DESIGN AND ANALYSIS OF TWO WHEELER CONNECTING ROD USING DIFFERENT MATERIAL

Design and stress analysis

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Abstract: Connecting Rods are used practically generally used in all varieties of automobile engines acting as an intermediate link between the piston and the crankshaft of an engine of an automobile. It is responsible for transmission the up and down motion of the piston to the crankshaft of the engine, by converting the reciprocating motion of the piston to the rotary motion of crankshaft. While the one end, small end the connecting rod is connecting to the piston of the engine by the means of piston pin, the other end, the bigger end being connected to the crankshaft with lower end big end bearing by generally two bolts. Generally connecting rods are being made up of stainless steel and aluminum alloy through the forging process, as this method provides high productivity and that too with a lower production cost. Forces generated on the connected rod are generally by weight and combustion of fuel inside cylinder acts upon piston and then on the connecting rod, which results in both the bending and axial stresses. The lateral bending stress are commonly called as whipping stress and this whipping stress forms the base of evaluation of performance of various materials that can be used for manufacturing of connecting rod. The conventional material used is steel which is designed using CAD tool which is CATIA V5 and subsequently analyzed for bending stress acting on it using ANSYS workbench 16.1 and this procedure is followed for different material which are cast iron grade 25 , carbon40 .

***IndexTerms* –Connecting rod, Catia, ANSYS, Failure,**

I. Introduction

The conversion of heat energy into mechanical energy is achieved by device called as internal combustion engine and connecting rod forms an integral part of this engine. The function of connecting rod is to transmit the gas pressure available at piston end to crank end for efficient conversion of heat energy into calculation of whipping stress due to inertia is first calculated for each material using analytical mechanical energy. During this process of energy conversion various forces and corresponding stresses are generated on the body of connecting rod and it forms the most stressed part of I. C engines. The two forces acting on connecting rod can be classified as buckling load due to gas pressure and lateral bending due to inertia forces.

The inertia forces cause bending of connecting rod which causes whipping stress acting on it. In this project the conventional material used for manufacturing of connecting that is steel is replaced by aluminum alloys such as 7075 and 6061 and High Strength carbon fiber. Computers employing microelectronics technology are called for aiding the geometric modeling of connecting rod which would provide the model for subjecting it to various materials.

Geometrically modeled connecting rod in CATIA V5 is subjected to different materials and the equivalent stress results are calculated software named as ANSYS which are compared with the equivalent stress acting on the conventional connecting rod material that is steel to provide a platform for evaluation and validation of design.

A.MATERIAL USED FOR FLYWHEEL:

1. Cast iron grade 25
2. Aluminum 7075
3. Carbon 40

II. SOFTWARE

The software will start (by default) with all toolbars docked to the edges of the main window. The toolbars contain buttons, which when clicked, open the various information windows or operate features in the software. The toolbars and windows can be freely moved around inside the main program window, to create your own screen layout.

A.INRODUCTION TO CATIA

CATIA started as an in-house development in 1977 by French aircraft manufacturer Avion Marcel Dassault, at that time customer of the CADAM software to develop Dassault's Mirage fighter jet. It was later adopted by the aerospace, automotive, shipbuilding, and other industries. Initially named CATI (*conception assistée tridimensionnelle interactive* – French for *interactive aided three-dimensional design*), it was renamed CATIA in 1981 when Dassault created a subsidiary to develop and sell the software and signed a non-exclusive distribution agreement with IBM. In 1984, the Boeing Company chose CATIA V2 as its main 3D CAD tool, becoming its largest customer. In 1988, CATIA V3 was ported from mainframe computers to UNIX. In 1990, General Dynamics Electric BoatCorp chose CATIA as its main 3D CAD tool to design the U.S. Navy's Virginia class submarine. Also, Lockheed was selling its CADAM system worldwide through the channel of IBM since 1978.In 1992, CADAM was purchased from IBM, and the next year CATIA CADAM V4 was published. In 1996, it was ported from one to four UNIX operating systems, including IBM AIX, Silicon Graphics IRIX, Sun Microsystems SunOS, and Hewlett-Packard HP-UX. In 1998, V5 was released and was an entirely rewritten version of CATIA with support for UNIX, Windows NT and Windows XP (since 2001). In the years prior to 2000, problems caused by incompatibility between versions of CATIA (Version 4 and Version 5) led to $6.1B in additional costs due to years of project delays in production of the Airbus A380. In 2008, Dassault Systèmes released CATIA V6. While the server can run on Microsoft Windows, Linux or AIX, client support for any operating system other than Microsoft Windows was dropped. In November 2010, Dassault Systèmes launched CATIA V6R2011x, the latest release of its PLM2.0 platform, while continuing to support and improve its CATIA V5 software. In June 2011, Dassault Systèmes launched V6 R2012. In 2012, Dassault Systèmes launched V6 2013x. In 2014, Dassault Systèmes launched 3DEXPERIENCE Platform R2014x and CATIA on the Cloud, a cloud version of its software.

B. INTRODUCTION TO ANSYS WORKBENCH

ANSYS mechanical is a finite element analysis tool for structural analysis including linear, nonlinear and dynamic studies. This computer simulation product provides finite elements to model behavior and supports material models and equation solvers for a wide range of mechanical design problems. ANSYS mechanical also includes thermal HYPER LINK and coupled analysis capabilities acoustics, piezoelectric, thermal –structural and thermos electric analysis.

III.DESIGN

SPECIFICATIONS OF A BAJAJ PULSAR 150 ENGINE:

Bore x stroke = 63.5 x 56.4 mm
Displacement =2 178.6 cc
Maximum power = 17.2 ps @ 8500 rpm
Compression ratio = 9.5 : 1
Auto-ignition temperature = 60

PRESSURE CALCULATIONS:

ACCORDING TO IDEAL GAS CONSTANT PV=MRT
Mass=Volume x Density
Volume = 178.6 cm^3 = 178.6 x 10^3 mm^3
Density of petrol = 737.22 x 10^{-9} kg/mm^3
Mass = 178.6 x 10^3 x 737.22 x 10^{-9} = 0.1316 kg
R specific = specific gas constant / molecular weight of petrol
Molecular weight of petrol = 114.228 g/mole
Specific gas constant = 8.3143
R specific = 8.3143 / 0.114228 = 72.786
PV = MRT
P = MRT / V
= 0.1316 X 72.786 X 288.15 / 178.6
= 15.49 mpa

DESIGN CALCULATIONS OF A CONNECTING ROD:

A connecting rod is a machine member which is subjected to alternating direct compressive and tensile forces. Since the compressive forces are much higher than the tensile force, therefore the cross- section of the connecting rod is designed as a strut and the rankine formula is used. A connecting rod subjected to an axial load W may buckle with x-axis as neutral axis in the plane of motion of the connecting rod,{or} y-axis is a neutral axis. The connecting rod is considered like both ends hinged for buckling about x-axis and both ends fixed for buckling about y-axis. A connecting rod should be equally strong in buckling about either axis.

Maximum force on the piston due to pressure fi = π / 4 $(d)^2$ x p
= π / 4 $(63.5)^2$ x 15.49
= 49087.28 N

Maximum angular speed $\omega_{max} = 2\pi N_{max}$ / 60
= 2π x 8500 / 60
= 890.11

Crank velocity = rω = 31.75 x 10^{-3} x 890.11 = 28.26 m/ sec

Buckling load (wb) = Fc x F.s
= 90542.29 x 5
= 245436.43 N

Radius of gyration kxx = $\sqrt{\frac{Ixx}{A}} = 1.78\,t$

Ixx = $419t^4$, A = 1 / $11t^2$

According to rankine formulae

WB = σ_c x a / 1 + a $[L / Kxx]^2$

For cast iron grade 25 σc = 6000 kgf / cm^2
= 600 N /mm^2

a = 1/1600 (for cast iron)

245436.43 = $(600x11t^2)/(1+((1/1600)x(112.8/1.78t)^2)$

754.51 = $11t^2/(1+(2.5/t^2))$

68.59= $t^4/t^2+2.5$

$T^4-68.59t^2-171.475=0$

$T^2=71$

T=6.5mm

DIMENSIONS OF THE CONNECTING ROD :

WIDTH OF THE SECTION B = 4t =4x6.5 =26mm

HEIGHT OF THE SECTION H = 5t =5x6.5=32.5mm

DEPTH NEAR THE BIG END H1 =1.2H = 1.2X32.5 =39mm

DEPTH NEAR THE SMALL END H2=0.85H=0.85x32.5 =27.625mm

LENGTH OF THE CONNECTING ROD = 2TIMES OF STROKE
=2X56.4
=112.8mm

Load on the big end bearing or crank pin = projected area x bearing pressure
= dc x lc x Pbc

dc = diameter of crank pin

lc = length of crank pin [1.0 to 1.25 dc]

Pbc = bearing pressure [10.8 to 12.6 N/mm^2]
= dc x 1.2 dc x 11
= 13.2 dc^2
13.2dc^2= 38542.29
dc = 60.98 mm
= 61 mm

Load on piston pin or small end bearing = projected area x bearing pressure
= dp x lp x Pbp
= dp x 1.5dp x 13
= 19.5dp^2
19.5dp^2 =2542.29
dp =50.12 mm = 51mm

Length of crank pin = 1.2 dc = 1.2 x 83 = 99.6 mm = 100 mm

Length of piston pin = 1.5 dp = 1.5 x 68.14 = 102.21 = 103 mm

Size of bolt for securing the big end cap :

Force on the bolts = π / 4 $(dcb)^2$ x at x hb

D_{cb} =core diameter of the bolts

σt = allowable tensile sterss for the material

n_b= No. of bolts

σt=150N/mm^2

= π / 4 $(dcb)^2$ x 60 x 2 = 94.26 $(dcb)^2$

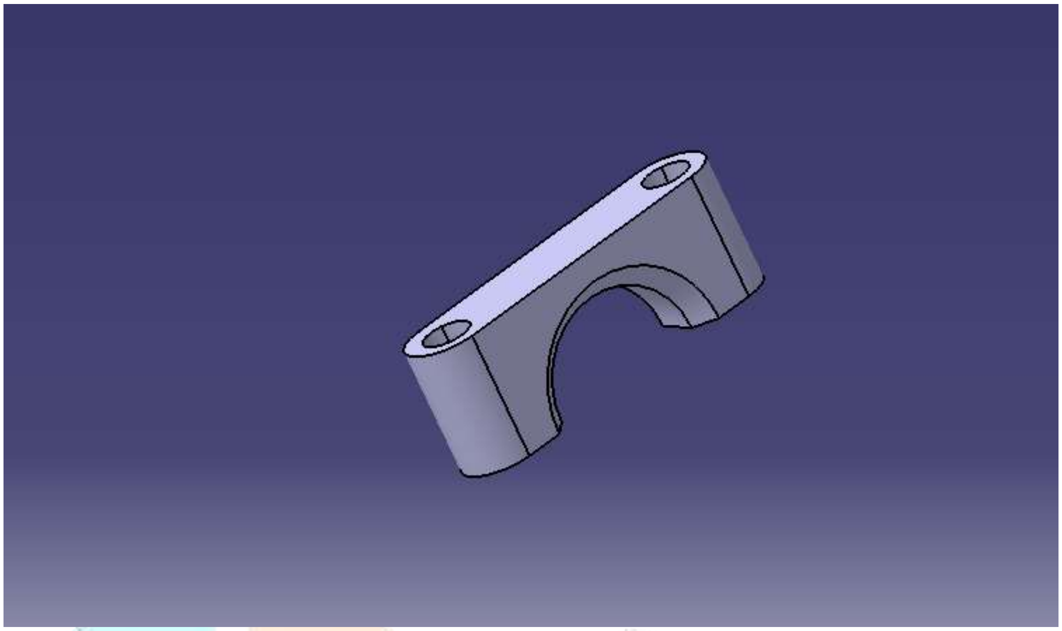
M_r = mass of reciprocating parts = 2.25

F_I = MR .ω^2.r [cosθ + [cos2θ/(l/r)]]

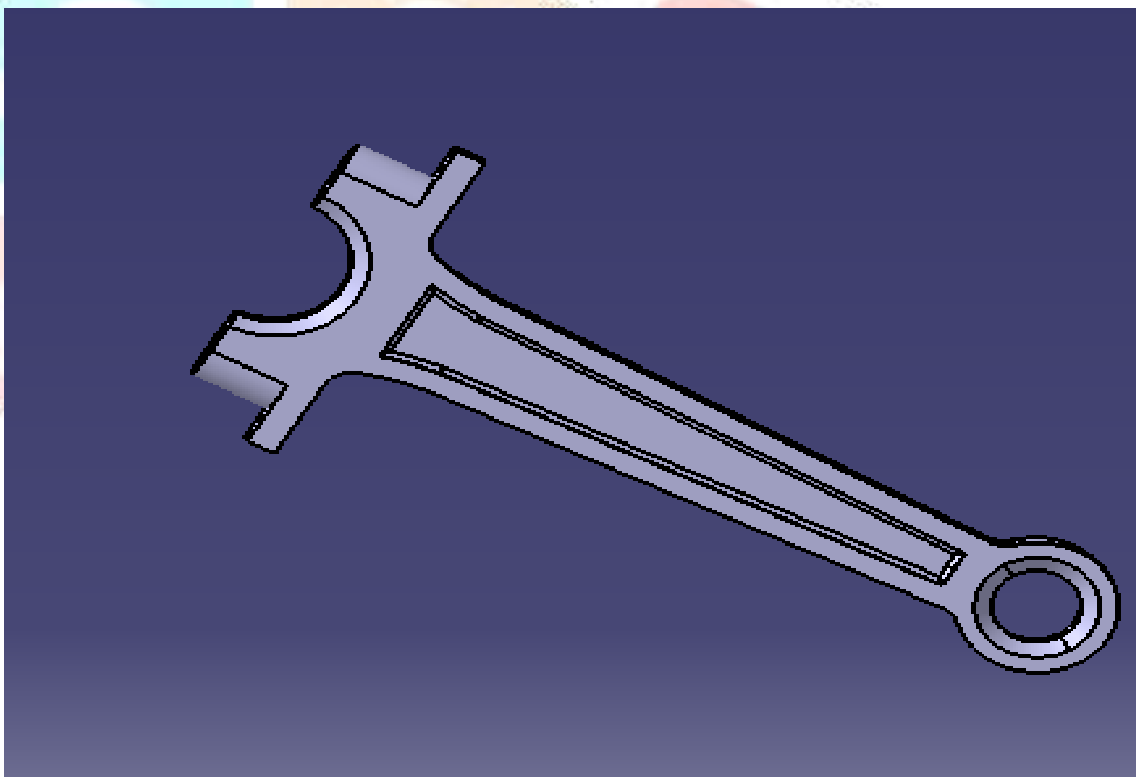
$= 2.25 \times 890.11^2 \times 0.03175[1+ (0.03175/0.127)]$
70749.53 N
$94.26(dcb)^2 = 70749.53$
dcb = 17.32 mm
nominal diameter of bolt db = dcb/0.84 = 28.03/0.84 = 20mm
ab = 600 kgf/cm^2 = 600x10/100 N/mm^2 = 60 N/mm^2
Mc =F1 x X/6
X = diameter of crank pin + 2 x thickness + nominal diameter + clearance bearing liner of a bolt
= dc + 2 x thickness + nominal diameter + clearance bearing liner of a bolt
= 61 + 2 x 3 + 33 + 3
= 103 mm
Maximum bending moment acting on the cap
Mc = Fi x X/6 = 70649.53 x 125 / 6 = 1.21×10^6
Setion modulus for the cap
$Zc = bc(tc)^2/6$
bc = length of the crank pin = 100mm
$= 100 \times (tc)^2/6 = 16.66(tc)^2$
Bending stress (σb) = Mc/ Zc
$60 = 1.47 \times 10^6/16.66(tc)^2$
$(tc)^2 = 1.47 \times 10^6/16.66 \times 60$
tc = 38.34 = 39mm
Mass of the connecting rod per meter length
Ml = volume x density
= area x length x density
$= 11(8.5 \times 10^{-3})^2 \times 0.1128 \times 8000 = 0.414$ kg
Maximum bending moment
$Mmax = Ml.\omega^2.r \times l^2/9\sqrt{3}$
$= 0.717 \times [2\pi \times 8500/60]^2 \times 0.03175 \times (0.1128)^2/9\sqrt{3}$
$= 18036.737 \times 8.1623 \times 10^{-4}$
= 8.5N.m = 8500 N.mm
Section modulus $Zxx = Ixx/(5t/2) = [419t^4/12] \times [2/5t] = 13.97t^3$
$= 13.97\ (8.5)^3$
= 8577.27 mm^3
Maximum bening stress due to inertia of bending forces :
ab(max) = Mmax/Zxx = 14720/8577.27 = 2.215 N/mm^2
Since, the maximum bending stress induced (2.215N/mm^2) is less than the allowable bending sterss of 60N/mm^2, therefore design is safe.
2.215N/mm^2 < 620N/mm^2
Inner diameter of small end (d1) = 51mm
Outer diameter of small end = d1 + 2tb + 2tm
Thickness of bush(tb) = 2 to 5 mm
Marginal thickness (tm) = 5 to 15 mm
= 51 + 2 x 2 + 2 x 5
= 65mm
Inner diameter of big end (d2) = 61mm
Outer diameter of the big end = d2 + 2tb + 2tm + 2db
= 61 +2 x 2 + 2 x 5 + 2 x 2 x 8
= 115mm

COMPONENTS DESIGNED IN THE CATIA

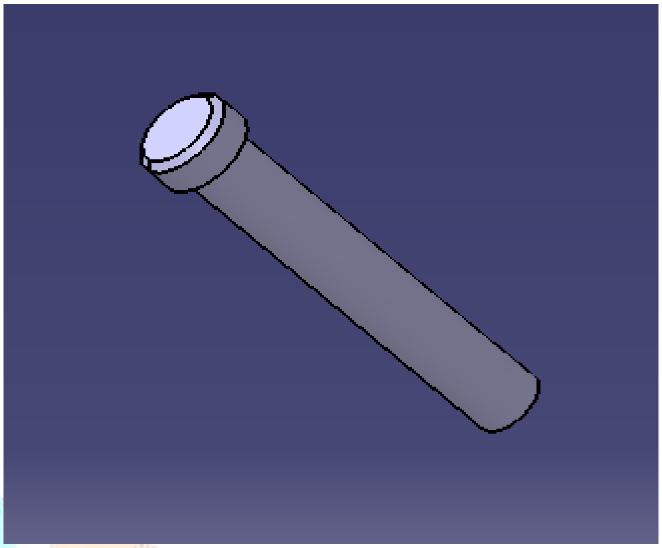
TOP END OFTHE CONNECTING ROD:



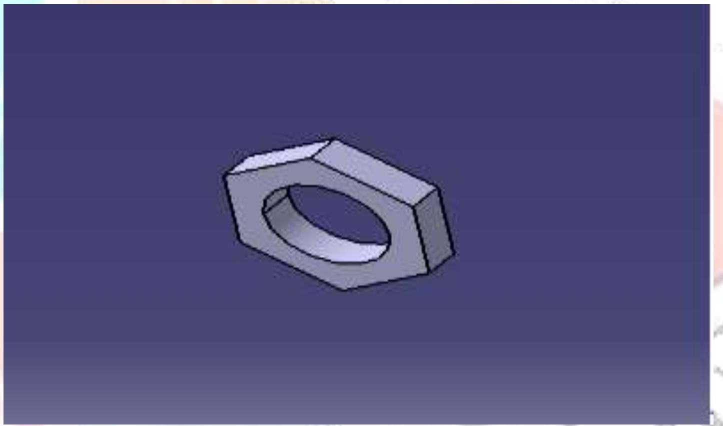
BOTTOM END OF THE CONNECTING ROD:



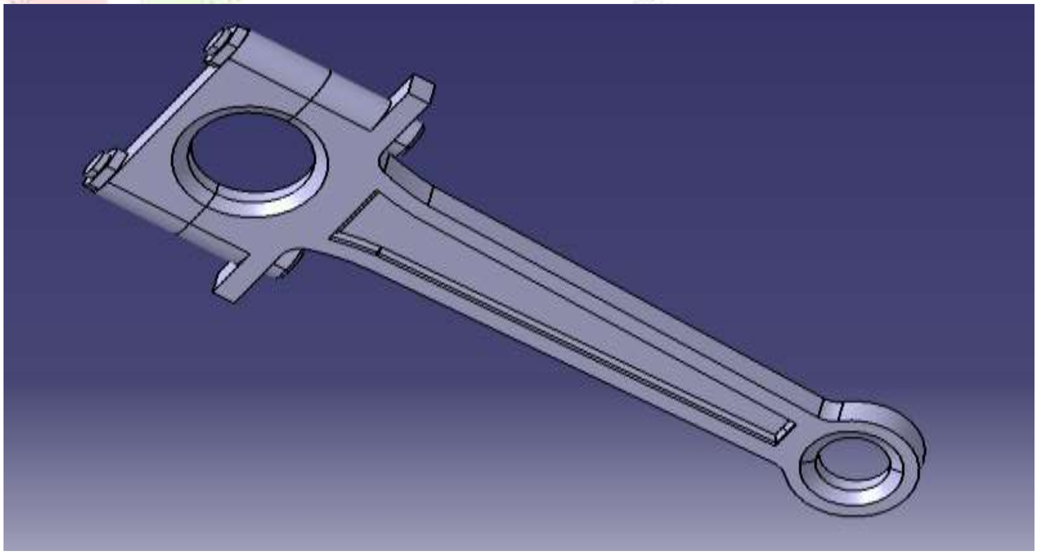
BOLT USED IN THE CONNECTING ROD:



NUT OF THE CONNECTING ROD:

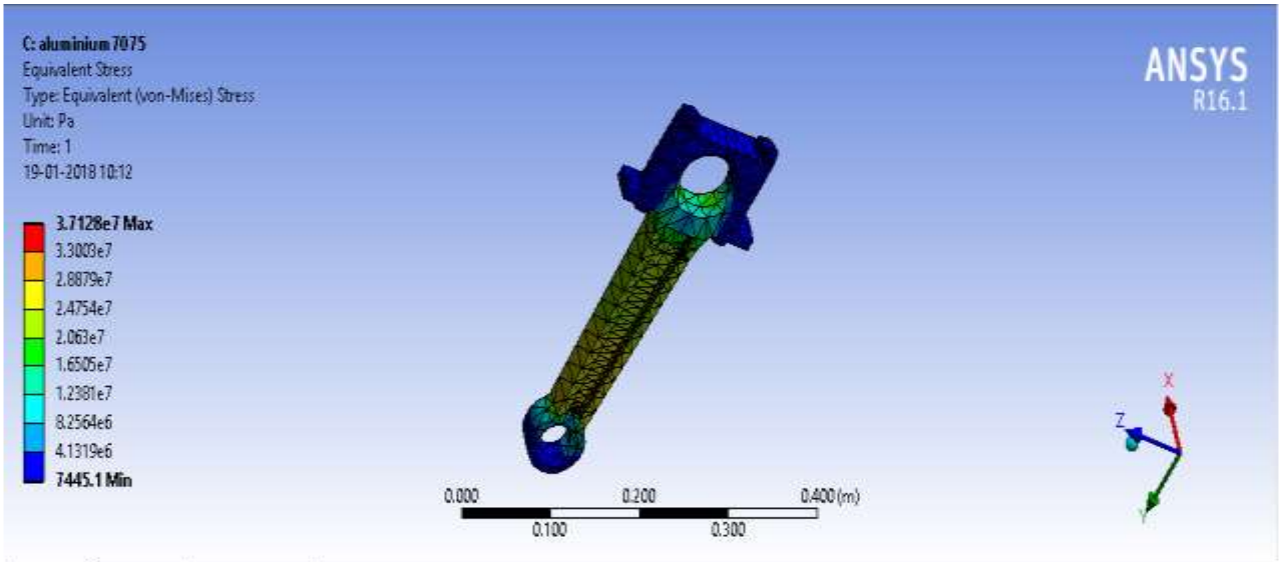


ASSEMBLY OF THE CONNECTING ROD:

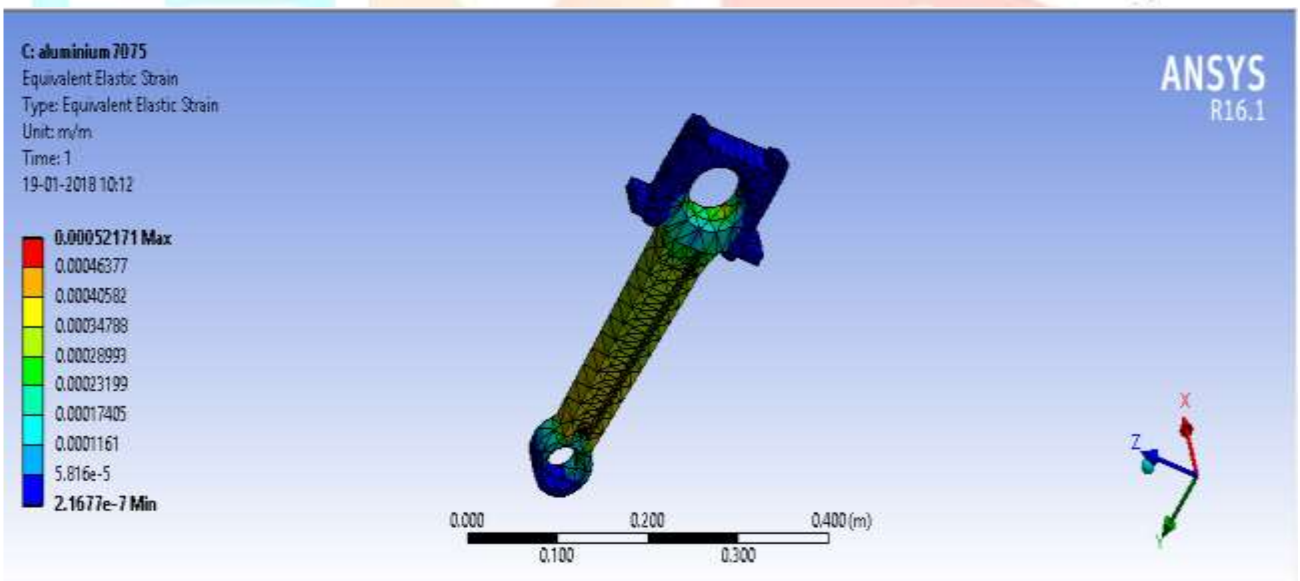


ANALYSIS OF CONNECTING ROD USING ANSYS WORK BENCH SOFTWARE WITH ALUMINIUM 7075:

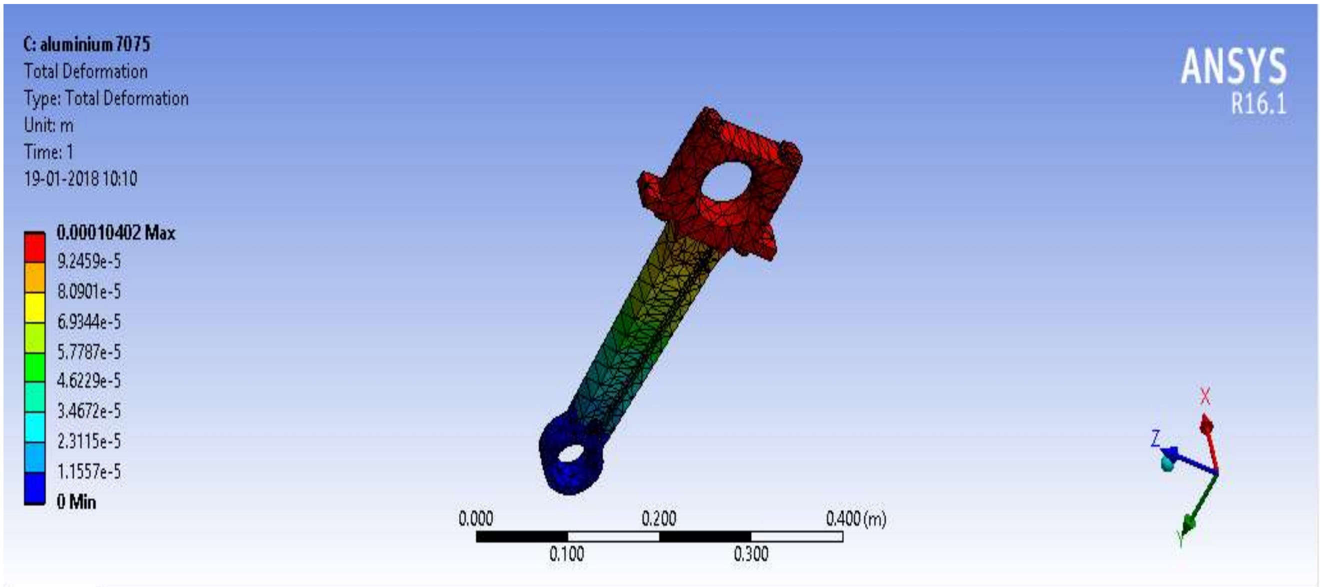
EQUIVALENT STRESS ANALYSIS OF ALUMINIUM 7075:



EQUIVALENT STRAIN ANALYSIS OF ALUMINIUM 7075:

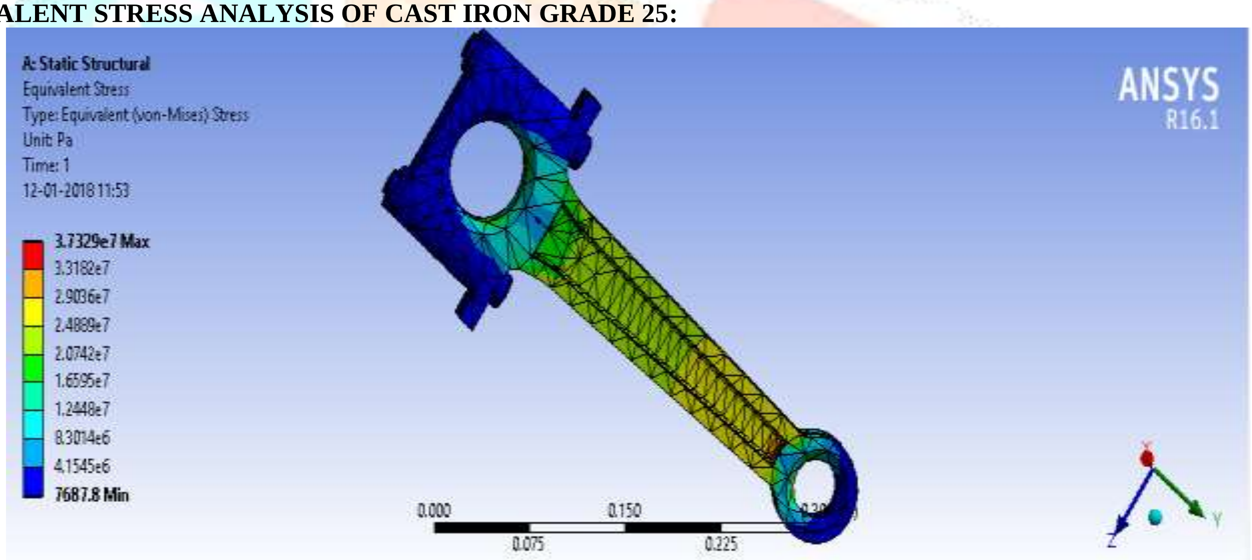


TOTAL DEFORMATION ANALYSIS OF THE ALUMINUM 7075:

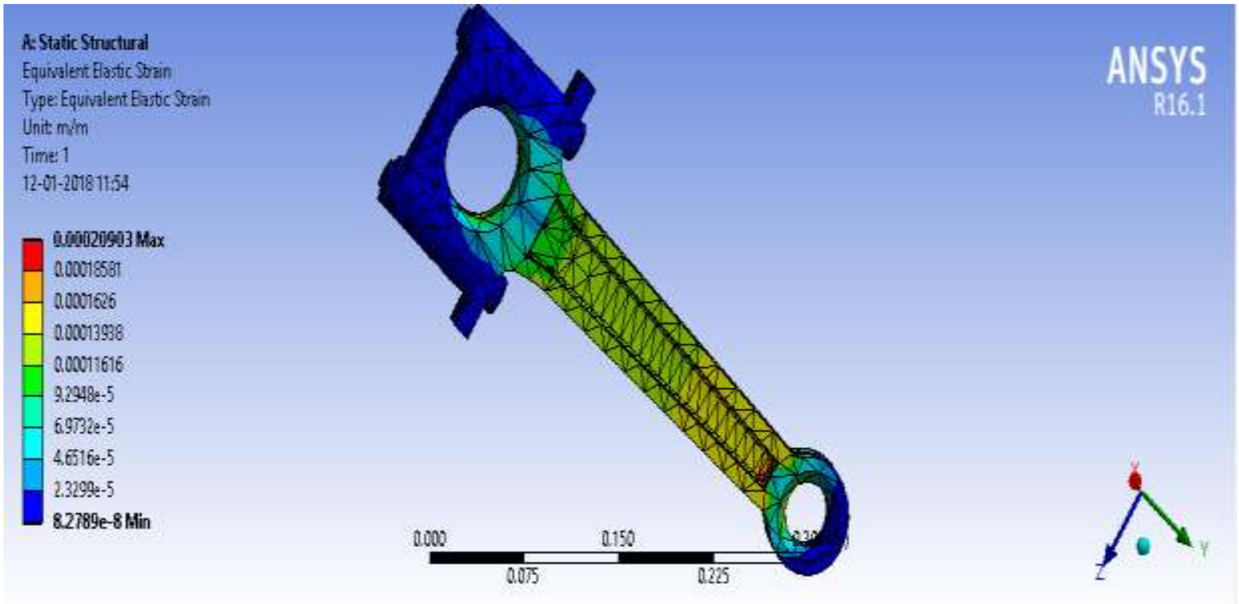


ANALYSIS OF THE CONNECTING ROD USING ANSYS WORKBENCH SOFTWARE WITH CAST IRON GRADE 25:

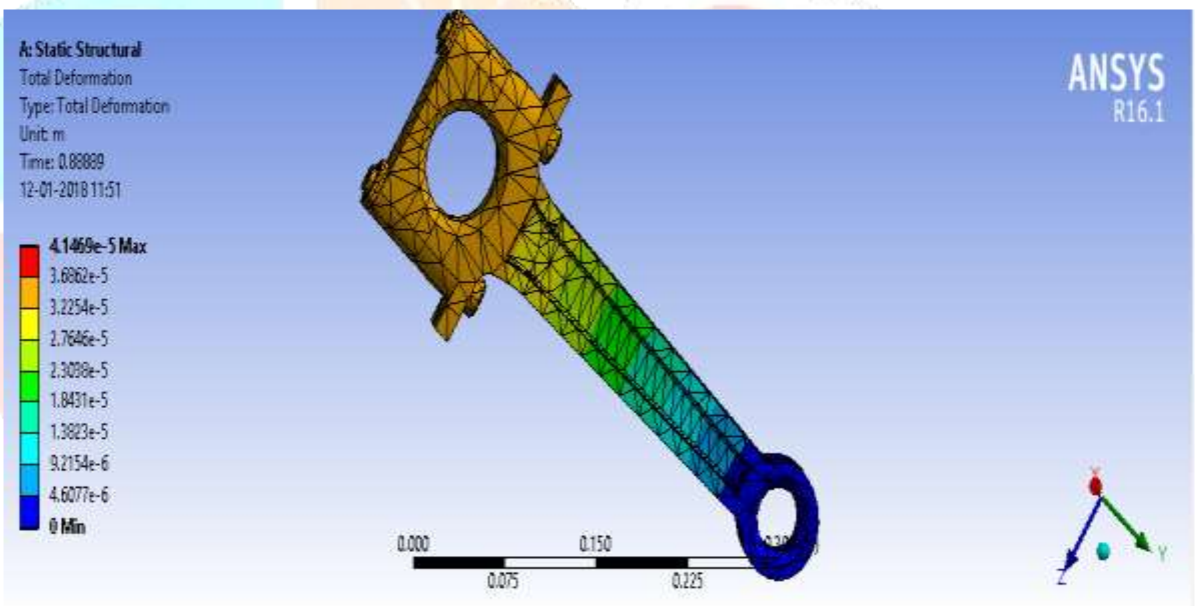
EQUIVALENT STRESS ANALYSIS OF CAST IRON GRADE 25:



EQUIVALENT STRAIN ANALYSIS OF THE CAST IRON GRADE 25:

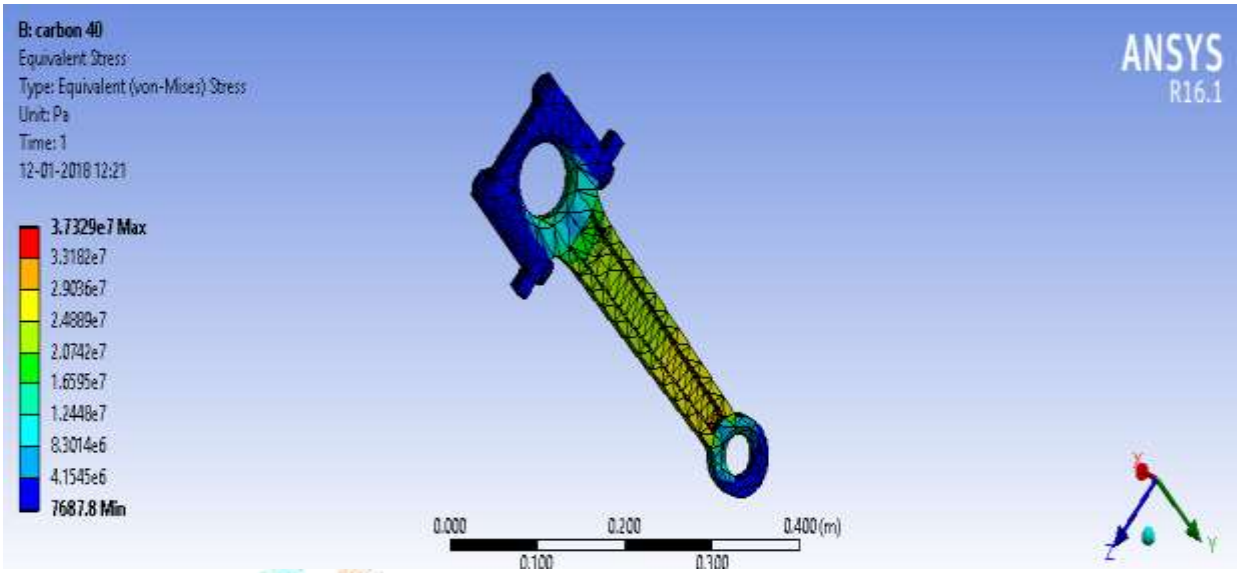


TOTAL DEFORMATION ANALYSIS OF THE CAST IRON GRADE 25:

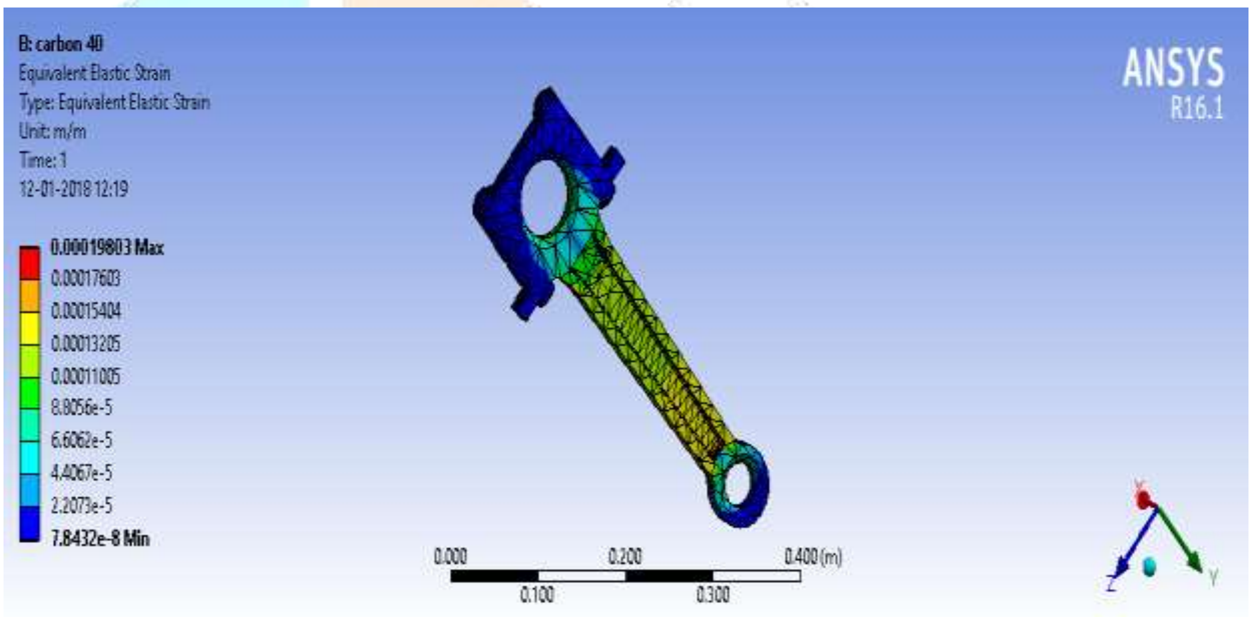


ANALYSIS OF THE CONNECTING ROD USING ANSYS WORKBENCH SOFTWARE USING CARBON40:

EQUIVALENT STRESS ANALYSIS OF THE CARBON40:

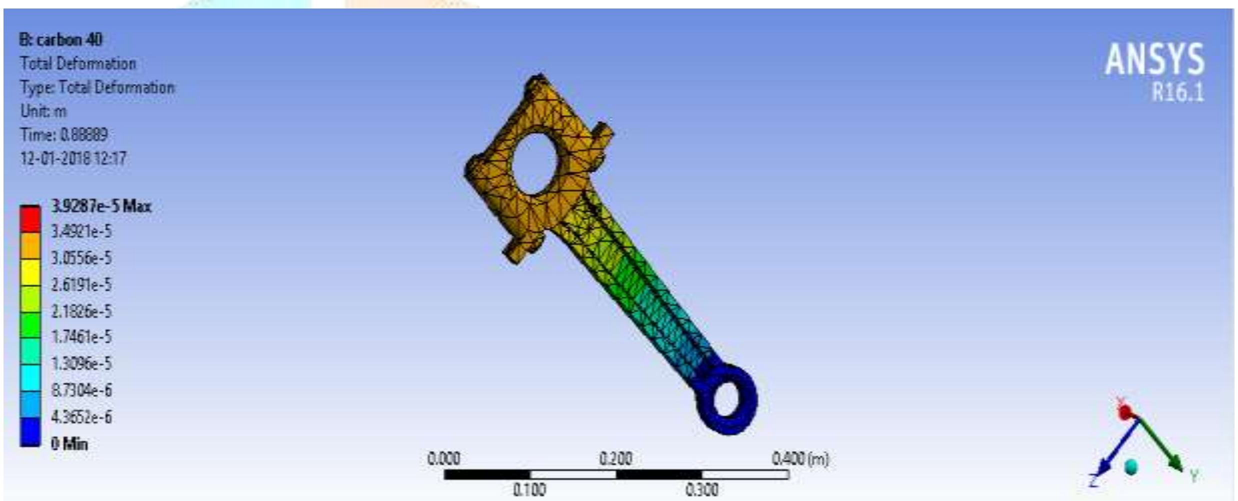


EQUIVALENT ELASTIC STRAIN OF THE CARBON40:



TOTAL DEFORMATION OF THE CARBON40:

Sl.NO	MATERIAL	STRESS	STRAIN	DEFORMATION	DENSITY	COST/KG
1.	ALUMINIUM 7075	3.7128 * e^7	0.00052171	0.00010402	7800kg/m^3	880
2.	CASTIRON GRADE 25	3.7329 * e^7	0.00020903	0.00004169	6600kg/m^3	730
3.	CARBON 40	3.7389 * e^7	0.00019803	0.00003925	7200kg/m^3	436



IV. RESULT:

V. CONCLUSION:

1. When compared to the aluminium 7075 the stress with holding capacity of the cast iron grade 25 & carbon40 are high.
2. The strain produced in the cast iron grade 25& carbon 40 is low when compared to the aluminium 7075.
3. The deformation produced is less when compared to the aluminium 7075 for the pressure of 15.5 mpa .
4. The weight of the connecting rod is reduced . Since the density of the aluminium 7075 is high and the density is directly proportional to the weight .
5. The cost of the aluminium 7075 is very high .

ADVANTAGE

1 . Stress withholding capacity is increased
2 . Strain acting on the material is decreased
3 . Deformation is reduced
4 . The connecting rod is produced at a moderate cost
5 . The weight of the connecting rod is reduced

VI. REFERENCE

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