Analysis and Optimization of Exhaust Valve of 4-Stroke CI Engine

[1]Suhail Tamboli, [2]Shivraj Salunkhe, [3]Omkar Shinde, [4]Prasad Pimple, [5]Prof. G.R. Nikhade
[1,2,3,4]Student, Mechanical Engineering, PVPIT Budhgaon, Maharashtra, India
[5]Assistant Professor, Mechanical Engineering, PVPIT Budhgaon, Maharashtra, India

Abstract—**Three roller sugar mill is the most vital part of sugar industry and sugar roller mill is used to separate the sucrose-containing juice from the cane. Cast iron has low elongation so that the roller teeth tend to break under high, concentrated loads, possibly saving other, more serious, extensive damage. The traditional cast iron shells are also low cost compared to more sophisticated materials. This study uses SBR alloy steel as an alternate material for roller grooves and compares it with the installed cast iron material to find the better grooves material. S. B. Reshellers Pvt. Ltd., after considerable research and development and various geometries, has patented a design proven to be highly efficient, known as 'Kamal' rollers. This paper discusses the operation of internal bleeding Kamal rollers and the benefits. The static analysis of crushing roller shaft grooves is carried out using ANSYS Workbench. The two material results for maximum shear stress on the Top, Feed, and Discharge roller are calculated analytically and compared with the results from software.**

Index Terms—**Sugar mill rollers; Max. Stress; ANSYS Workbench**

I. INTRODUCTION

The main objective of milling is to separate the sucrose containing juice from the cane. The prepared cane comes under three roller mills between top roller and feed roller where these rollers rotate and squeeze the prepared cane. The juice is thereby extracted and collected in a trough. The bagasse obtained after squeezing the juice is guided by means of trash plate to the opening between top roller and discharge roller and bagasse is squeezed once again in the set of three roller mills. In three roller mill, three rollers are arranged in triangular pattern for removing sucrose up to 96–97% max. There are grooved type three rollers and roller groove material is cast iron, used for better juicing efficiency.

The shaft of the roller is made up of forged steel and the shell of the roller is made up of cast iron. The shell is shrink fitted on the shaft. The arrangement of rollers in three roller mill has circumferential V grooving, with the main differences being the pitch and chevron grooves. Top roller is a critical component amongst all, as the drive torque, hydraulic load, and crushing load act on the top roller. Pressure acts at 160 kg/cm² on the top roller grooves.

The cast iron is compared with SBR alloy material to find the better grooves material. Wear due to corrosion as well as abrasion hampers the working of rollers and causes frequent re-grooving and shell changes. As arcing of the roller shells became standard practice for roughening the shell surface, the poor weldability of cast iron became a hindrance to surface engineering of the shells. Thus, shell material having higher strength, higher wear resistance, and better weldability was desirable. The top roller is most highly stressed, since it consumes about half of the mill torque. Out of total power, 50% is taken by the top roller, 35% by the discharge roller, and 15% by the feed roller.

II. PROBLEM STATEMENT AND OBJECTIVE

2.1. Statement

Analyses of crushing roller grooves are performed with two different materials. Analysis of three different rollers — top, feed, and discharge — is carried out. Analysis of grooves using cast iron and SBR alloy material is done both analytically and in ANSYS Workbench. The maximum shear stress and performance analysis of both materials are calculated to determine which material is better and most suitable for crushing roller grooves and arcing.

2.2. Objectives

1) Use good performance material for grooved roller for better efficiency.
2) Re-grooving can be completely avoided after first use.
3) Semi-smooth outer shell and analyze roller material strength and better groove.
4) Experimentation is done to find the maximum stress and deformation.

III. THEORETICAL APPROACH

In a conventional sugar mill, hydraulic load distribution is as per the following (refer Fig. 1):

a) Hydraulic Load applied on Top Roller is 100% for juice extraction.
b) Trash-plate absorbs maximum 25% of hydraulic load applied on top roller.
c) For juice extraction purpose maximum 75% of hydraulic load is used (i.e. 25% on Feed Roller and remaining 50% on Discharge Roller).

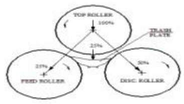

Fig. 1. Hydraulic load distribution on sugar mill rollers

The loads acting on the roller are due to crushing of sugar cane between top, feed, discharge roller and load due to the torque. The top roller is most highly stressed, since it consumes about half of the mill torque.

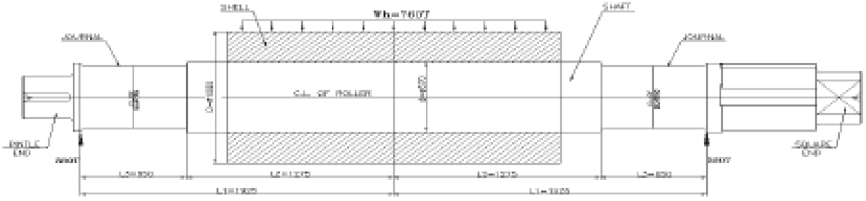

Fig. No. 3. Top, Feed & Discharge roller shaft.

Fig. 2. Three roller mill arrangement

The various terms relating to sugar mill rollers are as follows:

a) Shaft — A round forged steel bar on which cast iron shell or groove is fitted.
b) Roller journal — The polished surface at both ends of shell seat on which bearings are fitted.
c) Square end — The shaft end on which pinion and coupling are fitted.
d) Shell or groove — A hollow cast iron round that is shrunk-fitted on the shaft.

3.1. Analytical Calculation

Analytical calculation of roller groove material — cast iron and SBR alloy — to find stresses on the Top, Feed & Discharge roller in the sugar mill.

Cast Iron (ASTM40) Properties:

1) Roller Grooved material — Cast Iron (ASTM40)
2) Density — 7200 kg/m³
3) E — Modulus of Elasticity = 124 GPa
4) Poisson's ratio = 0.26
5) Syt — Yield strength in tension = 276 MPa
6) Sut — Ultimate tensile strength = 413 MPa
7) Pressure on top roller — 160 kg/cm²
8) Maximum weight of roller shaft with shell — 19.5 ton

Input Data (Cast Iron & SBR Alloy):

L_1 = 1200 mm
L_2 = 50 mm
L_3 = 4 mm (7 mm Arc)
D = 720 mm
d = 670 mm
HP = Mill power for drive = 670 HP
N = rpm of roller shaft = 4–5 rpm
W_h = Total hydraulic load = 760 ton

3.1.1. Calculation of Top, Feed & Discharge Roller Grooved

Torque transmitted = HP × 4500 / 2πN = 670 × 4500 / 2π × 4.5 = **119,963 kg-m**

3.1.2. Calculate the Maximum Stresses of Roller Groove and Arc

B.M. of roller grooved = $(276 \times 10^3) \times 1200 - (276 \times 10^3) \times (50/2)$

IV. DESIGN OF EXHAUST VALVE

4.1. Valve Specification

4-Stroke CI Engine — 450 cc

Valve Seat Angle: 30°
Gas Velocity: 2320 m/min
Mean Piston Speed: 210 m/min
Max. Gas Pressure: 6.25 N/mm²
Cylinder Bore Diameter: 140 mm
Stroke: 175 mm
Engine Speed: 1220 rpm
Exhaust Valve Temperature: 820°C
Length of Stem: 11.2 cm

4.2. Material Properties of Existing Valve

TABLE 1 Material Properties of Existing Valve

Material Properties	Symbol	Values for Ferritic AISI 409 Steel
Density	ρ	7800 kg/m³
Young's Modulus	E	220 GPa
Ultimate Tensile Strength	Sut	450 MPa
Yield Strength	Syt	250 MPa
Composition		Mn=1.0%, C=0.08%, Si=1.0%, Ti=0.75%, S=0.030%

4.3. Design

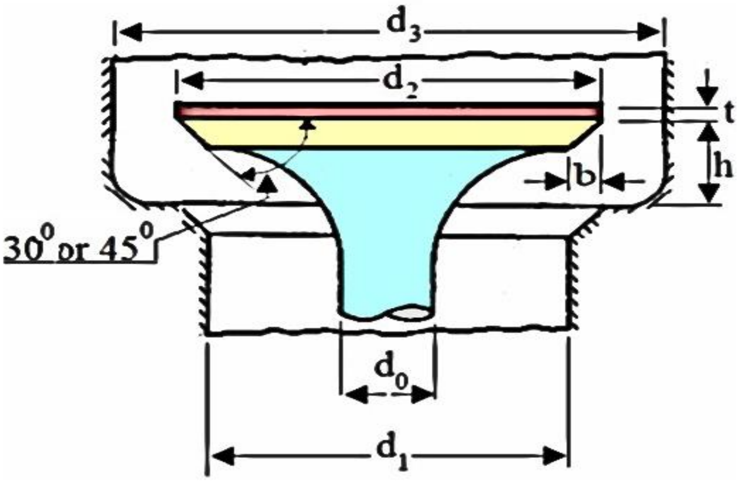

Fig. 1. Valve dimensions layout

The different parameters for the valve are calculated using the following formulae:

a) Port diameter: $d_1 = D / \sqrt{(Vg/Vp)}$
b) Valve lift: $h = d_1/4$
c) Port area: $A = (\pi/4)\, d_1^2$
d) Thickness of valve disc: $t = k_1\, d_1 \sqrt{(p\_c / Sut)}$
e) Valve head diameter: $d_2 = d_1 + 2(t \times \sin(90 - \alpha))$
f) Diameter of valve head opening area: $d_3 = \sqrt{(d_2^2 + d_1^2)}$
g) Width of seating: $b = 0.5(d_2 - d_1)$
h) Diameter of valve stem: $d_0 = (d_1/8) + 4$
i) Diameter check: $0.7854(d_3^2 - d_2^2) \geq 0.7854\, d_1^2$

$0.7854(66.25^2 - 51.14^2) \geq 0.7854(42.12)^2 \rightarrow 1759.55 \geq 1393.11$ — Design is safe.

4.4. Dimensions of Exhaust Valve

TABLE 2 Dimensions of Exhaust Valve

Design Parameter	Symbol	Dimension (mm)
Port Diameter	d_1	42.12
Valve lift	h	12.16
Thickness of valve disc	t	5.21
Diameter of valve head	d_2	51.14
Diameter of valve head opening area	d_3	66.25
Width of seating area	b	4.51
Diameter of valve stem	d_0	9.265

4.5. Forces on the Valve

Forces on the valve are due to gas pressure when it opens, the inertia force when the valve moves up, and the initial spring force to hold the valve in its seat against suction or negative pressure inside the cylinder.

a) Force due to gas pressure on the valve when it opens:

$F^G = (\pi/4)\, d_2^2\, p_c$ where: F^G = Gas Force; d_2 = Valve head diameter (mm); p_c = Cylinder pressure = 0.5 MPa

b) Inertia force when the valve moves up: F_A = mass × acceleration

Acceleration = $\pi^2\omega^2 h / 2\theta_L^2$; Cam Shaft Speed = ½ × 1220 × 2π/60 = 60.21 rad/sec; θ_L = 1.272 rad

Acceleration = $\pi^2 \times 60.21^2 \times 12.16 \times 10^{-3} / 2(1.272)^2 = 134.45$ m/s²

c) Initial spring force: $F_I = (\pi/4)\, d_2^2\, p_s$ where p_s = 0.03 MPa below atmosphere

d) Total Force on valve face: $F_T = F^G + F_A + F_I$

TABLE 3 Values of Force Acting on the Valve

Force Acting on Exhaust Valve	Values
Force due to gas pressure (F^G)	1027.03 N
Inertia force (F_A)	34.82 N
Initial spring force (F_I)	61.62 N
Total force (F_T)	1000.23 N

From the above table the inertia force represents the tensile force, whereas the compression force is the addition of initial spring force and force due to gas pressure, which is 1088.65 N.

V. ANALYSIS OF EXISTING VALVE

5.1. Transient Structural Analysis (9.25 mm Valve Radius)

Transient dynamic analysis is a technique used to determine the dynamic response of a structure under the action of any general time-dependent load. This type of analysis is used to determine the time-varying displacement, strain, and forces in a structure as the time scale of the loading is such that inertia or damping effects are considered important.

For the analysis of the valve, firstly the valve drawing is made with the evaluated dimensions, after which a CATIA model is created. Material properties of the existing material are added in the engineering data. Meshing of valve is done using tetrahedral elements and ANSYS is used for the required solutions. Following figures show the different results obtained during analysis.

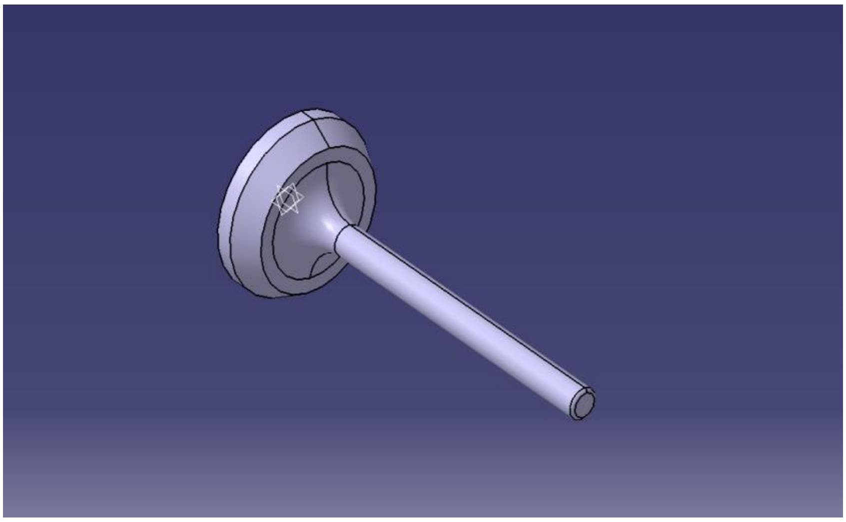

Fig. 2. CATIA model of Valve

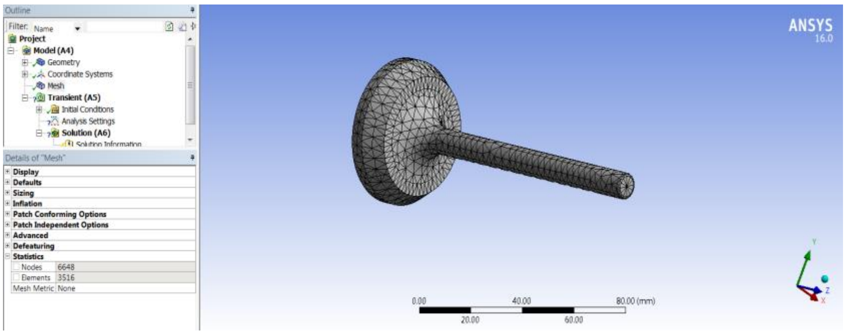

Fig. 3. Meshing of valve using tetrahedral element

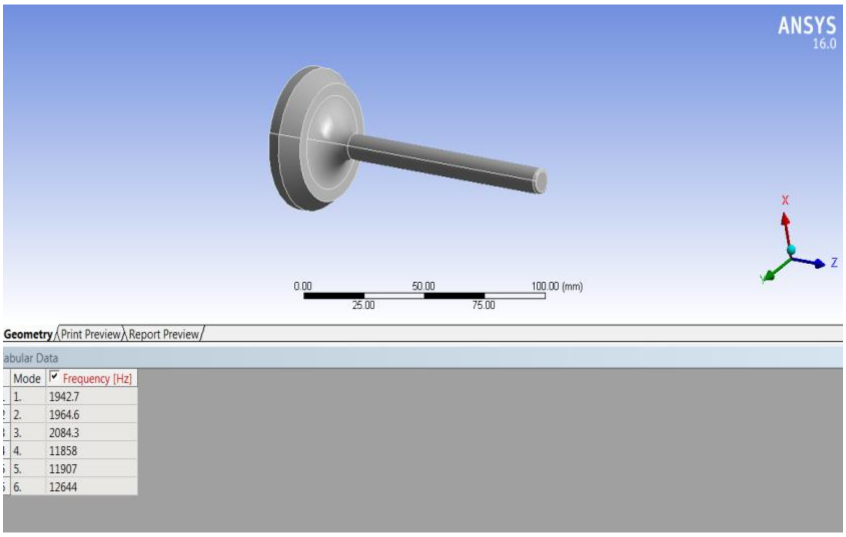

Fig. 4. Deformation in the valve

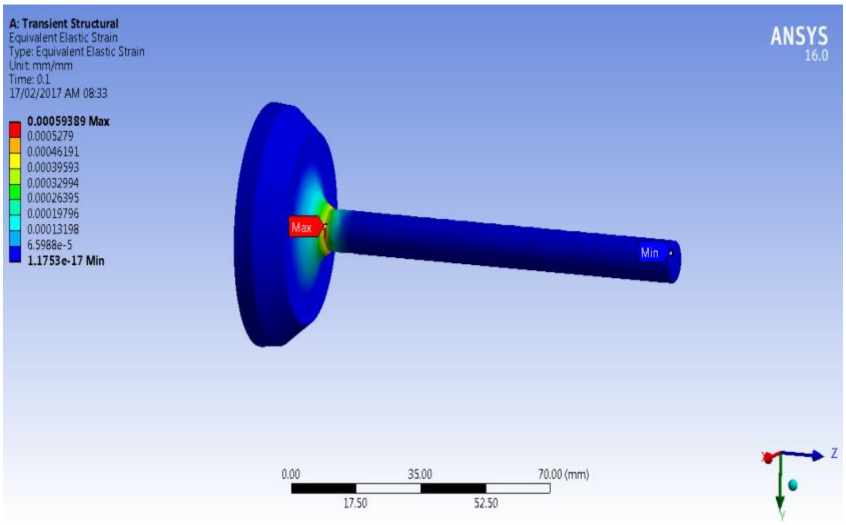

Fig. 5. Elastic strain in the valve

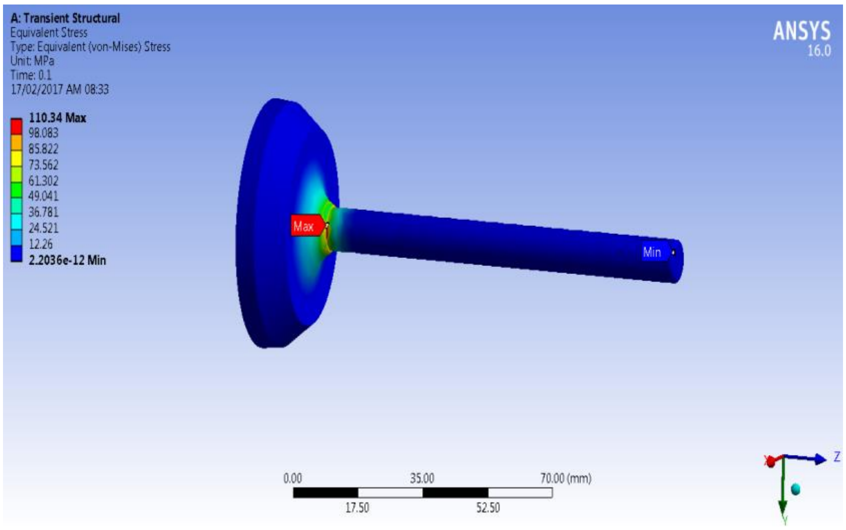

Fig. 6. Von-Mises stress in the valve

5.2. Modal Analysis (9.25 mm Valve Radius)

Modal analysis studies the dynamic properties of structures under vibration excitation. The modal analysis is obtained with different mode sets and their respective deformation. Each mode gives the vibration range in the form of frequency and maximum deformation at that frequency level. The maximum frequency is 12644 Hz, which is the natural frequency of the valve, above which resonance occurs and the structure experiences structural damage.

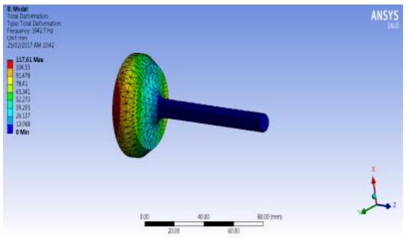

Fig. 7(a): 1st Mode shape

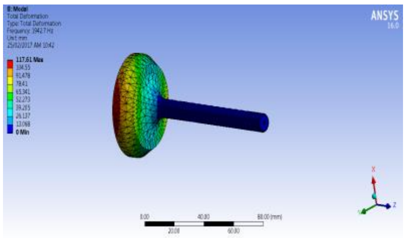

Fig. 7(b): 2nd Mode shape

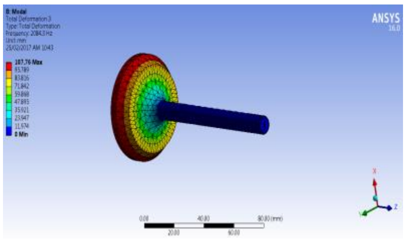

Fig. 7(c): 3rd Mode shape

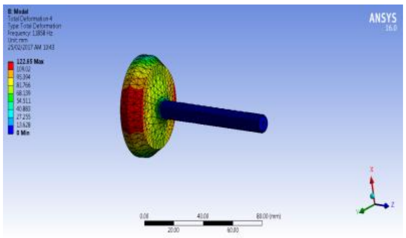

Fig. 7(d): 4th Mode shape

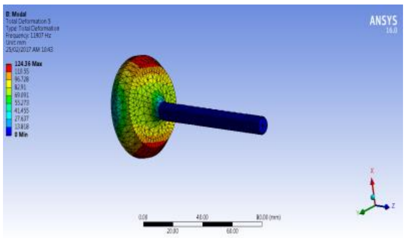

Fig. 7(e): 5th Mode shape

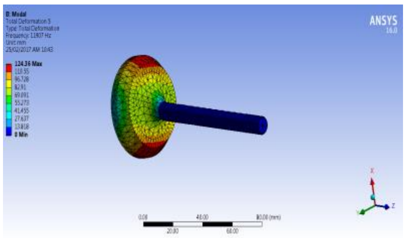

Fig. 7(f): 6th Mode shape

TABLE 4 Frequency Values and Their Deformation

Mode	Frequency (Hz)	Deformation (mm)
1	1942.7	117.61
2	1964.6	117.8
3	2084.3	107.76
4	11858	122.65
5	11907	124.36
6	12644	84.306

VI. OPTIMIZATION OF EXHAUST VALVE

6.1. Based on Valve Radius

From the analysis of the existing valve it is found that stress is concentrated at the valve radius. In order to reduce the stress, a number of trials are taken for the valve radius. The following table shows that the von-Mises stress obtained for the existing valve with 9.25 mm fillet radius is 110.34 MPa, which is above the allowable stress for the given material, causing failure. To obtain less stress at the fillet region, further valve radii are checked for stresses.

From the result table it can be seen that fillet 16.0 mm gives good results and the generated stresses are less compared to the existing fillet. Valve radius 16.00 mm is selected for further analysis. Stresses are 22.56% less and deformation is 11.39% less than the existing fillet. 16.0 mm is the maximum possible fillet that can be given to the valve.

TABLE 5 Valve Radius Trials for Modified Design

Trials	Fillet (mm)	Stress (MPa)	Deformation (mm)	Strain (mm/mm)
Trial 1	0	137.31	0.013765	0.0012702
Trial 2	2	133.67	0.0050015	0.00077069
Trial 3	4	124.50	0.0059952	0.0006869
Trial 4	6	120.63	0.0049696	0.00064063
Trial 5	8	115.79	0.0052576	0.00061465
Trial 6	9.25 (Existing)	110.34	0.0050293	0.00059389
Trial 7	12	98.751	0.0043375	0.00052021
Trial 8	14	88.489	0.0035392	0.00050017

Trials	Fillet (mm)	Stress (MPa)	Deformation (mm)	Strain (mm/mm)
Trial 9 (Selected)	16	85.449	0.0044562	0.00047561

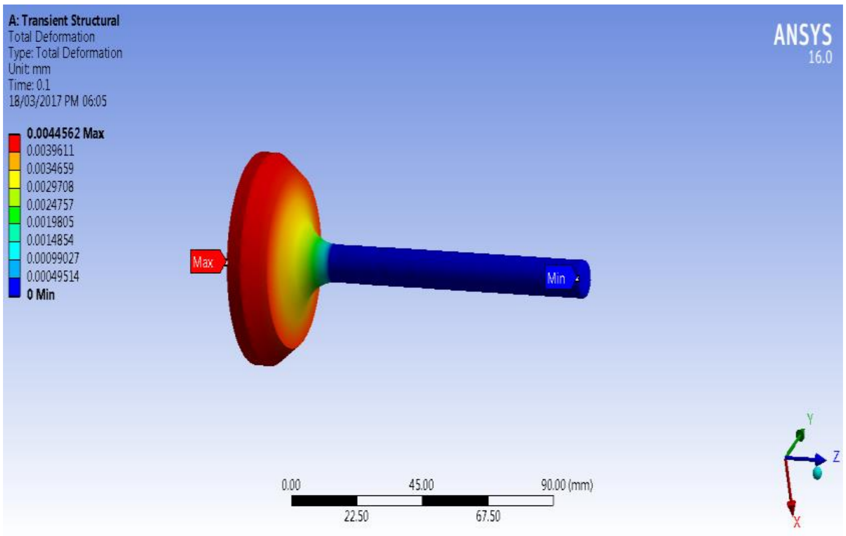

Fig. 8. Deformation in the valve (16 mm radius)

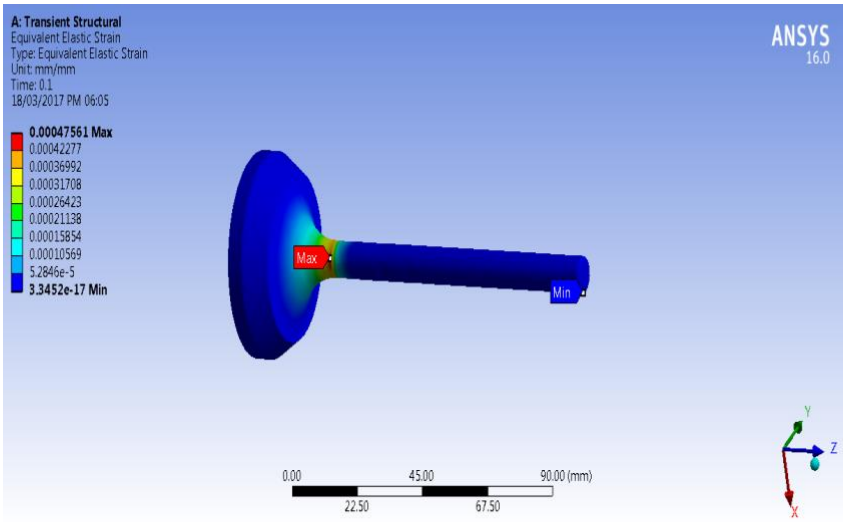

Fig. 9. Elastic strain in the valve (16 mm radius)

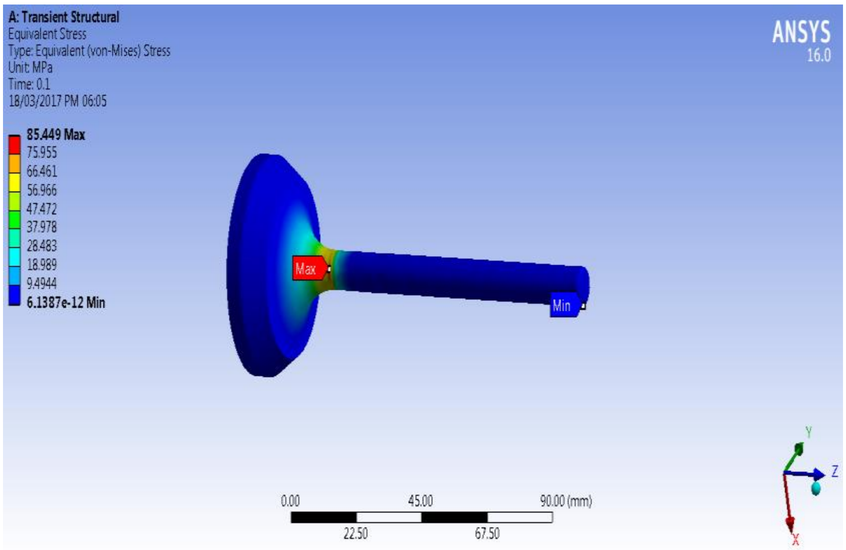

Fig. 10. Von-Mises stress in the valve (16 mm radius)

6.2. Based on Material

For providing alternate material for the valve, different materials are evaluated. Material properties are edited in the engineering data of ANSYS and further analysis is done to obtain deformation, strain and von-Mises stresses. The following table shows the different materials used as alternatives with their properties.

TABLE 6 Properties of Alternative Valve Materials

Material	Density ρ (kg/m³)	Young's Modulus E (GPa)	Yield Strength Syt (MPa)	Composition
ASTM A890	7800	230	520	Fe=58.1–65.9%, Cr=24–26%, Ni=6–8%, Mo=4–5%, Mn=0–1.5%, Si=0–1%
AISI 1541 Carbon Steel	7900	190–210	380–650	Fe=97.82–98.29%, Mn=1.35–1.65%, C=0.360–0.440%, S=0.05%, P=0.04%
Super Alloy 21-2N Valve Steel	7600	215	440	Cr=20.35%, Mn=8.5%, Ni=2.13%, C=0.55%, Mo=0.50%, Si=0.25%

TABLE 7 Results Based on Materials for Valve Radius 16.00 mm

Parameter	AISI 409	ASTM A890	AISI 1541	21-2N
Deformation (mm)	0.00445	0.00368	0.00319	0.00398
Elastic Strain (mm/mm)	0.00047	0.00052	0.00045	0.00056
Von-Mises Stress (MPa)	85.449	92.227	79.765	99.705
Allowable Stress (MPa)	83.33	173	133.33	146.67
Volume (mm³)	33675	33675	33675	33675
Density (kg/m³)	7500	7790	7600	7850
Weight (g)	252.56	262.33	255.93	264.35

From the above result it can be seen that Material AISI 1541 gives better results without failure and can therefore be used as alternative material to the existing material. AISI 409 steel (existing material) has low allowable stress but also shows reasonably good results. ASTM A890 has stress under the allowable limit but the stress values are overly safe. Hence AISI 1541 is selected for further experimentation.

VII. EXPERIMENTAL TESTING

For the testing of the exhaust valve of selected material AISI 1541 with valve radius 16.00 mm, a valve is tested on a universal testing machine.

TABLE 8 Specification of Universal Testing Machine

Specification	Value
Max Load Capacity	100 KN
Load Accuracy	Within ±1%
Test Space — Tensile	550 mm
Test Space — Compression	500 mm
Piston Stroke	200 mm
Dimensions	750 × 600 × 2100 mm
Power Supply	Three-Phase, 240V-50Hz

The valve with the selected material is tested in order to validate the results obtained from ANSYS. Experimental observations are given below:

TABLE 9 Observation Table

Weight (gm)	Material	Deformation (mm) Trial 1	Trial 2	Trial 3	Average
224.4	AISI 1541	0.0034	0.0033	0.0034	0.0034

VIII. RESULTS AND DISCUSSION

After performing the testing it was found that the final design valve of AISI 1541 material with 16.00 mm valve radius gives deformation of 0.0034 mm, which is less than 0.0050 mm for the existing valve. From the FEA results, the Von-Mises stress obtained from the existing valve is 110.34 MPa, which is above the allowable stress of 83.33 MPa. The final design gives stress of 79.76 MPa which is below the allowable stress for that material. The deformation obtained from the existing valve with FEA is 0.00502 mm, whereas the deformation of the new valve with AISI 1541 material is 0.00319 mm. The error between FEA deformation of the new valve and the experimental deformation is only 6.17%.

IX. CONCLUSION

Based on the transient structural analysis and the experimentation from the present study, the valve radius is the vital parameter to avoid failure of the valve. A valve with fillet radius 16.00 mm shows safe results. The exhaust valve with AISI 1541 material shows 36.56% less deformation and 27.71% less stress compared to the existing design.

X. FUTURE SCOPE

a) With the help of FFT, vibration analysis can be studied.
b) Thermal analysis can be done.

REFERENCES

[1] Lucjan Witek, "Failure and thermo-mechanical stress analysis of the exhaust valve of diesel engine," Engineering Failure Analysis, vol. 66, pp. 154–165, 2016.
[2] S. M. Jafari, H. Mehdigholi, and M. Behzad, "Valve Fault Diagnosis in Internal Combustion Engines Using Acoustic Emission and Artificial Neural Network," Hindawi Publishing Corporation Shock and Vibration, Vol. 2014, Article ID 823514, 9 pages.
[3] Kum-Chul Oh et al., "A Study of Durability Analysis Methodology for Engine Valve Considering Head Thermal Deformation and Dynamic Behaviour," R&D Center, Hyundai Motor Company, 2014 SIMULIA Community.
[4] Yuvraj K. Lavhal & Prof. Jeevan Salunke, "Overview of Failure Trend of Inlet & Exhaust Valve," International Journal of Mechanical Engineering and Technology (IJMET), ISSN 0976–6340, Vol. 5, Issue 3, March 2014, pp. 104–113.
[5] B. E. Gajbhiye, V. V. Bhoyar, S. B. Chawale, R. S. Ambade, "Vibration Testing and Performance Analysis of IC Exhaust Valve Using Finite Element Technique," International Journal of Research in Advent Technology, E-ISSN: 2321-9637, Vol. 2, No. 2, February 2014.
[6] Naresh Kr. Raghuwanshi et al., "Failure Analysis of Internal Combustion Engine Valves: A Review," International Journal of Innovative Research in Science, Engineering and Technology, ISSN: 2319–8753, Vol. 1, Issue 2, December 2012.
[7] Sanoj T. & S. Balamurugan, "Thermo Mechanical Analysis of Engine Valve," International Journal of Science and Research (IJSR), ISSN (Online): 2319-7064, Volume 3, Issue 5, May 2014.
[8] A. S. More & S. P. Deshmukh, "Analysis of Valve Mechanism — A Review," IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE), ISSN (e): 2278-1684, ISSN (p): 2320–334X, pp. 06–09.
[9] H. J. C. Voorwalda, R. C. Coisse, M. O. H. Cioffi, "Fatigue Strength of X45CrSi93 stainless steel applied as internal combustion engine valves," Procedia Engineering, vol. 10, pp. 1256–1261, 2011.
[10] P. Forsberg, P. Hollman, S. Jacobson, "Wear mechanism study of exhaust valve system in modern heavy duty combustion engines," Wear, vol. 271, pp. 2477–2484, 2011.
[11] Nurten Vardar et al., "Investigation of Exhaust Valve Failure in Heavy-duty Diesel Engine," Gazi University Journal of Science, GU J Sci, 23(4): 493–499, 2010.

[12] Jerzy Jaskolski, Rudolf Krzyzak, "The Temperature and Stress Fields of Valves of IC Engine," Journal of KONES Powertrain and Transport, Vol. 14, No. 3, 2007.