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Finite Element Investigation of Horizontal Shaft with Point Load, Axial Stress and Turning Moment

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ABSTRACT

The study, finite element investigation of horizontal shaft with point load, axial stress and turning moment was successfully achieved. Solid shaft of 60 mm diameter and 200mm long was created using Autodesk Inventor. The material assigned to the created shaft was Stainless Steel 400C to improve strength and stiffness. Under finite element analysis software, shaft was subjected to turning moment of 100 N mm, axial stress of 30MPa and point load of 900N with constraints. Simulation results showed that the Von Mises stress was found to be 339.3 MPa. With material yield strength of 240MPa, the failure of shaft due to yielding is possible under the given conditions and would be more predominant when rotational speed of shaft is increased. In addition, the 1st and 3rd principal stresses were found to be 526.8MPa and 169.7MPa respectively. These results indicated that the shaft would fail due to tensile stress rather than compressive stress. Also, maximum stress induced in shaft was found to be 526.5MPa. The maximum displacement was observed to be 0.00748mm and allowable value for power transmission shaft is 0.13mm, this indicated higher shaft stability with safety factor of 15ul. Findings, suggested that horizontal shaft subjected to point load, axial stress and turning moment would fail due to tensile stress rather than compressive stress. Hence, engineering material with high tensile strength should be used in shaft design and manufacturing to improve reliability and performance.

Keywords ---- *Axial stress, Shaft, Point load, Turning moment, Stress, Finite element analysis*

INTRODUCTION

Horizontal shaft is mostly used to transmit rotary power under loading conditions. Horizontal machine shaft are integral part of machine itself, could also function as a support beam to other machine elements such as pulley, gear, bearing, brackets, etc. Horizontal shaft may be solid or hollow circular cross section depending upon the application. But the paper adopted a solid shaft with constant cross section due to its wide industrial applications (Khurmi and Gupta, 2012).

Deflection of horizontal shaft with respect to shaft diameter was compared by Christopher and Michael (2013). It was noted that 30 mm diameter shaft would experienced larger displacement/deflection while 50 mm diameter shaft experienced smaller displacement/ deflection.

Finite element analysis here involves the use of simulation to predict and understand maximum stress, strain and displacement of a horizontal shaft under axial load, point load, turning moment and constraints. Finite element uses finite element method, which is a numerical technique that cuts the structure of the shaft into several elements and then reconnects the elements at point called nodes.

According to Sara and Engel (2017), as cited in Ugwuegbu and Ewurum (2022), it was stated that very machine shaft designer must ensure that the shaft geometry satisfy the requirements of the material strength and shaft-supported components. In addition, shaft failure arises from fluctuated load conditions or combined torsion and bending load conditions which usually cause shaft deflection/displacement. Deflection/displacement analysis relies on the overall shaft geometry and the work condition. In general, shafts deflect linearly as a beam and angularly as a torsion bar.

Review of related literatures done by the authors revealed that some of the load conditions that could necessitate horizontal shaft failure or results to complex stresses were point load, axial load and twisting moment. Hence, the paper aimed at studying finite element investigation of horizontal shaft with point load, axial stress and turning moment.

METHODOLOGY

A horizontal solid shaft with constant cross sectional diameter of 60 mm and length of 200mm was created using Autodesk Inventor. Stainless Steel 400C material was assigned to the created shaft model. The horizontal shaft model was imported into finite element analysis software and was subjected to turning moment of 100 N mm, axial stress of 30MPa and point load of 900N with fixed constraints. Simulation results for maximum stress, strain, displacement were noted and reported as below.

MESHING

Meshing was used to divide the shaft model into section with nodes of 1970 and elements of 1129. 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 and table 4.

TABLES AND FIGURES

Table 1: Mesh settings and general objective and settings for Shaft

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	10/15/2023, 6:55:54 PM
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	SHAFT
Designer	EWURUM TENNISON
Cost	$25.00
Date Created	10/15/2023

Table 2: Physical and Material Properties for Shaft

Material	Stainless Steel 400C	
Density	7.75 g/cm^3	
Mass	0.565168 kg	
Area	43026.8mm^2	
Volume	561168 mm^3	
Center of Gravity	x=-2.5 mm y=0 mm z=0 mm	
Name	Stainless Steel	
General	Mass Density	7.75 g/cm^3
	Yield Strength	240MPa
	Ultimate Tensile Strength	540MPa
Stress	Young's Modulus	193GPa
	Poisson's Ratio	0.3 ul
	Shear Modulus	74.2308 GPa
Part Name(s)	SHAFT	

Table 3: Model Operating Conditions

Load Type	Moment
Magnitude	100 N mm
Vector X	100.000 N mm
Vector Y	0.000 N mm
Vector Z	0.000 N mm

Table 4: Model Operating Conditions Force

Load Type	Force
Magnitude	900.000 N
Vector X	0.000 N
Vector Y	-864.023 N
Vector Z	251.920 N

Table 5: Model Operating Conditions Pressure

Load Type	Pressure
Magnitude	30.000 MPa

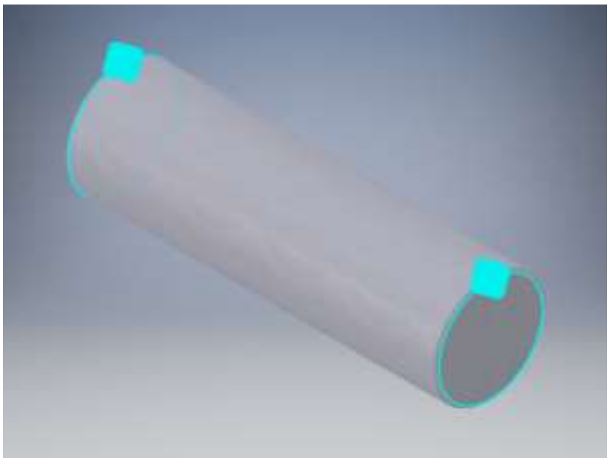

Fig 1.0(a): Shaft Model with Constraints

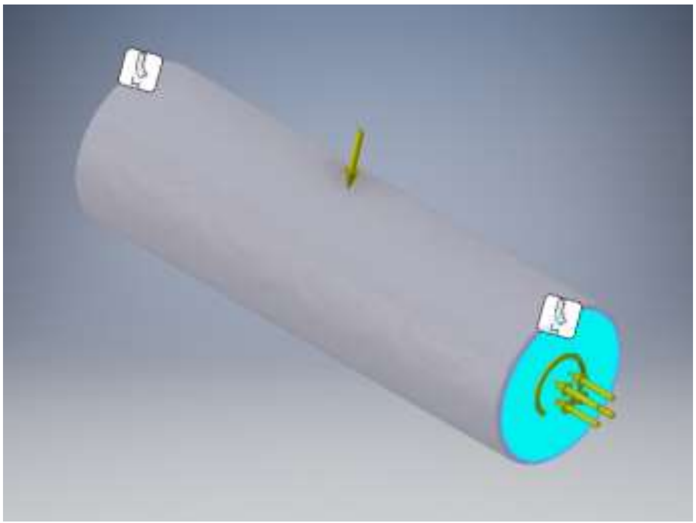

Fig 1.0(b): Shaft with Point load, Axial Stress, Twisting moment and Constraints

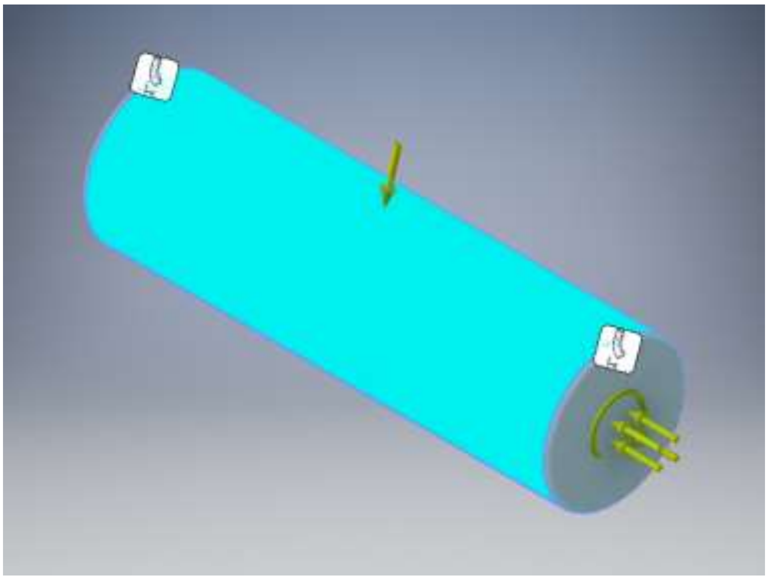

Fig 1.0(c): Shaft with Point load, Axial Stress, Twisting moment and Constraints

EQUATIONS

The stress components in an element are given as below.

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

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

$$(\tau_{xy})_n = \frac{E}{2(1+v)}(b_n + d_n) \ldots\ldots(3)$$

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

The displacement field is shown below.

$$a_n = \frac{\partial u_n}{\partial x} \ \ldots(4)$$

$$e_n = \frac{\partial v_n}{\partial y} \ \ldots(5)$$

$$b_n + d_n = \frac{\partial u_n}{\partial y} + \frac{\partial v_n}{\partial x} \ldots.(6)$$

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

The principal strains are given below

$e_x = \frac{1}{E}\left[\sigma_x - \frac{1}{m}(\sigma_y + \sigma_z)\right] \ldots\ldots(7)$ (Rajput, 2008).

$$e_y = \frac{1}{E}\left[\sigma_y - \frac{1}{m}(\sigma_x + \sigma_z)\right] \ldots\ldots(8)$$

$$e_z = \frac{1}{E}\left[\sigma_z - \frac{1}{m}(\sigma_x + \sigma_y)\right] \ldots\ldots(9)$$

Von Mises Stress can be given as below.

$Von - mises\ stress = \sqrt{\sigma_x{}^2 - \sigma_x\sigma_y + \sigma_y{}^2} \ldots.(10)$

RESULTS

The following results were gotten when simulation was run in finite element software.

Table 6: Reaction Force and Moment on Constraints for Shaft

Constraint Name	**Reaction Force**		**Reaction Moment**	
	Magnitude	**Component (X,Y,Z)**	**Magnitude**	**Component (X,Y,Z)**
Fixed Constraint:1	73591.1 N	73585.6 N	7.08845 N m	0 N m
		864.033 N		-2.68495 N m
		-251.896 N		-6.56028 N m

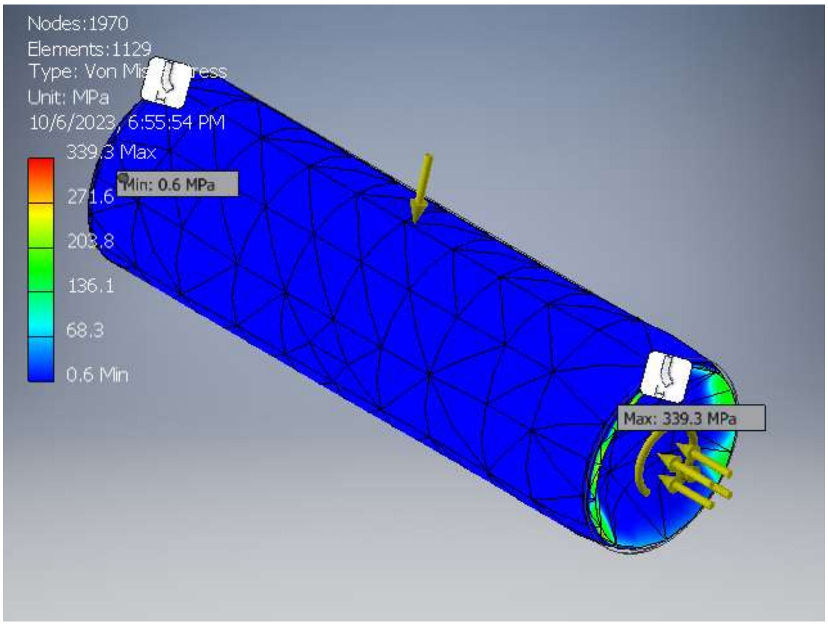

Fig 2: Von Mises Stress for Shaft

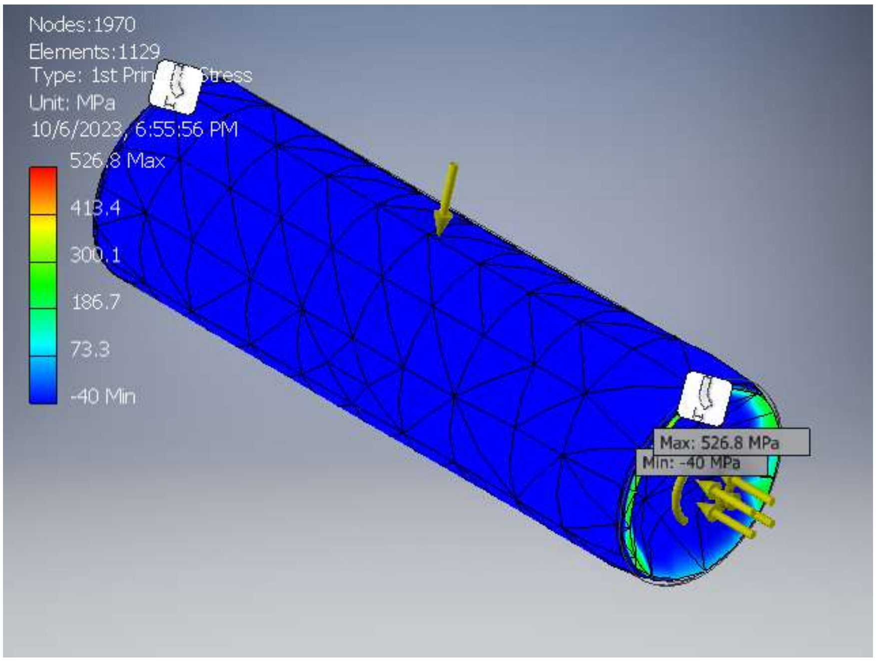

Fig 3: 1st Principal Stress for Shaft

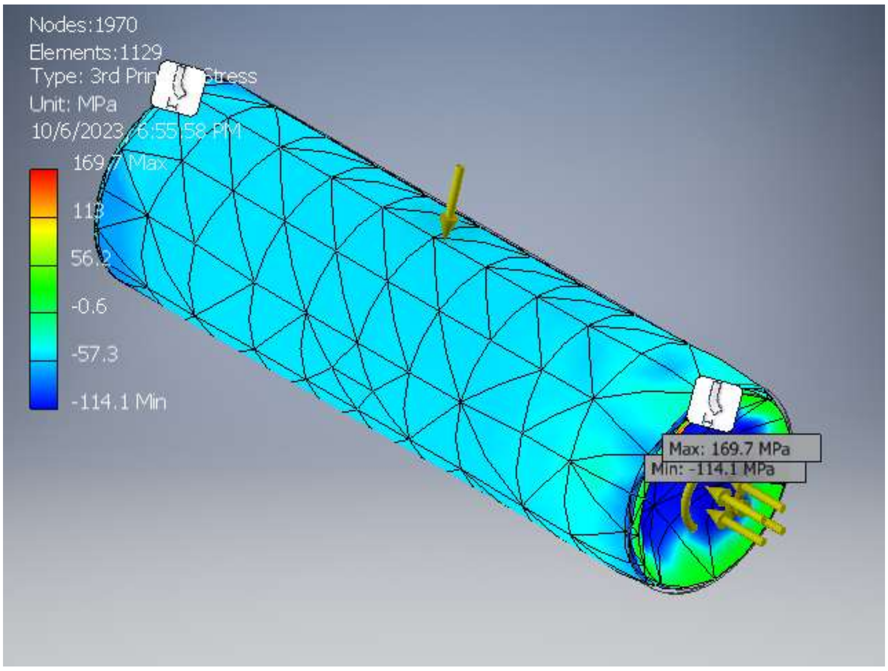

Fig 4: 3rd Principal Stress for Shaft

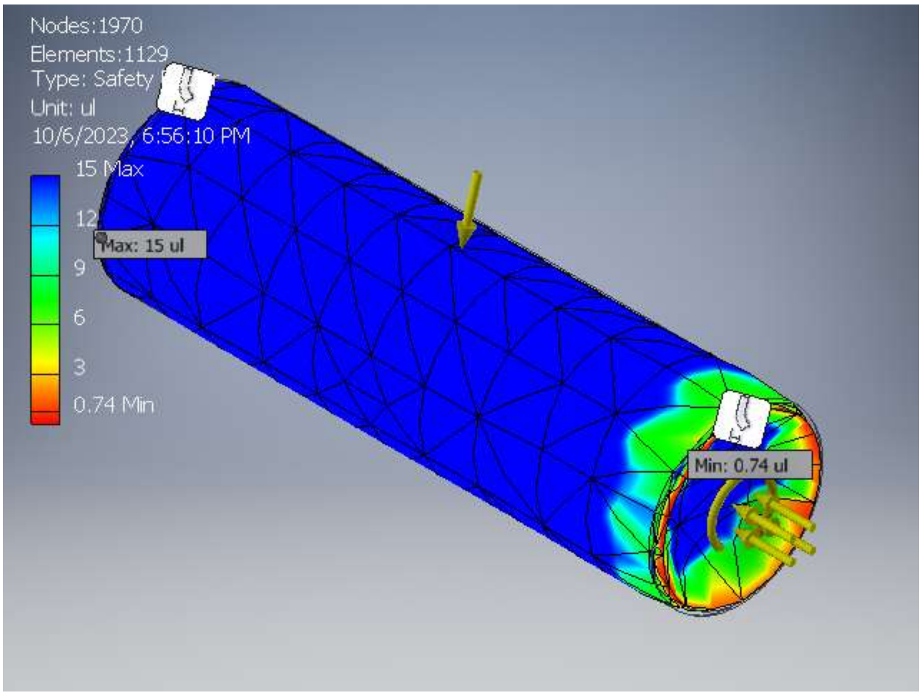

Fig 5: Safety Factor for Shaft

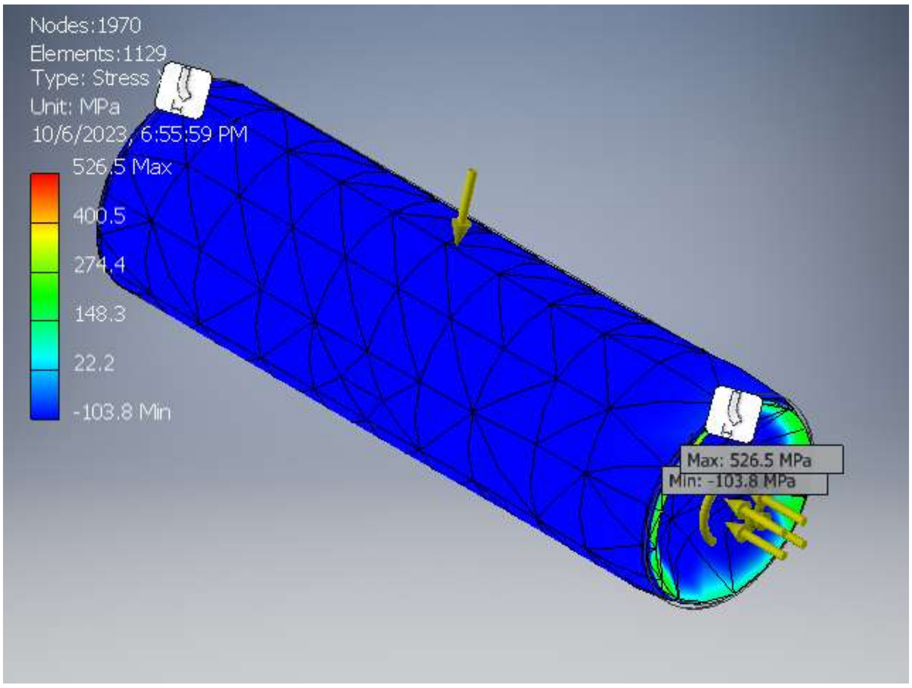

Fig 6: Maximum Stress on Shaft

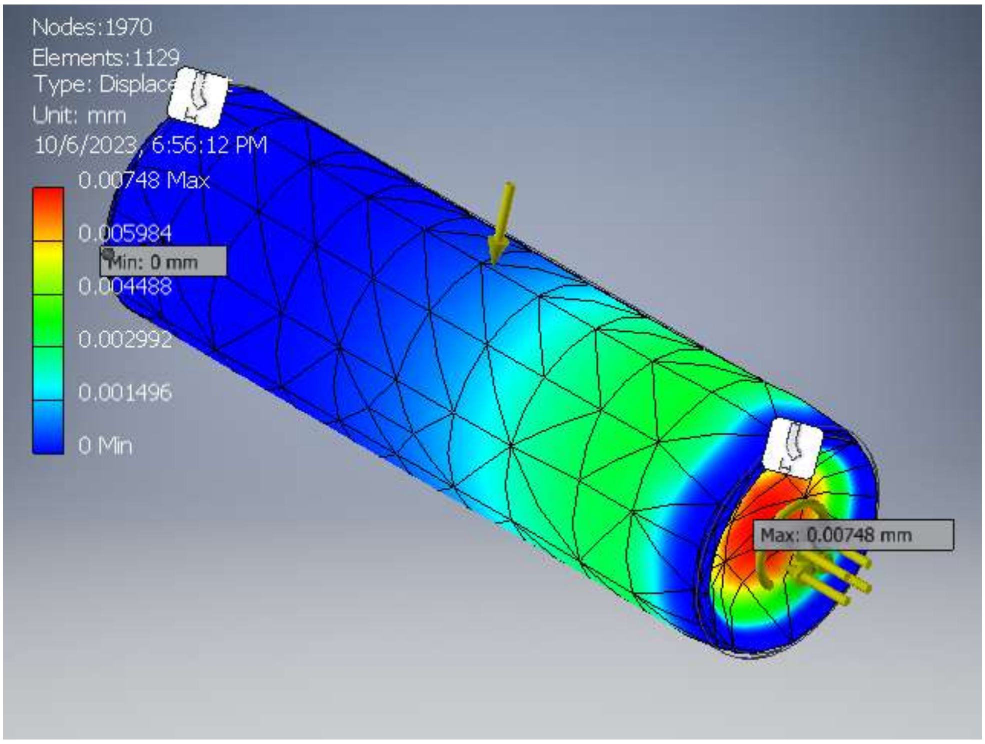

Fig 7: Maximum Displacement on Shaft

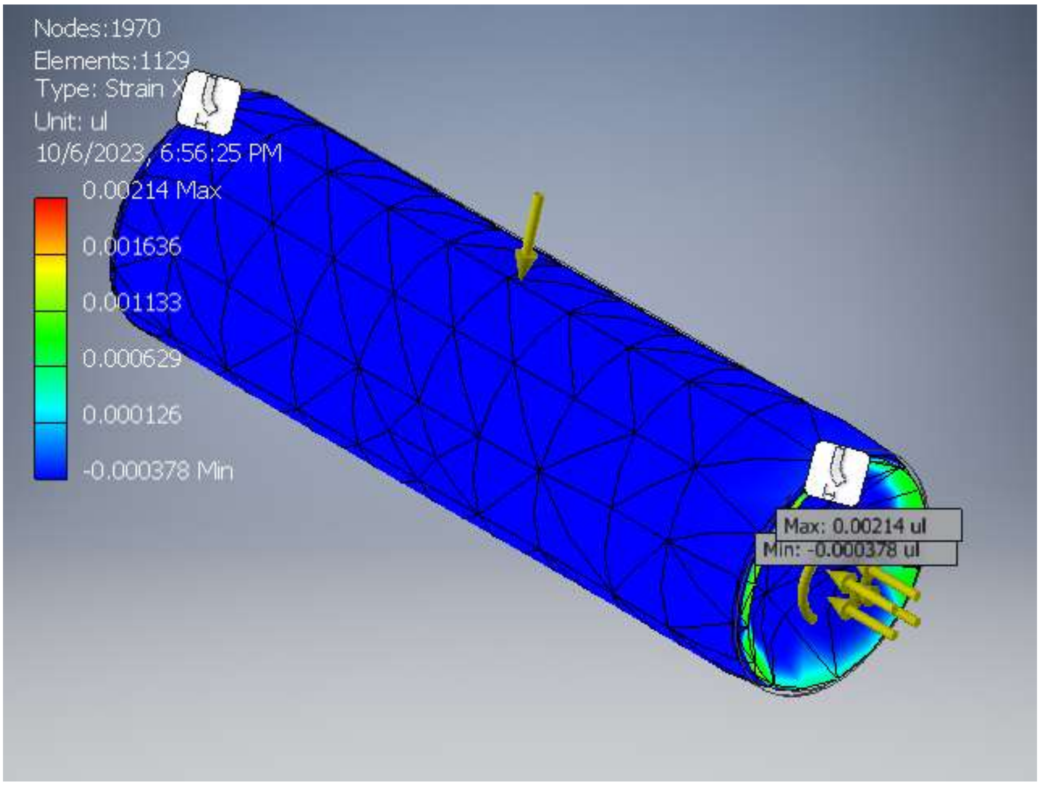

Fig 8: Maximum Strain on Shaft

Table 7: Output Summary of Shaft

Name	**Minimum**	**Maximum**
Volume	565168 mm^3	
Mass	4.52134 kg	
Von Mises Stress	0.560528 MPa	339.332 MPa
1st Principal Stress	-40.0395 MPa	526.807 MPa
3rd Principal Stress	-114.087 MPa	169.727 MPa
Displacement	0 mm	0.00747958 mm
Safety Factor	0.736742 ul	15 ul
Stress XX	-103.845 MPa	526.524 MPa
Stress XY	-32.3227 MPa	40.44 MPa
Stress XZ	-64.5603 MPa	80.0157 MPa
Stress YY	-48.7756 MPa	180.49 MPa
Stress YZ	-28.1602 MPa	34.8072 MPa
Stress ZZ	-54.2728 MPa	198.085 MPa
X Displacement	-0.00747953 mm	0.000216751 mm
Y Displacement	-0.000735132 mm	0.000813541 mm
Z Displacement	-0.00104259 mm	0.00106588 mm
Equivalent Strain	0.00000567914 ul	0.00176177 ul
1st Principal Strain	-0.0000479255 ul	0.00214155 ul
3rd Principal Strain	-0.000575295 ul	0.00000615499 ul
Strain XX	-0.00037788 ul	0.00213965 ul
Strain XY	-0.000217718 ul	0.000272394 ul

Strain XZ	-0.000434862 ul	0.000538966 ul
Strain YY	-0.000226988 ul	0.00024403 ul
Strain YZ	-0.00018968 ul	0.000234452 ul
Strain ZZ	-0.000486012 ul	0.000196761 ul

DISCUSSION

Finite element investigation of a horizontal shaft with point load, axial stress and turning moment was achieved here. A solid shaft with constant cross sectional diameter of 60 mm and length of 200mm was created using Autodesk Inventor.

Stainless Steel 400C material was assigned to the created shaft and was imported into finite element analysis software and was subjected to turning moment of 100 N mm, axial stress of 30MPa and point load of 900N with fixed constraints, as shown in Fig 1(a) to Fig (c). Simulation results for maximum stress, strain, displacement and principal stresses were noted and reported as shown in Fig 2 to Fig 7.

According to Fig 2, results showed that the Von Mises stress was found to be 339.3 MPa. Since, yield strength of the assigned material is 240 MPa, it suggested that failure of shaft due to yielding is possible under the given conditions and would be more predominant when rotational speed of shaft is increased.

According to Fig 3 and Fig 4, the 1st and 3rd principal stresses were found to be 526.8MPa and 169.7MPa respectively. These results indicated that the shaft would fail due to tensile stress rather than compressive stress. Also, maximum stress induced in shaft was found to be 526.5MPa, according to Fig 6. Since the ultimate tensile strength of assigned material was 540MPa, therefore the design was safe.

Similarly, According to Fig 8, maximum deformation/strain induced in shaft was found to be 0.00214 and this is below allowable value. Fig 7 showed that the maximum displacement was observed to be 0.00748mm and allowable value for power transmission shaft is 0.13mm, this showed higher stability with safety factor of 15ul.

CONCLUSION

According to the findings, it can be deduced that horizontal shaft subjected to point load, axial stress and turning moment would fail due to tensile stress rather than compressive stress. Hence, engineering material with high tensile strength should be used in shaft design and manufacturing to improve reliability and performance.

RECOMMENDATIONS

The following recommendations are suggested based on the study:

1) Shaft material must have higher tensile strength rather than compressive strength, since failure due to tensile stress is predominant.

2) This research could also be done in future using different geometry design, sizes, loads and other advanced software for generalization.

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