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FINITE ELEMENT ANALYSIS OF HELICAL SPRING FOR SEVERE SERVICE APPLICATIONS

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ABSTRACT

The study, Finite Element Analysis of helical spring for severe service applications was successfully investigated. The diameter of the open helical spring is 20 mm; wire diameter is 2mm, spring height is 50mm, pitch of 5mm with 10 revolutions. The helical spring model was prepared with the aid of inventor software and imported to; Finite Element Analysis software where stress, strain and displacement were predicted. Helical spring model was subjected to pressure load of 20.00 MPa with fixed constraints. Results revealed that, the maximum 1st and 3rd principal stresses were found to be16.6502 MPa and 0.0196886 MPa respectively; this suggested that the helical spring model failure would be due to compressive stress rather than tensile stress under the given loading condition. Maximum displacement was found to be 0.000185399 mm at maximum stress of 13.0785 MPa along XZ axis within a safety factor of 15. This result, suggested that to improve reliability and stability of operation, excessive loading of helical spring along XZ axis must be avoided. Von Mises stress was found to be 25.5624 MPa and yield strength of the assigned material was 103.4 MPa, which indicated that failure of helical spring model due to yielding is not possible. The researchers made the following recommendations: Excessive loading of helical springs along XZ axis must be avoided to reduce stress and displacement within permissible limit, helical spring material must have higher compressive strength rather than tensile strength, since failure due to compressive stress is lower, etc.

Keywords: Stress, Helical Spring, Displacement, Finite Element Analysis, Pressure Load, Constraints.

I. INTRODUCTION

Background to the Study

Helical spring is one of the most fundamental flexible mechanical elements used in several industrial applications like balances, brakes, vehicle suspensions and engine valves to satisfy functions like applying forces, storing or absorbing energy, providing mechanical systems with the flexibility and maintaining a force or a pressure. In addition, helical springs serve as the elastic member for most common types of vibration absorbers (Sreenivasulu, Yaswanth, Sukumar, Gouse Basha, ArunKumar, Heamanth & Krishna, 2019).

Sreenivasulu et al. (2019) opined that the possibility of resonance and excessive vibration (or surging) are reduced in helical springs because volute springs have a uniform pitch, more damping due to helical structure and an increasing natural period of vibration (instead of a constant period of vibration as in a cylindrical spring) as each helical closes. For design and selection of springs for practical purposes, deflection of the spring under axial load and maximum stresses induced are two major factors.

Spring is an elastic or resilient body, whose function is to deflect or deform when load is applied and recover its original shape when load is removed. When the load or shock vibrations are exerted on the spring it compresses and absorbs the vibration and reduces the amplitude of disturbances (Singh , Amilkanthwar, Walli , Jasoliya &Patel, 2017).

Finite element analysis here involves the use of simulation to predict and understand how helical spring may behave under severe physical applications of loads and constraints. Finite element uses finite element method, which is a numerical technique that cuts the structure of the helical spring into several elements and then reconnects the elements at point called nodes.

Imaizumi et al. (as cited in Gordon, 2010) stated spring designers tailor the shape and size of axial compression springs to meet the design requirements without accounting for non-negligible off-axis or multi-axial loading; however, as more complex designs incorporate larger degrees of multi-axial load carrying capability, the effects of uniform and non-uniform bending and shear stress must be considered.

There is need to evaluate the stress and displacement behavior of helical springs to ensure reliability and stability of operation in mechanical machines and devices. Hence, the paper aimed at studying finite element analysis of helical spring for severe service applications.

Statement of the Problem

In order to minimize, sudden failure of helical springs under severe applications due to excessive stress and displacement, there is a need to evaluate the stress and displacement behavior under a given load condition.

According to Imaizumi et al. (as cited in Gordon, 2010) stated that spring designers tailor the shape and size of axial compression springs to meet the design requirements without accounting for non-negligible off-axis or multi-axial loading; however, as more complex designs incorporate larger degrees of multi-axial load carrying capability, the effects of uniform and non-uniform bending and shear stress and displacement must be considered. It is on this note that the researchers aimed at studying finite element analysis of helical spring for severe service applications.

Purpose of the Study

The general purpose of the study is to determine the finite element analysis of helical spring for severe service applications.

Significance of the Study

The result of this study will be beneficial to industrial spring designers/production engineers in the following ways:

1) Production engineers can improve helical spring safety; reduce operational noise and avoid sudden spring failure by choosing spring material whose allowable stress is greater than induced stress.
2) The knowledge of displacement can be used to improve the design life of helical springs by making appropriate provisions for expansion.

Scope of the Study

This research will focus on determining the finite element analysis of helical spring for severe service applications. So, all efforts were directed towards the general objectives. Stress and displacement evaluations followed Finite Element Analysis. The researchers are members of Federal Polytechnic Nekede, within South East of Nigeria. Results may be subject to variations within other parts of the World.

II. LITERATURE REVIEW

Sreenivasulu et al. (2019) investigated modeling and analysis of helical springs using CATIA-V5R19 and ANSYS 16.0. They concluded that testing the behavior of springs before manufacturing, reduces investment cost. In addition, helical springs serve as the elastic member for most common types of vibration absorbers.

Singh et al.(2017) evaluated design and analysis of helical compression spring used in suspension system by finite element analysis method. They stated that spring is an elastic or resilient body, whose function is to deflect or deform when load is applied and recover its original shape when load is removed. When the load or shock vibrations are exerted on the spring it compresses and absorbs the vibration and reduces the amplitude of disturbances.

Gordon (2010) studied stress approximation techniques for helical compression spring subjected to lateral loading. They stated that spring designers tailor the shape and size of axial compression springs to meet the design requirements without accounting for non-negligible off-axis or multi-axial loading; however, as more complex designs incorporate larger degrees of multi-axial load carrying capability, the effects of uniform and non-uniform bending and shear stress must be considered.

III. METHODOLOGY

The researchers considered an open helical spring model with assigned material soft brass to increase fatigue, ductility and resilience as shown in **Figure 1.0.** The diameter of the open helical spring is 20 mm; wire diameter is 2mm, spring height is 50mm, pitch of 5mm with 10 revolutions. The helical spring model was prepared with the aid of inventor software and imported to; Finite Element Analysis software where stress and displacement were predicted. Helical spring model was subjected to pressure load of 20.00 MPa with fixed constraints.

Design Analysis

The stress components in an element are given as below.

$$(\sigma_x)_n = \frac{E}{(1+v)(1-2v)}[(1-v)a_n + ve_n] \ldots..(1.0) \text{ (as cited in Onyenobi etal., 2022)}$$

$$(\sigma_y)_n = \frac{E}{(1+v)(1-2v)}[va_n + (1-v)e_n] \ldots..(2.0)$$

$$(\tau_{xy})_n = \frac{E}{2(1+v)}(b_n + d_n) \ldots..(3.0)$$

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

The displacement field is shown below.

$$a_n = \frac{\partial u_n}{\partial x} \ \ldots(4.0)$$

$$e_n = \frac{\partial v_n}{\partial y} \ \ldots(5.0)$$

$$b_n + d_n = \frac{\partial u_n}{\partial y} + \frac{\partial v_n}{\partial x} \ldots.(6.0)$$

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

The principal strains are given below

$e_x = \frac{1}{E}\left[\sigma_x - \frac{1}{m}(\sigma_y + \sigma_z)\right] \ldots..(7.0)$ (as cited in Onyenobi etal., 2022).

$$e_y = \frac{1}{E}\left[\sigma_y - \frac{1}{m}(\sigma_x + \sigma_z)\right] \ldots..(8.0)$$

$$e_z = \frac{1}{E}\left[\sigma_z - \frac{1}{m}(\sigma_x + \sigma_y)\right] \ldots..(9.0)$$

Von Mises Stress can be given as below.

$Von - mises\ stress = \sqrt{\sigma_x{}^2 - \sigma_x\sigma_y + \sigma_y{}^2} \ldots. (10.0)$

The torsional shear stress is given as below:

$$\text{Shear stress, } \tau = \frac{8\ \text{W.D}}{\pi d^3} \ldots. (11.0)$$

$$\text{W = axial load on the spring}$$

Direct shear stress due to load is given as below:

$$\text{Shear stress} = \frac{4\ \text{W}}{\pi d^2} \ldots (12.0)$$

Deflection of the coil spring can be calculated as shown below:

Deflection, $\delta = \frac{8\ W \times D^3}{G \times d^4} \ldots. (13.0)$

Energy stored in helical spring can be given as below:

$$U = \frac{\tau^2}{4\ K^2 \times G} \times V \ldots.(14.0)$$

$$\text{V = volume of spring wire}$$

$$\text{K = Wahl's stress factor}$$

Where $K = \frac{4C-1}{4C-4} + \frac{0.615}{C}$

IV. RESULTS AND PRESENTATIONS

Meshing

Meshing was used to divide the helical spring model into section with nodes of 19245 and elements of 10212. 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. See **Figure 2.0 and Table 2.0.**

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Table 1.0: General Objective and Settings

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	11/28/2022, 11:26 AM
Detect and Eliminate Rigid Body Modes	No

Table 2.0: Mesh Settings

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

Constraints

Fixed constraints were applied at the helical spring ends. This applies a constraint where all translational degrees of freedom are fixed. See **Figure 3.0 and Table 3.0.**

Table 3.0: 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	0.491132 N	0.468991 N	0 N m	0 N m
		0.145802 N		0 N m
		0 N		0 N m
Fixed Constraint:2	115.585 N	115.585 N	2.52385 N m	0 N m
		0 N		-0.88126 N m
		0 N		2.365 N m

Table 4.0: General Material Properties

Part Number	COIL SPRING
Designer	EWURUM TENNISON
Cost	$40.00
Date Created	11/28/2022
Material	Brass, Soft Yellow
Density	8.47 g/cm^3
Mass	0.0246526 kg
Area	4299.89 mm^2
Volume	2910.58 mm^3

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Center of Gravity	x=-0.0222245 mm y=20.2918 mm z=-0.0000309621 mm	
Name	Brass, Soft Yellow	
General	Mass Density	8.47 g/cm^3
	Yield Strength	103.4 MPa
	Ultimate Tensile Strength	275 MPa
Stress	Young's Modulus	109.6 GPa
	Poisson's Ratio	0.331 ul
	Shear Modulus	41.1721 GPa
Part Name(s)	COIL SPRING	

Table 5.0: Operating conditions

Load Type	Pressure
Magnitude	20.000 MPa

Table 6.0: Overall Result Summary

Name	Minimum	Maximum
Volume	2910.58 mm^3	
Mass	0.0246526 kg	
Von Mises Stress	0 MPa	25.5624 MPa
1st Principal Stress	-11.4904 MPa	16.6502 MPa
3rd Principal Stress	-23.8238 MPa	0.0196886 MPa
Displacement	0 mm	0.000185399 mm
Safety Factor	4.045 ul	15 ul
Stress XX	-21.4649 MPa	1.65521 MPa
Stress XY	-8.14206 MPa	12.624 MPa
Stress XZ	-12.9039 MPa	13.0785 MPa
Stress YY	-13.432 MPa	2.33225 MPa
Stress YZ	-1.41861 MPa	1.55148 MPa
Stress ZZ	-13.2588 MPa	3.25041 MPa
X Displacement	-0.000184905 mm	0 mm
Y Displacement	-0.0000271829 mm	0.00000876453 mm
Z Displacement	-0.0000225309 mm	0.0000264479 mm

Equivalent Strain	0 ul	0.000206959 ul
1st Principal Strain	-0.000000569571 ul	0.000180762 ul
3rd Principal Strain	-0.000180328 ul	0.0000000000000018782 ul
Strain XX	-0.000128548 ul	0.0000174468 ul
Strain XY	-0.0000988784 ul	0.000153308 ul
Strain XZ	-0.000156708 ul	0.000158827 ul
Strain YY	-0.0000341218 ul	0.0000257141 ul
Strain YZ	-0.0000172278 ul	0.0000188415 ul
Strain ZZ	-0.0000357182 ul	0.0000232556 ul

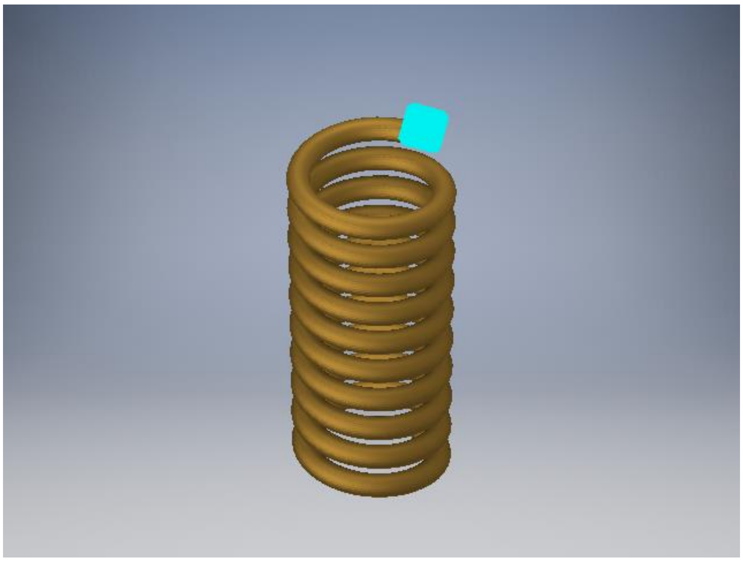

Figure 1.0. Open Helical Spring Model

Figure 2.0. Meshing of Helical Spring

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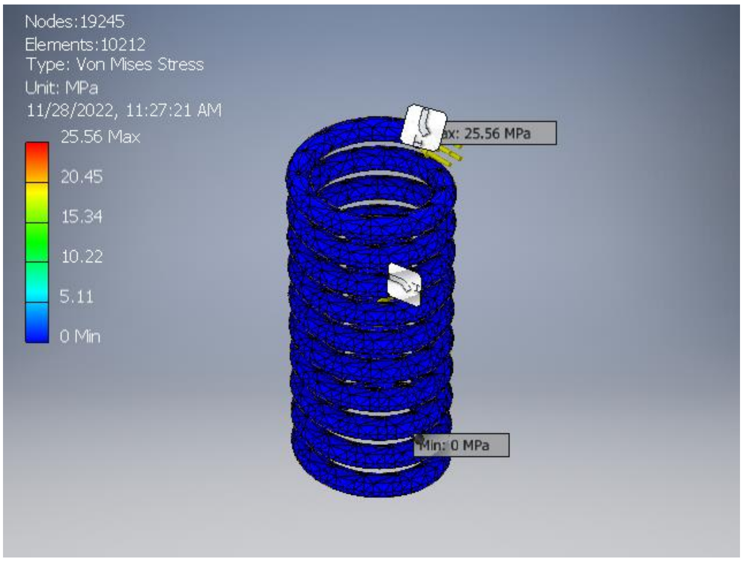

Figure 3.0. Von Mises Stress

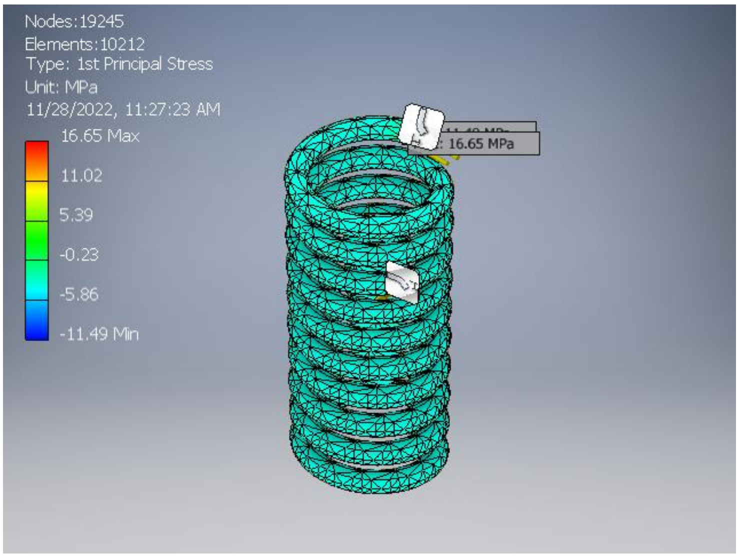

Figure 4.0. 1st Principal Stress

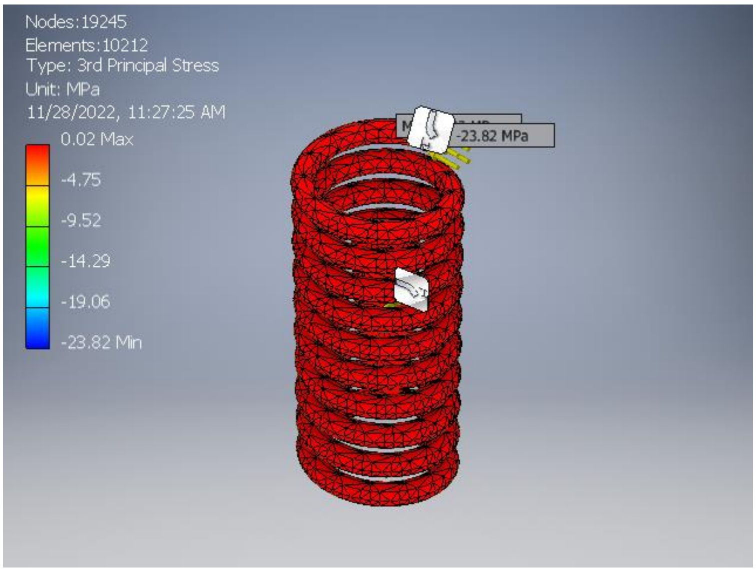

Figure 5.0. 3rd Principal Stress

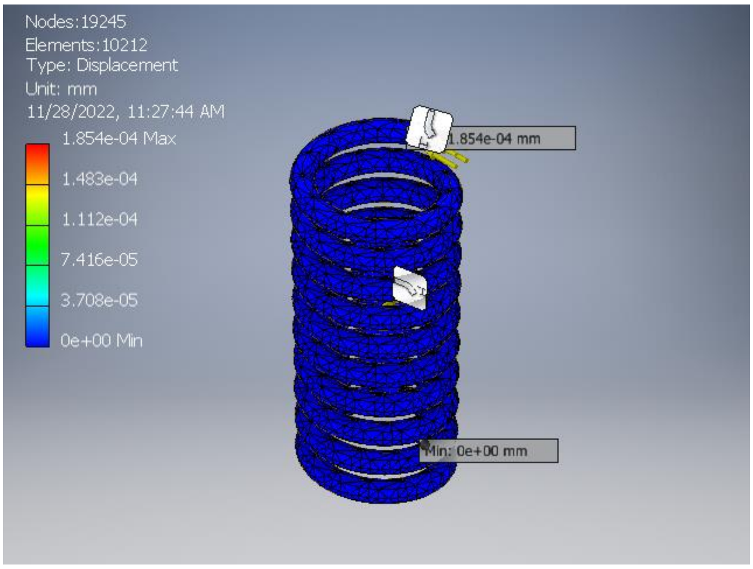

Figure 6.0. Displacement

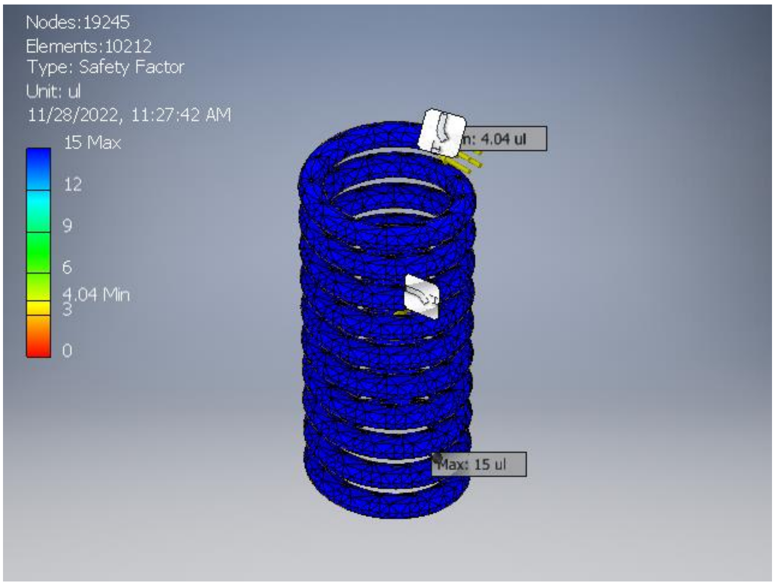

Figure 7.0. Safety Factor

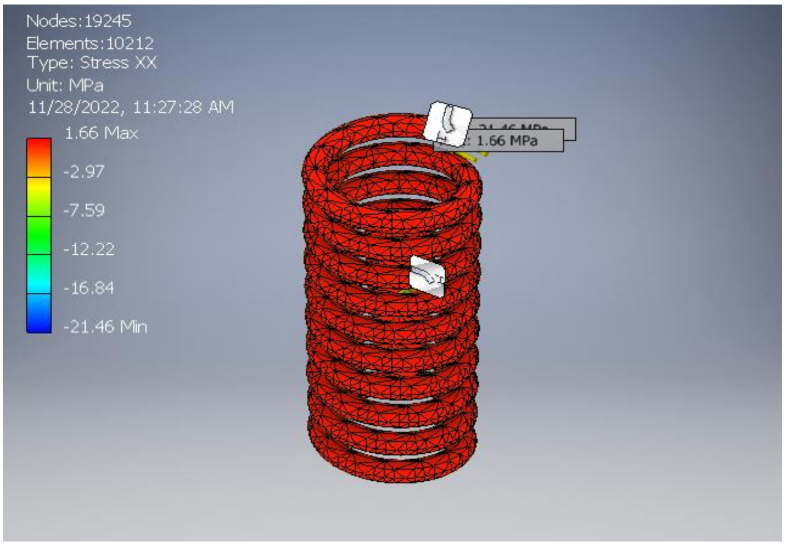

Figure 8.0. Stress along XX

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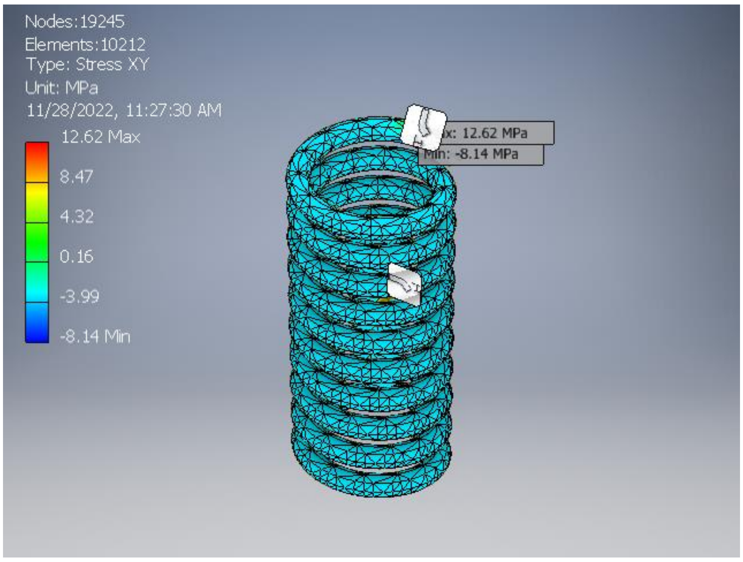

Figure 9.0. Stress along XY

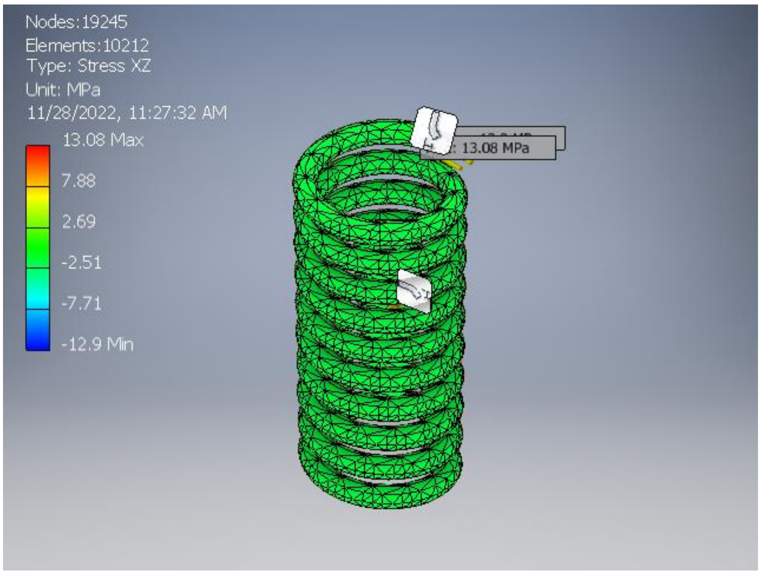

Figure 10.0. Stress along XZ

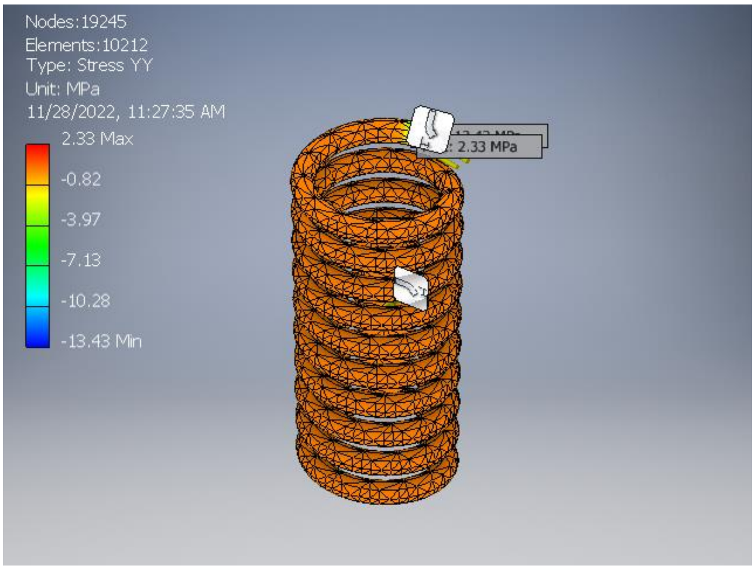

Figure 11.0. Stress along YY

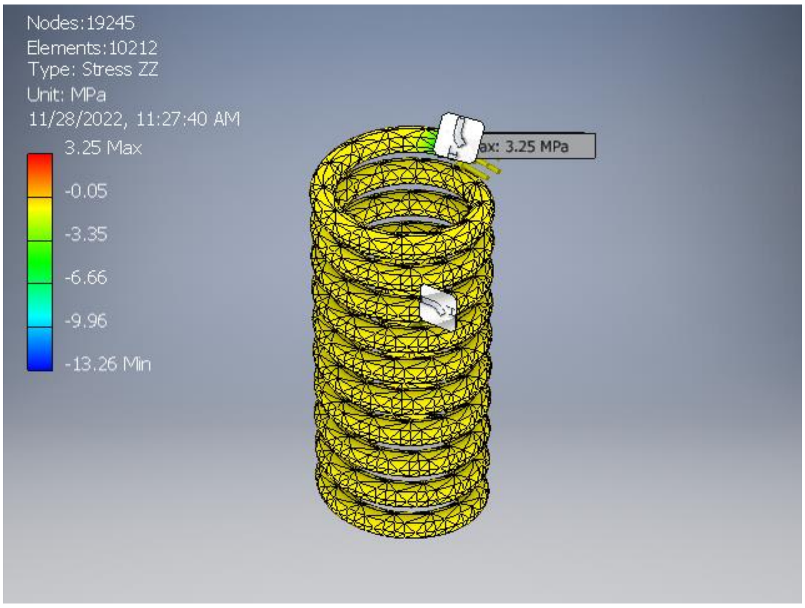

Figure 12.0. Stress along ZZ

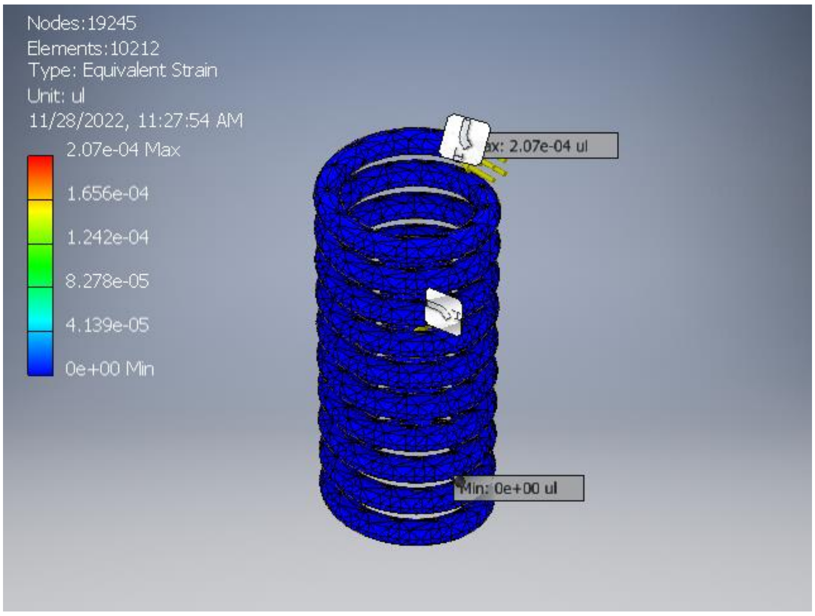

Figure 13.0. Equivalent Strain

V. DISCUSSION

Finite Element Analysis of helical spring for severe application was investigated. The diameter of the open helical spring is 20 mm; wire diameter is 2mm, spring height is 50mm, pitch of 5mm with 10 revolutions. The helical spring model was prepared with the aid of inventor software and imported to; Finite Element Analysis software where stress and displacement were predicted. Helical spring model was subjected to pressure load of 20.00 MPa with fixed constraints.

Furthermore, the helical spring model was subjected to finite element analysis to predict stress, strain and displacement under pressure load of 20.00MPa, fixed constraints. Von Mises stress was found to be 25.5624 MPa and yield strength of the assigned material was 103.4 MPa, which indicated that failure of helical spring model due to yielding is not possible.

Also, maximum 1st and 3rd principal stresses were found to be 16.6502 MPa and 0.0196886 MPa respectively , this suggested that the helical spring model failure would be due to tensile stress rather than compressive stress under the given loading condition. In addition, maximum displacement was found to be 0.000185399 mm at maximum stress of 13.0785 MPa along XZ axis within a safety factor of 15. This result, suggested that to improve reliability and stability of helical spring, excessive loading of spring along XZ axis must be avoided. (See **Figure 3.0 – 13.0**).

VI. CONCLUSION

According to the findings, it can be deduced that the values of stress and displacement acting on helical spring during operation, influences mechanical machines and devices reliability and stability and hence, control limits must be set to stabilize performance.

VII. RECOMMENDATIONS

The following recommendations are suggested based on the study:

1) Excessive loading of helical springs along XZ axis must be avoided to reduce stress and displacement within permissible limit.

2) Helical spring material must have higher compressive strength rather than tensile strength, since failure due to compressive stress is lower.

3) This research can also be done in future using different helical spring designs models and other advanced software for generalization.

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