Design, Analysis & Weight Reduction of Shell of Refrigerator Compressor

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Abstract - *The refrigerator is a device used for cooling purpose. It is one of the home appliances using mechanical vapour compression cycle in its process. Performance of the systems become the main issue and many researches are still ongoing to evaluate and improve the efficiency of any used system. Refrigerator consist of various important part such as, compressor, condenser and evaporator. These all components play main role in efficiency of refrigerator. If such components are having the heavy system then it may decrease the overall efficiency of refrigerator as they consume the more amount of power. Hence in this project we are working on the shell and piston of compressor in refrigerator. As we known that the compressor used in refrigerator is a reciprocating type. Connecting rod is the intermediate link between the piston and the crank shaft. And is responsible to transmit the push and pull from the piston pin to crank pin, thus converting the reciprocating motion of the piston to rotary motion of the crank. The loads acting on this piston and crank shaft are cyclic in nature. Further in this project we will use the modelling and analysis software and ensure the weight reduction and optimisation of system. The modelling software is CATIA and analysis software is ANSYS.*

***Key Words*: Optimisation, Pressure vessel, Compressor, Efficiency, Piston**

1. INTRODUCTION

The refrigerator is used for cooling purpose. The domestic refrigerator contains the sealed compressor. This compressor is sealed because to avoid the noise. It also consists of crank shaft, connecting rod, piston, etc. In this project the optimization of piston and compressor shell will be done. This will decrease the overall weight of the system and also increase the efficiency of the system. Due to the low weight of system like Shell, piston and crank shaft the power consumption of system will be decrease and it will conclude on the increase in the efficiency of the system. This project is unique and very less work is being done on refrigerator compressor.

1.1 Technical Specifications of compressor shell used for study

Thus, Specification of the compressor shell are tabulated below:

Sr. No.	Parameter	Values
1	Diameters ID OD	 140 mm 148 mm
2	Length of Pressure vessel	190 mm
3	Internal Pressure	247 Psi
4	Thickness	4 mm
Material Specifications of MS		
5	Tensile Strength, Yield	590 Mpa
6	Force	8 kg

Table 1.1 – Specifications of existing shell

Fig. 1.1 – Photographic view of existing shell

2. LITERATURE REVIEW

[1] Ashwani Kumar, Shaik Imran Behmad, Pravin P Patil et al The main objective of this research work is to investigate and analyse the stress distribution of piston at actual engine condition. This research work suggests a new type of SiC reinforced ZrB2 composite material that can sustain at higher temperature (1680 K) and pressure (18 MPa).

[2] Jatender Datta, Dr. Sahib Sartaj Singh et al The paper shows the behavior of piston made of Carbon Graphite and Aluminum Alloy 2618 applied heat power value of 200 Watt. The result of Temperature distribution and resultant temperature gradient was found and the main motive is to find the comparison between both of materials of piston.

[3] K Ramesh Babu, G Guru Mahesh and G Harinath Gowd et al In this paper the authors have studied the variation of Isotherms and heat flux with respect to radius, height of piston, liner, cylinder head and thermal analysis. First thermal analysis was done and analyzed the temperature distribution over the convectional engine and copper coated convectional engine. In the second stage structural analysis was carried out using the thermal loads obtained in the first stage. Three different types of materials were taken for analysis.

[4] Dilip Kumar Sonar, Madhura Chattopadhyay et al The authors had studied a piston which is designed using CATIA V5R20 software. Complete design is imported to ANSYS 14.5 software then analysis is performed. Aluminium alloy has been selected for structural and thermal analysis of piston. Results are shown and a comparison is made to find the most suited design.

3. PROBLEM DEFINITION AND METHODOLOGY

A. Problem Definition

We have to reduce the overall weight of the system. The load bearing components like piston and crank shaft are used in refrigerations and are also heavy in assembly. The Shell of the compressor is also heavy. This heavy component decreases the overall efficiency. Due to the heavy weight components the power consumption of system increases and thus it is not good as it increases the running cost of the system. The material requirement also increases as the component is heavy. Hence to overcome all this problem this system should be redesign for optimisation.

B. Objectives

1. To study the current system in detail with its specification and all required considerations.
2. To design & optimize the existing material for compressor shell and piston.
3. To optimize system according to one of the following:

a) Changing dimensions of system and keeping material same as it is.

b) Keeping same dimensions and changing material of components.

c) Changing both material as well as dimensions of component.

4. Modelling of new design with help of CATIA software.
5. To analyse the optimized components to study the stress on the system.
6. To compare existing & optimized piston & shell of compressor.

C. Methodology

The following are important steps for completion of objectives -

1. Check design of various existing components in compressor.

2. Creating geometric model and finite element model of existing components of compressor using CATIA software.

3. Analysis of shell of compressor and piston by using ANSYS software.

4. Simulations for Model Analysis.

5. Optimization of compressor assembly for weight reduction.

6. Comparison between existing and optimized design.

4. DESIGN & ANALYSIS OF EXISTING COMPRESSOR SHELL

In compressor the shell carries the whole system in it. It consists the motor, cylinder, piston and other required components. The motor, piston cylinder arrangement is mounted on a spring and bolt arrangement. The weight of this system is nearly 8 Kg.

After considering the factor of safety as 2.5 the load on pressure vessel will be
$8 \times 2.5 = 20$ Kg
Thus converting 20 Kg in Force, we get,
$20 \times 9.81 = 196.2$ N
Material - MS
For given material we have to select standard properties of that material, such as
Syt= 590 Mpa
ρ= 7860 kg/m^3
Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for pressure vessel.
Allowable Stress (σall) = Syt / Fs
= 590/2.5
= 236 Mpa.
Thus, the load acting on the shell is at its bottom plate, and other outer surface is used only for preventing the noise of the Compressor. Thus, this load acting on the system after considering factor of safety is 20 Kg, after converting it into the Force we get the Force of 197 N. Thus, the shell will be designed according to this force.
Considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows

$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa

Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.

Deflection on the Shell is given as follows
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm

The there is scope for geometric optimization if we Compare the Stress.

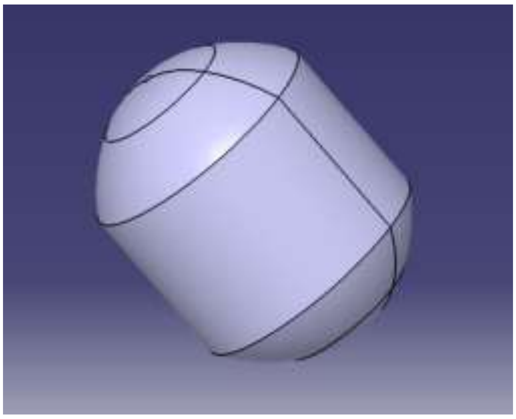

Fig. 4.1 – Geometric model of shell

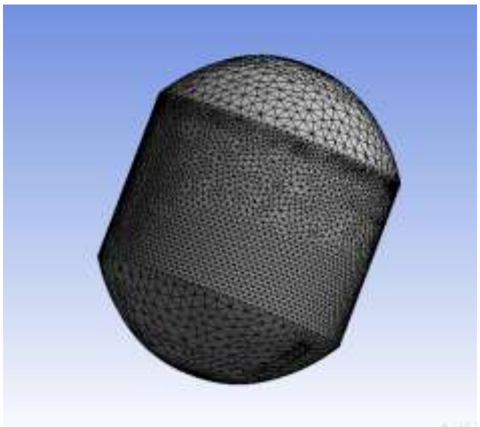

Fig. 4.2 – Meshing of shell

Fig. 4.3 – Boundary conditions for shell

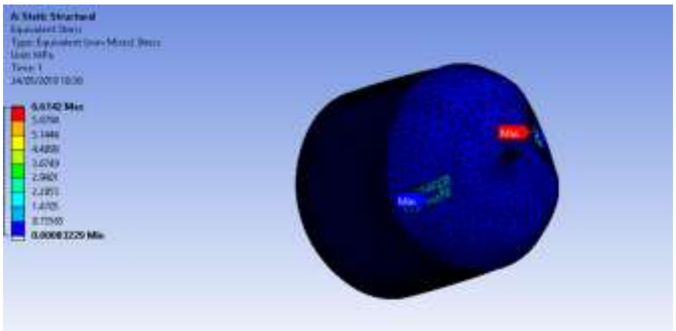

Fig. 4.4 – Stress analysis of shell

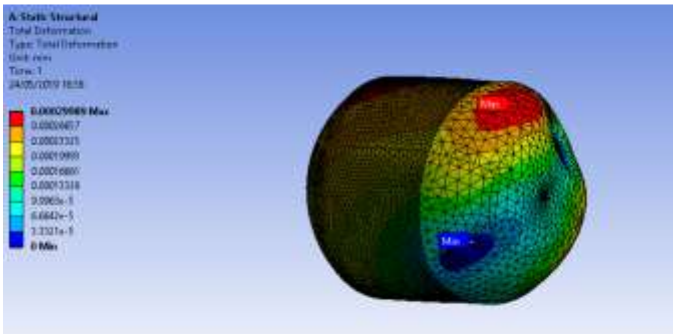

Fig. 4.5 Deformation analysis of shell

5. DESIGN & ANALYSIS OF OPTIMIZED SHELL

In compressor the shell carries the whole system in it. It consists the motor, cylinder, piston and other required components. The motor, cylinder piston arrangement is mounted on a spring and bolt arrangement. Weight of this system is nearly 8 Kg. After considering the factor of safety as 2.5 the load on pressure vessel will be

$8 \times 2.5 = 20$ Kg

Thus converting 20 Kg in Force We get,

$20 \times 9.81 = 196.2$ N

Case 1) Changing dimensions of system and keeping material same as it is:

In this case the dimensions of the system are changed by iterative method and material is kept same as that of existing system.
Consider factor of safety as 2.5,
Material - MS
For given material we have to select standard properties of that material, such as
E= 2.10×10^5 Mpa
Syt= 590 Mpa
ρ= 7860 kg/m^3
Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for shell.
Allowable Stress (σall) = Syt / Fs
= 590/2.5
= 236 Mpa.
Thus, the load acting on the shell is at its bottom plate, and other outer surface is used only for preventing the noise of the Compressor. Thus, this load acting on the system after considering factor of safety is 20 Kg, after converting it into the Force we get the Force of 197 N. Thus, the shell will be designed according to this force.
Sample iteration
Outer Diameter = 148 mm
Inner Diameter = 140 mm
Thickness 4 mm
Considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows
$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa
Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.
Deflection on the Shell is given as follows,
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm

Same like this, 12 such iterations are carried on. The following table shows the iterations

Thick	Inner	Outer	Stress	Final Weight	Difference
4	140	148	9.8	4.74	0.00
3.5	141	148	12.8	4.17	0.57
3	142	148	17.4	3.59	1.15
2.5	143	148	25.0	3.01	1.73
2	144	148	39.1	2.42	2.32
1	146	148	156.3	1.22	3.52
4	137	145	9.6	4.60	0.14
3.5	138	145	12.6	4.05	0.69
3	139	145	17.1	3.49	1.25
2.5	140	145	24.6	2.92	1.82
2	141	145	38.5	2.35	2.39
1	143	145	153.8	1.19	3.55

Table 5.1 - Iteration Table for changing dimensions & keeping same material

Case 2) Keeping same dimensions and changing material of components:

In this case the dimensions of existing systems are kept as it is and material and its properties are changed. While selecting these three materials the availability, cost and required strengths are checked. The three materials used in this system are as follows:

1. A 36 hot rolled
2. Stainless steel 304
3. Aluminium alloys

Iteration (1):
Material – A 36 (Hot rolled)
For given material we have to select standard properties of that material, such as
E= 2×10^5 Mpa
Syt= 400 Mpa
ρ= 7800 kg/m3
Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for optimised roller.
Allowable Stress (σall) = Syt / Fs
= 590/2.5
= 236 Mpa.

Considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows
$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa
Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.
Deflection on the Shell is given as follows,
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm
Iteration (2):
Material – Stainless Steel 304
For given material we have to select standard properties of that material, such as

E= 2×105 Mpa
Syt= 505 Mpa
ρ= 8030 kg/m3
Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for optimised roller.
Allowable Stress (σall) = Syt / Fs
= 505/2.5
= 202 Mpa.

Considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows
$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa
Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.
Deflection on the Shell is given as follows,
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm
Mass of the pressure vessel:
Density = Mass/ Volume
Volume = $\prod \times h (r_1^2 - r_2^2) + 4/3 * \prod \times (r_1^3 - r_2^3)$

$= 0.00060350$ m^3

Mass= 0.00060350 × 8030

= 4.8 Kg

Iteration (3):

Material: Aluminium alloys

Density	2800 kg/m3
Tensile strength, Ultimate	900 Mpa
Tensile Strength, Yield	600 Mpa

Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for optimised roller.
Allowable Stress (σall) = Syt / Fs
= 600/2.5
= 240 Mpa.

Thus, Stress on pressure vessel
Considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows
$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa
Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.

Deflection on the Shell is given as follows,
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm
Mass of the pressure vessel:

Density = Mass/ Volume
Volume = $\prod \times h (r_1^2 - r_2^2) + 4/3 * \prod \times (r_1^3 - r_2^3)$

$= 0.00060350$ m^3

Mass= 0.00060350 × 2800

= 1.69 Kg

Case 3) Changing both material as well as dimensions of component:

In this case the dimensions as well as material and its properties are changed. While selecting these three materials the availability, cost and required strengths are checked. The three materials used in this system are as follows:

1. A 36 hot rolled
2. Stainless steel 304
3. Aluminium alloys

Iteration (1):

Material – A 36 (Hot rolled)
For given material we have to select standard properties of that material, such as
E= 2×10^5 Mpa
Syt= 400 Mpa
ρ= 7800 kg/m3
Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for optimised roller.
Allowable Stress (σall) = Syt / Fs
= 590/2.5
= 236 Mpa.

Thus, Stress on shell
Considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows
$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa

Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.
Deflection on the Shell is given as follows,
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm
Mass of the pressure vessel:
Density = Mass/ Volume
Volume = $\prod \times h (r_1^2 - r_2^2) + 4/3 * \prod \times (r_1^3 - r_2^3)$

$= 0.00060350 \ m^3$

Mass= 0.00060350 × 8030

= 4.7 Kg

Same like above iteration following 12 iterations was carried out, and tabulated below.

Thick	Inner	Outer	Stress	Final Weight	Difference
4	140	148	9.8	4.71	0.03
3.5	141	148	12.8	4.14	0.60
3	142	148	17.4	3.57	1.17
2.5	143	148	25.0	2.99	1.75
2	144	148	39.1	2.40	2.34
1	146	148	156.3	1.21	3.53
4	137	145	9.6	4.57	0.17
3.5	138	145	12.6	4.02	0.72
3	139	145	17.1	3.46	1.28
2.5	140	145	24.6	2.90	1.84
2	141	145	38.5	2.33	2.41
1	143	145	153.8	1.18	3.56

Table 5.2 - Iteration table for changing both dimensions as well as material

Iteration (2):
Material – Stainless Steel 304
For given material we have to select standard properties of that material, such as
E= 2×105 Mpa
Syt= 505 Mpa
ρ= 8030 kg/m3
Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for optimised roller.
Allowable Stress (σall) = Syt / Fs
= 505/2.5
= 202 Mpa.

Thus, Stress on pressure vessel

Considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows

$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa
Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.
Deflection on the Shell is given as follows,
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm
Mass of the pressure vessel:

Density = Mass/ Volume
Volume = $\prod \times h (r_1^2 - r_2^2) + 4/3 * \prod \times (r_1^3 - r_2^3)$

$= 0.00060350 \ m^3$

Mass= 0.00060350 × 8030

= 4.8 Kg

Same like above iteration following 10 iterations was carried out, and tabulated below.

Thick	Inner	Outer	Stress	Final Weight	Difference
4	140	148	9.8	4.85	-0.11
3.5	141	148	12.8	4.26	0.48
3	142	148	17.4	3.67	1.07
2.5	143	148	25.0	3.07	1.67
2	144	148	39.1	2.47	2.27
1	146	148	156.3	1.25	3.49
4	137	145	9.6	4.70	0.04
3.5	138	145	12.6	4.14	0.60
3	139	145	17.1	3.56	1.18
2.5	140	145	24.6	2.98	1.76
2	141	145	38.5	2.40	2.34
1	143	145	153.8	1.21	3.53

Table 5.3 - Iteration table for changing both dimensions as well as material

Iteration (3):

Material: Aluminium alloys

Density	2800 kg/m3
Tensile strength, Ultimate	900 MPa
Tensile Strength, Yield	600 MPa
Melting Point	1370-1430°C

Considering uniformly distributed load & FOS as 2.5
We have to calculate actual FOS for optimised roller.
Allowable Stress (σall) = Syt / Fs
= 600/2.5
= 240 Mpa.

Thus, Stress on pressure vessel considering the force is exerted in the circular plate at bottom of compressor, we get the stress as follows
$M_{max} = (P/4 \times \prod) \times (1 + \mu) \ln (r/R)$
$= (197/ 4 \times \prod) \times (1+0.3) \ln (74/20.5)$
$= 26.16$ N-mm
$\sigma = (6M) / t^2$
$= (6 \times 26.16) / 4^2$
$= 9.81$ Mpa
Thus, of we compare this stress with allowable then it is very less. Thus, the Optimization can be achieved in the system.
Deflection on the Shell is given as follows,
$\delta = 0.217 \times F \times r^2 / E \times t^3$
$= 0.217 \times 197 \times 74^4 / 2.1 \times 10^5 \times 4^3$
$= 0.017$ mm
Mass of the pressure vessel:

Density = Mass/ Volume
Volume $= \prod \times h (r_1^2 - r_2^2) + 4/3 * \prod \times (r_1^3 - r_2^3)$

$= 0.00060350 \ m^3$

Hence, Mass = 1.7 Kg

Same like above iteration following 12 iterations was carried out, and tabulated below:

Thick	Inner	Outer	Stress	Final Weight	Difference
4	140	148	9.8	1.69	3.05
3.5	141	148	12.8	1.49	3.25
3	142	148	17.4	1.28	3.46
2.5	143	148	25.0	1.07	3.67
2	144	148	39.1	0.86	3.88
1	146	148	156.3	0.44	4.30
4	137	145	9.6	1.64	3.10
3.5	138	145	12.6	1.44	3.30
3	139	145	17.1	1.24	3.50
2.5	140	145	24.6	1.04	3.70
2	141	145	38.5	0.84	3.90
1	143	145	153.8	0.42	4.32

Table 5.4 - Iteration table for changing both dimensions as well as material

In above study we are more weight reductions. But if we think about the cost of the aluminum alloys then it is more than the steels.

All Iterations According to the all three cases are tabulated below:

	Thick	Inner	Outer	Stress	Final Weight	Diff.
MS	4	140	148	9.8	4.74	0.00
	3.5	141	148	12.8	4.17	0.57
	3	142	148	17.4	3.59	1.15
	2.5	143	148	25.0	3.01	1.73
	2	144	148	39.1	2.42	2.32
	1	146	148	156.3	1.22	3.52
	4	137	145	9.6	4.60	0.14
	3.5	138	145	12.6	4.05	0.69
	3	139	145	17.1	3.49	1.25
	2.5	140	145	24.6	2.92	1.82
	2	141	145	38.5	2.35	2.39
	1	143	145	153.8	1.19	3.55
MS	4	140	148	9.8	4.74	0.00
SS 304	4	140	148	9.8	4.85	-0.11
A 36	4	140	148	9.8	4.71	0.03
AL ALL.	4	140	148	9.8	1.69	3.05
SS 304	4	140	148	9.8	4.85	-0.11
	3.5	141	148	12.8	4.26	0.48
	3	142	148	17.4	3.67	1.07
	2.5	143	148	25.0	3.07	1.67
	2	144	148	39.1	2.47	2.27
	1	146	148	156.3	1.25	3.49
	4	137	145	9.6	4.70	0.04
	3.5	138	145	12.6	4.14	0.60
	3	139	145	17.1	3.56	1.18

	2.5	140	145	24.6	2.98	1.76
	2	141	145	38.5	2.40	2.34
	1	143	145	153.8	1.21	3.53
	4	140	148	9.8	4.71	0.03
	3.5	141	148	12.8	4.14	0.60
	3	142	148	17.4	3.57	1.17
	2.5	143	148	25.0	2.99	1.75
	2	144	148	39.1	2.40	2.34
A 36	1	146	148	156.3	1.21	3.53
	4	137	145	9.6	4.57	0.17
	3.5	138	145	12.6	4.02	0.72
	3	139	145	17.1	3.46	1.28
	2.5	140	145	24.6	2.90	1.84
	2	141	145	38.5	2.33	2.41
	1	143	145	153.8	1.18	3.56
	4	140	148	9.8	1.69	3.05
	3.5	141	148	12.8	1.49	3.25
	3	142	148	17.4	1.28	3.46
	2.5	143	148	25.0	1.07	3.67
	2	144	148	39.1	0.86	3.88
AL ALL.	1	146	148	156.3	0.44	4.30
	4	137	145	9.6	1.64	3.10
	3.5	138	145	12.6	1.44	3.30
	3	139	145	17.1	1.24	3.50
	2.5	140	145	24.6	1.04	3.70
	2	141	145	38.5	0.84	3.90
	1	143	145	153.8	0.42	4.32

Table 5.5 - Iteration table according to all three cases

After studying all above iterations, the following shell was selected. It is as follows:

Thickness = 1 mm
Outer Diameter = 148 mm
Inner diameter = 146 mm
Material = Mild steel

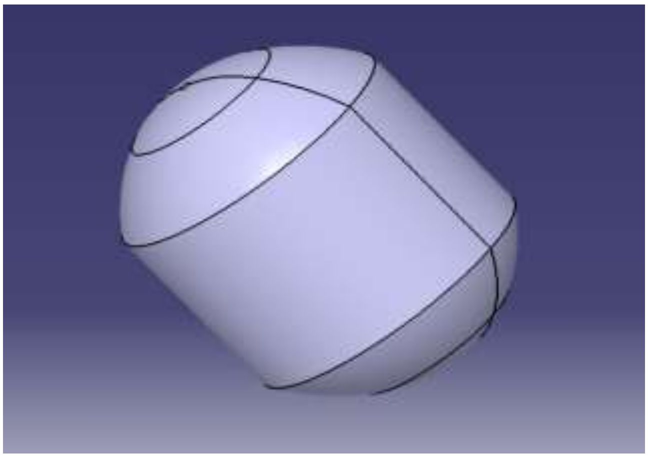

Fig. 5.1 – Geometric model of optimized shell

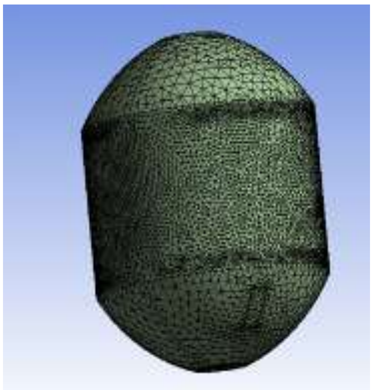

Fig. 5.2 – Meshing of optimized shell

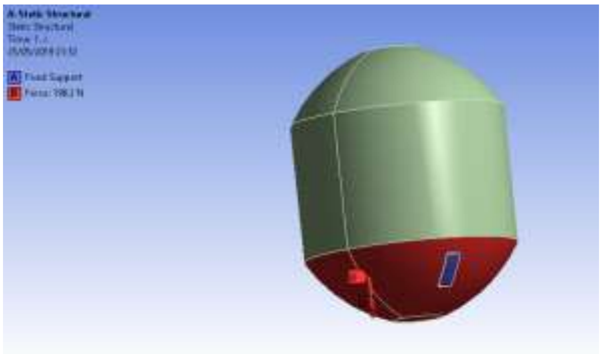

Fig. 5.3 – Boundary conditions for optimized shell

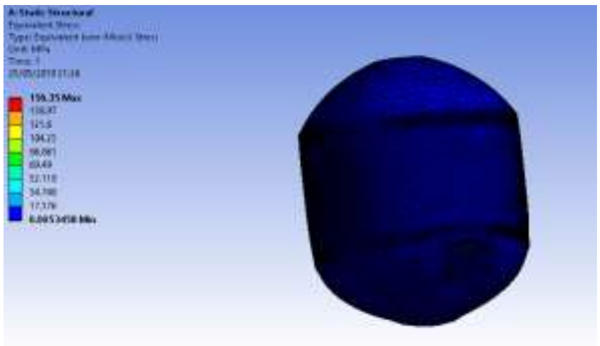

Fig. 5.4 – Stress analysis of optimized shell

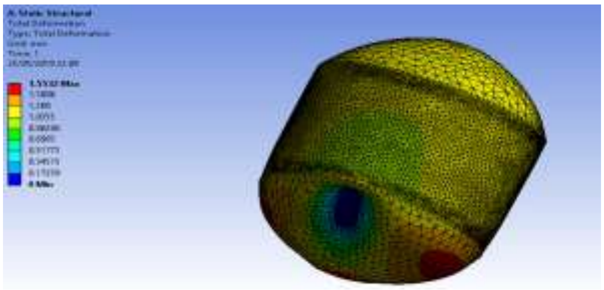

Fig. 5.5 – Deformation analysis of optimized shell

6. RESULT AND DISCUSSIONS

After studying all above iterations, the following shell was selected. It is as follows:

Thickness = 1 mm

Outer Diameter = 148 mm

Inner diameter = 146 mm

Material = Mild steel

Total weight reduction of 3.52 Kg was obtained. It means total optimisation of 74.89 % is achieved.

Comparison of Stress on basis of Theoretical and analytical results:

Dimensions	Design calculations Results		Ansys Results	
	Stress	Deflection	Stress	Deflection
Thickness = 1 mm Outer Diameter = 148 mm Inner diameter = 146 mm	156.3 Mpa	1.11 mm	156.35 Mpa	1.55 mm

Table 6.1 - Theoretical & analytical comparison of shell

7. CONCLUSION

Optimization was achieved on Shell of compressor. Some of components of system like motor assembly and vents are kept as it is due to its proper design. Design calculations, analysis model, and optimized system are compared on stress basis. The weight reduction achieved on shell does not affect the load carrying capacity of system. **3.53 Kg** weight reduction is achieved by optimize design than existing design. **74.41 % of material was saved** on optimized system than existing system which further save cost of system.

8. FUTURE SCOPE

- The use of composite materials can be done for more weight reduction. This weight reduction can also bring a lightest compressor.
- The skotch yoke mechanism is used to get rotary motion from the piston, this system can be optimised.
- Use of heat treatments can also increase the surface strength and thus again thickness can be reduced.
- Vibration analysis of whole system can be increase working capacity of system

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