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Design and Stress Evaluation of Rotary Gas Impeller for Performance Improvement

[1]Efosa Obaseki, [2]Olewuezi Chinedum Emmanuel, [3]Ifo Chike Valentine, [4]Ekanem Oto-Obong Aniefiok & [5]Ewurum Tennison. I

[1, 2, 4&5] Department of Mechanical Engineering, School of Engineering Technology, Federal Polytechnic Nekede, Owerri, Nigeria

[3]Maintenance Unit, Nigerian Breweries PLC, Enugu State, Nigeria

Email & phone number: tennymech@gmail.com, +2348161513849

ABSTRACT

In this paper, design and stress evaluation of rotary gas impeller for performance improvement was conducted. The authors adopted engineering design principles and finite element analysis to achieve better results. The model was prepared with the aid of inventor software and imported to; Finite Element Analysis software where the operational stresses was predicted. The created impeller model had a stepped barrel or stepped rotor retaining four rectangular vanes or blades with gas holes and it is mounted on an end threaded shaft which is usually coupled to an electric motor. The bigger barrel diameter was 30mm with length of 40mm and small barrel diameter was 20mm with length of 50mm. Small barrel was threaded within a length of 20mm with right hand ANSI metric thread profile for coupling to the prime mover or electric motor. Barrels together with blades retained central and side holes respectively for suction and discharge of gases. Rectangular blade dimensions were 2mm×4mm. The gas impeller was subjected to turning moment of 100 N-mm and 2 MPa gas pressure under fixed rotational constraints. Results showed that the Von Mises stress was found to be 42.7555 MPa. Since, yield strength of the assigned material is 689 MPa, it suggested that failure of gas impeller material due to yielding load is not possible under the given conditions. Also, the 1st and 3rd principal stresses were found to be 48.0505 MPa and 13.7433 MPa respectively. These indicated that the blade material would fail due to tensile stress rather than compressive stress. Stress concentration was found to be much at the, weld heads and tails of blades, as well as threaded barrel section, according to the study. In addition, the maximum displacement/deformation of impeller due to gas pressure was found to be 0.0305028 mm. This suggested that the material used showed lower deformation response and could improve service life of gas pumps. Also, the maximum induced stress was found to be 40.6269 MPa whereas the ultimate tensile strength of material used for the gas impeller was 861.25 MPa. This suggested that the design was safe and not over stressed. Authors suggested that gas impeller material must have high tensile strength rather than compressive strength, since failure due to tensile stress is predominant.

Keywords ---- Gas impeller, Stress, Finite element analysis, Turning moment, Constraints

1.0 INTRODUCTION

Gas impeller is a rotating component of a centrifugal compressor that agitates, sucks and accelerates gas outwards from the center of rotation; when coupled to a prime mover such as electric motor or mechanical gas turbine, thus transferring energy from the motor that drives the compressor to the gas being pumped. According to Khurmi and Gupta (2012) as cited by Onyenobi et al., (2022) in Ibezim et al., (2024) maintained that torsional vibration and deformation of blades are always a challenging problem for Engineers in gas impeller.

Open impeller is a wheel or rotor with a series of backward curved vanes or blades and it is mounted on a shaft which is usually coupled to an electric motor. Mohamed, Moey, Ibrahim, Yazdi, & Merdji (2022) as cited in Onyenobi et al., (2022) contributed that the gas impeller blade is a critical component in a centrifugal compressors or gas pumps that determine the efficiency of gas pumps.

The designed gas impeller here is a step barrel or step rotor with four rectangular vanes or blades with gas holes and it is mounted on an end threaded shaft which is usually coupled to an electric motor. Impeller blades are critical component in a centrifugal compressor that determines performance and reliability of the compressor, since the gases bombard it surface. This necessitated the need for, gas impeller stress evaluations to ensure reliability and safety of operation. Reviewed literatures revealed that excessive gas impeller stress might ruin compressor or gas pump performance and also lower service life. Hence, the paper aimed at studying the design and stress evaluation of rotary gas impeller for performance improvement.

2.0 METHODOLOGY

The researchers created a rotary gas impeller model using Autodesk Inventor with an assigned material being Stainless Steel 400C; to reduce corrosion and peeling effects. The created impeller model had a step barrel or step rotor retaining four rectangular vanes or blades with gas holes and it is mounted on an end threaded shaft which is usually coupled to an electric motor. The bigger barrel diameter was 30mm with length of 40mm and small barrel diameter was 20mm with length of 50mm. Small barrel was threaded within a length of 20mm with right hand ANSI metric thread profile for coupling to the prime mover or electric motor. Barrels together with blades retained central and side holes respectively for suction and discharge of gases. Rectangular blade dimensions were 2mm×4mm. Gas impeller model was imported to Finite Element Analysis software where stresses were evaluated. Fixed rotating constraint was used with a turning moment of 100 N-mm and gas pressure of 2 MPa acting at the blade faces. Results were recorded as reported as shown below.

2.0.1 MESHING

Meshing was used to divide the gas impellers into section with nodes of 16626 and elements of 1998. 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 Fig 2.0.**

Table 1: General objective and settings:

Design Objective	Single Point
Study Type	Static Analysis
Last Modification Date	4/18/2024, 3:25 PM
Detect and Eliminate Rigid Body Modes	No

Table 2: 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

Table 3: Mechanical Properties of Material

Name	Stainless Steel, 440C	
General	Mass Density	7.75 g/cm^3
	Yield Strength	689 MPa
	Ultimate Tensile Strength	861.25 MPa
Stress	Young's Modulus	206.7 GPa
	Poisson's Ratio	0.27 ul
	Shear Modulus	81.378 GPa
Part Name(s)	GAS IMPELLER	

Table 4: Physical Properties of Material

Material	Stainless Steel, 440C
Density	7.75 g/cm^3

Mass	0.141569 kg
Area	18208.6 mm^2
Volume	18267 mm^3
Center of Gravity	x=0.166718 mm y=18.1352 mm z=0.29756 mm

2.0.2 Operating Conditions

Table 5 : Moment

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

Table 6: Pressure

Load Type	Pressure
Magnitude	2.000 MPa

The tables 1 to 6 above indicated the setting conditions in finite element software, physical and mechanical properties of the assigned material, with the operating conditions generated by finite element software used.

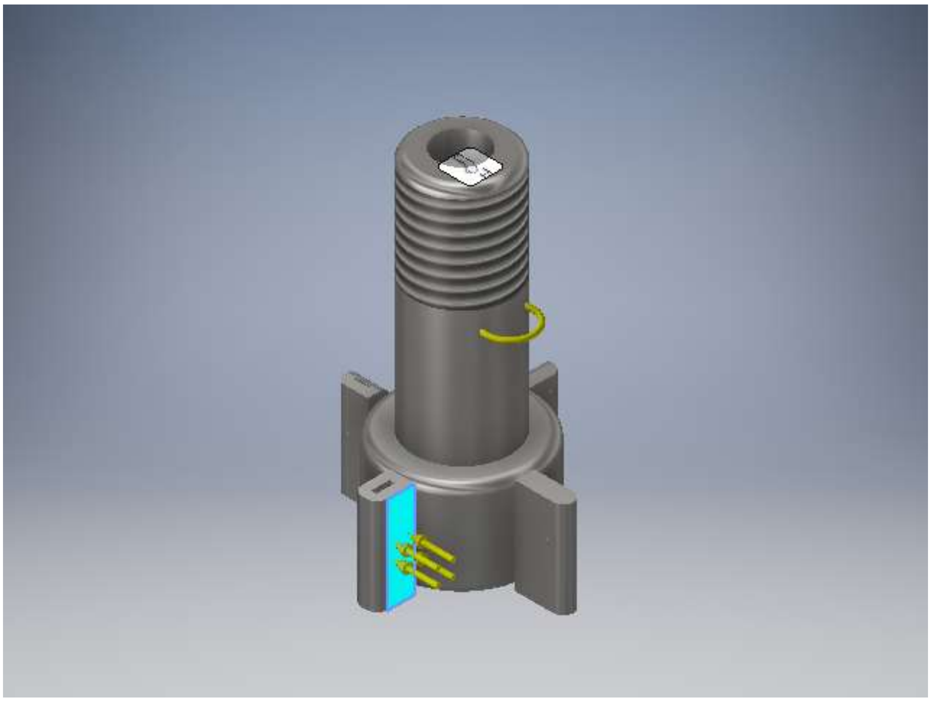

Fig1.0(a). Gas Impeller Model with Constraint, Turning Moment and Gas Pressure

Fig1.0 (b). Gas Impeller Model with Constraint, Turning Moment and Gas Pressure

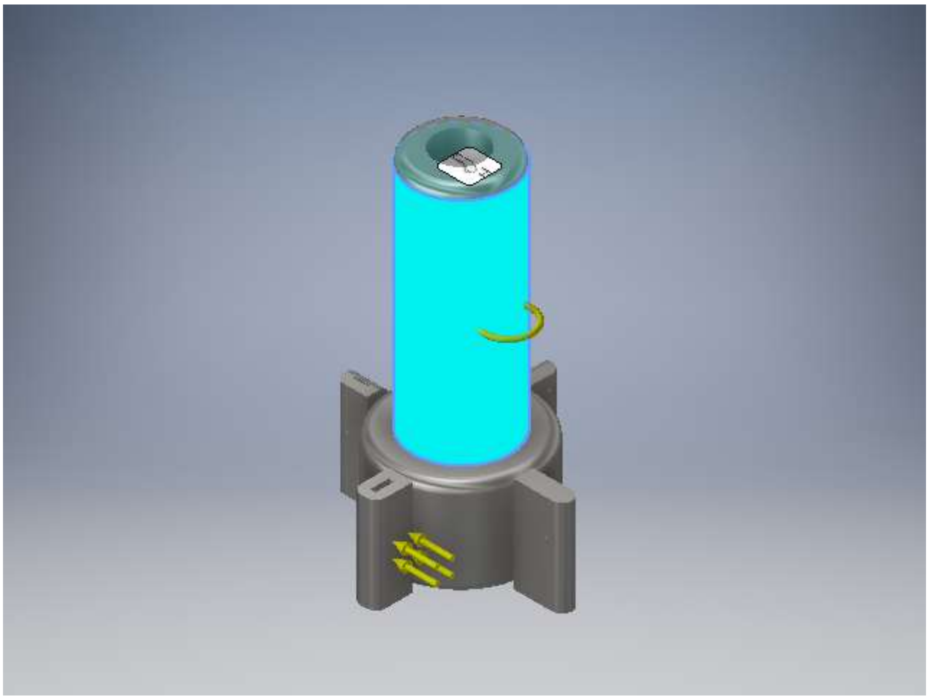

Fig1.0 (c). Gas Impeller Model with Constraint, Turning Moment and Gas Pressure

3.0 MODEL ANALYSIS

The stress components in impeller elements are given as below.

$(\sigma_x)_n = \frac{E}{(1+v)(1-2v)}[(1-v)a_n + ve_n] \ldots..(1.0)$ (Westmann, 2004)

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

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

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

The displacement field is shown below.

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

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

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

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

The principal strains are given below

$e_x = \frac{1}{E}\left[\sigma_x - \frac{1}{m}\left(\sigma_y + \sigma_z\right)\right] \; ..... (7.0)$ (Rajput, 2008).

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

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

Von Mises Stress can be given as below.

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

$Angular \; velocity \; of \; rotor \; barrel, \omega = \frac{2\pi N}{60}$ rad/s …….(11) (Rajput, 2008)

Where N = rpm of rotor rotor;

$Work \; done \; per \; seconds = Torque \times angular \; velocity \; ......$ (12.0)

$Discharge = \pi D_2 B_2 \times V_{f2} ......$ (13.0)

$$B_2 = width \; of \; impeller \; at \; exit$$

$$D_2 = diameter \; of \; impeller \; at \; exit$$

$$V_{f2} = velocity \; of \; flow \; at \; exit = u_2$$

4.0 RESULTS

Table 7: 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	231.764 N	231.764 N	14.5062 N m	0 N m
		0 N		4.15502 N m
		0 N		13.8984 N m

According to table 7, the gas impeller under screwed fixed constraints was subjected to gas pressure of 2MPa. It was observed that the maximum reaction moment and force at the fixed points was 14.5062 N m and 231.764N, this result suggested that the impeller barrel would require a fastening nut whose nominal diameter is not less than 24mm to achieve effective stability.

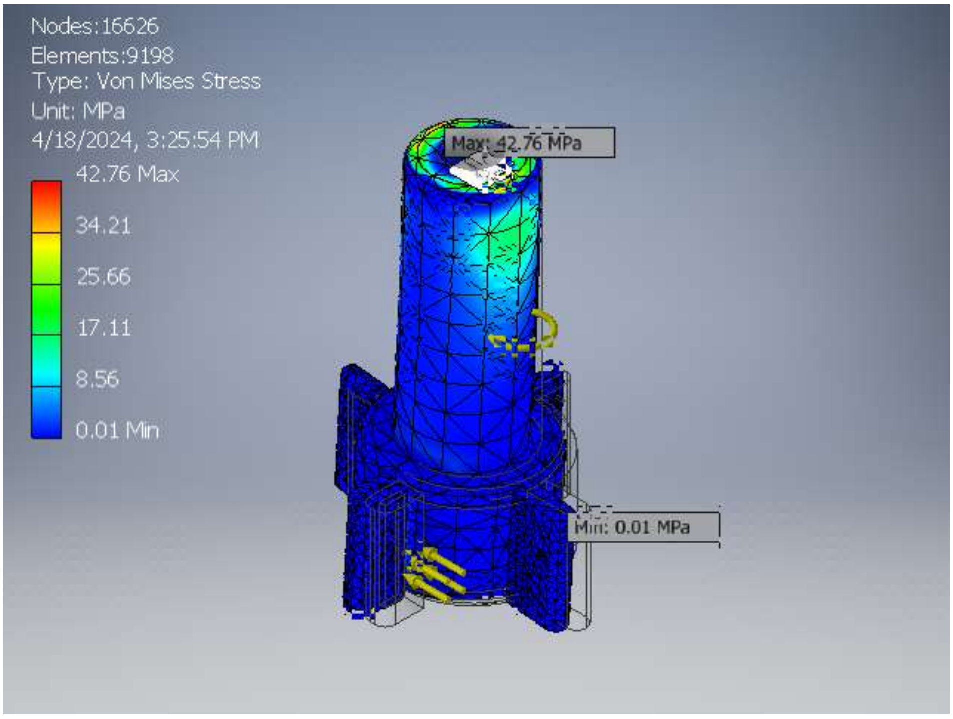

Fig2.0. Von Mises Stress

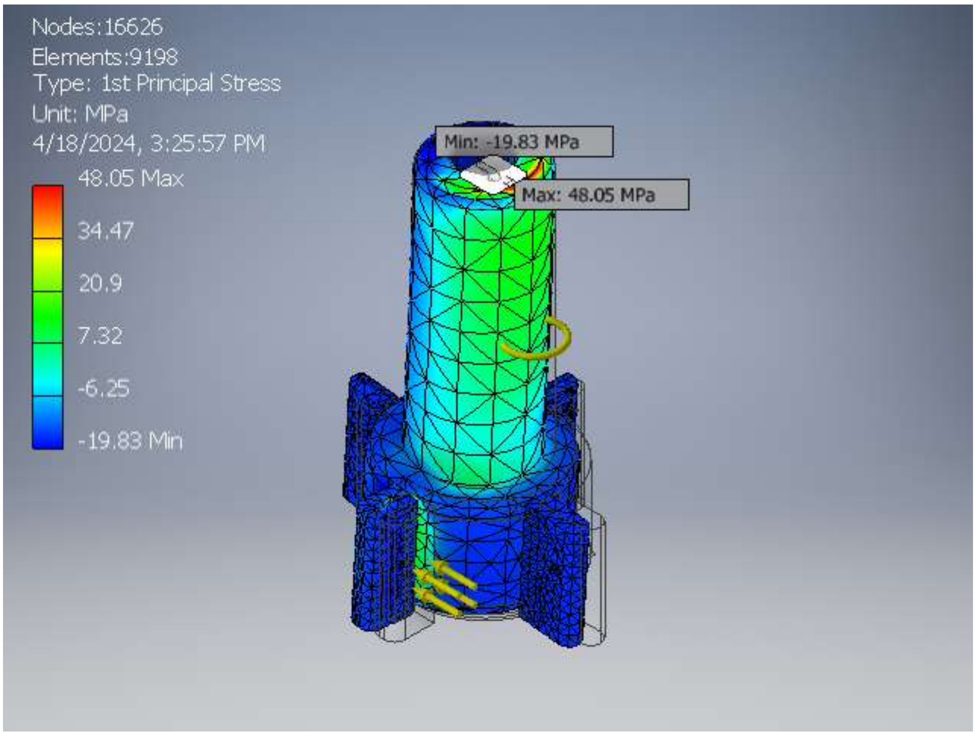

Fig3.0. 1st Principal Stress

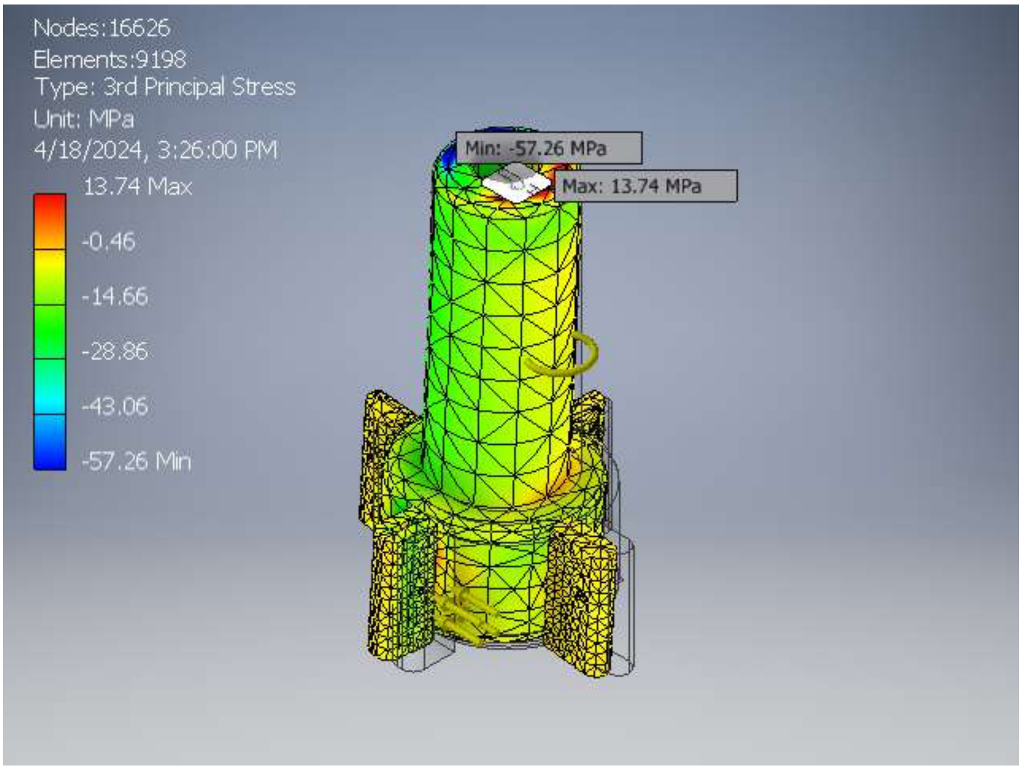

Fig4. 3rd Principal Stress

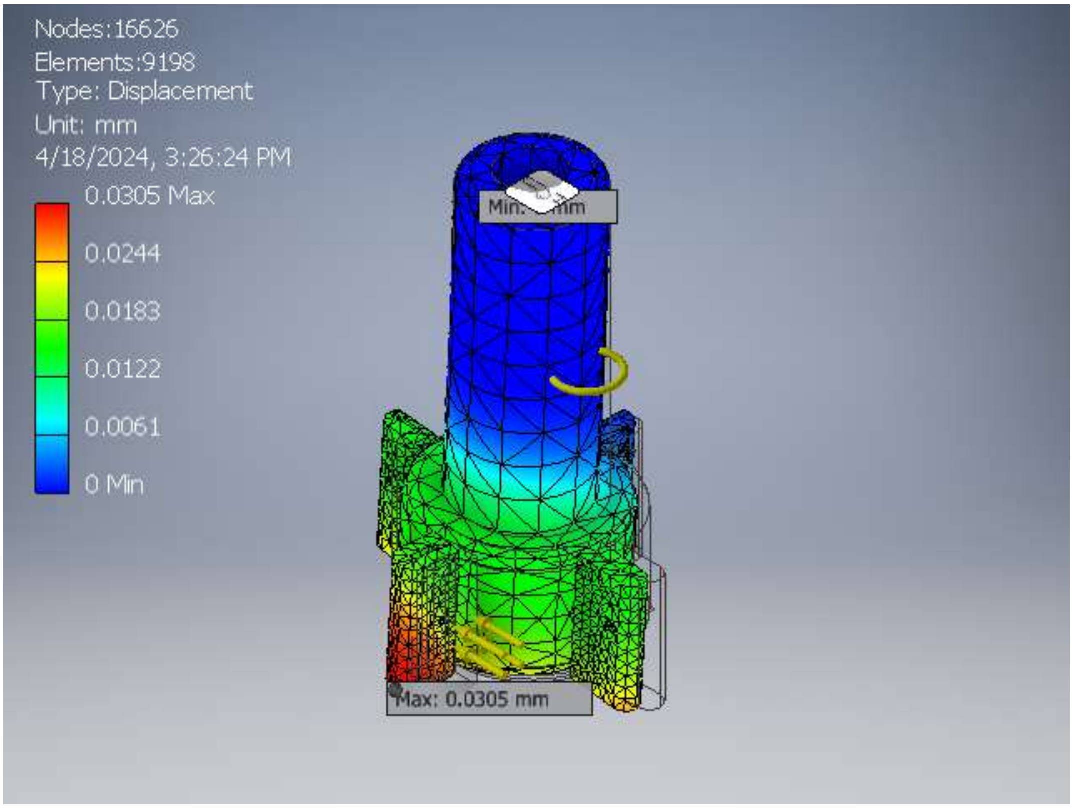

Fig5.0(a). Maximum Displacement

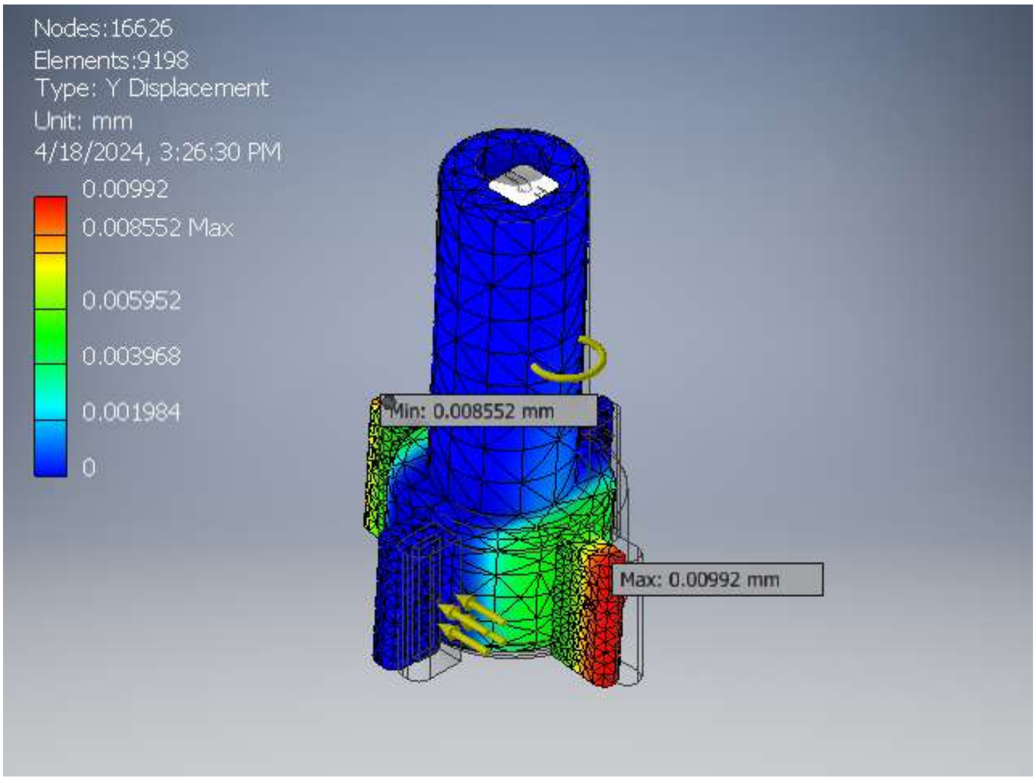

Fig 5.0(b). Maximum Displacement

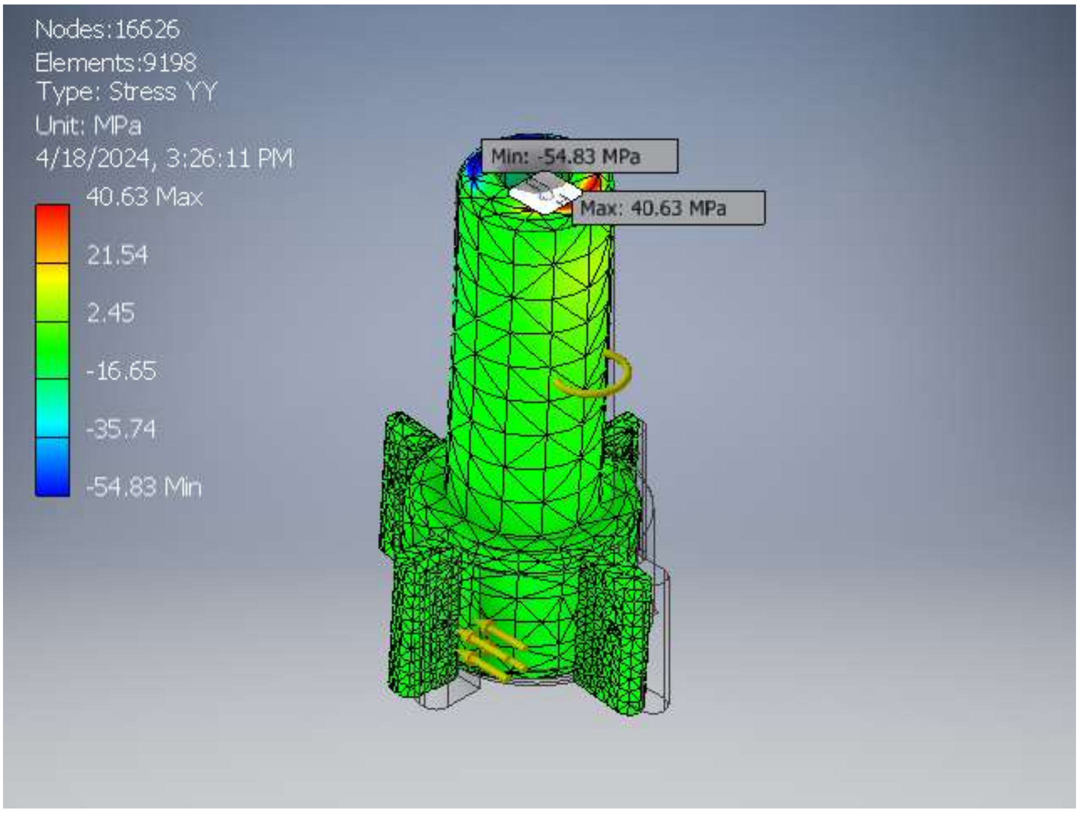

Fig6.0. Maximum Stress Induced

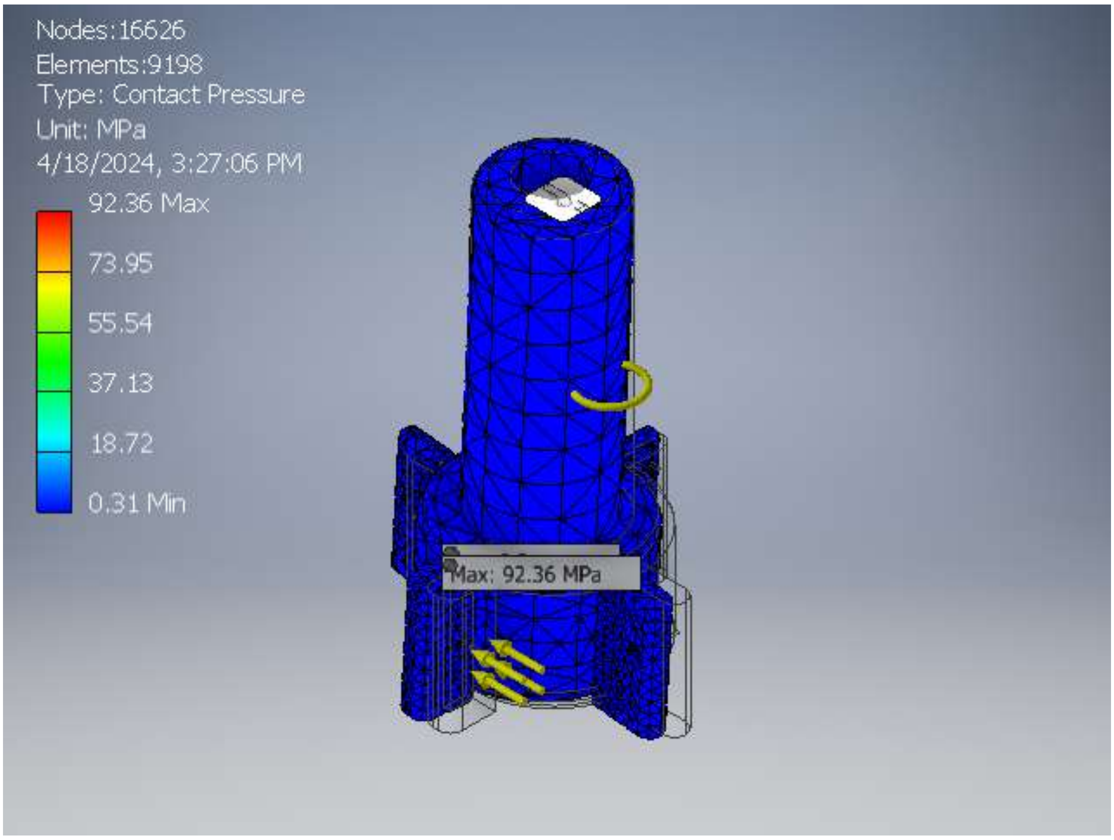

Fig7.0. Maximum Contact Pressure

According to **Fig 2**, results showed that the Von Mises stress was found to be 42.7555 MPa. Since, yield strength of the assigned material is 689 MPa; it suggested that failure of gas impeller material due to yielding load is not possible under the given conditions.

According to **Fig 4** and **Fig 4**, the 1st and 3rd principal stresses were found to be 48.0505 MPa and 13.7433 MPa respectively. These results indicated that the blades material would fail due to tensile stress rather than compressive stress; this result, suggested that to improve reliability and stability of operation, excessive loading of impeller blades must be avoided. Stress concentration was found to be much at the, weld heads and tails of blades, as well as threaded barrel section, according **fig 5.0 to 6.0.**

Fig 5.0 showed that the maximum displacement/deformation of impeller due to gas pressure was found to be 0.0305028 mm. This suggested that the material used showed lower deformation response and could improve service life of gas pumps. Also, the maximum induced stress was found to be 40.6269 MPa where as the ultimate tensile strength of material used for the gas impeller was 861.25 MPa. This suggested that the design was safe and not over stressed.

Table 8: Result Summary

Name	**Minimum**	**Maximum**
Volume	18266.5 mm^3	
Mass	0.141565 kg	
Von Mises Stress	0.00518181 MPa	42.7555 MPa
1st Principal Stress	-19.8304 MPa	48.0505 MPa
3rd Principal Stress	-57.2624 MPa	13.7433 MPa
Displacement	0 mm	0.0305028 mm
Safety Factor	15 ul	15 ul
Stress XX	-24.9972 MPa	32.2859 MPa

Stress XY	-19.3213 MPa	10.5766 MPa
Stress XZ	-8.90444 MPa	8.73269 MPa
Stress YY	-54.8265 MPa	40.6269 MPa
Stress YZ	-15.0842 MPa	12.1686 MPa
Stress ZZ	-22.5993 MPa	27.2046 MPa
X Displacement	-0.0304515 mm	0.0000382809 mm
Y Displacement	-0.00992044 mm	0.00855242 mm
Z Displacement	-0.00544593 mm	0.00627197 mm
Equivalent Strain	0.0000000212946 ul	0.000180752 ul
1st Principal Strain	-0.0000034957 ul	0.000207923 ul
3rd Principal Strain	-0.000220902 ul	0.0000000161587 ul
Strain XX	-0.00011846 ul	0.000153751 ul
Strain XY	-0.000118713 ul	0.0000649842 ul
Strain XZ	-0.0000547104 ul	0.0000536551 ul
Strain YY	-0.000205936 ul	0.000154183 ul
Strain YZ	-0.00009268 ul	0.000074766 ul
Strain ZZ	-0.0000978954 ul	0.00011665 ul
Contact Pressure	0 MPa	92.3609 MPa
Contact Pressure X	-35.4892 MPa	6.64727 MPa
Contact Pressure Y	-4.85898 MPa	7.03539 MPa
Contact Pressure Z	-85.208 MPa	37.6959 MPa

5.0 CONCLUSION

According to the findings, it can be deduced that the values of stress and displacement acting on rotary gas impeller during operation, influences impeller reliability and stability and hence, control limits must be set to stabilize overall performance.

6.0 RECOMMENDATIONS

The following recommendations are suggested based on the study:

1) Excessive loading of gas impeller blade along YY axis must be avoided to reduce stress and displacement within permissible limit.
2) Gas impeller blade material must have higher tensile strength rather than compressive strength, since failure due to tensile stress is predominant.
3) This research can also be done in future using different impeller blade designs and other advanced software for generalization.

7.0 LIMITATION OF THE STUDY

In the course of carrying out this research, design and stress evaluation of rotary gas impeller for performance improvement, using finite element simulation method, 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 electricity, material sourcing, etc.

8.0 ACKNOWLEDGEMENTS

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9.0 FUNDING

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10.0 DISCLOSURE OF CONFLICT OF INTEREST

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

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