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Original Article

Mathematical Design and Simulation of Double Acting Reciprocating Compressor

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Abstract *- Many compressor applications require maximum compressed air discharge in the minimum time. The system analysis considered parameters such as efficiency, free air delivery, time for filling the tank, and running costs. Based on the instruments utilized in sectors that typically run at a maximum pressure of 9 kg/cm^2, compressed air is utilized. The analysis involved designing a double acting reciprocating compressor based on the available specifications of a single acting reciprocating compressor (Model - SS 05090 HN), ELGI make, based on the tools used in the industries to achieve the same specification to pressure 9 Kg/cm^2 at 925 rpm for piston displacement of 0.606 m^3/min(21.4 cfm). To fill a receiver tank of 220 liters for the single acting compressor, it takes 29.87 seconds, whereas the double acting compressor takes only 9.4964 seconds. This demonstrates that the discharge rate of the double acting compressor is almost twice, and the running cost is reduced by Rs. 242.13 /- for eight hours of operation per day. This analysis highlights the superior performance, cost-effectiveness, operational efficiency, and financial viability of the double acting reciprocating air compressor, showing that it yields optimal results compared to the single acting reciprocating compressor.*

Keywords *- Optimum design, Reciprocating compressor, Free air delivery.*

1. Introduction

A comprehensive understanding of compressor technology is essential to meet the growing demand. Most of the existing research has focused on single acting reciprocating compressors, which are commonly employed in many industrial applications due to their simpler design and lower manufacturing cost. Studies have extensively explored thermal management, energy efficiency, noise reduction, and advanced control systems to optimize the performance of these compressors. Moreover, significant research has been dedicated to the use of advanced materials aimed at improving durability and reducing weight.

However, despite these advancements, double acting reciprocating compressors - which allow suction and compression on both sides of the piston, effectively doubling the air delivery for the same crankshaft speed - have received comparatively less attention. While the double acting design theoretically improves volumetric efficiency and output, few studies have presented detailed component-level design and analysis, including piston, cylinder, valves, and crankshaft assembly, which is one of the important needs for any compressor industry. This need has given the path for designing the double acting air compressor by selecting a suitable material [6].

2. Literature Review

Budagyan, A.P. et al. [1] and Plastinin focused on the design and development of reciprocating compressors, closely examining the effects of temperature variations on cooling and overall operation. Additionally, they emphasized the importance of optimizing the compressors' basic geometric dimensions. A. Shahrivar and A. R. Abdolmaleki [2] noted that the longevity and functionality of a connecting rod system can be significantly impacted by improper material selection. To withstand dynamic and cyclic stresses, materials with high mechanical strength, hardness, and excellent tensile and fatigue properties are essential. Studies have shown that alloys like titanium and forged steel are preferred due to their reliability under wear and stress. Poor material choices lead to early failure, reduced engine efficiency, and increased maintenance costs.

Shenoy P. S. [3] highlighted the crucial role of connecting rod caps and bolts in securing the crankshaft-rod assembly under high dynamic and cyclic stresses caused by piston acceleration and motor power transmission. Research has focused on fatigue analysis, optimal material selection, and structural improvements to prevent failure. These factors are especially critical in high-performance applications such as natural gas compression, chemical processing, and cryogenic systems. Gu Z. and associates [4] confirmed that improper material selection in reciprocating compressors severely affects the performance and lifespan of connecting rods. Materials with high tensile strength, hardness, and fatigue resistance are necessary to endure dynamic and cyclic loads. Titanium and forged steel alloys are often selected for their superior mechanical and wear properties. Optimized material selection is crucial, as poor choices can lead to early failure, reduced efficiency, and increased maintenance. B.J. Kim et al. [6] pointed out that reciprocating compressors play a vital role in efficiently compressing gases in industries such as chemical processing and natural gas. Components like the connecting rod, rod cap, crankshaft, crank pin, and bolts face complex dynamic loading. Research shows these components endure severe cyclic loads due to rapid piston motion and mechanical inputs from electric or diesel drives. These dynamic forces reduce fatigue life and reliability, highlighting the importance of robust design and material choices. Heinz P. Bloch et al. [7] showcased a double acting free piston expander designed to recover power in transcritical CO_2 cycles. This design enhances system efficiency by extracting energy from both sides of the piston. Studies confirm its effectiveness in improving performance in refrigeration and heat pump applications. Griza et al. [8] emphasized how advanced modeling and experimental techniques improve understanding of failure processes, stress distribution, and material behavior. These techniques enhance bolt design, reliability, and service life in critical structural applications. K. Thriveni et al. [9] conducted a static study of a crankshaft from a single-cylinder, four-stroke IC engine using ANSYS for FEA and CATIA V5 for modeling. The results matched theoretical values and showed a Von Mises stress of 15.83 MPa and a shear stress of 8.271 MPa. Future work will involve material variation and dynamic analysis.

Khare S. et al. [10] used Finite Element Analysis (FEA) to propose modified connecting rod designs to reduce stress concentrations. Optimizing component shape and material distribution significantly enhances fatigue life and durability. Author previously developed a double acting air compressor to reach 9 kg/cm² at 925 rpm more efficiently than a single acting compressor. Using CATIA V5R10, components were modeled and analyzed, confirming their performance within safe stress limits. Zhang X. et al. [11] investigated connecting rod performance through material optimization, improved manufacturing processes, and structural redesign. Research focused on fatigue resistance, lightweight materials, and FEA. Advances in forging and alternative alloys improve performance, longevity, and efficiency. In a research, non-linear finite element analysis and experimental methods to study fretting wear-induced fractures in turbine blade tang holes was applied. Fretting wear accelerates fatigue failure by acting as a stress concentrator. Their model accurately predicted fretting fatigue behavior, validated by experimental results. Fadag et al. [12] explored fretting fatigue crack growth in Ti–6Al–4V alloy using finite element methods. Their study showed that localized contact stresses significantly accelerate crack initiation and growth. FEM results closely matched experimental data, underlining the importance of accounting for fretting in components like connecting rods and bolts. He B. and Ilman M. N. [13] identified fatigue as the primary failure mode in connecting rods. Contributing factors include poor material selection, design flaws, and overload-induced bending. Additional issues, such as bolt misalignment and spalling, increase failure risk. Cracks typically originate in stress concentration areas. The literature stresses using premium materials, precise assembly, and well-designed structures to enhance reliability. Juarez C. [14] analyzed connecting rod failure in a diesel generator. Fatigue cracks originated at stress concentration zones due to unfinished surfaces and residual machining marks. Metallographic and fractographic studies revealed the impact of poor maintenance and inconsistent materials. The findings underscore the importance of precision manufacturing and proper material selection. Jan Tuhovcak, Jiří Hejčík, and Miroslav Jícha [15] emphasized the role of valve characteristics in reciprocating compressor efficiency. A numerical method based on energy balance is proposed as a simpler alternative to CFD for evaluating performance. The method was validated using p-V diagrams and performance testing of a 4-cylinder semi-hermetic compressor. P. Grolier [16] proposed an analytical formula for calculating volumetric efficiency in reciprocating compressors.

This model considers clearance volume and suction gas temperature, aiding early-stage compressor design by simulating capacity and compression work. Nitin Pawar et al. [18] emphasized the importance of water pump housing strength in high-performance engines. Studies using CREO and ANSYS demonstrated the effects of thermal and structural stresses on emissions and efficiency. Double acting compressors were tested to validate models, showing that efficiency is affected by speed, flow rate, and pressure. A quick return mechanism can reduce input power, enhance discharge, and improve overall efficiency-beneficial for oil refineries, gas pipelines, and chemical/refrigeration plants. Kush Patel, Hely Patel, Zeel Patel, Arun Kumar, and Bose Babu [21] focused on redesigning compressor pistons for better efficiency and durability. Enhancements included surface coatings for heat dissipation, new materials like high-strength alloys, and improved design. Experimental results showed reduced energy use and improved stability, promoting sustainable pneumatic systems. Hrishikesh Ganesh Jarang and Dr. R. S. Deshpande [22] designed a compact double acting reciprocating gas compressor aimed at achieving higher pressures faster than single acting ones. Design considerations included guideways, piston head, and stroke length. The design process supported the development of compact compressors for varied industrial applications.

Hiren K. Leuva and Prof. S. A. Shah [23] analyzed the stress distribution in a piston used in a reciprocating oil-less gas booster compressor. A solid model was created using Creo Parametric 2.0 and analyzed in ANSYS 15.0. The results were consistent with analytical calculations and confirmed the need for proper piston design. A. Almasi [24] highlighted recent advances in optimizing compressor design using tools like CATIA, ANSYS, and Pro/E. Thermal analysis and improved coatings reduce thermal stress, while effective lubrication and condition monitoring enhance reliability. Ch. Venkata Rajam et al. [25] evaluated stress distribution in pistons using CATIA V5R16 and ANSYS 11.0. Design optimization reduced ring land width, barrel thickness, and volume, resulting in increased efficiency without compromising structural strength. Rohit Tamrakar et al. [26] optimized piston design using PRO-E and global sensitivity analysis. Structural and thermal simulations identified ideal dimensions to minimize mass and improve performance under specific loading conditions. Jan Tuhovcak et al. [16] reiterated the influence of valve characteristics on suction and discharge flow behavior. An energy-balance-based numerical tool was used to simulate compressor performance, validated with a four-cylinder semi-hermetic compressor. A thermal analysis of pistons made from different materials using Creo 5.0 and ANSYS. Aluminum alloy was found to be lightweight and deformation-resistant; cast iron had high safety but was brittle; titanium alloy offered high strength but was costly; copper alloy had low heat resistance and safety. Kush Patel, Hely Patel, Zeel Patel, Arun Kumar, and Bose Babu [22] explored piston redesign to improve performance, efficiency, and reliability. Surface coatings and high-strength composites reduced friction and thermal stress, while experimental studies confirmed increased energy efficiency and component longevity. These advancements contribute to sustainable engineering.

2.1. Literature Gap

There are still a number of significant gaps in the literature despite the substantial study on reciprocating compressors. With little investigation of integrated system-level behavior under dynamic real-time conditions, the majority of research concentrates on analyzing individual components, such as connecting rods or pistons. Advanced materials and coatings under various industrial conditions lack long-term experimental validation. Furthermore, the combined effects of mechanical, thermal, and fatigue stresses during long operating cycles are frequently ignored by optimization techniques. A few comparisons exist between new and classic designs in large-scale industrial settings. These discrepancies underscore the necessity of comprehensive, interdisciplinary methods to improve compressor sustainability and dependability. Performance and cost-effectiveness: A comparative analysis of free air delivery, Energy (electrical unit) consumption, operational efficiency and financial viability.

3. Design of Experimental Setup

3.1. Specifications of the Available Single Acting Reciprocating Compressor

Model: - ELGI Equipment Ltd.
Model: - SS05090 HN ELGI make
Piston Displacement (cfm): - 21.4
Free air Delivery (cfm): - 15.6
Motor Power (kW): - 3.7
Compressor (rpm): - 925
Tank capacity litres: - 220
Overall dimension (L× B × H) mm: - 1595 × 545 × 1085
Net weight (Kg): - 256

The design of the double acting reciprocating compressor was based on calculations performed using the parameters mentioned below, with reference to the Design Data Book [27]. During these calculations, the following parameters were kept constant: 925 rpm, 9 kg/cm² of air pressure, and 21.4 cfm of piston displacement.

3.2. Cylinder Design

Specification

1. Material: FG300 (ferrous castings).
2. Tensile stress: 300N/mm^2
3. Brinell hardness: 180 to 230 (HB).
4. Factor of safety: 10
5. Permissible stress: 30 N/mm^2

3.2.1. Diameter and Stroke Length

According to [1], the stroke length to diameter ratio must lie in the range of 1.5 to 1.7. For the calculation, the length of the stroke is 150 mm, and the bore diameter is 93.75 mm.

L/D ratio= 150/93.75 =1.6.

The L/d ratio lies in the mentioned range. Hence, the stroke length and bore are accepted.

3.2.2. Thickness of the Cylinder Wall

The thickness of the cylinder wall is calculated by Birnie's equation, which is mentioned in Equation (1) [2].

$$t = 0.5D\left[\left(\frac{\sigma+(1-\mu)p_{max}}{\sigma-(1+\mu)p_{max}}^{0.5} - 1\right)\right] \quad (1)$$

After putting in the specified values, the thickness is,

t = 0.4168 mm

For the CI Material, added 6.5 mm

Total thickness = 0.4168 + 6.5

= 6.9168 mm

According to [1], the thickness is considered to be 8mm.

3.2.3. Design of Cylinder Head

Table 1. The various stress calculations for the failure limit of the cylinder

Sr. No	Name of Stress	Value(N/mm$^{2)}$
1	Thermal stresses	24
2	Hoop stress	5.17
3	Tensile stress	2.586
4	Bending stress	0.489

Thickness of Flat Cylinder Head

$$t_h = 0.31D\left(\frac{p_{max}}{\sigma_t}()^{0.5}\right)$$

$$= 0.31 * 93.75\left(\frac{0.8829}{30}\right)^{0.5}$$

$$= 4.98mm \approx 5mm$$

3.2.4. Design of Piston

The design of the piston by using Grashof's formula, [12].

Thickness of Crown

$$t_1 = 0.43D\sqrt{\frac{p}{f}}$$

$$t_1 = 5.315mm$$

Where, p = Fluid pressure = 0.8299 bar

f = tensile strength (aluminium) = 206 MN/mm^2

FOS = 4

$$f = \frac{206}{4} = 51.5N/mm^2$$

Design of Piston Ring

- The radial thickness of the CI piston ring-

$$t_r = D\sqrt{\frac{3 * p_r}{\sigma}}$$

Where, σ is the allowable strength for CI= $\sigma = 82N/mm^2$
Radial pressure = pr = 0.03432 N/mm^2
Hence, t = 3.3219 mm.

The Depth of the Piston Ring

$$h = 0.7t_r \ to \ t_r$$
$$= 0.7 * 3.3219 \ to \ 3.3219$$
$$= 2.325 \ to \ 3.3219mm$$

The Diameter of Piston Rod

$$d = D\left[\frac{p}{\sigma_t}\right]^{0.5}$$

$$\therefore Design\ stress\sigma_t = \frac{785}{10} = 78.5N/mm^2$$

$$\text{Diameter of piston rod, d} = 93.75 * \left[\frac{0.8829}{78.5}\right]^{0.5}$$
$$= 9.94 \approx 10mm$$

Table 2. The various stress calculations for the failure limit of the piston

Sr. No	Name of Stress	Value(N/mm^2)
1	Stress in a column	34.33

3.2.5. Design of Connecting Rod

For a double acting compressor, the gas force on the piston is given by,

$P_{gas} = p$ Area of cross section$_{max}$= 6094.58N

The force on the rod end side is given by,

$$= 6094.58 - \frac{\pi}{4}d^2 * p$$
$$= 6094.58 - \frac{\pi}{4}(16)^2 * 0.8829$$
$$= 5917.06N$$

Volume of Piston,

$$v = [A_1 * B] - [A_2(B - E)] - A_3 \qquad [1]$$

$$v = \left(\frac{\pi}{4}D^2 * B\right) - \left\{\left[\frac{\pi}{4}(D - 2E)^2 - \frac{\pi}{4}32^2\right](B - E)\right\} - \left\{\frac{\pi}{4}16^2 B\right\}$$

$$v = \left(\frac{\pi}{4}93.75^2 * 30\right) - \left\{\left[\frac{\pi}{4}(93.75 - 2 * 10)^2 - \frac{\pi}{4}32^2\right](30 - 10)\right\} - \left\{\frac{\pi}{4}16^2 * 30\right\}$$

$$v = 131703.92mm^3$$

$$v = 131703.92 * 10^{-9}m^3$$

Density of CI $\rho = 7200kg/m^3$

$$mass = volume * density$$

$$m = 131703.92 * 10^{-9} * 7200$$
$$m = 0.9482682kg$$

The inertia force due to the reciprocating mass is given by the relation.

$$p_{inertia} = -m\omega^2 r\left[\cos\theta + \frac{\cos 2\theta}{n}\right] \qquad [1]$$
$$\therefore p_{inertia} = 889.28N$$
$$F_{inertia} = 100.32N$$

3.2.6. Design of the Receiver Tank

Available capacity is 220 litres.

1 litre = 0.001m^3

220 litres = 0.001 × 220 m^3

= 220 × 10^6 mm^3

Volume = Π/4 d^2. L

= Π/4 × (1.5 d) × d^2

= Π/4 × 1.5 × d^3

220 × 10^6 = Π/4 × 1.5 d^3

d = 571.58 mm

Diameter of tank, d = 570mm

But the length of the tank, l = 1.5 d

l = 1.5 × 570 = 857.37 mm

l = 865 mm

Real volume = $\frac{\pi}{4}d^2 l$

= Π/4 (570)^2 × 865

= 220.7 × 10^6 mm^3

= 220.7 × 10^{-3} m^3

= 220.7 litres

Hence, the selected l and d are correct.

The thickness of the tank is obtained by the Hoop stress formula.

$$\sigma_H = \frac{pd}{2t_s}$$

$$75 = \frac{0.8829 * 570}{2t_s}$$

$\therefore t_s = 3.37mm \approx 3.5mm$ Hoop Stress = 75 (N/mm^2)

3.2.7. Design of V-belt Drive

Considering centre distance c = D + d
c = 187.5 + 125 = 312.5 mm
Nominal inside length for 'A' = 610 mm
Nominal pitch length for 'A' = 645mm
Standard length
Nominal inside length = 1168 mm
Nominal pitch length = 1204 mm

3.2.8. Design of Crank Shaft

The dimension of the crank web can be found empirically as follows,

Width b = (1.1 – 1.2) d_{pin}
b = 59.8 – 69.6 mm
Selecting width b = 60 mm
Thickness, t = (0.6 – 0.75) d_{pin}
= 31.8 to 37.5 mm
Selecting thickness, t = 32 mm

The crank web at the TDC position is subjected to bending moment and direct compressive stresses.

Bending stress,

The bending moment on the web,

$$M_1 = \frac{P}{2}\left(\frac{l}{2} - \frac{l_{pin}}{2} - \frac{t}{2}\right)$$

$$= \frac{p * \frac{\pi}{4} d^2}{2}\left(\frac{164}{2} - \frac{60}{2} - \frac{32}{2}\right)$$

$$= 109.702 * 10^3 N - mm \quad [1]$$

And the section modules

$$Z_1 = \frac{1}{6} bt^2$$

$$= \frac{1}{6} * 60 * (32)^2$$

$= 10240 mm^3$ ∴ $Bending$ stress=$\sigma_{b1} = \frac{M_1}{Z_1}$

$$= \frac{109702.48}{10240} = 10.713 N/mm^2$$

$$\sigma b_1 = \frac{p}{b*t}$$

The bending stress will be tensile stress on the face 1.2 and compressive stress on the face 3 – 4.

Calculation Result of Double Acting Air Compressor

Maximum pressure = 8.829 bar = 0.8829 N/mm²
RPM of compressor = 925 rpm
cylinder bore = 93.25 mm
cylinder length = 205 mm
Cylinder thickness = 8 mm
Material of cylinder = FG 300
Factor of Safety for cylinder = 10
Stroke length = 150 mm
Radius of crank = 75 mm
L/D Ratio = 1.6
cylinder flange thickness = 9.6 mm
No of studs = 4
Size of studs = 8
L/r Ratio = 3
Length of connecting rod = 225 mm
Thickness of Piston = 30 mm
piston ring depth = 4.875 mm

The vertical distance from the piston's crown to the upper edge of the first piston ring is 6.420 mm.

The measurement from the piston's base to the lower edge of the second piston ring = 6.420 mm.

The space separating the two piston rings measures = 7.410 mm

Number of piston rings = 2
Diameter of small end of taper bore = 12.87 mm
Diameter of the big end of the taper bore = 16 mm
Thickness of piston ring = 10 mm

The thickness of the structural web within the piston is measured at 10 mm.

Diameter of the boss of the taper bore = 32 mm
Diameter of piston rod = 16 mm
Length of piston rod = 235 mm
Volume of piston =131703.92×10^{-9} m^3
Mass of piston = 0.9482 kg
Outer diameter of piston pin = 15 mm
The axial length of the small end is 30 mm
External diameter of small end = 24 mm
Internal bore diameter of small end = 18.75 mm
External diameter of big end = 80 mm
Internal bore diameter of big end = 64 mm
Crank pin dia = 58 mm
crank pin length = 60 mm
Wall thickness of the insert (bush) = 3 mm
Length of air receiver tank = 865 mm
Diameter of air receiver tank = 570 mm
Capacity of tank = 220.7 litres
Thickness of tank = 3.5 mm
Thickness of air tank cover = 21.18 mm
Length of wall of cover = 28.46 mm
Power of motor = 3.7 KW
RPM of motor = 1500
Frame Designation = PM 112 m
Diameter of bigger V-belt pulley = 187.5 mm
Diameter of small pulley = 125 mm
Centre distance of belt drive = 400 mm
Length of belt = 1295 mm
Nominal pitch length = 1331 mm
Selection of V belt = A type
Speed of belt = 9.08 m/sec.
No. of belts of capacity 1.99 KW = 2 No

The Drafted View of a Double Acting Air Compressor

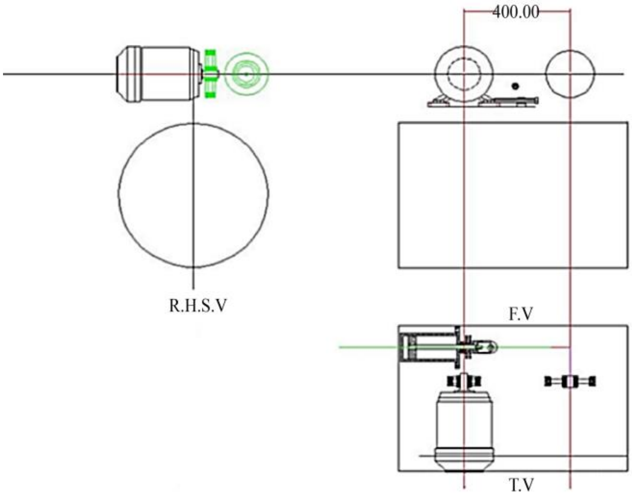

Fig. 1 Drafted view of a double acting air compressor

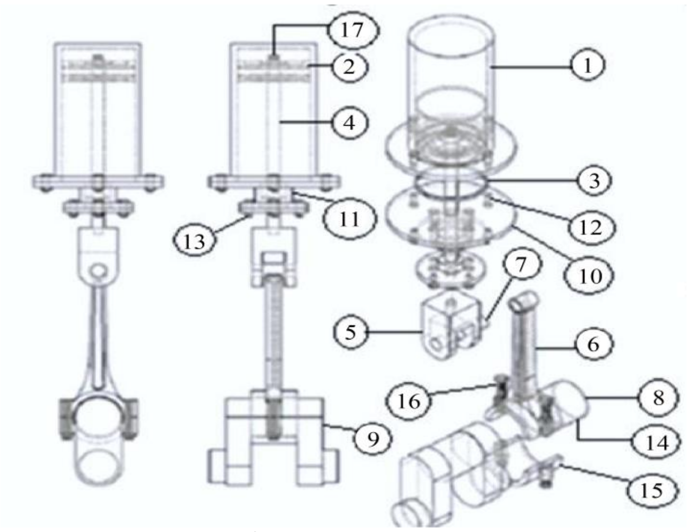

Fig. 2 Part assembly of a double acting reciprocating compressor

Table 3. List of components with quantity

Sr. No.	Name of Component	Qunaitutiy	Material
1	Cylinder	1	Grey CIFG300
2	Piston	1	Al alloy
3	Piston Ring	2	Al alloy
4	Piston Rod	1	C-40
5	Cross Head	1	Grey CI
6	Connecting Rod	1	Grey CI
7	Guden Pin	1	C-14
8	Crank Pin	1	Grey CI
9	Crank Shaft	1	Carbon Steel C-14
10	Flange	2	Gray CI
11	Bush	1	Rubber
12	Cylinder Bolt	8	40Cr IM028
13	Cylnder Nut	8	40Cr IM028
14	Crank Shaft Bush	1	Rubber
15	Cap	1	Grey CI FG300
16	Bolt Cap	1	40Cr IM028
17	Piston Rod Nut	1	40Cr IM028

3.3. CAD Modelling

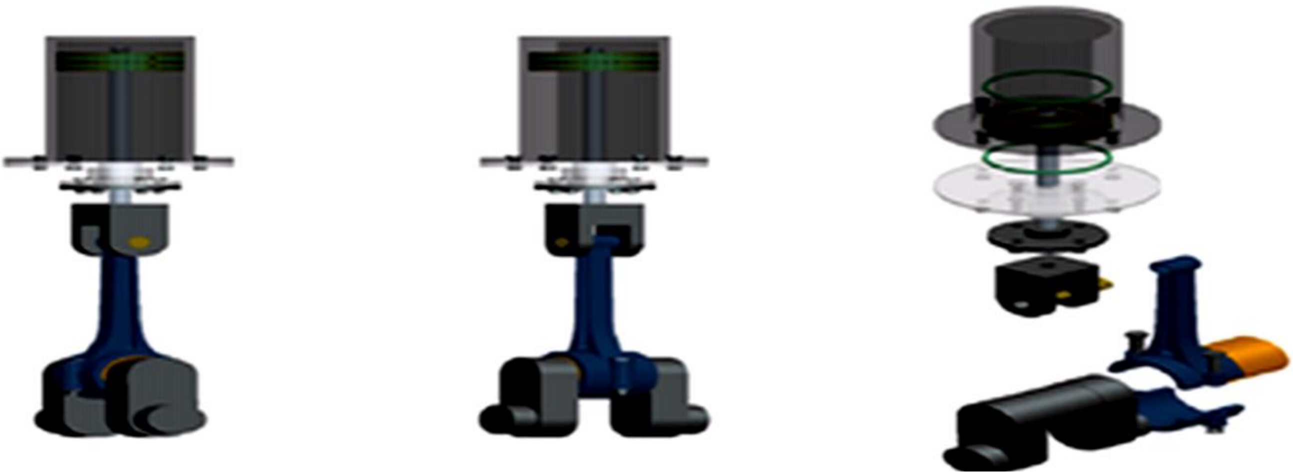

Fig. 3 CAD model of double acting reciprocating compressor using Pro-E tool

4. Simulation Result

4.1. Cylinder

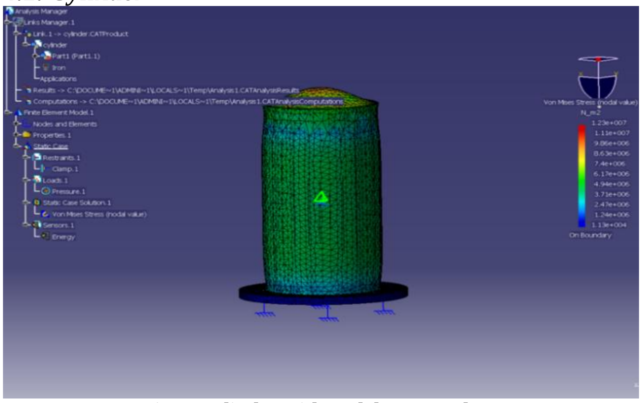

Fig. 4 Cylinder with nodal stress values

4.2. Piston

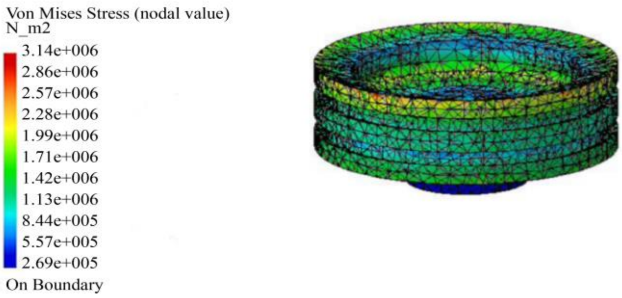

Fig. 5 Piston with nodal stress values

4.3. Piston Rod

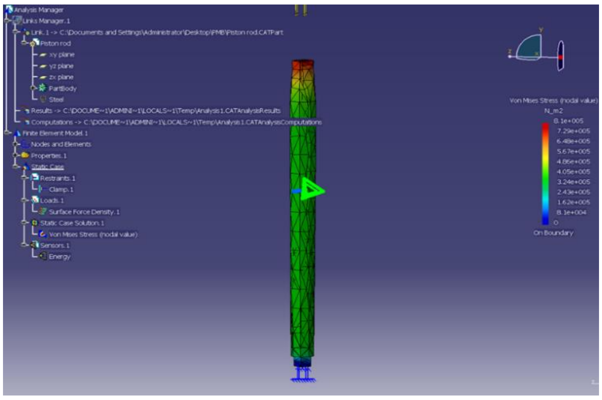

Fig. 6 Piston rod with nodal stress values

4.4. Connecting Rod

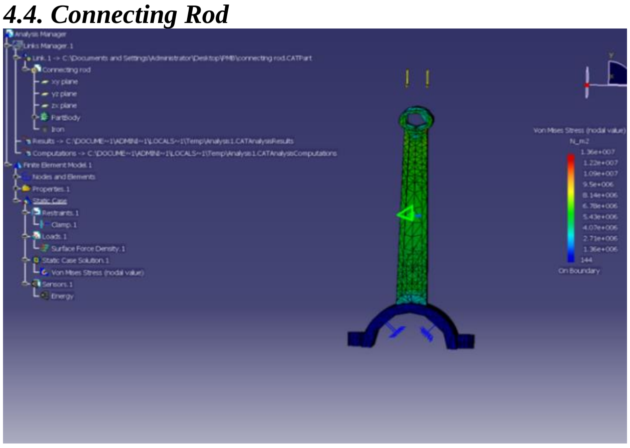

Fig. 7 Connecting rod with stress nodal values

4.5. Crank Shaft

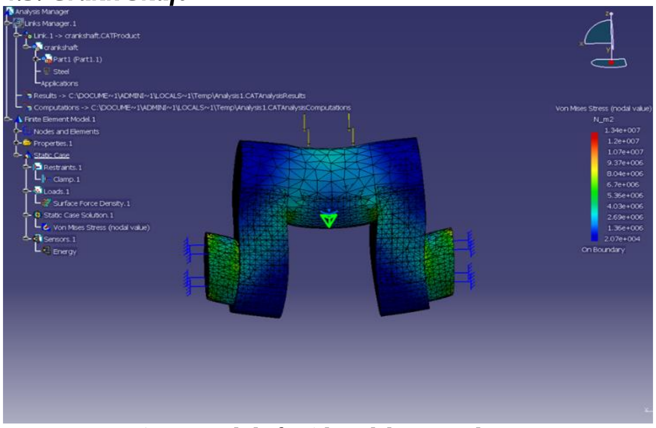

Fig. 8 Crankshaft with nodal stress values

5. Performance Analysis

5.1. Power Required

$$IP = \frac{n}{n-1} p_1 * V_1 \left[\left(\frac{p_2}{p_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= 6.59\text{KW}$$

5.2. Cost Analysis

Receiver tank capacity = 220 liters = 220 × 10^{-3} m^3

FAD for single acting air compressor = 15.6 cfm = 0.4419 m^3/min, and Input power required to run the single acting compressor = 3.7 kW.

Time required to fill the receiver tank of capacity 220 litres for a single acting compressor.

$$= \frac{220*10^{-3}}{0.4419} = 0.4978 \text{ minutes } = 29.87 \text{ seconds}$$

Electrical energy unit needed = $3.7 * \frac{0.4978}{60}$

$$= 0.03069 kWh$$

Cost of energy @ Rs. 6 per kwh =0.03069×6

=Rs. 0.18414/-

Number of tanks filled in 8 hrs.= $\frac{3600}{29.87} * 8$= 964.178

The cost of energy for 8 hrs a day = $\frac{0.18414}{29.87} * 3600 * 8$

= Rs.177.543/- (For filling 964.178 tanks)

For a Double Acting Compressor

FAD for double acting air compressor = 49.30 cfm = 1.39 m^3/min, and Input power required to run the double acting compressor = 6.591 kW.

Time required to fill the receiver tank of the same capacity, 220 liters, for a double acting compressor.

$$= \frac{220*10^{-3}}{1.39} = 0.1582 \text{ minutes}$$

$$= 9.4964 \text{ seconds}$$

$$\text{Electrical energy unit needed} = 6.591 \times \frac{220*10^{-3}}{1.39}$$

= 0.01738 kWh (approx.)

Cost of energy @ Rs. 6 per kwh = 0.0173863 × 6= Rs 0.1043/-

The running cost of the double acting compressor to fill the receiver once in 9.4964 seconds is Rs 0.1043/-

For filling 1 tank Double acting compressor saves = 0. 0.18414- 0.1043= Rs. 0.07984

Number of tanks filled in 8 hrs.= $\frac{3600}{9.4964} * 8 = 3032.728$

The cost of energy saving for a double acting compressor as compared to a single acting compressor for 8 hours a day =3032.728*0.07984 = Rs. 242.13/-

The total initial cost of a double acting compressor [13] is Rs. 129275/-

And the initial cost of a single acting compressor [13] is Rs. 83400/-

Difference in the initial cost = Rs. 45875/-

5.3. Payback Period

If the compressor runs for 8 hrs. per day, then the payback period

$$=\frac{45875}{242.13} = 189.46 = 190 \text{ days}$$

The power required to run the single acting air compressor to generate 9 kg/cm2 pressure is 3kW (5hp). For the same pressure, the power required for a double acting air compressor is about 6.591 kW, which is nearly twice that of a single acting.

Free air delivery of a single acting air compressor is 15.6 cfm, but for a double acting air compressor, free air delivery is 49.30 cfm, which is quite more than that of a single acting air compressor. So, the time required to fill the receiver tank is reduced to that of a single acting compressor. The time required for filling the receiver tank once for a single acting compressor is 29.87 seconds, while for a double acting compressor it is 9.4964 seconds. Also, the electrical unit needed for a single acting compressor is equal to 0.03069 kW, while for a double acting compressor it is equal to 0.01738 kWh, which is the advantage of a double acting air compressor. This also reduces the fluctuation of the pressure supplied to pneumatic tools under operation to speed up the work and also enhance the quality of work. This is otherwise affected by large fluctuations of pressure supplied to pneumatic tools.

The cost of energy for 8 hrs. a day by using a double acting compressor, the running cost of Rs 242.13/- will be saved. The initial cost taken from [13] for a single acting compressor is Rs. 83400/-, and for all the components of a double acting compressor, it will cost Rs. 129275/-. A doubleacting compressor will cost Rs. 45875/- more as compared to a single acting compressor, but the running cost is much less as compared to a single acting compressor, so to recover this extra initial cost, a payback period of 190 days will be required.

Hence, the double acting air compressor can be used in place of a single acting compressor for industrial use, which generates the same pressure of 9 kg/cm^2.

6. Result and Discussion

After studying the single acting compressor and the newly designed double acting compressor, the following parameters were analyzed.

6.1. Cylinder Stress

After conducting a mathematical and simulation study of a double acting compressor cylinder, shown in Figure 9, the tensile stress of the cylinder was found to be 16.66 N/mm².

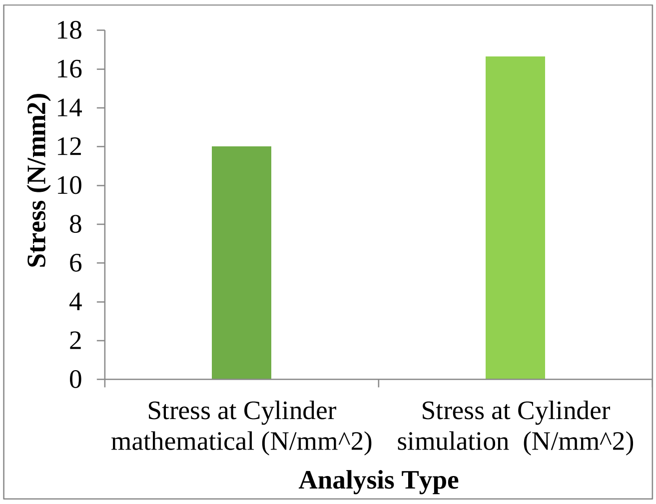

Fig. 9 Comparison between mathematical and simulation cylinder stress analysis

6.2. Piston Stress between

The mathematical and simulation study of the double acting compressor piston, shown in Figure 10, revealed a tensile stress of 34.33 N/mm². The difference between the simulation and mathematical readings is 0.96%.

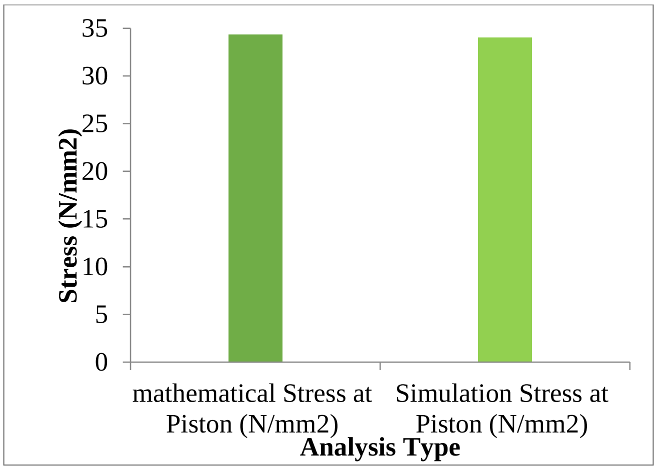

Fig. 10 Comparison between mathematical and simulation piston stress analysis

6.3. Piston Rod

After the mathematical and simulation study of a double acting compressor piston rod as shown in Figure 11, the tensile stress of the piston was found to be 78.5 N/mm². The difference between the simulation and mathematical readings is 8.28%.

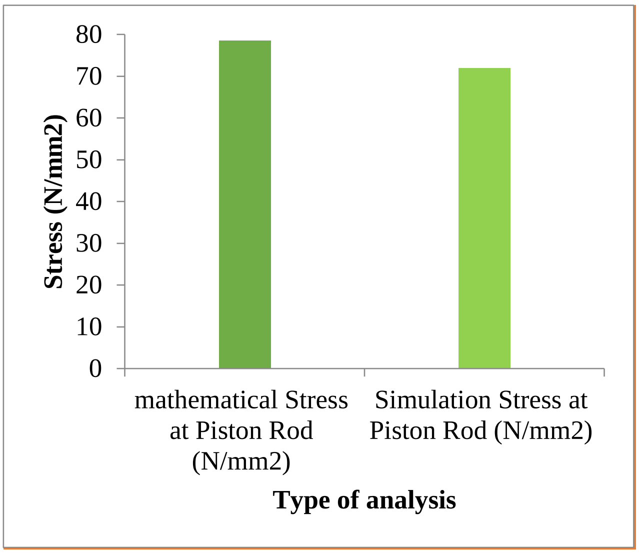

Fig. 11 Comparison between mathematical and simulation piston rod stress analysis

6.4. Crank Shaft

After a mathematical and simulation study of a double acting compressor crankshaft (as shown in Figure 12, the tensile stress of the crankshaft was found to be 13.4 N/mm². The difference between the simulation and the mathematical readings is 20%.

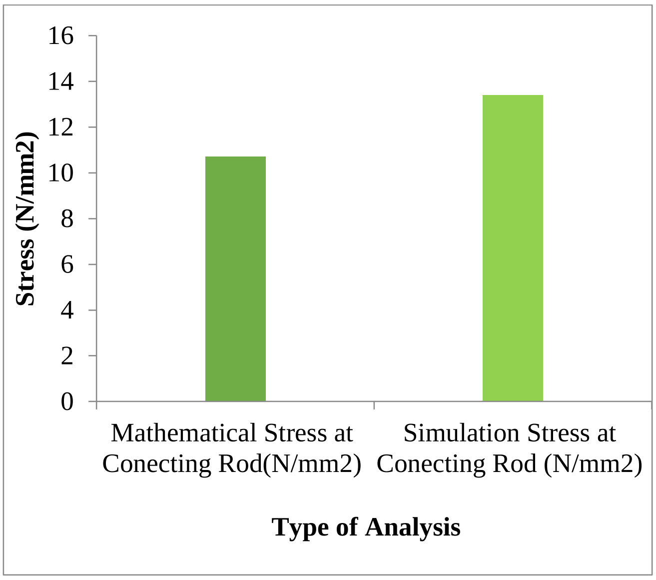

Fig. 12 Comparison between mathematical and simulation crankshaft stress analysis

6.5. Connecting Rod

After a mathematical and simulation study of a double acting compressor connecting rod, as shown in Figure 13, the tensile stress of the connecting rod was found to be 13.6 N/mm². The difference between the simulation and the mathematical reading is 11.76%.

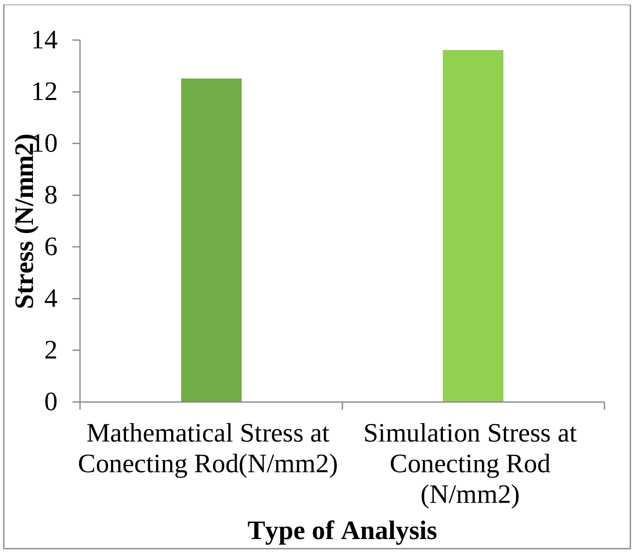

Fig. 13 Comparison between mathematical and simulation connecting rod stress analysis

6.6. Free Air Delivery

The free air delivery of a single acting air compressor is 15.6 cfm, but for a double acting air compressor, the free air delivery is equal to 49.3 cfm, which is more than that of a single acting compressor. The only advantage of a double acting compressor is that it can reduce the time it takes to fill the air receiver.

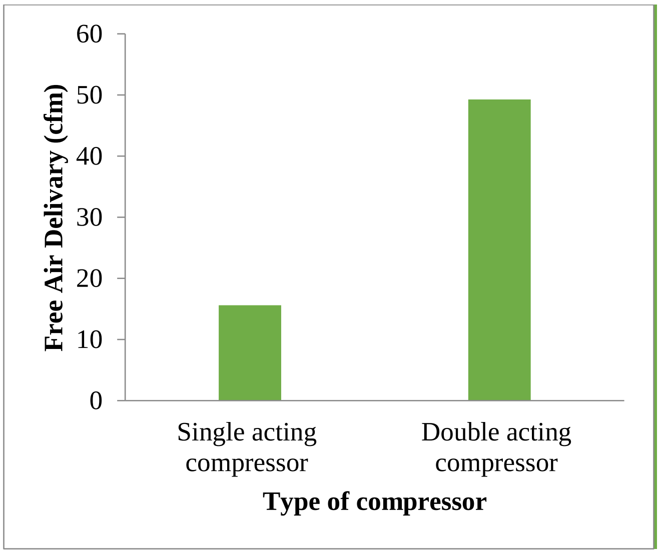

Fig. 14 Comparison between FAD for single and double acting compressors

6.7. Running Cost

After running for 8 hours per day, the results showed that the operating cost for the double acting compressor, as compared to the single acting compressor, saves Rs. 242.13 per day.

As per the standard protocol, the machine's lifespan is considered to be 5 years. After analyzing the five-year operation of both single acting and double acting compressors, including capital and operating costs, it was found that the double acting compressor saves Rs. 441887/-

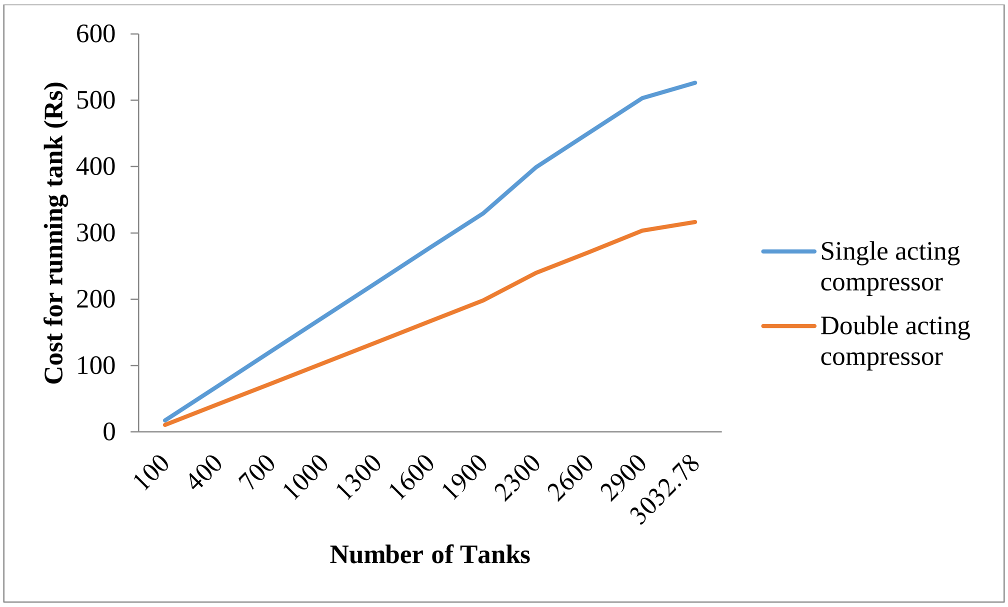

Fig. 15 Running cost for single and double acting compressors

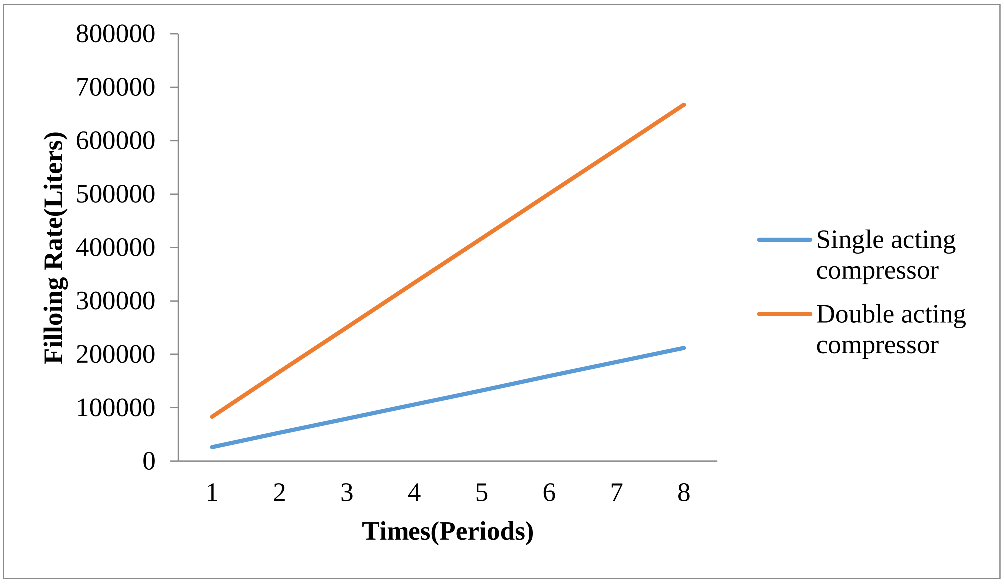

Fig. 16 Filling rate of single-and double acting compressors

6.8. Filling Rate of Compressor

After 8 hours of operation, the air output of both compressors was measured. The single acting reciprocating compressor produced 212,080 liters of air, while the double acting reciprocating compressor generated 667,040 liters. This demonstrates the efficiency advantage of the double acting compressor, which is designed to provide the required operating pressure. The significant difference in air output highlights the double acting compressor's capability to perform more work within the same time frame.

7. Conclusion

The analysis involved designing a double acting reciprocating compressor based on the available specifications of a single acting reciprocating compressor. The results predicted that the stresses that will be developed during its running condition can be reduced, and the life of the double acting compressor can be improved. This demonstrates that the discharge rate of the double acting compressor (3032.78) is more than that of the single acting compressor (964). Additionally, the running cost was reduced by Rs. 242.13 /- for eight hours of operation per day when using the double acting compressor. After analyzing the five-year operation of both single acting and double acting compressors, including capital and operating costs, it was found that the double acting compressor saves Rs. 441887/-. This analysis highlights the superior performance and cost-effectiveness, energy (electrical unit) consumption, operational efficiency and financial viability of the double acting reciprocating air compressor, showing that it yields optimal results compared to the single acting reciprocating compressor.

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