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Analytical and CAD Modeling of Double-Acting Reciprocating Compressor Components

Pramod K Jadhao[1], Dr Sachin R Karale[2], Nitin N Pawar[3], Brijkishor Sah[4]
[1,2] Mechanical Engineering Department G H Raisoni University Amravati India
[3]Alamuri Ratnamala Institute of Engineering and Technology, Thane, India
[4]Datta Meghe College of Engineering, Thane, India
Correspondence e-mail:
pramodkjadhao1974@gmail.com ,sachin.karale@raisoni.net ,nitinpawar1702@gmail.com,kishor1006.bs@gmail.com

Abstract:

The design and performance analysis of a double-acting reciprocating air compressor, created to satisfy industrial needs for high-volume compressed air output in a short amount of running time, are presented in this study. In order to maintain a working pressure of 9 kg/cm^2, the study entails the methodical engineering of crucial parts, such as the piston bore diameter, stroke length ratio, flange assembly, bolts, studs, and piston rings. A commercially available single-acting reciprocating compressor (Model SS 05090 HN, ELGI manufacture) with a piston displacement of 21.4 CFM and an operating speed of 925 rpm is the basis for the design framework. Switching to a double-acting mechanism greatly increases the compressor output and efficiency. The components were modeled, stress analyses were performed, and geometric assessments were carried out using Computer-Aided Design (CAD) tools. According to the investigation, the double-acting arrangement enhances operational effectiveness, cost-effectiveness, and economic viability and achieves outstanding performance. The results confirm that double-acting compressors are a more efficient option in industrial settings than traditional single-acting systems.

Introduction

Kim BJ et al. [2001] In sectors like natural gas and chemical processing, reciprocating compressors are essential because they enable it to produce liquid gases and air more easily. The crankshaft, crank pin, connecting rod, rod cap, and bolts are some of the parts that make up these compressors. The crankshaft and connecting rod system produce rotary motion from reciprocating piston motion. The literature emphasizes how fast piston acceleration and deceleration, along with the mechanical loads from power sources like electric motors and diesel engines, subject connecting rod caps and bolts to complicated dynamic forces. These stresses greatly impact the compressor components' fatigue life and performance dependability. Shenoy P S [2006] and Sivaprasad S [2010] In reciprocating compressor mechanisms, the connecting rod cap and connecting bolts are essential because they help secure the crankshaft-rod assembly under high dynamic stresses. Their vulnerability to varying stresses during operation due to piston acceleration, deceleration, and inertial forces is highlighted in the literature. Additionally, these parts are subjected to cyclic strain from power transfer, frequently from diesel or electric motors. Studies have concentrated on fatigue analysis, material selection, and structural optimization to improve durability and avoid failure. The safe and effective operation of compressors used in the natural gas, chemical processing, and cryogenic gas production industries depends on their integrity. He B. [2013], Xu XL[2007], Ilman M N

[2013], and Juarez C [2016] Several studies have investigated the failure mechanisms of connecting rod systems, highlighting fatigue as a primary cause. Research emphasizes that factors such as improper material selection, poor design, and overload bending significantly contribute to premature failure. Additional issues, including misaligned or improperly adjusted bolts, spalling, and assembly deficiencies, have also been identified as critical contributors. Fatigue failures often originate from stress concentrations or surface defects exacerbated by cyclic loading. The literature consistently suggests that improving material quality, optimizing design parameters, and ensuring precise assembly procedures are essential for enhancing the durability and reliability of connecting rod systems under operational stresses. Gu Z et al. [2005] and Shahrivar A. and Abdolmaleki A R [2006]. Improper material selection can critically affect the performance and durability of a connecting rod system. The literature emphasizes the need for materials with high mechanical strength, hardness, and superior tensile and fatigue properties to withstand dynamic and cyclic loads. Studies have shown that alloys such as forged steel and titanium are preferred due to their reliability under stress and wear conditions. Poor material choice leads to premature failure, reduced engine efficiency, and increased maintenance costs, highlighting the importance of optimal material selection. P.K. Jadhao et al. [2012] Auther compared compressor types, emphasizing the double-acting reciprocating compressor's superior performance. Its ability to deliver higher output in less time supports its adoption over single-acting models, particularly in high-pressure applications.Zhang X et al. [2011] Numerous studies have explored enhancing connecting rod performance through material optimization, manufacturing techniques, and structural design. Research focuses on lightweight composites, fatigue resistance, and finite element analysis to reduce stress concentration. Advancements in forging methods and alternative alloys improve durability, efficiency, and overall engine performance. Khare S et al. [2012]. Several studies have suggested altered connecting rod designs utilizing finite element analysis (FEA) to reduce stress concentrations. Improving geometry and material distribution in a compressor component highly increases fatigue life and durability. Studies show that these FEA-driven enhancements successfully lower maximum stress, resulting in more dependable and effective engine parts. Lee B et al. [2011] The study examined fretting wear-induced fracture in turbine blade tang holes using a combination of experimental and non-linear finite element method (FEM) techniques. Fretting wear, which acts as a stress raiser and hastens fatigue failure, was discovered on the fracture surface. The close match between FEM predictions and experimental data validated the correctness of the model in evaluating fretting fatigue mechanisms. Recent connecting bolt research highlights the integration of fracture analysis, microstructure characterization, and numerical simulations employing fracture mechanics. Griza et al. [13] and Fadag et al. [14] These methods improve knowledge of material behavior under load, stress distribution, and failure processes. In crucial structural applications, sophisticated modeling and experimental methodologies facilitate enhanced bolt design, dependability, and service life. Nitin Pawar et al. [2023] emphasize the significance of water pump housing strength in high-performance engines, which several research studies have emphasized. Studies demonstrate the effects of thermal and structural stresses on engine emissions and efficiency. Precise analysis and optimization are made possible by modeling software like CREO and simulation tools like ANSYS, which demonstrate significant gains in stress distribution and deformation reduction compared to traditional designs.

Mathematical Analysis:

Bore to stroke selection:

The volume swept per revolution for a double-acting cylinder is calculated using the formula $V = \frac{\pi}{4} D^2 L$ where D is the bore diameter, and L is the stroke length. Given a constant swept volume of v=3.4610×$10^{(-4)}$ m^3**.** The ideal bore diameter is found by calculating the bore-to-stroke length ratio. The design of essential parts, including the cylinder, piston ring, and flange, depends heavily on this ratio. From the graph of 'L' versus 'D' shown in Figure No. 1, it is evident that varying the stroke length between 1.5 and 2 meters minimally impacts piston diameter. Therefore, selecting a piston diameter of 0.015 m is reasonable. However, this results in a longer stroke length, around 1.9 to 2 m, which increases the overall compressor size. Therefore, the calculation values are selected L = 150mm & D = 93.75mm.

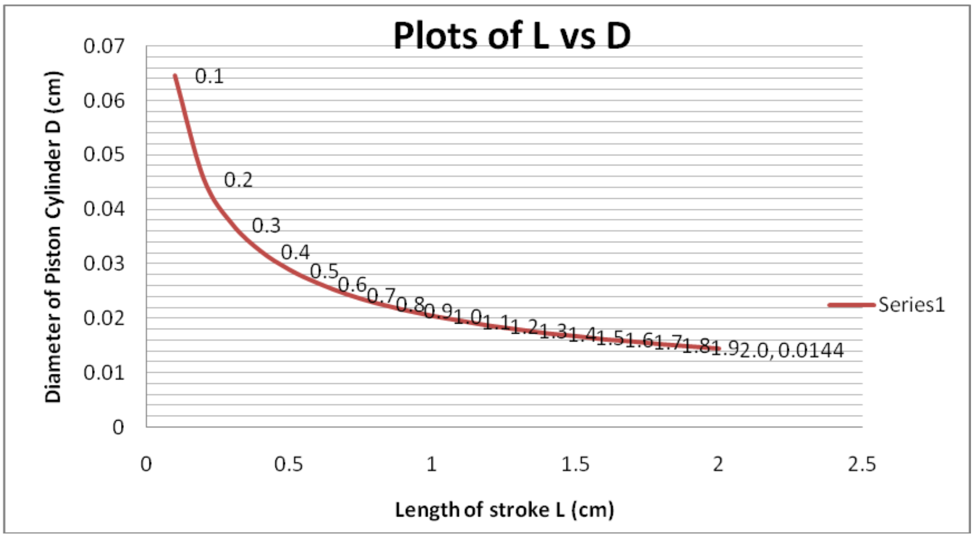

Figure No. 1. Graph of bore diameter and stroke length

Thickness of cylinder Flange:

The Flange thickness

$$t_2 = (1.2 \text{ to } 1.4) \times t$$
$$= (1.2) \times 8$$
$$t_2 = 9.6 \text{ mm}$$

Design of cylinder head: -

Thickness of flat cylinder head:-

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

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

$$= 4.98mm \approx 5mm$$

Number of studs:-

$$i = 0.015 \text{ D} + 4$$
$$= 0.015 \times (93.75) + 4$$
$$= 5.406 \approx 6$$

It should be a multiple of 4

We can take 4-stud.

The root diameter of studs or bolts

$$d_r = D \cdot \sqrt{\frac{p}{i\sigma_{t_{bolt}}}}$$

Selecting carbon steel C40 for studs according to the design data book, the range of Tensile strength = (570 – 667) N/mm^2, a selecting tensile strength= 600 N/mm^2, and factor of safety = 10.

Allowable tensile strength of bolt material

$$\sigma_{t_{bolt}} = \frac{600}{10} = 60N/mm^2$$

$$\therefore d_r = 93.75 * \sqrt{\frac{0.8829}{4*60}}$$

$$= 5.68mm$$

Failure of Bolt

$$\text{Tensile Load } = \frac{\pi}{4}D^2 * p_{max}$$

$$= \frac{\pi}{4}(93.75)^2 * 0.8829$$

= 6094.05823N

Bolts are used. Hence, this must be equal to

$$4 * \frac{\pi d_r^2}{4} f_t = 6094.5823$$

$$4 * \frac{\pi * (5.68)^2}{4} f_t = 6094.5823$$

$$\therefore f_t = 60.13N/mm^2$$

The induced stress is more than the permissible 60 N/mm^2. Hence, the diameter of the bolt selected is 8 mm. Selecting hexagonal bolt M8 – Jam – 8mm (without slot). [R.K. Purohit, 1999].

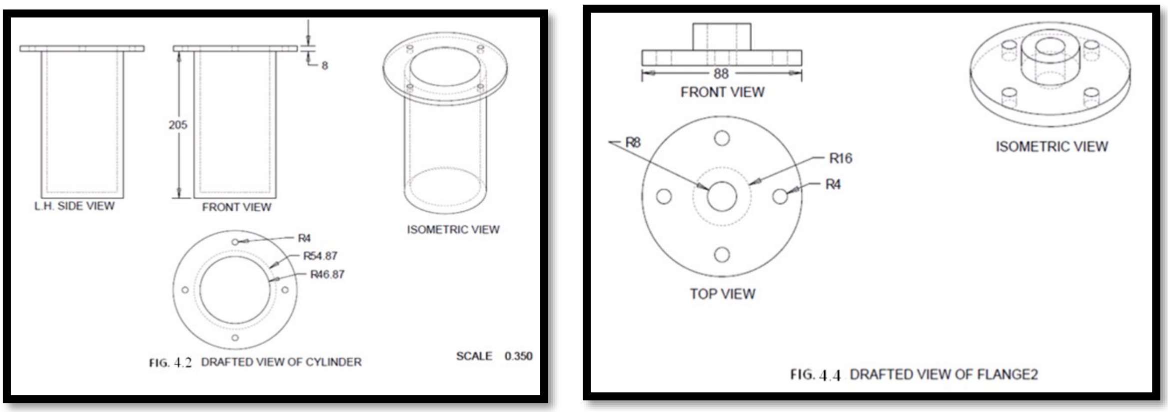

Figure No.2 Drafting image of Flange, bolt, and stud

Design of Tank (vessel):-

Pressure vessels are vessels of different shapes and constructions used to store or supply liquids, vapor, or gases. They are subjected to internal or external pressure of more than 0.7 atmospheric gauge. Pressure vessels are extensively used in thermal power plants, nuclear power plants, process and chemical industries, and for water, steam, gas and air supply in various industries. The fusion welding process fabricated pressure vessels from steel plates that were welded together. These vessels are classified based on geometric shapes such as cylindrical, conical, and spherical.

In the present design, the tank capacity is 220 liters.

$$1 \text{ liter} = 0.001 m^3$$
$$220 \text{ liters} = 0.001 \times 220 \ m^3$$
$$= 220 \times 10^{-3} \ m^3$$
$$= 220 \times 10^{6} \ mm^3$$
$$\text{Volume} = \frac{\pi}{4} d^2 . L$$
$$= \frac{\pi}{4} \times (1.5 \ d) \times d^2$$
$$= \frac{\pi}{4} \times 1.5 \times d^3$$
$$220 \times 10^6 = \frac{\pi}{4} \times 1.5 \ d^3$$
$$d = 571.58 \text{ mm}$$

Dia of tank d = 570mm

But length of tank l = 1.5 d

$$l = 1.5 \times 570 = 857.37 \text{ mm}$$
$$l = 865 \text{ mm}$$

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

$$= \frac{\pi}{4} (570)^2 \times 865$$
$$= 220.7 \times 10^6 \ mm^3$$
$$= 220.7 \times 10^{-3} \ m^3$$
$$= 220.7 \text{ liters}$$

The selected l and d is correct.

Considering the material for the tank C40, having the tensile strength 570 – 667 N/mm^2. Since the cylinder is subjected to variable load the recommended F.O.S. is 8.

Therefore design stress $= \frac{600}{8}$

$$\sigma_H = 75 \ N/mm^2$$

Thickness of tank is obtained by the Hoop stress formula.

$$\sigma_H = \frac{pd}{2t_s}$$
$$75 = \frac{0.8829 * 570}{2t_s}$$
$$\therefore t_s = 3.37mm \approx 3.5mm$$

Verifying the thickness of cylindrical shells by using equation

$$t_s = \frac{pd}{2fj - P}$$
$$= \frac{0.8829 * 570}{(2 * 75 * 1) - 0.8829}$$
$$= 3.37mm \approx 3.5mm$$

t_s = 3.5 mm for the shell.

Design of V-belt Drive:-

It has been decided to use two V-belts of cross-section 'A,' each having a power transmission capacity of 3.75 KW, which is against the requirement of 3.75 KW. The recommended belt speed is 15 m/sec, and the equivalent pitch diameter is 100 mm.

Equivalent pitch diameter, $d_e = d \times k_d$

Where, d pitch diameter of the smaller pulley (mm). k_d = small diameter factor. Since the speed Ratio is 3, the value of k_d = 1.14

Equivalent pitch diameter

$$D_e = d \times k_d$$
$$\therefore d = \frac{d_c}{k_d} = \frac{100}{1.14} = 87.719$$
$$d \approx 88mm$$

The minimum diameter of the pulley recommended = 63 mm, hence the diameter of 88 mm.

The equivalent pitch diameter of a drive pulley

= 3 × equivalent pitch diameter = 3 X 100

= 300 mm

D = 3 × 88 = 264 mm

After the belt design, the following remarkable selection was made.

Minimum pitch diameters for section **'A'** with **'V'** groove angle 38° =125 mm.
It is verified with the recommended standard pulley diameter,
Recommended diameter = 125 mm (preference 1)
Bigger pulley diameter = 125 × 1.5 = 187.5 mm.
Selecting motor 3.7 kW – 1500 rpm.
Maximum rim speed permitted for M.S. CI pulley = 25 m/sec
The belt of section **A** is recommended for 0.4 to 4.0 kW rating.

Hence, a section is okay for 3.7 kW. The maximum number of strands permitted is 6, but we selected 2.

Velocity:

$$v = \left(\frac{2\pi N}{60}\right) * \left(\frac{125}{2}\right)$$
$$= \frac{2\pi * 2850}{60} * \frac{125}{2} * \frac{1}{1000}$$
$$= 18.6532\ m/sec$$

Length of Belt:

Center distance c = D + d

c = 187.5 + 125

= 312.5 mm

Nominal inside length for 'A' = 610 mm

Nominal pitch length for 'A' = 645mm

$$\therefore L = \frac{\pi(D+d)}{2} + \frac{D-d^2}{4c} + 2c$$

$$\therefore L = \frac{\pi(187.5+125)}{2} + \frac{(187.5-125)^2}{4 * 312.15} + 2 * 312.15$$

=1118.99 mm

Standard length

Nominal inside length	=	1168 mm
Nominal pitch length	=	1204 mm

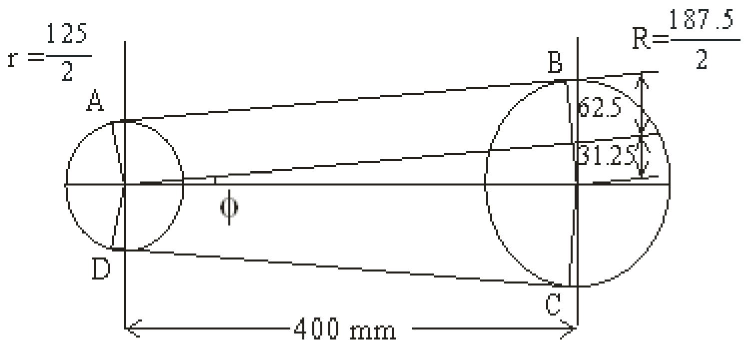

Figure No. 3: Block Diagram V-Belt drive

Design consideration of Piston Ring:-

Piston rings are typically manufactured from chrome-plated grey cast iron or alloy cast iron to enhance wear resistance and must comply with IS 5791 standards.

The radial thickness of the CI piston ring is given by,

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

Where σ is the allowable strength for CI in the 82 to 110 N/mm2 range. Hence $\sigma = 82\ N/mm^2$. The radial pressure of the piston ring on the cylinder barrel is 0.03432N/mm^2.

t_r = 3.3219 mm

The depth of the piston ring,

$$h = 0.7t_r$$

=2.325 to 3.3219mm

The minimum depth h of the piston ring,

$$h = \frac{D}{10i}$$

$$h = \frac{93.75}{10*2} = 4.875mm$$

Where i is the number of rings,

Selecting the maximum value of h = 4.875 mm.

Result and Discussion:

Belt:

Pulley distances were adjusted per the mathematical computations shown in Table No. 1. A belt length of 1331 mm was chosen for the experimental setup to satisfy the design specifications. This arrangement guarantees appropriate tension, alignment, and effective power transfer within the system.

Table No. 1: Centre distance between pulley and standard length of belt

Sr. No.	Center Distance	The belt inside Length (mm)	Standard length (mm)
1	312.5	1118.99	1168
2	400	1293.29	1331
3	520	1681.27	1730

Bolt and Flange:

The cylinder flange has a thickness of 9.6 mm. The design parameters of the stud/bolt, as shown in the table below, represent the optimum values for both flange and bolt design. The tensile failure of the bolt is observed at a stress value of 60.13 N/mm^2, and the permissible stress is 60 N/mm^2. The stress capacity is safe.

Table No. 2: Bolt parameter

Sr. No	Parameter	Quantity
1	Number of studs	4

2	Diameter of studs	5.68 mm
3	Tensile Load	6094.58 N
4	Tensile stress	60.13 N/mm^2

Figure No. 4 illustrates the CAD model of the bolt and flange assembly corresponding to the above-mentioned design parameters. The diagram provides a detailed visualization of the optimized geometry, including the flange thickness of 9.6 mm and the bolt dimensions, ensuring structural compatibility and effective load distribution within the assembly.

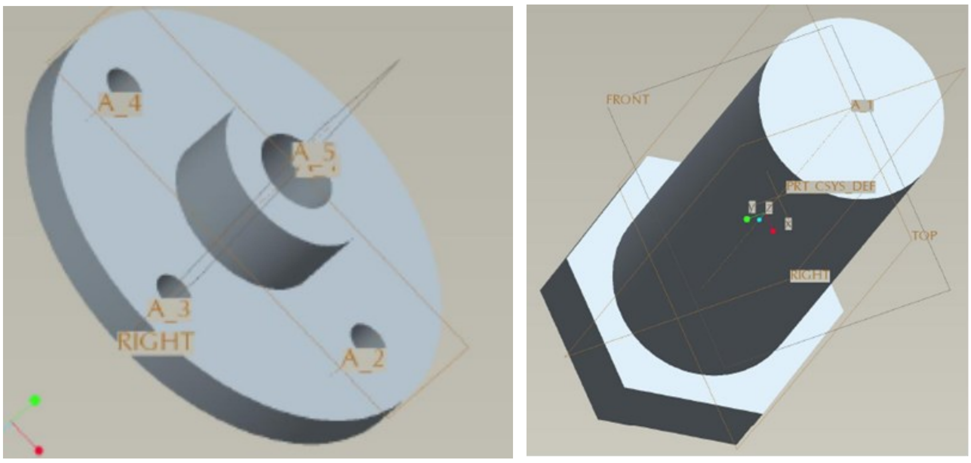

Figure No.4: CAD model of Flange and Bolt

Tank

The table below displays the parameters determined for the double-acting compressor's tank. The hoop stress was 75 N/mm2, and the safety factor was 8. These numbers show that, in operational conditions, the tank design guarantees sufficient strength and dependability.

Table No. 3: Design parameter of air tank

Sr. No	Parameter	Quantity
1	Capacity	220.7 litters
2	Diameter	571.58 mm
3	Length	857.37 mm
4	Hoop stress	75 N/mm^2
5	Thickness	3.37 mm

Piston Ring

The radial thickness of the ring is determined to be 3.32 mm, ensuring adequate sealing and durability. The allowable stress for the piston ring material is 82 N/mm^2, maintaining structural integrity under operating conditions.

Table No. 4: Design parameter of piston ring

Sr. No	Parameter	Quantity
1	Radial thickness	3.3219 mm
2	Depth	4.875 mm
3	Length	857.37 mm

4	Hoop stress	75 N/mm^2
5	Thickness	3.37 mm

Figure No. 5 displays the CAD model of the piston ring as shown below. This model illustrates the design features based on the calculated parameters, including a radial thickness of 3.32 mm. The geometry ensures optimal performance under an allowable stress limit of 82 N/mm^2 for efficient compressor operation.

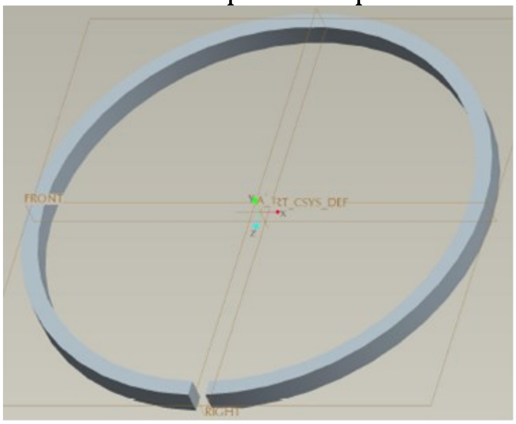

Figure No. 5: CAD module of Piston Ring

The bar chart compares component diameters, with the tank having the largest at over 550 mm. Cylinder bore, V-belt, and pulleys show moderate sizes, which is crucial for system functionality. The bigger pulley supports effective transmission. The stud has the smallest diameter and is suited for fastening rather than flow or motion roles.

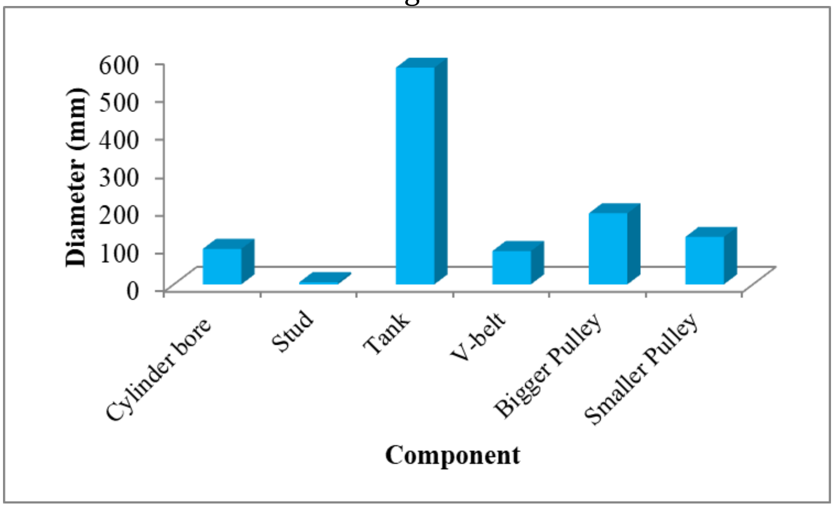

Figure No. 6: Diameter of different component

The piston ring exhibits the highest tensile stress at around 88 N/mm^2, followed by the tank and bolt at approximately 82 N/mm^2 and 67 N/mm^2, respectively. The flange has the lowest tensile stress, indicating its lesser demand for structural stress.

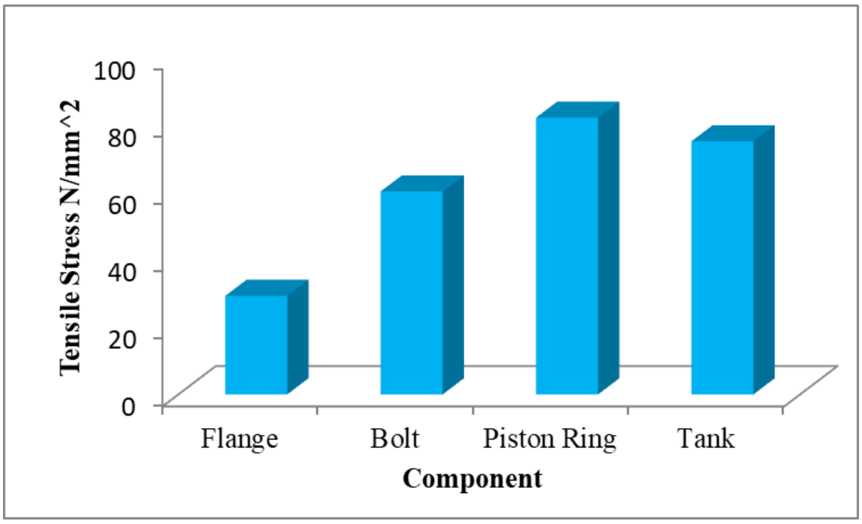

Figure No.7: Tensile Stress of different component

Conclusion:

The current study effectively shows that a double-acting reciprocating compressor is a more practical and efficient substitute for the conventional single-acting design. Extensive mathematical research and CAD modeling have assessed and improved important compressor components. The structural safety of crucial components under operating conditions was validated by stress analysis. A range of bore-to-stroke ratios were investigated to find the most effective configuration. Furthermore, real working conditions were considered when designing the flange and pulley, guaranteeing dependability in real-world applications. Additionally, the piston ring was accurately sized by mathematical computations, which improved overall performance and sealing effectiveness. These results point to the viability and enhanced efficiency of the suggested double-acting compressor design.

References:

1. Kim BJ, Kim KW. Thermo-Elasto hydrodynamic Analysis of Connecting Rod Bearing in Internal Combustion Engine. Journal of Tribology.2001;123:473-8. DOI: 10.1115/1.1353181.
2. Shenoy P S, Fatemi A. Dynamic analysis of loads and stresses in connecting rods. Proceedings of the Institution of Mechanical Engineers -- Part C.2006;220:615-24. DOI: 10.1243/09544062JMES105.
3. Sivaprasad S, Narasaiah N, Das SK, Das G, Tarafder S, Gupta KK, et al. Investigation on the failure of air compressor. Engineering Failure Analysis.2010; 17:150-7. DOI: 10.1016/j.engfailanal.2009.04.016.
4. He B, Shi G, Sun J, Chen S, Nie R. Crack analysis on the toothed mating surfaces of a diesel engine connecting rod. Engineering Failure Analysis.2013;34:443-50. DOI: 10.1016/j.engfailanal.2013.09.004.
5. Xu XL, Yu ZW.Failure Analysis of a Diesel Engine Connecting Rod. Journal of Failure Analysis & Prevention.2007;7:316-20. DOI: 10.1007/s11668-007-9058-9.
6. Ilman MN, Barizy RA. Failure analysis and fatigue performance evaluation of a failed connecting rod of reciprocating air compressor. Engineering Failure Analysis.2015;56:142-9. DOI: 10.1016/j.engfailanal.2015.03.010.

7. Juarez C, Rumiche F, Rozas A, Cuisano J, Lean P. Failure analysis of a diesel generator connecting rod. Case Studies in Engineering Failure Analysis.2016;7:24-31. DOI: 10.1016/j.csefa.2016.06.001.
8. P.K. Jadhao , K.K. Jadhao, S. S. Bhagwat, Design, Modeling and Analysis Of Double Acting Reciprocating Compressor Components, International Journal of Scientific & Engineering Research Volume 3, Issue 4, April-2012 ISSN 2229-5518.
9. Gu Z, Yang S, Ku S, Zhao Y, Dai X. Fracture splitting technology of automobile engine connecting rod. International Journal of Advanced Manufacturing Technology.2005;25:883-7. DOI: 10.1007/s00170-003-2022-2.
10. Shahrivar A, Abdolmaleki AR. Failure of a screw compressor shaft. Engineering Failure Analysis.2006;13:698-704. DOI: 10.1016/j.engfailanal.2004.10.021.
11. Zhang X, Cai Q, Zhou G, Chen Q, Xiong Y. Microstructure and mechanical properties of V-Ti-N microalloyed steel used for fracture splitting connecting rod.Journal of Materials Science.2011; 46:1789-95. DOI: 10.1007/s10853-010-5002-6.
12. Khare S, Singh OP, Bapanna Dora K, Sasun C. Spalling investigation of connecting rod. Engineering Failure Analysis.2012;19:77-86. DOI: 10.1016/j.engfailanal.2011.09.007.
13. Lee B, Suh J, Lee H, Kim T. Investigations on fretting fatigue in aircraft engine compressor blade.Engineering Failure Analysis.2011;18:1900-8. DOI: 10.1016/j.engfailanal.2011.07.021.
14. Griza S, Bertoni F, Zanon G, Reguly A, Strohaecker TR. Fatigue in engine connecting rod bolt due to forming laps. Engineering Failure Analysis.2009;16:1542-8. DOI: 10.1016/j.engfailanal.2008.10.002.
15. Fadag HA, Mall S, Jain VK.A finite element analysis of fretting fatigue crack growth behavior in Ti–6Al –4V.Engineering Fracture Mechanics.2008;75:1384-99. DOI: 10.1016/j.engfracmech.2007.07.003.
16. Pawar, Nitin & Jadhao, Kiran & Bhotra, L & Kumar, Santosh. (2022). Design and Analysis of Pump Housing for Water Cooled Engine System. 9. 318-330.
17. R.K. Purohit, *Thermal Engineering*, Scientific Publishers (India) Jodhpur 3rd Edition 1999, PP. 1177 – 1209.