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Design and Analysis of Piston for Two Stages Reciprocating Air Compressor

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Abstract: Compressed air is used for 900 hp locomotive brake system. This paper presents the design of piston for first stage and second stage of the two stage reciprocating air compressor. Also presents the static stress and nonlinear nodal stress analysis of the piston for first stage and second stage of two stage reciprocation air compressor. This design intend to support in modify of two stages reciprocating air compressor in 900 hp locomotive in practical usage. Therefore, the maximum final discharge (free air delivery), stroke and pressure of the high stage of the compressor were assumed as same as the practical usage of compressor about 760 liter per minute. The discharge is depending on the bore and stroke of the compressor. From this parameter, designed the bore of the piston to get require discharge.

Keywords: Design of Piston, Analysis of Piston, Reciprocating Compressor, Two Stage Air Compressor.

I. INTRODUCTION

Compressors are power consuming thermodynamic devices to compress the gases and vapors from low pressure to high pressure. According to the second law of thermodynamics, this is possible when work is done on the gas by as an external agency, such as prime movers, electric motor and internal combustion engine, etc., using direct and indirect transmission. Out of work received by the compressor, some is absorbed in overcoming the friction; some will lost as radiation, some will lost in cooling the fluid employed to cool the compressor and delivering the gas at high pressure. Thus, the compressor sucks gas at low pressure, compresses and delivers it at high pressure to the storage vessel called receiver from where it may be carried by a pipeline to where ever it is desired [7]. Small reciprocating compressors from 5 to 30 horsepower are commonly seen in automotive applications and are typically for intermittent duty. Larger reciprocating compressors well over 1000 hp are commonly found in large industrial and petroleum applications [6]. Most multi-stage compressors use intercoolers, which are heat exchangers that remove the heat of compression between the stages of compression. Intercooling affects the overall efficiency of the machine. As mechanical energy is applied to a gas for compression, the temperature of the gas increases. After-coolers are installed after final stage of compression to reduce the air temperature.

II. BACKGROUNDOF DESIGN

Multistage compression refers to the compression process completed in more than one stage a part of compression occurs in one cylinder and subsequently compressed air is sent to subsequent cylinders for further compression. In case it is desired to increase the compression ratio of compressor then multi-stage compression becomes inevitable.

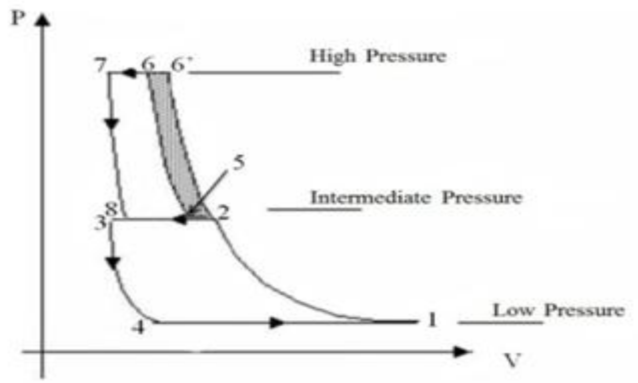


Fig.1. PV diagram for tow stage compressor.

Fig.1 shows the multistage compression occurring in two stages. Here first stage of compression occurs in cycle 1-2-3-4 and after first stage compression partly compressed enters second stage of compression and occurs in cycle 2-6-7-8 [1]. In that figure, process 1-2, 5-6,7-8 and 3-4 are polytrophic process. And then, process 2-5,6-7 and 4-1 are isothermal process. In this design, two stage reciprocating air compressor were driven by 900hp diesel locomotive engine via belt. By the ratio of the pulley, the minimum revolution of the compressor is 400 rpm the maximum revolution of the compressor is 985 rpm.

III. DESIGN CALCULATION

A single acting reciprocating compressor runs at 400 revolutions per min and takes in air at 1 bar and 33.2℃ and compresses it in 2 stages to 4 bar. The free air delivery is

0.00506 m^3/sec. There is an intercooler between each stage, which returns the air to 33.2˚C each stage has one piston with a stroke of 64mm.

The compression ratio per stage,

$$r_s = \sqrt[s]{r_t} \quad (1)$$

Where: s = number of stages

r_t = overall compression ratio($P_{final}/P_{initial}$)

Temperature of the end of the first stage compression is determined by following formula:

$$T_2 = (P_2/P_1)^{n-1/n} \times T_1 \quad (2)$$

Where: T_1 = room temperature

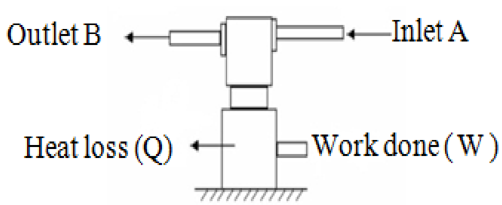
P_2 = low stage outlet pressure

P_1 = ambient pressure

$$\text{Mass of air; } m = (P \times V)/(R \times T) \quad (3)$$

The indicated power of the each stage is shown as follow;

$$I.P = mRnT/(n-1)[(P_2/P_1)^{n-1/n} - 1] \quad (4)$$



Steady flow energy equation;

$$H_A + P_{(in)} = H_B + Q_{(out)}$$

$$Q_{(out)} = P_{(in)} - mC_p(T_B - T_A) \quad (5)$$

Intercooling Effect;

$$Q_{(out)} = mC_p(T_2 - T_5) \quad (6)$$

Where;

$Q_{(out)}$ = Heat loss from intercooler

m = mass of air

C_p = specific heat at constant pressure

T_2 = low stage outlet temperature

T_5 = high stage inlet temperature

$$\text{The ideal isothermal power} = m_1 R T_1 \ln(P_2/P_1) \quad (7)$$

$$\eta_{iso} = \text{Isothermal power/Actual power} \quad (8)$$

Where;

η_{iso} = Isothermal Efficiency

R = Characteristic gas constant

Free Air Delivery = Swept Volume × Speed

Swept Volume = FAD/ Speed

Swept Volume = Bore Area × Stroke

$$D = \sqrt{(\text{Swept Volume} \times 4)/(\text{Stroke} \times \pi)} \quad (9)$$

A. Design of Piston Parts

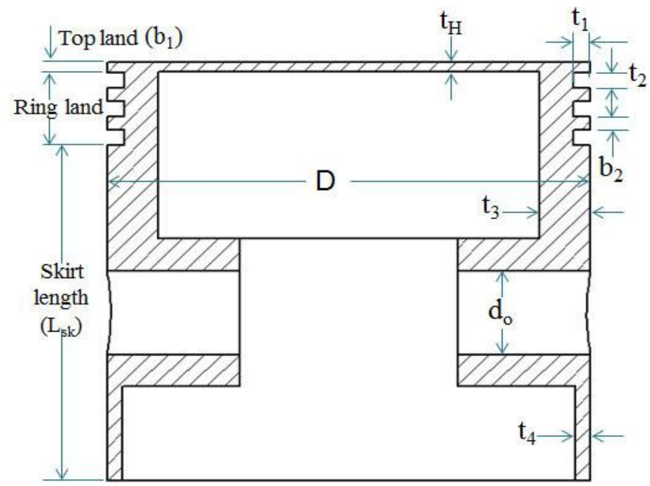


Fig.2. Design parameter of the piston.

Where,

D = diameter of the piston

t_H = thickness of piston head

t_1 = radial thickness of rings

t_2 = axial thickness of rings

t_3 = thickness of the piston barrel

t_4 = the wall thickness of the open end of the piston

b_1 = Width of top land

b_2 = Width of other land

d_o = outer diameter of the piston pin

L_{sk} = length of piston skirt

i. Design of piston head: Thickness of piston head (fig.2), according to Grashoff's formula,

$$t_H = \sqrt{3P.D^2/16\sigma_{th}} \quad (10)$$

Where,

P = Maximum gas pressure in N/mm^2,

D = Cylinder bore or outside diameter of the piston in mm and

σ_{th} = Permissible bending (tensile) stress for the material of the piston in MPa or N/mm^2 .

It may be taken as 35 to 40 MPa for grey cast iron, 50 to 90 MPa for nickel cast iron and aluminum alloy and 60 to 100 MPa for forged steel. In this design, choose the aluminum alloy. So, permissible bending (tensile) stress is 90 MPa.

ii. Design of Piston Rings: The radial thickness (t_1) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring.

$$t_1 = D\sqrt{3P_w/\sigma_{bt}} \quad (11)$$

Where,

P_w=pressure of gas on the cylinder wall in N/mm^2, [0.025 to 0.042 N/mm2].

σ_{bt} = allowable bending (tensile) stress inMPa , It value may be taken from 85 MPa to 110 Pa for cast iron rings[2].

t_1=radial thickness.
P_w=Pressure of the gas on the cylinder wall in N/mm^2. Its value is limited from 0.25 N/mm^2 to 0.042 N/mm^2.

In this design select the allowable bending (tensile) stress σ_{bt} is 98 MPa and Pressure of the gas on the cylinder wall is 0.042 N/mm^2.[2]

iii. Design Ofaxial Thickness Of Rings: The axial thickness (t_2) of the rings may be taken as $0.7t_1$ to $1.0t_1$. The minimum axial thickness may also be obtained from the following empirical relation:

$$t_2 = D/10n_R \tag{12}$$

$$n_R = \text{no: of rings}$$

In this design, used three number of rings.

iv. Width of Top Land: The width of top land (i.e the distance from the piston to the first ring groove) is made larger than other ring lands to protect the top ring from high temperature conditions existing at the top of the piston.

$$b_1 = t_H \text{ to } 1.2t_H \tag{13}$$

v. Width Of Other Ring Land: The width of other ring lands(i.e. the distance from the top of the piston to the first ring groove) is made larger than other ring lands to protect the top ring from high temperature conditions existing at the top of the piston,

$$t_3 = 0.03D+b+4.5 \tag{14}$$

vi. Maximum Thickness Of The Pistion Barrel: It is a cylindrical portion of the pistion. The maximum thickness(t_3) of the piston barrel may be obtained from the following empirical relation:

$$b = t_1 + 0.4$$

Where,
b= Radial depth of piston ring groove which is taken as 0.4mm larger than the radial thickness of the piston ring(t_1).

$$t_3 = 0.03D+t_1+4.9$$

$$t_4 = 0.25t_3 \text{ to } 0.35t_3 = 0.3t_3 \tag{15}$$

vii. The wall thickness towards the open end of the piston

$$L_{sk} = 0.65D \text{ to } 0.8D = 0.725D \tag{16}$$

Viii. Length of Piston Skirt: The portion of the piston below the ring section is known as piston skirt.

$$L = L_{sk} + 3t_2+2b_2+b_1 \tag{17}$$

The total length of the piston (L) is given by
L = Length of skirt+ Length of ring section+ Top land

$$L = L_{sk} + 3t_2+2b_2+b_1 \tag{18}$$

ix. Side Thrust Due To Gas Load: The side thrust (R_c) on the cylinder liner is usually taken as 1/10 of the maximum gas load on the piston.

$$Rc = F/10 \tag{19}$$

$$F = P \times \pi D^2/4 \tag{20}$$

Where,
F = the maximum gas load on the piston
P = Maximum gas pressure in N/mm^2

x. Maximum Bending Stress of The Piston

$$\sigma_B = P \times (D_i/2t_H)^2 \tag{21}$$

$$D_i = D - 2(s + t_3+\Delta t_r)$$
$$= D - 2(t_3 - t_1+t_3+\Delta t_r)$$
$$= D-2(2t_3-t_1+\Delta t_r) \tag{22}$$

Where,
D_i - Piston inner diameter in mm.
s - Thickness of crown wall (t_3-t_1).
Δt_r - Radial clearance of ring in piston (for compression ring- 0.7 to 0.95 mm)
t_H - the thickness of the piston head

B. Design of Piston Pin

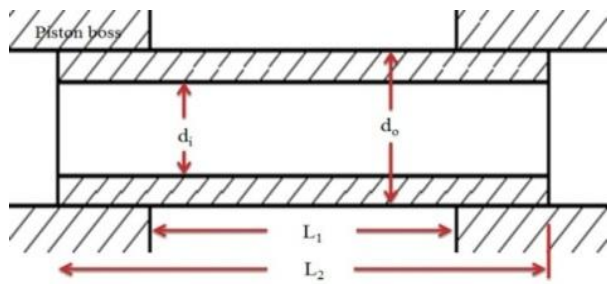


Fig.3. Design parameter of the piston pin.

Where,
d_o = Outter diameter of the piston pin
d_i = Inner diameter of the piston pin
L_1 = Length of the piston pin for connecting rod
L_2 = Length of the piston pin

Fig.3 presents the material used for the piston pin is usually case hardened steel alloy is 84 MPaand heat treated alloy steel is 140 MPa.[3] The piston pin should be design for the maximum gas load the piston. The bearing area of the piston pin should be about equally divided between the piston pin bosses and the connecting rod bushing. Thus the length of the pin in the connecting rod bushing will be about 0.45 of the cylinder bore or piston diameter (D), allowing for the clearance of the pin etc. The piston pin may be checked in bending by assuming the gas load to be uniformly distributed over the length (l_1) with supports at the center of the bosses at the two ends. The length of the piston pin (l_2) is taken as;

$$l_2 = (l_1+D)/2 \tag{23}$$

The maximum bending moment at the center of the pin,

$$M = F.D/8 \tag{24}$$

Where, F- Maximum gas load on the piston

$$d_o = [32M/]^{\frac{1}{3}} \tag{25}$$

$$d_i = k\, d_o \tag{26}$$

Where,

d_o - outside diameter of piston pin

d_i - inside diameter of piston pin

k - ratio of inner diameter to outer diameter (k = 0.6)

C. Result table of piston and piston pin

Table 1 and 2 describe the result of first stage piston pin and piston. Table 3 and 4 express the result of second stage piston pin and piston.

TABLE I: Result Table Of First Stage Piston Pin

Sr.No	Specification	Symbol	Result	Unit
1	Outer diameter of the piston pin	d_o	18	mm
2	Inner diameter of the piston pin	d_i	12	mm

TABLE II: Result Table Of First Stage Piston

Sr.No	Specification	Symbol	Result	Unit
1	Diameter of the piston	D	122	mm
2	Thickness of piston head	t_H	2.5	mm
3	Length of piston	L	110.2	mm
4	Maximum gas pressure on piston	P	0.2	N/mm^2
5	Maximum gas load on the piston	F	2337.9	N
6	Side thrust on the cylinder liner	R_c	233.79	N
7	Maximum bending stress of piston	σ_B	48.12	N/mm^2

TABLE III: Result Table Of Second Stage Piston Pin

Sr.No	Specification	Symbol	Result	Unit
1	Outer diameter of the piston pin	d_o	16	mm
2	Inner diameter of the piston pin	d_i	10	mm

Table IV: Result Table Of Second Stage Piston

Sr.No	Specification	Symbol	Result	Unit
1	Diameter of the piston	D	85	mm
2	Thickness of piston head	t_H	2.5	mm
3	Length of piston	L	78.57	mm
4	Maximum gas pressure on piston	P	0.4	N/mm^2
5	Maximum gas load on the piston	F	2269.8	N
6	Side thrust on the cylinder liner	R_c	226.98	N
7	Maximum bending stress of piston	σ_B	36.25	N/mm^2

Fig.4 and 5 express the 2D drawing and dimension of piston and piston pin of first stage and second stage. Fig.6 and 7 show the 3D drawing of piston and piston pin of first stage and second stage.

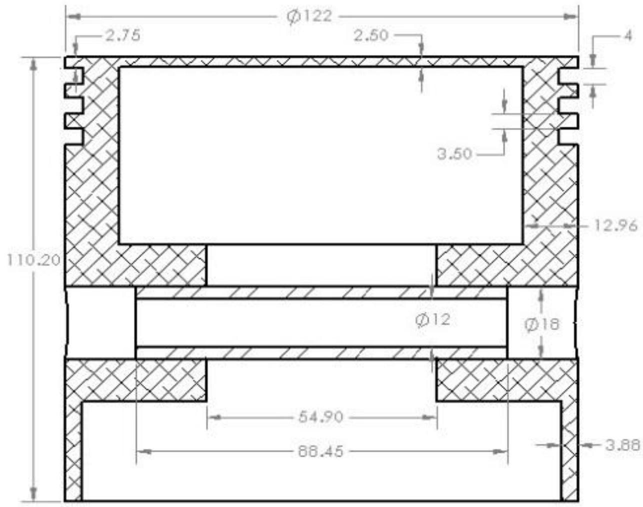


Fig.4. Dimension of the piston and piston pin for first stage.

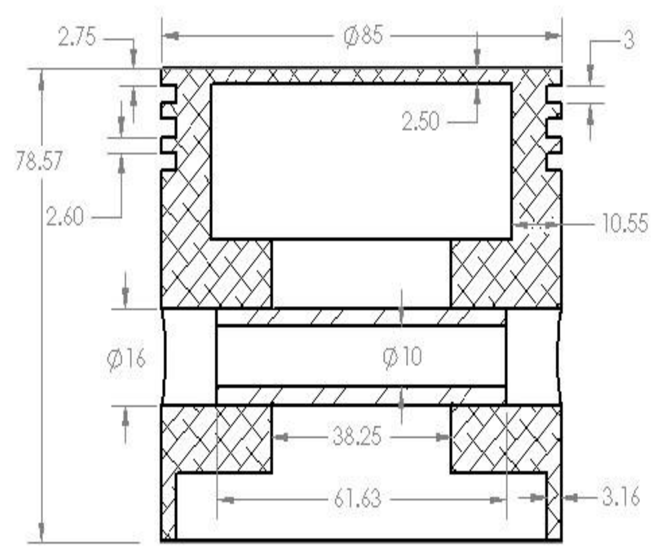


Fig.5. Dimension of the piston and piston pin for second stage.

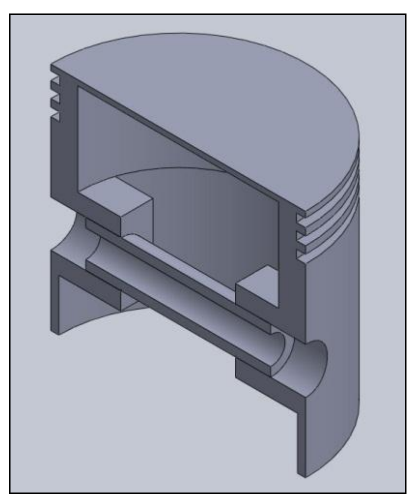


Fig.6. 3D drawing of the piston and piston pin for first stage.

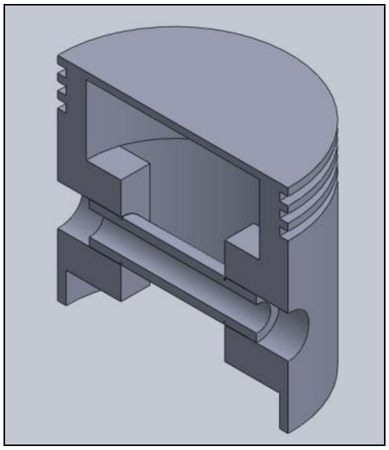


Fig.7. 3D drawing of the piston and piston pin for second stage.

D. Analysis of Piston and Piston Pin

To estimate the following stress, Solid Works software has been used. The material is used for the piston is 1060aluminum alloy. For piston, the maximum pressure in the cylinder is applied on the piston. The maximum pressure $0.2N/mm^2$ is on first stage piston head and $0.4N/mm^2$ is acting on second stage piston head. The contiguous area between piston pin and piston boss are fixed geometry for stress simulation [4] analysis as shown in figs.8 to 13.

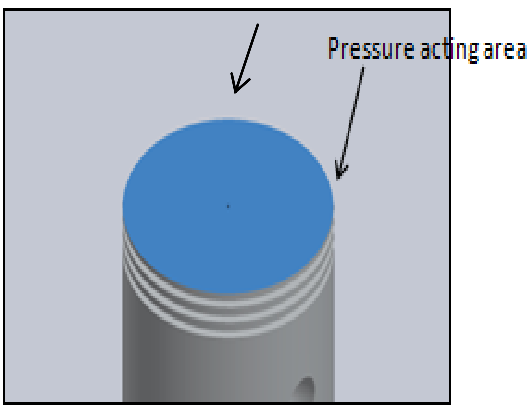


Fig.8.Pressure acting area on the piston head.

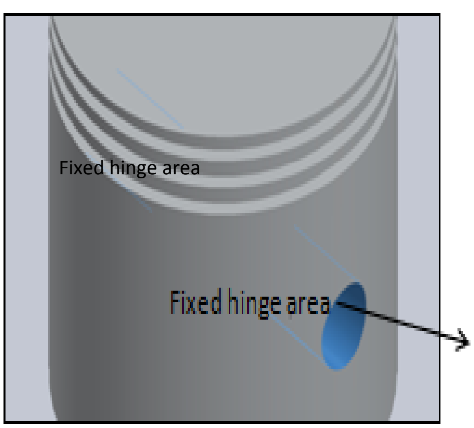
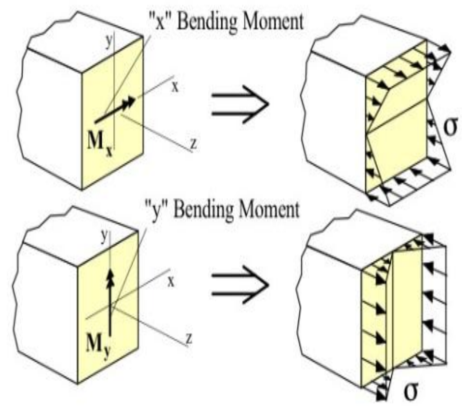


Fig.9. Fixed area on the piston.

Bending stress is happened on the perpendicular plain from exacted pressure. By this concept, pressure exacted from the Y-axis and result comes out from the Z-X plane. In this paper, show the normal stress parallel to the Z-axis.



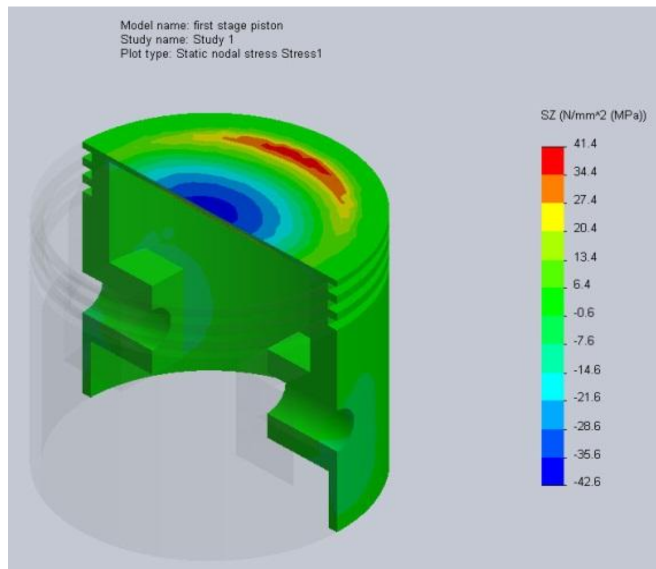


Fig.10. Stress analysis of first stage piston.

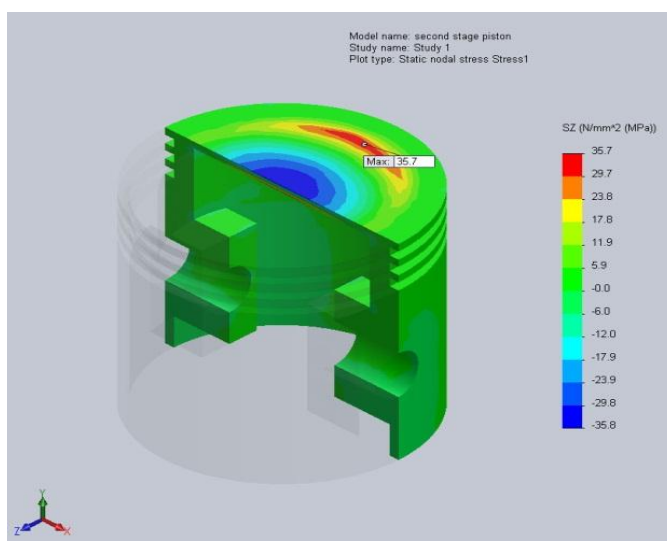


Fig. 11. Stress analysis of second stage piston.

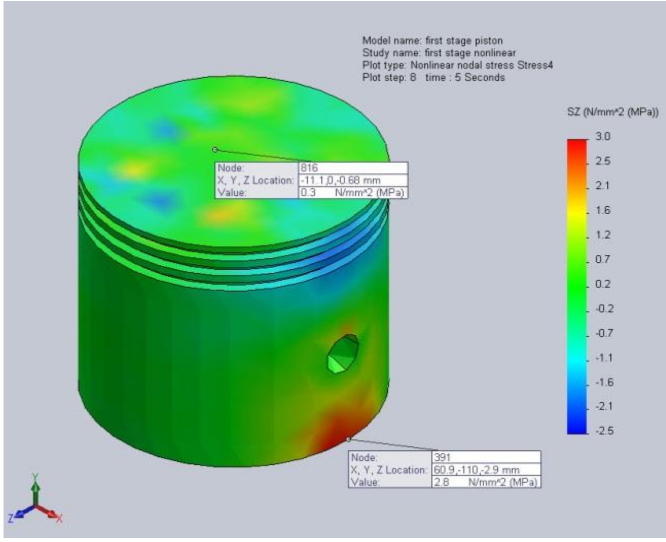


Fig.12.Nonlinear nodal stress analysis of first stage piston.

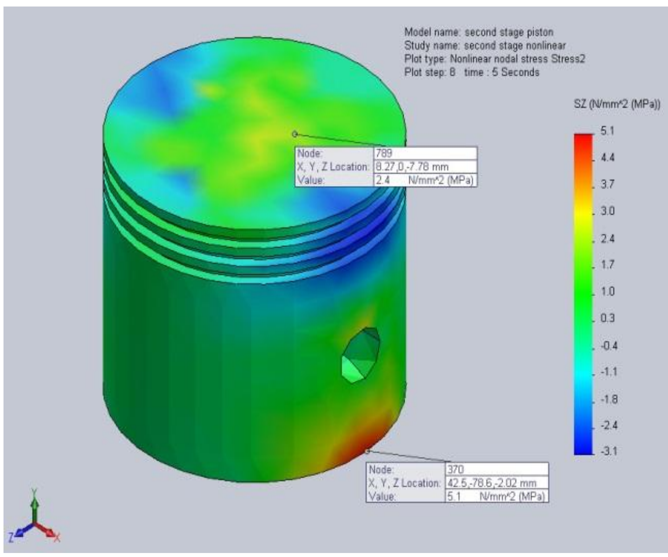


Fig.13.Nonlinear nodal stress analysis of second stage piston.

IV. RESULST AND DISCUSSION

Table V: Comparison of Bending Stress

Stage	Calculation result	Software result	Permissible Bending stress
1st stage piston	48.12MPa	41.4MPa	50MPa
2nd stage piston	36.25MPa	35.7MPa	50MPa

It can be seen from the stress analysis of piston of two stage reciprocating air compressor that the stresses produced during the operations are less as compared to the design stress. The maximum bending stress of the piston can be determined by this study. For first stage piston, maximum bending stress is 41.4MPa from software and 48.12MPa from calculation which is less than permissible bending stress 50MPa. For second stage piston, maximum bending stress is 35.7MPa from software and 36.25MPa from calculation result which is less than permissible bending stress 50MPa.So the design is safe. For nonlinear nodal stress analysis, variable time is 5 second and stress increment is 1 N/mm^2. And then, sample nodes were getting from the place of maximum stress and head of piston. In this design, the material is used for the piston is 1060aluminum alloy. In actual production, material can be change to get closely enough for permissible bending stress. Finally, Result as shown in figs.14 and 15.

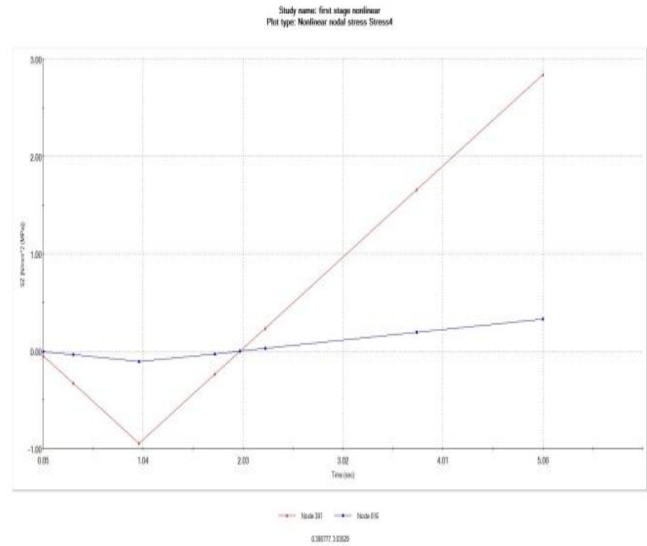


Fig.14. Result of nonlinear nodal stress analysis for first stage piston.

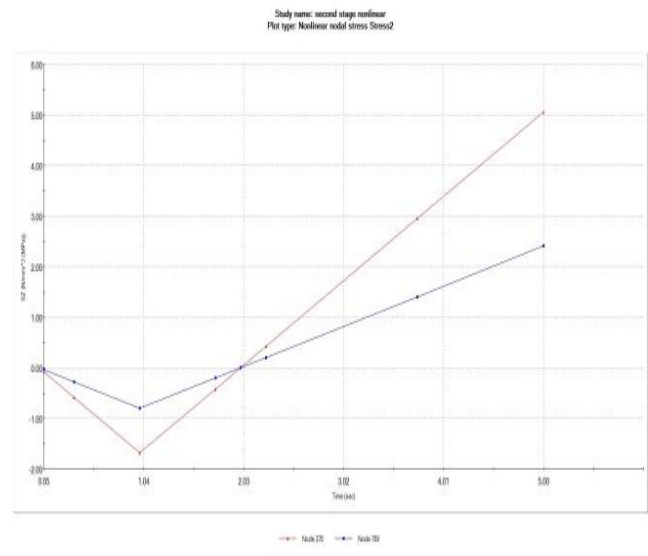


Fig.15. Result of nonlinear nodal stress analysis for second stage piston.

V. CONCLUSION

In this paper designed the piston of the two stage reciprocating air compressor. The piston diameter is designed by minimum revolution of the compressor to get the require discharge; because of the driven engine may be idle condition. The maximum bending stress of the piston is designed by the maximum gas pressure in the cylinder at the maximum revolution of the compressor. By analysis, using Solid works software and simulates the static stress and nonlinear nodal stress analysis to compare the result of software and result of design calculation. The other machines parts of two stage reciprocating air compressor will be calculate and compare with the result of simulation for further study.

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