iJRASET
International Journal For Research in
Applied Science and Engineering Technology

INTERNATIONAL JOURNAL FOR RESEARCH

IN APPLIED SCIENCE & ENGINEERING TECHNOLOGY

Volume: 11 **Issue:** IV **Month of publication:** April 2023

DOI: https://doi.org/10.22214/ijraset.2023.51021

International Journal for Research in Applied Science & Engineering Technology (IJRASET)
ISSN: 2321-9653; IC Value: 45.98; SJ Impact Factor: 7.538
Volume 11 Issue IV Apr 2023- Available at www.ijraset.com

Design & Optimization Gudgeon Pin for C.I. Engine

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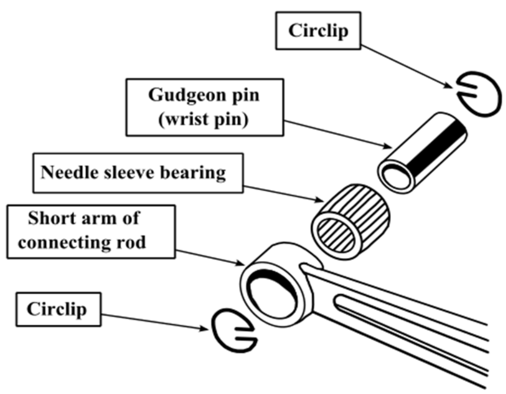
Abstract: Premature wear of the Gudgeon pin is the major concern for the company. Gudgeon pin connects the piston and the small end of the connecting rod of IC engines. This paper deals with Stress analysis and Fatigue analysis of Gudgeon pin used in diesel engine. In stress analysis frictional stresses and von-Mises stresses coming on Pin are determined using finite element analysis tool ANSYS 16. Effect of different factors on frictional stresses and Von-Mises stresses such as change in internal diameter of pin and application of diamond-like carbon (DLC) coating are analyzed. In Fatigue analysis, fatigue life of Pin is determined using fatigue analysis tool FEMFAT 5.0 b. Effect of change in surface roughness of connecting rod small end bush and change in internal diameter of pin on fatigue life is analyzed.

Keywords: Gudgeon Pin, Design, Analysis, Construction and working of Gudgeon pin, Fatigue analysis, NCODE ANSYS tool, CATIA.

I. INTRODUCTION

Excessive premature wear of the Gudgeon pins initiated this FEA investigation. The purpose was to develop a design variation that would remove this failure mode. Piston pin or Gudgeon pin or wrist pin connects the piston and the small end of the connecting rod of IC engines. Gudgeon pin is generally hollow and made from case hardening steel heat treated to produce a hard ware resisting surface. Though simple in appearance, without moving parts, it must be recognized as a precision engineered component. This is because it has to satisfy several conflicting requirements: It must combine strength with lightness; it must be close fitting but with freedom of movement, and it must resist wear without scuffing. The Gudgeon pin used in this study is of 24 mm outer diameter and 13 mm internal diameter made of 17Cr3 material. The expected operating life of pins is 3000 hours but the test results showed that the pin diameter gets reduced by 40 microns in merely 475 hours which is not desired. In this paper finite element analysis is performed on piston assembly which consists of piston, Gudgeon pin and connecting rod small end bush using FEA tool ANSYS. Contact analysis technique is used to analyse pin and bush in which frictional contacts are established in between piston, pin and bush. Piston assembly is then analyzed against the maximum combustion pressure and the frictional stresses and maximum Von-Mises stresses coming on the pin's outer surface are determined. Iterations have been performed for redesigning the pin by reducing pin's internal diameter and by application of diamond-like carbon (DLC) coating on pin. The effects of these redesigns on frictional stresses and on Von-Mises stresses are analyzed. At last fatigue analysis is performed on piston assembly using fatigue tool FEMFAT5.0b. Fatigue life of pin is determined with rough bush and with increased surface finish of bush. Also effect of reduced internal diameter of pin on the fatigue life is analyzed. In internal combustion engines, the gudgeon pin (UK, wrist pin US) connects the piston to the connecting rod and provides a bearing for the connecting rod to pivot upon as the piston moves.[1] In very early engine designs (including those driven by steam and also many very large stationary or marine engines), the Gudgeon pin is located in a sliding crosshead that connects to the piston via a rod. A Gudgeon is a pivot or journal. The origin of the word Gudgeon is the Middle English word gojoun, which originated from the Middle French word goujon. Its first known use was in the 15th century.



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II. PROBLEM STATEMENT

The function of the piston is to absorb the energy released after the combustion and to produce useful mechanical energy. When the combustion of fuel takes place in heavy diesel engine cylinder, high temperature and pressure develops. Because of high speed and at high loads, the piston is subjected to high thermal and structural stresses. The investigations indicate that the greatest stress appears on the upper end of the piston and stress concentration is one of the main reason for fatigue failure. Due to stress concentration and high thermal load the upper end of the piston, crack generally appears. This crack may even split the piston.

The main objectives are

1) To investigate the maximum stress using stress analysis
2) To investigate the maximum temperature using thermal analysis.
3) To investigate Stiffness of the piston crown to reduce the deformation.

III. OBJECTIVE

1) The objective is to identify the optimal combination of piston pin shape and Manufacturing.
2) To make this objective a reality, a computer-aided design (CAD) of the physical testing done by Ansys along with the various piston pin designs will be demonstrated using the Catia V5 software.
3) Objective to import this model into Stress is that to conduct quasi-static FE (Finite element) analysis.
4) Main objective is adjust the material data used in the analysis by various iterative methods.
5) After the analysis and Gudgeon pin will be manufactured and testing on it with compare to design results.

IV. MATERIAL & METHOD

Material used for Gudgeon Pin is stainless steel. In this project we use another material for Gudgeon Pin, i.e., Aluminum alloy.
Stainless steel is notable for its corrosion resistance, and it is widely used for food handling and cutlery among many other applications.
Stainless steel does not readily corrode, rust or stain with water as ordinary steel does. However, it is not fully stain-proof in low-oxygen, high-salinity, or poor air-circulation environments.
There are various grades and surface finishes of stainless steel to suit the environment the alloy must endure. Stainless steel is used where both the properties of steel and corrosion resistance are required.

A. Chemical Composition

(mass %)

Steel grade	C (carbon)	Si (Silicon)	Mn (Manganese)	P (Phosphorus)	S (Sulfur)	Cr (chromium)	Cu (Copper)	N (Nitrogen)	Others
JFE Standard	0.025 Max.	1.00 Max.	1.00 Max.	0.040 Max.	0.030 Max.	20.00~ 23.00	0.30~ 0.80	0.025 Max.	Ti 8X(C%+N%)~ 0.80%
Typical	0.01	0.1	0.2	0.03	0.002	20.8	0.4	0.01	Ti/0.3

B. Mechanical Properties

Steel grade	0.2% proof stress (N/mm^2)	Tensile stress (N/mm^2)	Elongation (%)	Mean r-value
JFE443CT	305	483	31	1.3
SUS430	320	490	29	1.0
SUS304	260	645	60	1.0

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C. Aluminum Alloy

Aluminum alloys (or aluminum alloys; see spelling differences) are alloys in which aluminum (Al) is the predominant metal. The typical alloying elements are copper, magnesium, manganese, silicon, tin and zinc. There are two principal classifications, namely casting alloys and wrought alloys, both of which are further subdivided into the categories heat-treatable and non-heat-treatable. About 85% of aluminum is used for wrought products, for example rolled plate, foils and extrusions. Cast aluminum alloys yield cost-effective products due to the low melting point, although they generally have lower tensile strengths than wrought alloys. The most important cast aluminum alloy system is Al–Si, where the high levels of silicon (4.0–13%) contribute to give good casting characteristics. Aluminum alloys are widely used in engineering structures and components where light weight or corrosion resistance is required. Alloys composed mostly of aluminum have been very important in aerospace manufacturing since the introduction of metal-skinned aircraft. Aluminum-magnesium alloys are both lighter than other aluminum alloys and much less flammable than alloys that contain a very high percentage of magnesium.

V. CALCULATION

Design Consideration, Design parameter and Design calculation:

A. Design Consideration for Piston & Pin

In designing a piston for an engine, the following points should be taken into consideration:

1) It should have enormous strength to withstand the high pressure.
2) It should have minimum weight to withstand the inertia forces.
3) It should form effective oil sealing in the cylinder.
4) It should provide sufficient bearing area to prevent undue wear.
5) It should have high speed reciprocation without noise.
6) It should be of sufficient rigid construction to withstand thermal and mechanical distortions.
7) It should have sufficient support for the piston pin.

B. Design Parameter

1) Thickness of piston head (tH)
2) Heat flows through the piston head (H)
3) Radial thickness of the ring (t1)
4) Axial thickness of the ring (t2)
5) Width of the top land (b1)
6) Width of other ring lands (b2)

Bearing Considerations - :

Given Data - :

Bore Diameter	= D = 68.5 X 10^{-3} m
Stroke Length	= L = 72 X 10^{-3} m
Maximum Gas Pressure	= 25 bar or
P	= 2.5 N / mm^2
Mean Effective Pressure	= 0.75 N / mm^2
FuelConsumption	= 0.15 kg / BP / ht
	= 0.15/3600 kg / BP / ht
	= 41.7 X 10^{-6}
Speed	= 2500 r.p.m.
HCV	= 42 X 10^3 KJ / Kg (High Calorific Value)
Maximum Power (37 bhp @ 5000 r.p.m.)	= 37 X 0.745 KN

1) Piston head or Crown on basis of strength

$t_H = \sqrt{3p.D^2}$

$= \sqrt{3 \times 5 \times (68.5)^2 / 38 \times 56}$

Considering σt for cast iron = 38 mpa

= 38 N / mm^2

= 7.6 mm

= 8 mm

Since engine is 4 stroke engine, therefore the number of working stroke per minut

n = N / 2

n = 5000 / 2

n = 2500

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And cross-sectional area of cylinder (A)

$$A = \pi D^2 / 4$$

$$A = \pi (68.5)^2$$

$$A = 3683.41 \text{ mm}^2$$

We know that indicated power,

$$Ip = PmLAn / 60$$

$$Ip = 0.75 \times 72 \times 10^{-3} \times 3683.41 \times 2500 / 60$$

$$Ip = 8287.6 \text{ W}$$

$$Ip = 8.287 \text{ kW}$$

We know that heat flowing to piston head (H)

$$H = C \times HCV \times m \times BP$$

C = Constant representing that portion of the heat supplied to the engine which is applied by piston, which is normally taken as 0.05.

Therefore $H = 0.05 \times 42 \times 10^3 \times 41.7 \times 10^{-6} \times 27.56 \text{ kW}$

$$H = 2.41 \text{ kW}$$

$$H = 2410 \text{ W}$$

2) Therefore thickness of the piston head on the basis of heat dissipation

$$t_H = H / 12.56\, k\, (T_C - T_E)$$

$$= 2410 / 12.56 \times (46.6) \times 220$$

$$= 0.01871 \text{ m}$$

$$= 18 \text{ mm}$$

(For cast iron value of k is 46.6 W/m°c)

Taking the larger or the two values we shall adopt

$$t_H = 18 \text{ mm}$$

Radial Ribs - :

The radial ribs may be 4 in numbers, the thickness of ribs varies from $t_H / 3$ to $t_H / 2$.

Therefore thickness of rib $t_R = 18 / 3$ to $18 / 2$

$= 6$ to 9 mm

Let us adopt $t_R = 7.5$ mm

3) Piston Ring

Let us assume there are 4 rings out of which 3 are compression rings and one is an oil ring, we know that radial thickness of piston ring.

$$t_L = D \times \sqrt{3Pw / \sigma t}$$

$$t_L = 68.5 \times \sqrt{3 \times 0.035 / 90}$$

$$t_R = 2.33 \text{ mm}$$

Here assume $Pw = 0.035 \text{ N / mm}^2$,

$\sigma_t = 90$ mpa. For piston pin outer

And

Axial thickness of piston ring $t_2 = 0.7 \times t_1$

$$t_2 = 1.63 \text{ mm}$$

$$t_2^* \ 2 \text{ mm}$$

Minimum axial thickness of piston ring,

$$t_2 = D / 10 \times n_i$$

$$t_2 = 68.5 / 10 \times 4$$

$$t_2 = 1.7125 \text{ mm}$$

Distance from top of the piston to the 1st ring groom

$b_1 = t_H$ to $1.2\, t_H$

$b_1 = 18$ to 21.6 mm

width of other ring,

$b_1 = 0.75\, t_2$ or t_2

$b_1 = 1.5$ or 1.71 mm

Gap between free ends of rib,

$G_1 = 3.5\, t_1$ to $4\, t_1$

$G_1 = 3.5 \times 2$ to 4×2

$G_1 = 7$ to 8 mm

G_2 = Gap between second ring

$G_2 = 0.002\, D$ to $0.004\, D$

$G_2 = 0.002 \times 68.5$ to 0.004×68.5

$G_2 = 0.137$ to 0.274 mm

By considering mean,

Therefore $G_1 = 7.5$ mm

$G_2 = 0.2055$ mm

4) Piston Pin,

Let, D_0 : Outside diameter of pin in mm.

L_1 : Length of pin in the bush of small end of C.R in mm.

P_{b1}: Bearing pressure at small end of C.R bushing in N / mm²

at value for bronz bushing is taken as 25 Ψ / mm²

Therefore load on pin due to bearing pressure = bearing pressure X bearing area

$$= P_{b1} \times D_0 \times L_1$$

$$= 25 \times D_0 \times 0.45 \times 6.85$$

Because $L_1 = 0.45\, D_0$

$$= 770.62\, D_0 \text{ mm}$$

Therefore we know the maximum load on the piston due to gas pressure or maximum gas load,

$$= \pi / 4 \times D^2 \times P$$

$$= \pi / 4 \text{ X } 68.5^2 \text{ X } 2.5 \text{ N / mm}^2$$

$$P = 9213.21 \text{ N}$$

From above we can find that

$$770.62\ D_0 = 9213.21$$

$$D_0 = 11.95 \text{ mm}$$

$$D_0 = 12 \text{ mm}$$

$$Di = 0.6 \text{ X } D_0$$

$$Di = 7.17 \text{ mm}$$

Let us piston pin made up of alloy steel. For which the bending stress (σ_b) may be taken as 540 mpa,

Maximum bending moment at centre of pin.

$$M = P_D / 8$$

$$M = 9213.21 \text{ X } 68.5 / 8$$

$$M = 7888811 \text{ N.mm}$$

$$M = 78.88 \text{ X } 10^3 \text{ N.mm}$$

We also know that max bending moment

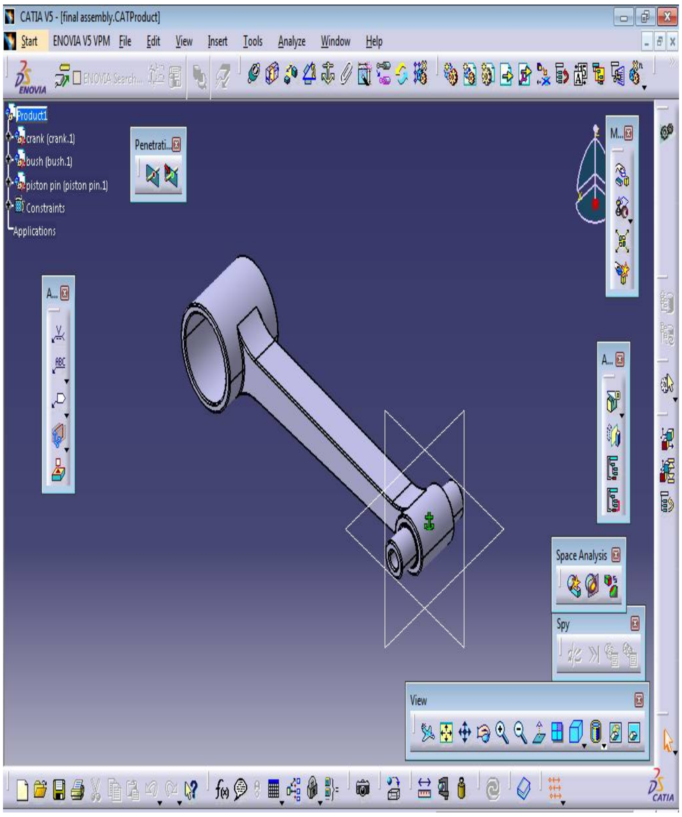
$$M = \pi/32\ [\ D_0^4 - D_i^4 / D_0] \text{ X } \sigma_b$$

$$48.88 \text{ X } 10^3 = \pi/32\ [\ 12^4 - 7.1^4 / D_0] \text{ X } \sigma_b$$

$$\sigma_b = 532.941 \text{ N / mm}^2$$

$$\sigma_b = 532.941 \text{ Mpa}$$

VI. CATIA DESIGN



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VII. UTM MACHINE TESTING

A. Test Result

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- Testing of Metal, Rubber, Plastic, Gasket, Foam, Dental.
- Calibration of Durometer, Temp. Sensors, Vernier & Load cells.
- Failure analysis

TEST REPORT

PRAJ/20 -03/358B

Date: 20/03/20

Student Name :
Address : Pune.
Reference :
Service Requirements : Testing of samples.

1.0 Tensile Strength
(As per ASTM D 638-2003 Standard)

Sr. No.	Sample Identification	Tensile Strength (MPa)
1	AL Sample	349.34
2	MS Sample	697.28
3	SS Sample	790.62
4	Composite Sample	267.48

2.0 Compression Strength
(As per ASTM D 695-2002)

Sr. No.	Deflection in mm	Load at various deflection (N)			
		AL Sample	MS Sample	SS Sample	Composite Sample
1	0.15	558.60	1136.80	984.90	940.80
2	0.30	1989.40	3322.20	3400.60	2195.20
3	0.45	4449.20	6027.00	6365.10	3777.90
4	0.60	7173.60	8859.20	9476.60	5566.40
5	0.75	10236.10	11686.50	12940.90	7389.20
6	0.90	13916.00	14709.80	16640.40	9192.40
7	1.05	17742.90	17836.00	20570.20	8879.80
8	1.20	21746.20	20981.80	24764.60	7918.40
9	1.40	26783.40	25666.20	26420.80	---

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- Failure analysis

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Date: 20/03/20

3.0 Izod Impact Strength
(As per ASTM D 256-2003)

Sr. No.	Sample Identification	Impact Value (J)	Izod Impact Strength (Kj/m^2)
1	Composite Sample	17.20	252.79

TEST CONDUCTED BY

SACHIN
TEST ENGINEER

for A.M.BHAGAT
PROPRIETOR

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NOTE: 1. Sampling is done by the party. Test Results pertain to the sample(s) received for the testing without prejudice of its lot or batch.
2. Report shall not be used in case of legal matters.

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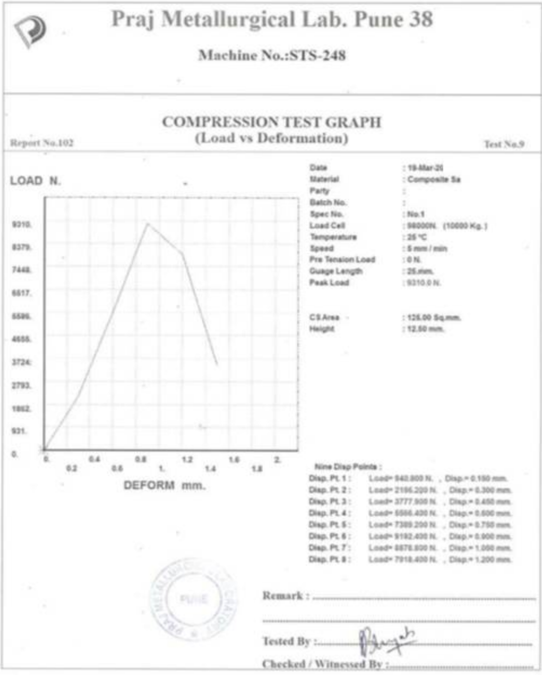

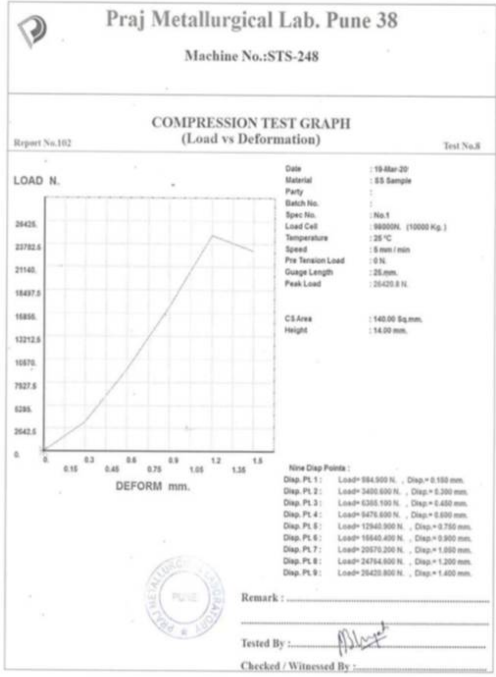

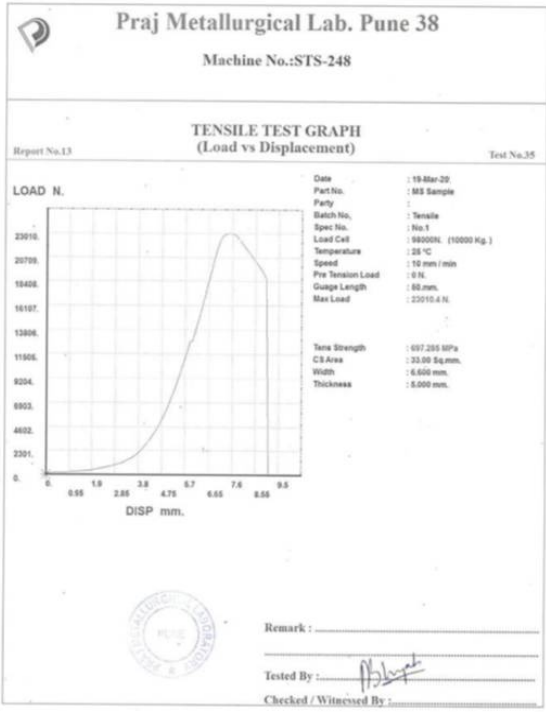

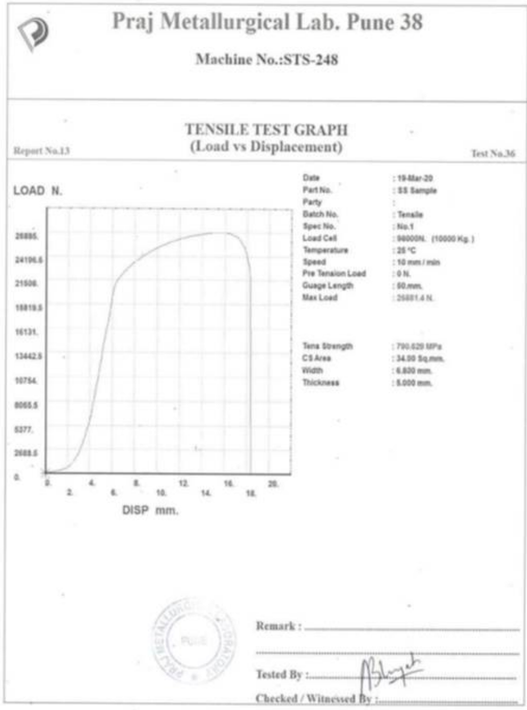

VIII. CONCLUSION

Sr. No	Material	Mass (kg)	Total Deformation (mm)	Von Misses Stress (Mpa)
1.	Carbon Fiber	1.6842	0.13806	200.94
2.	Aluminium alloy	1.7134	0.062008	133.43
3.	Stainless steel	1.8293	0.027231	143.9
4.	Mild steel	1.827	0.03646	106.96

IX. FUTURE SCOPE

Piston pin Design models are simulated on iteration based and it requires more number of iterations to check whether design is safe or not and to validate the models with the allowable. Instead of the above process, DOE – Design of Experiments concept can be used to optimize the design within short time and to get better optimized parameters. DOE should be carried in Ansys workbench. In Ansys workbench modelling can be done from Catia or Design Modeller using parametric model options. DP stands for design points, optimization can be done in workbench based on the required outputs namely deformations and stress with in prescribed limits.

Piston is one of the most important components of engine. It is a part in motion which is present in cylinder. In the engine the expansion of gas occurs in cylinder up to crankshaft through connecting rod. The piston lasts this gas pressure and inertial forces at work and this may lead to crack formation and piston wear.

The study reports show that stress concentration is highest at upper portion and this is one of the main reasons for crack formation and wear. This paper describes stress distribution on piston head of an IC engine by using finite element method. It is achieved by CAD and CAE software. Our main purpose is to study the static behaviour of piston head and analyze the stress distribution.In an automobile Industry piston is found to be most important part of the engine which is subjected to high mechanical and thermal stresses.

Due to very large temperature difference between the piston crown and cooling galleries induces much thermal stresses in the piston. Besides the gas pressure, piston acceleration and piston skirt side force can develop cycle of mechanical stresses which are superimposed on the thermal stresses. Due to this reason thermo-mechanical stresses are one of the main causes of the failure of the piston. Thus it has become very important to discuss the thermal and mechanical stresses to improve the quality and performance of the piston. In spite of all the improvements and advancements in the technologies there exists large number of defective or damaged pistons.

Thermal and mechanical fatigue plays a prominent role in the designing of pistons. Large numbers of complex fatigue tests are carried out by piston manufacturers but this involves very high cost and time .Thus finite element analysis is carried out for stresses, temperature gradient, and deformation and fatigue characteristics. In this paper, a detailed stress analysis of piston is done under various thermal and structural boundary conditions which are applied to the finite element model of the piston. Structural, thermal and coupled thermo-mechanical stresses and temperature gradient are obtained from the analysis. Life and Factor of safety for the piston are obtained from fatigue analysis.

Running conditions for piston pin boss bearing have become very severe due to the high combustion pressure and piston temperature increase over the past ten years. The aim of this paper was to analyze the friction and lubrication characteristic of piston pin boss bearings and a connecting rod small end bearing. Effects of different lubrication models, pin structures, and thermal deformation on the lubrication were discussed. The lubrication characteristics and performance parameters including oil film pressure distribution, asperity contact pressure, the minimum oil film thickness, the maximum oil film pressure, and friction power loss were listed. The results showed that the minimum oil film thickness was very different and the maximum oil film pressure was nearly the same. A parabola profile of pin bore can reduce the wear to some extent, and a flare profile intensified wear in some places and caused the wear to be concentrated on a smaller area. Reducing the inner diameters will reduce the wear of the pin boss. However, in a realistic design of the pin, avoiding high inertial force of the piston system and satisfying the demand for reliability of the pin, increasing the inner diameters and reliability is a 'trade off' problem.

X. ACKNOWLEDGEMENTS

The author would like to thank following guides for his constant encouragement and able guidance.
Prof. R.R Kulkarni
Prof. Kedar Bhagwat
Siddhant Collage of Engineering, Sudumbare, Pune.

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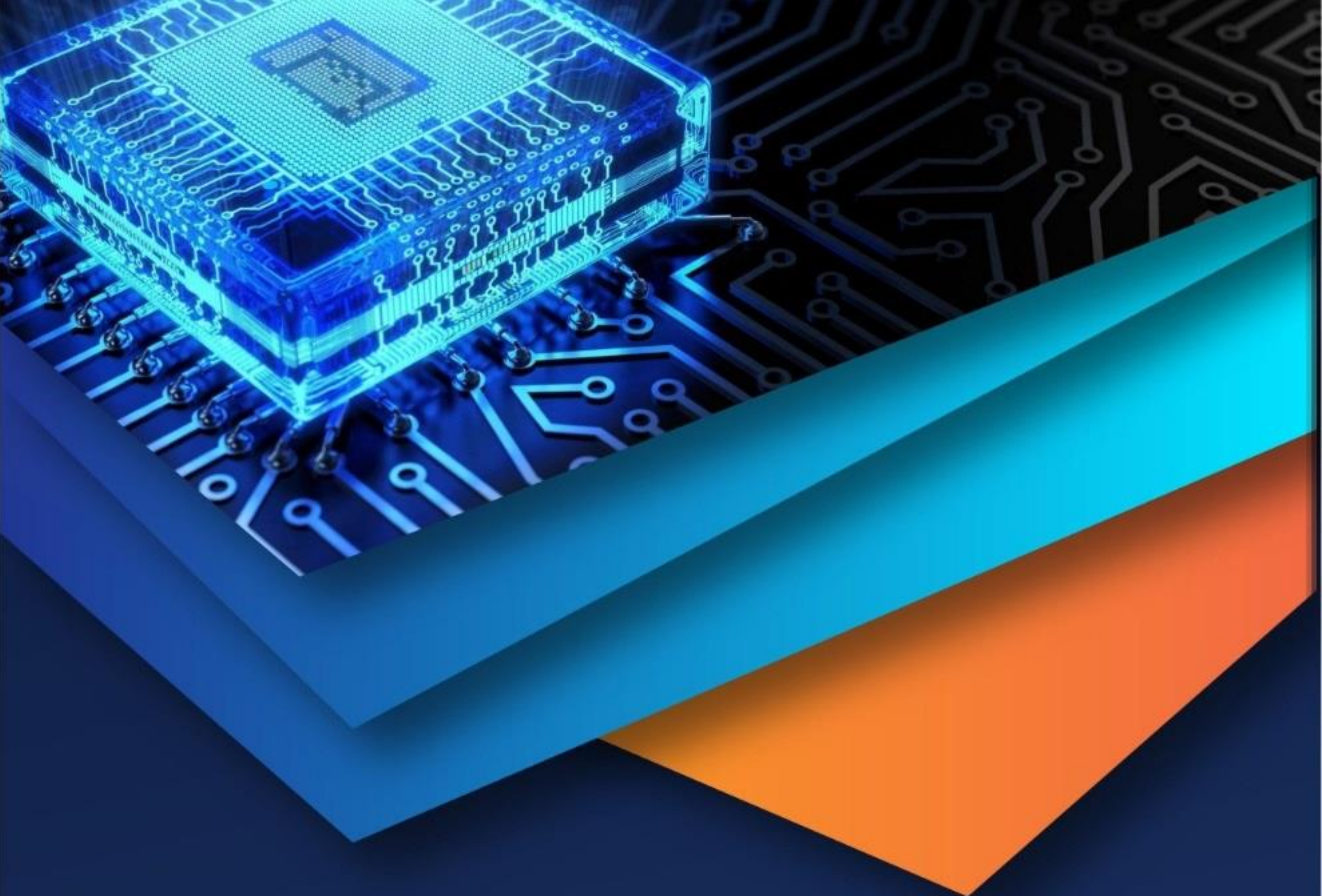

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