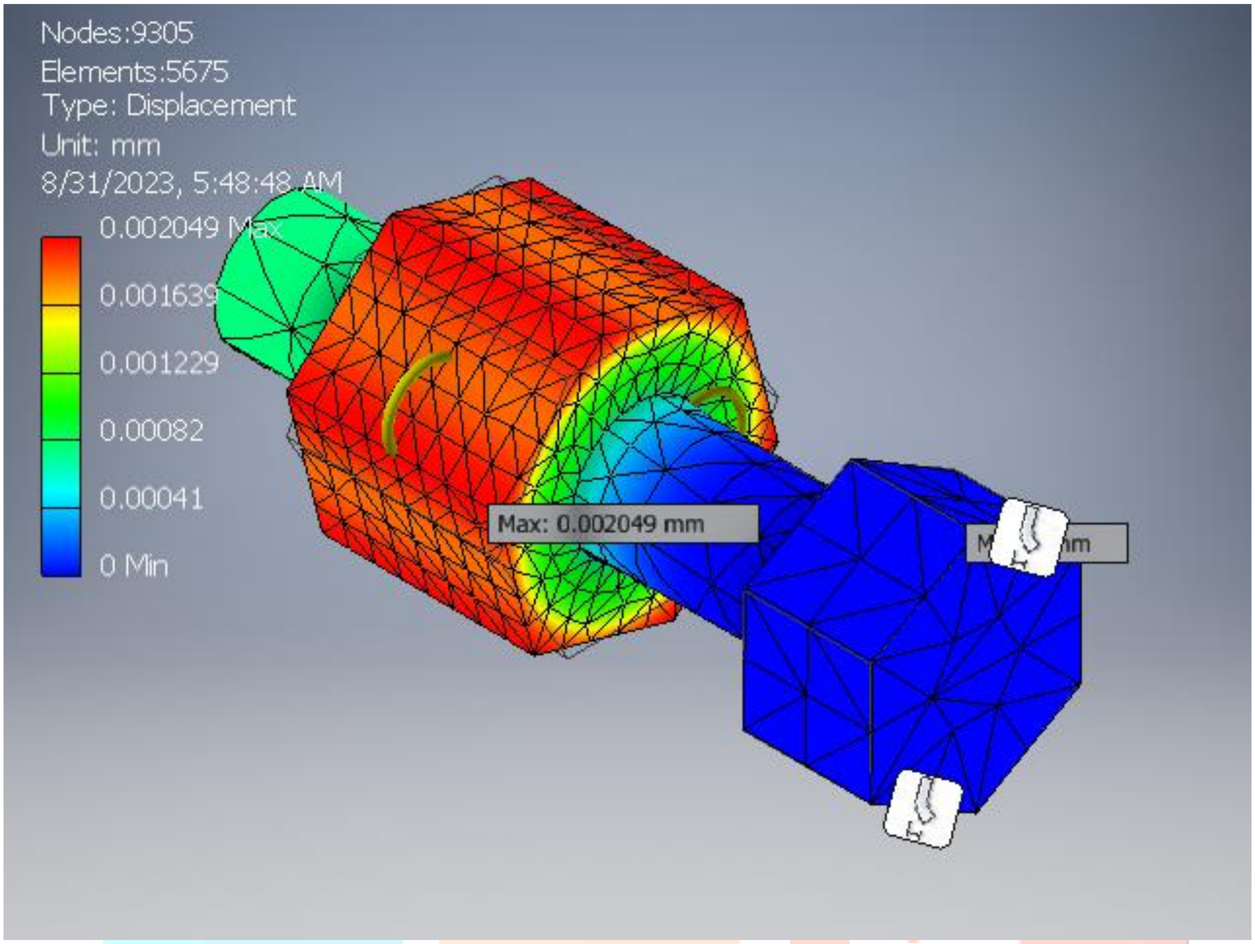

Fig12.0. Displacement

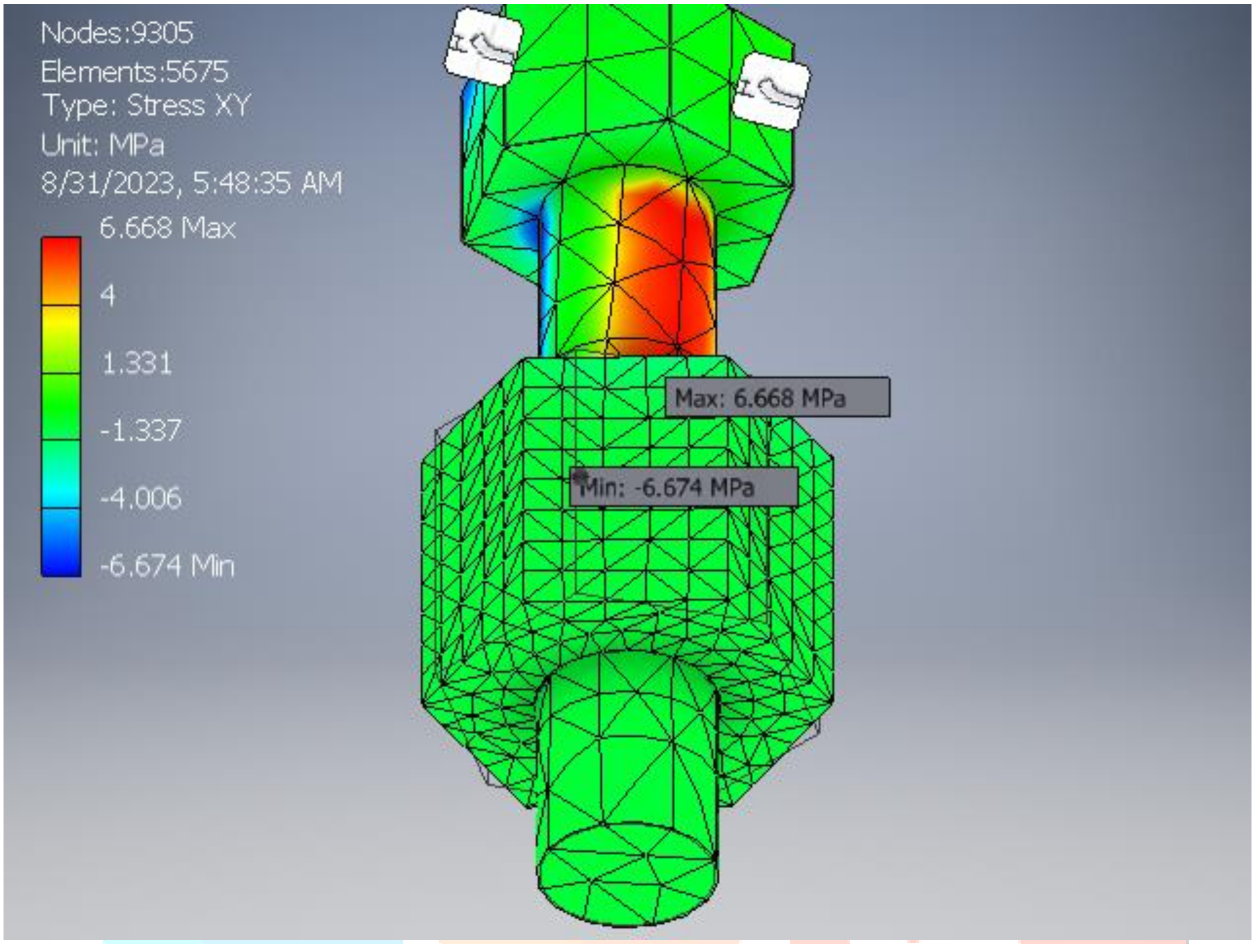

Fig13.0. Maximum Fastening Stress

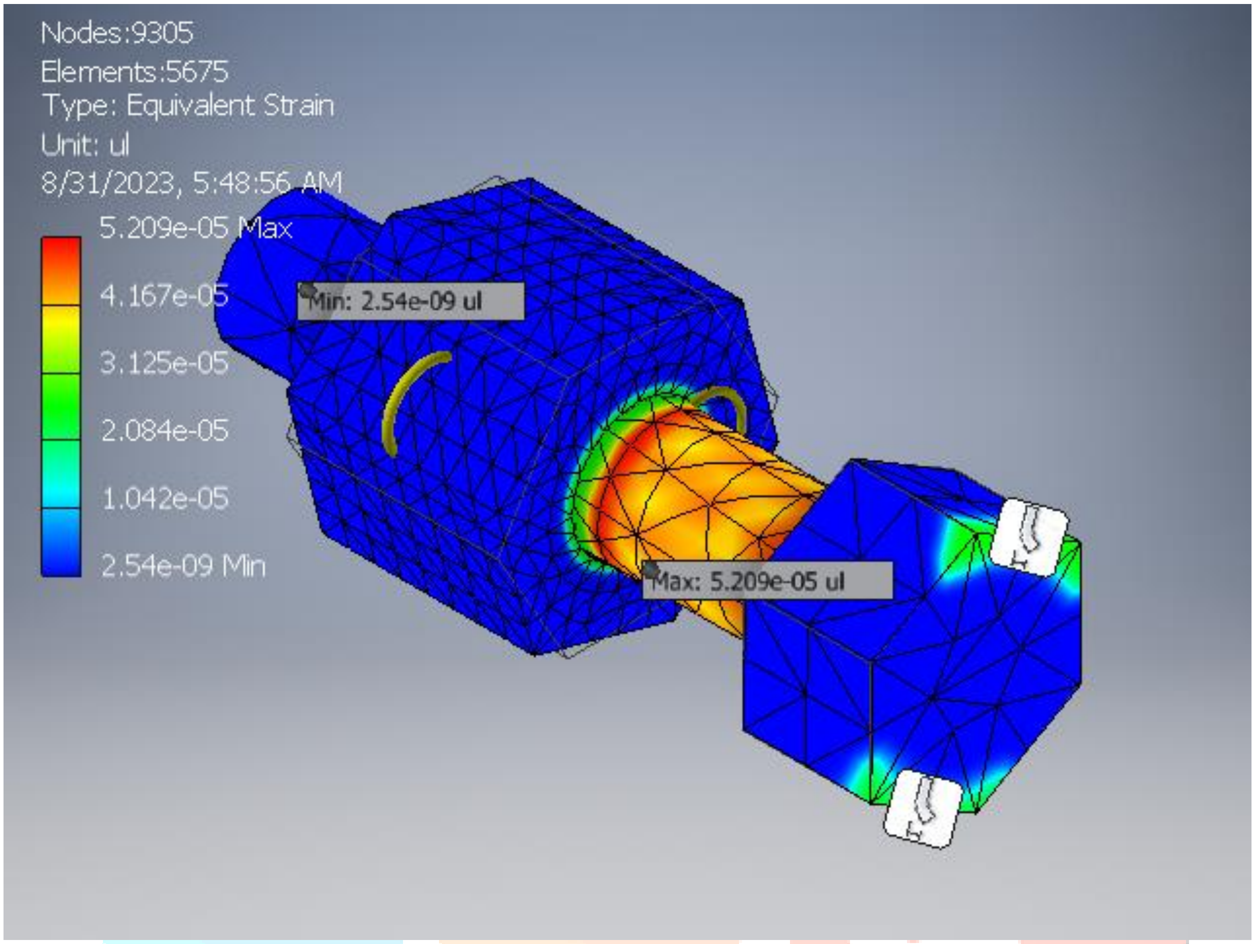

Fig14.0. Maximum Fastening Strain

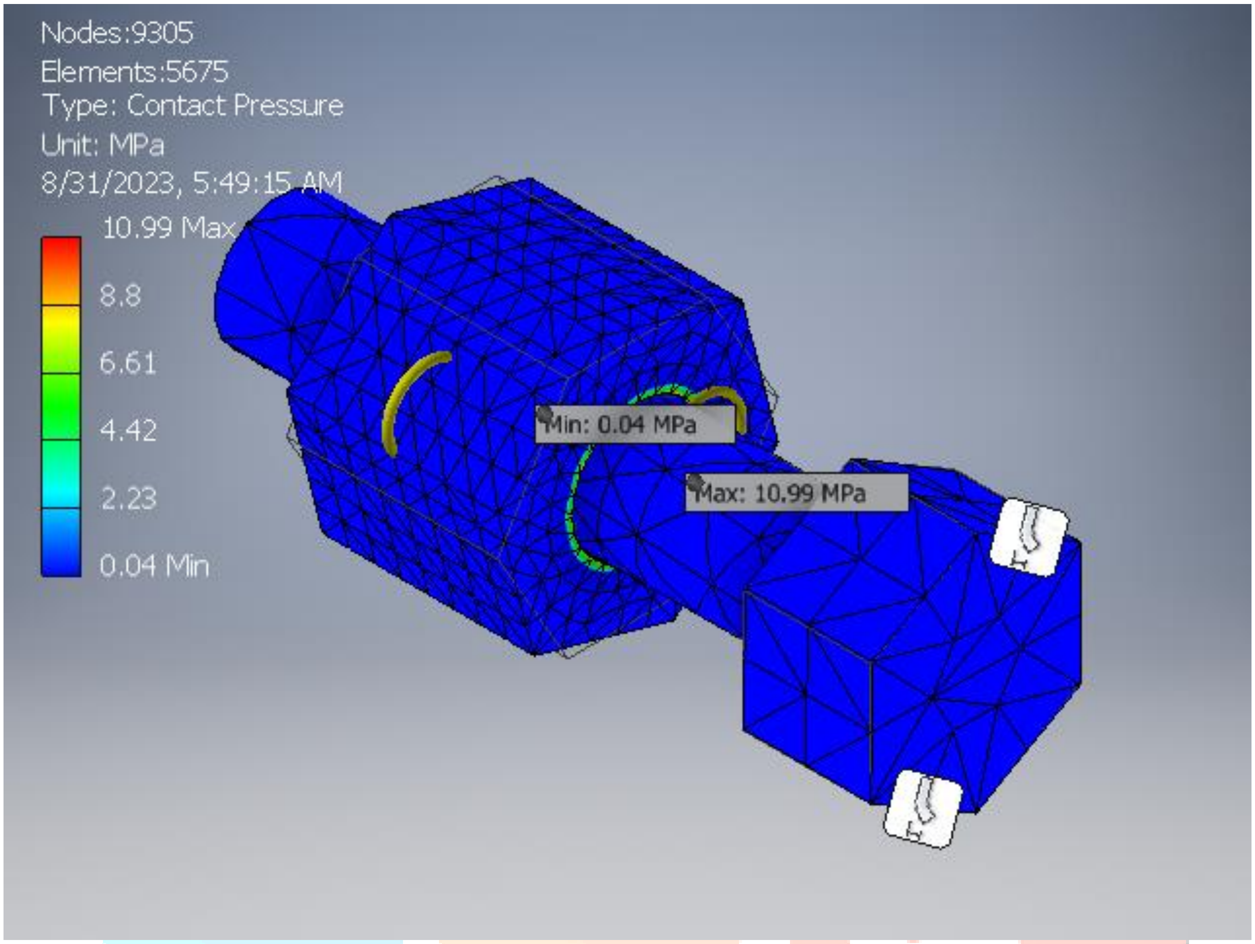

Fig15.0. Maximum Contact Pressure

Table 7.0: Output Summary

Name	Minimum	Maximum
Volume	2112.57 mm^3	
Mass	0.0165836 kg	
Von Mises Stress	0.000582701 MPa	12.1411 MPa
1st Principal Stress	-0.0365136 MPa	7.01954 MPa
3rd Principal Stress	-7.06585 MPa	0.0464872 MPa
Displacement	0 mm	0.00204884 mm
Safety Factor	15 ul	15 ul
Stress XX	-0.615814 MPa	0.675825 MPa
Stress XY	-6.67448 MPa	6.66822 MPa
Stress XZ	-6.96849 MPa	6.48625 MPa
Stress YY	-1.99658 MPa	5.70647 MPa
Stress YZ	-1.81121 MPa	2.97822 MPa
Stress ZZ	-4.1688 MPa	2.13776 MPa
X Displacement	-0.0000164193 mm	0.0000165525 mm
Y Displacement	-0.00203833 mm	0.00204884 mm
Z Displacement	-0.00177355 mm	0.00176557 mm

Equivalent Strain	0.00000000254013 ul	0.0000520856 ul
1st Principal Strain	0.00000000172952 ul	0.0000453404 ul
3rd Principal Strain	-0.0000448675 ul	-0.00000000194701 ul
Strain XX	-0.00000955983 ul	0.00000997207 ul
Strain XY	-0.0000429503 ul	0.00004291 ul
Strain XZ	-0.0000448422 ul	0.000041739 ul
Strain YY	-0.0000126175 ul	0.0000305516 ul
Strain YZ	-0.0000116551 ul	0.0000191648 ul
Strain ZZ	-0.0000204142 ul	0.0000128724 ul
Contact Pressure	0 MPa	10.9924 MPa
Contact Pressure X	-0.486563 MPa	0.65049 MPa
Contact Pressure Y	-10.585 MPa	10.4827 MPa
Contact Pressure Z	-10.3553 MPa	9.70701 MPa

Name	Steel, High Strength, Low Alloy	
General	Mass Density	7.85 g/cm^3
	Yield Strength	275.8 MPa
	Ultimate Tensile Strength	448 MPa
Stress	Young's Modulus	200 GPa
	Poisson's Ratio	0.287 ul
	Shear Modulus	77.7001 GPa
Part Name(s)	NUT AND BOLT	

DISCUSSION

Design and finite element investigation of bolt and nut for fastening stress reduction were investigated. researchers created a bolt of M6× H27 with a hexagonal head of 12mm, and a hexagonal nut of M12× H10 both having a right hand ANSI metric thread profile, with assigned material being Steel, High Strength, Low Alloy to reduce wear and tear as shown in **Fig1.0**. The bolt was first subjected to fastening moment of 240 N mm, with the nut being subjected to fixed constraints. Thereafter, the nut was subjected to fastening moment of 240 N mm with the bolt being subjected to fixed constraints. According to **Fig3.0** and **Fig 10.0**, results showed that the Von Mises stress was found to be 12.32 MPa when turning moment was at bolt head and 12.14 MPa when turning moment was at the Nut. Yield strength of the assigned material changed from 250 MPa, to 275.8 MPa respectively. This indicated that fastening at the Nut would reduce failure due to yielding. Similarly, the ultimate tensile strength of assigned material reduced from 540 MPa to 448 MPa when fastening moment was applied at bolt head and Nut side respectively. This suggested that failure due to tensile stress would be predominant when fastening at the Nut side, as shown in **Table 5.0** and **Table 7.0.**

Also, maximum fastening stresses were observed to be 7.109 MPa and 6.66822 MPa when fastening moment was applied at bolt head and Nut side respectively. This suggested that fastening at the Nut side would reduce fastening stress or thread wear, as shown in **Fig 6.0** and **Fig 13.0**.

Fig 5.0 and **Fig 12.0** showed that the maximum displacements were observed to be 0.00190463 mm and 0.00204884 mm when fastening moment was applied at bolt head and Nut side respectively. This indicated that fastening at the Nut side will give lesser revolution to achieve a tightened bolt and nut joint. In addition, the maximum contact pressures were observed to be 10.7241 MPa and 10.9924 MPa when fastening moment was applied at bolt head and Nut side respectively. This indicated that slippage is minimized at nut side fastening with high induce stress at the bolt shank, according to **Fig 8.0** and **Fig 15.0**.

CONCLUSION

According to the findings, it can be deduced that the bolt and nut fastening stress could be reduced by applying fastening/ turning moment at the nut side rather than the bolt head.

RECOMMENDATIONS

The following recommendations are suggested based on the study:

1) Fastening at nut side should be adopted to reduce stress and maximize advancement.
2) Bolt and nut design and manufacturing materials must have higher tensile strength rather than compressive strength or yield strength, since failure due to tensile stress is predominant.
3) This research could also be done in future using different bolt and nut sizes, turning moments, designs and other advanced software for generalization.

LIMITATION OF THE STUDY

In the course of carrying out this research, Design and finite element investigation of bolt and nut for fastening stress reduction, the researchers encountered many hindrances which might cause deviations from actual results. Some of the hindrances includes: design difficulties, cost of FEA analysis, lack of stable electricity, material sourcing, etc.

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DISCLOSURE OF CONFLICT OF INTEREST

This research article is original and the corresponding author hereby confirms that all co-authors participated actively in the development of the paper and has read and approved the manuscript with no ethical issues and with declaration of no conflict of interest.

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