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FINITE ELEMENT ANALYSIS OF KNUCKLE JOINT WITH DIFFERENT MATERIALS USING ANSYS

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ABSTRACT
kunckle*joint is used to connect two rods whose axes either coincide or intersect and lie in one plane. The knuckle joint assembly consists of Single eye, Double eye or fork and Knuckle pin. Knuckle joint is utilized to transmit axial tensile force and allows for a certain amount of angular movement between the rods around the pin's axis. Yield strength is the requirement for the selection of material for the rods because they are susceptible to tension. The pin is subjected to shear stress and bending stress. Therefore, strength is criterion for while a double eye forms at the other end. Pin holds both ends together.*
The ends of the rod are in the shape of octagons for grasping purpose. The pin now keeps the two eyeballs together when they are being pulled apart. This paper mainly focuses on Finite element analysis of knuckle joint with different materials using Ansys. Plain carbon steel (30C8), Gray cast iron, stainless steel and Aluminium alloy are used and analyzed for total deformation, Shear stress and Equivalent stress calculation. The main objective and goal of this project is to design and analyze the structural deformations in a Knuckle joint. Modeling of a knuckle joint is done in CATIA V5R21 and analysis is performed in ANSYS (Workbench V2022R2).

Keyword: *Knuckle joint, CATIA V5R21, ANYSIS Workbench V2022R2 etc….*

1. Introduction

A knuckle joint is a mechanical joint that helps to connect two rods that are under the action of tensile loads. In some cases, it is possible to join the rods with knuckle join if they are under compressive load with proper guided supports. In this article, we will discuss how to calculate the design parameters of the knuckle joint by considering the different modes of failure in this joint. It is used to connect two rods in which one is a single eye and the second one is a double eye or fork. The single eye has one hole which is inserted into double eye or fork and the fork has two holes in which knuckle pin will insert. The Knuckle Pin is used to hold and grip the single eye and double eye end together. In the knuckle pin at the bottom, there is a hole for the taper pin and There is one more component name is a collar. The collar has two holes to insert taper pin. The collar hole and knuckle Pinhole are being matched. For gripping purposes wow we bring all the holes in such a way that hole of the knuckle pin coincides with the holes of the collar and The tapper pin is inserted in the collar hole through the hole of the knuckle pin.

1.1 Methods of Failure in Knuckle Joint:

1. Failure of the knuckle pin in shear
2. Failure of the single eye or rod end in tension
3. Failure of the single eye or rod end in shearing
4. Failure of the single eye or rod end in crushing
5. Failure of the forked end in tension
6. Failure of the forked end in shear
7. Failure of the forked end in crushing

1.2 Knuckle Joint Application

It is Used to join the coaches of the train easily. It has application in the valve mechanism of a reciprocating engine. It also connects the connecting rod between wheels of locomotives. Used to the joint between the tie rod joint of a roof truss and bridge structure. Another application is Tie rod joint of the jib crane. Knuckle joint also used in Link of roller chain, bicycle chain, and chain straps of watches and so on. This joint is mainly used in the mechanical and automobile sectors whose axis lies on the same line (Joint supports skeleton for wheel assembly.

1.3 Knuckle Joint Application

It is Used to join the coaches of the train easily. It has application in the valve mechanism of a reciprocating engine. It also connects the connecting rod between wheels of locomotives. Used to the joint between once the knuckle joint is made it has long tool life. The joint has good mechanical rigidity (has very little possibility to bend). It can withstand a huge tensile load. It reduces impact shock and has high system rigidity. e manufacturing process of the knuckle joint is simple and the setup process is also simple.

1.4 Knuckle Joint Disadvantages

The angular movement can be done in only one plane or axis. It cannot withstand large compressive loads which is major drawback. When Compared to universal joint it has less flexibility

2. Problem

It is required to design a knuckle joint to connect two circular rods subjected to an axial tensile force of 35 KN. The rods are co-axial and a small amount of angular movement between their axis is permissible. Design the knuckle joint and specify the dimensions of its components. The material for the joint is 30c8 (Syt = 400 N / mm²). Assume F.O.S as 4.

Solution:

Given, P = 35 KN = 35000 N

$\sigma t = Syt / FOS = 400 / 4 = 100$ N/mm²

$\sigma c = Syt / FOS = 400 / 4 = 100$ N/mm²

$\tau = (0.5\ Syt) / FOS = (0.5 \times 400) / 4 = 50$ N/mm²

The joint is designed by considering the various methods of failure as discussed below

1. Failure of the solid rod in tension

Let d = Diameter of the rod.

We know that the load transmitted (P),

$P = (\pi / 4) \times d^2 \times \sigma t$

$35000 = (\pi / 4) \times (d^2) \times 80$

$d^2 = 35000 / 62.83$

$d^2 = 557.05$

d = 23.61 mm say 25 mm.

∴ d = Diameter of the rod = 25 mm.

Now the various dimensions are fixed as follows:

Diameter of knuckle pin,

$d_1 = d = 25$ mm

Outer diameter of eye,

$d_2 = 2d = 2 \times 25 = 50$ mm

Diameter of knuckle pin head and collar,

$d_3 = 1.5\ d = 1.5 \times 25 = 37.5$ mm

Thickness of single eye or rod end,

$t = 1.25\ d = 1.25 \times 25 = 31.25$ mm

Thickness of fork,

$t_1 = 0.75\ d = 0.75 \times 25 = 18.75$ mm

Thickness of pin head,

$t_2 = 0.5\ d = 0.5 \times 25 = 12.5$ mm

2. Failure of the knuckle pin in shear

Since the knuckle pin is in double shear, therefore load (P),

$35000 = 2 (\pi / 4) \times (d_1)^2 \tau$

$35000 = 2 (\pi / 4) \times (25)^2 \tau$

$\tau = 35000 / 1413.7167$

∴ $\tau = 35.6$ Mpa

∴ τ (50 MPa) > 35.6 Mpa. Induced shear stress is less than permissible shear stress.

3. Failure of the single eye or rod end in tension

The single eye or rod end may fail in tension due to the load. We know that load (P),

$35000 = (d_2 - d_1)\ t \times \sigma t$

$35000 = (50 - 25)\ 31.25 \times \sigma t$

$\sigma t = 35000 / 781.25$

∴ $\sigma t = 44.8$ Mpa

∴ σt (100 MPa) > 44.8 Mpa. Induced tensile stress is less than permissible tensile stress.

4. Failure of the single eye or rod end in shearing

The single eye or rod end may fail in shearing due to the load. We know that load (P),

$P = (d_2 – d_1)\ t \times \tau$

$35000 = (50-25)\ 31.25 \times \tau$

$\tau = 35000 / 781.25$

$\therefore \tau = 44.8$ Mpa

∴ τ (50 MPa) > 44.8 Mpa. Induced shear stress is less than permissible shear stress.

5. Failure of the single eye or rod end in crushing

The single eye or rod end may fail in crushing due to the load. We know that load (P),

$P = d_1 \times t \times \sigma c$

$35000 = 25 \times 31.25 \times \sigma c$

$\sigma c = 35000 / 781.25$

$\therefore \sigma c = 44.8$ Mpa

∴ σc (100 MPa) > 44.8 Mpa. Induced compressive stress is less than permissible compressive stress.

6. Failure of the forked end in tension

The forked end may fail in tension due to the load. We know that load (P),

$P = (d2 – d1)\ 2\ t_1 \times \sigma t$

$35000 = (50 – 25)\ 2 \times 18.75 \times \sigma t$

$\sigma t = 35000 / 937.5$

$\therefore \sigma t = 37.33$ Mpa

∴ σt (100 MPa) > 37.33 Mpa. Induced tensile stress is less than permissible tensile stress.

7. Failure of the forked end in shear

The forked end may fail in shearing due to the load. We know that load (P),

$P = (d_2 – d_1)\ 2\ t_1 \times \tau$

$35000= (50 – 25)\ 2 \times 18.75 \times \tau$

$\tau = 50000 / 937.5$

$\therefore \tau = 37.33$ Mpa

∴ τ (50 MPa) > 37.33 Mpa. Induced shear stress is less than permissible shear stress.

8. Failure of the forked end in crushing

The forked end may fail in crushing due to the load. We know that load (P),

$P = d_1 \times 2\ t_1 \times \sigma c$

$35000 = 25 \times 2 \times 18.75\times \sigma c$

$\sigma c = 35000 / 937.58$

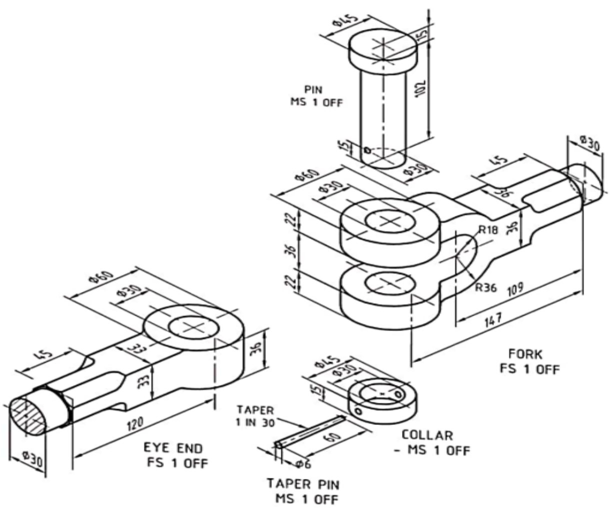
$\therefore \sigma c = 37.33$ Mpa

∴ σc (100 MPa) > 37.33 Mpa. Induced compressive stress is less than permissible compressive stress.

From above, we see that the induced stresses are less than the given design stresses, therefore the above calculated dimensions are safe and good to go. Considering all the possible modes of failure in the knuckle joint, we have calculated the induced stress and seen that they are within the specified permissible stress values in the given problem statement.

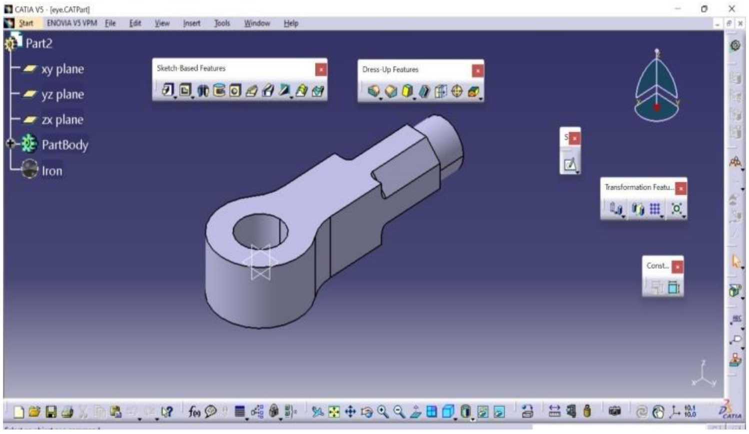
3. Geometric modelling of knuckle joint

Drawing a solid model is the initial step in any machine assembly analysis using a FEM programme. Utilizing 3D software, the knuckle joint is modelled. In this case, CATIA V5 will be used for modelling. Modeling and drafting of all parts are done in accordance with the dimensions determined from the knuckle design.



KNUCKLE JOINT

The French company Assault Systems created the multi-platform CAD/CAM/CAE commercial software suite known as CATIA (Computer Aided Three-dimensional Interactive Application). CATIA, a piece of product lifecycle management software by Assault Systems, was created in the C++ programming language.



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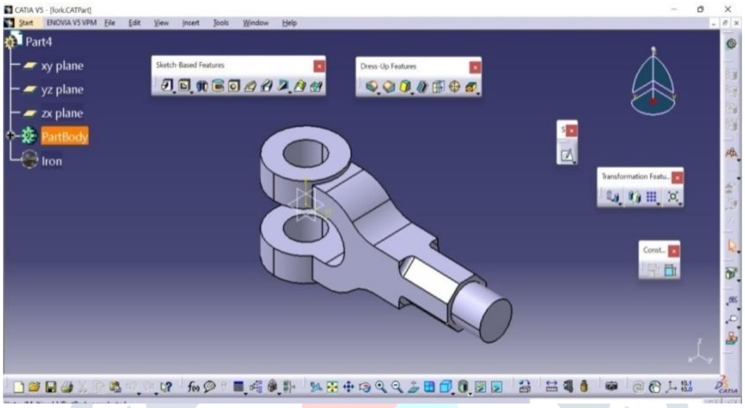
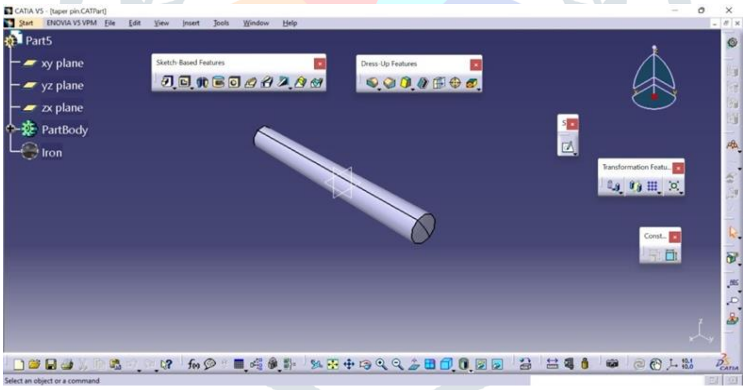
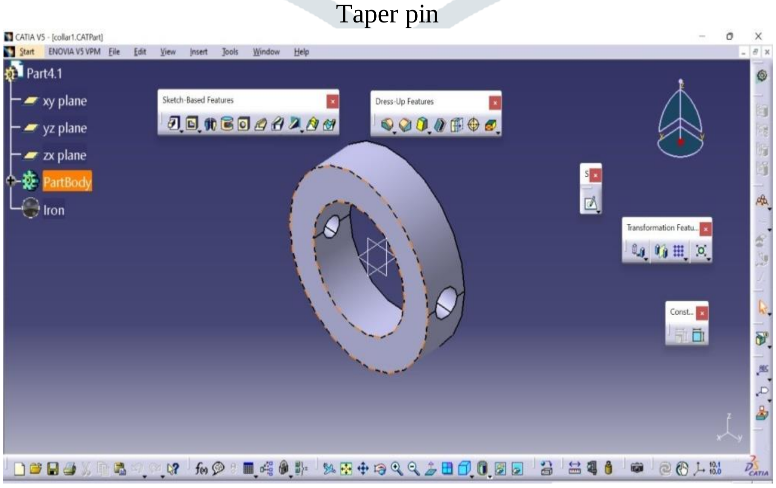


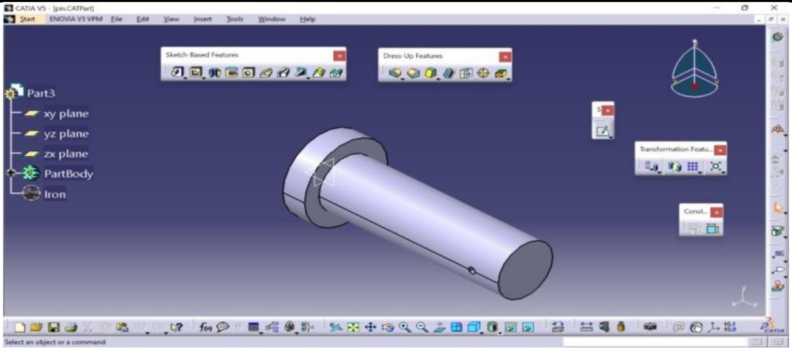
fork



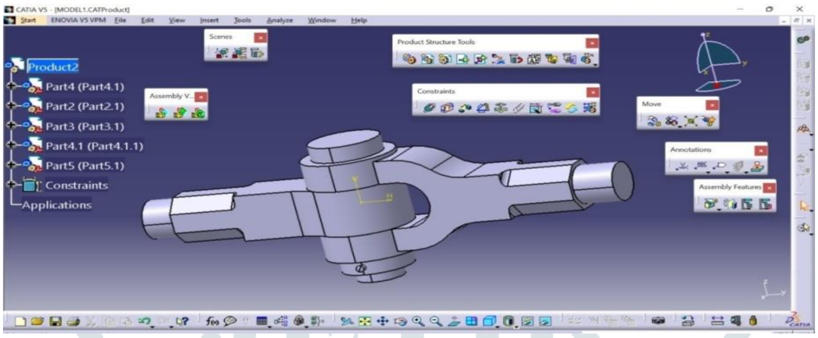
Taper pin



Collar



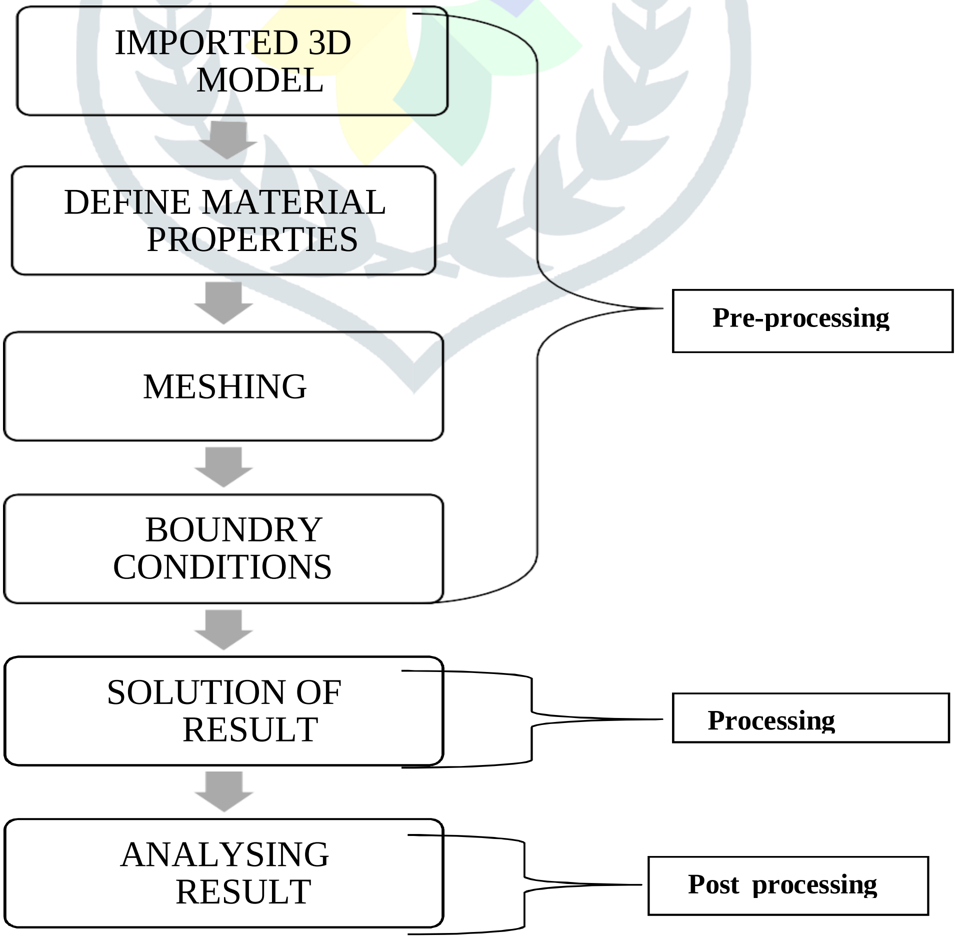
Pin



Knuckle Joint Assembly

4. ANALYSIS

Finite element method is a numerical method for solving problem of engineering and physics. It is also referred to as finite element analysis (FEA). FEA is a numerical method that offer a means to find approximation solution. FEA applied in engineering is computational tool for performing engineering. During working condition knuckle pin is subjected to high stress. As pin is flexible element which can be easily replaced. we can take pin for analyzing purpose. we using ANSYS software for analyzing knuckle pin.

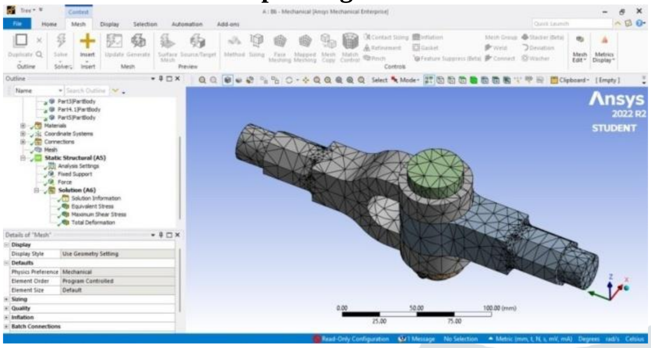


Flow chart in FEA

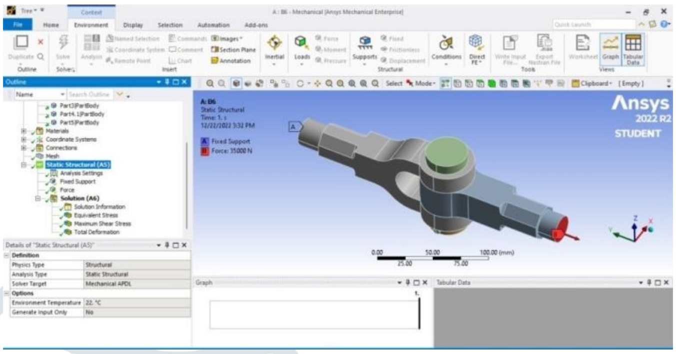
Steps in FEA:

1. **Preprocessing:** first step in preprocessing is to import or create model and define material properties then mesh generation is done. Then boundary conditions areapplied.
2. **Processing:** This process is done by software
3. **Post Processing:** In this step view the results of the solution. The result can be viewed in various formats: graph, value, animation etc.

3.4.2 Fama-Mcbeth two pass regression



Boundary conditions



statistics of mesh

Nodes	Elements
16701	8879

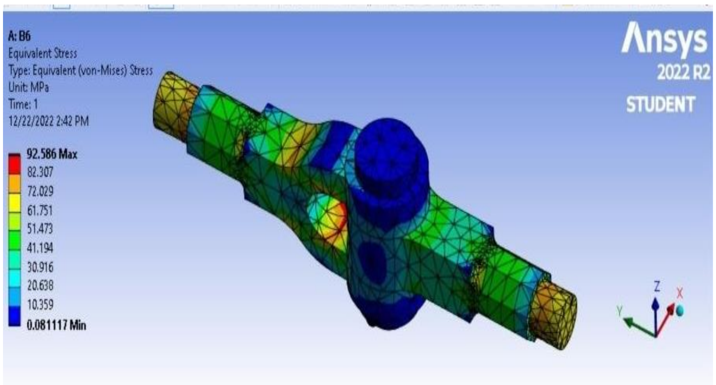
Material properties

The material properties and some common input data used for 30C8 are mentioned in Table 5.

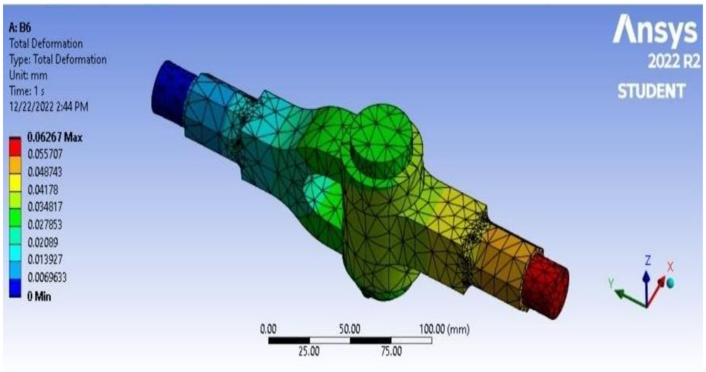
Mechanical properties	Plain carbon steel 30C8	Stainless Steel	Aluminium alloy	Gray CastIron
Density (Kg/m^3)	7800	7850	2770	7200
Compressive yield strength (MPa)	400	2.07e+008	2.8e+08	190e-6
Tensile yield strength (MPa)	400	2.07e+008	2.8e+08	190e-6
Young's modulus (Pa)	2e+11	1.93e+011	7.1e+10	1e+006
Poisson ratio	0.3	0.31	0.33	0.23
Bulk modulus (Pa)	1.6e+11	1.693e+011	6.9608e+10	6.1728e+05
Shear modulus (Pa)	7.69e+11	7.366e+010	2.6692e+10	4.065e+005

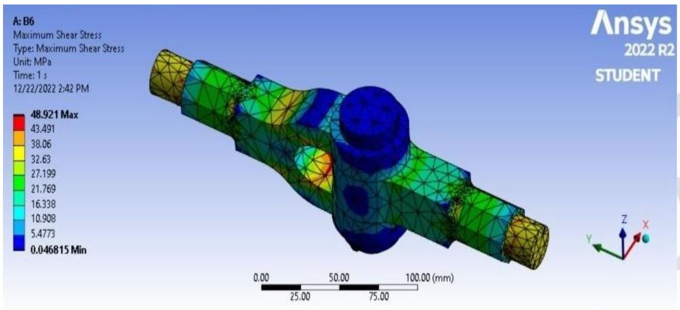
Analysis results of different materials

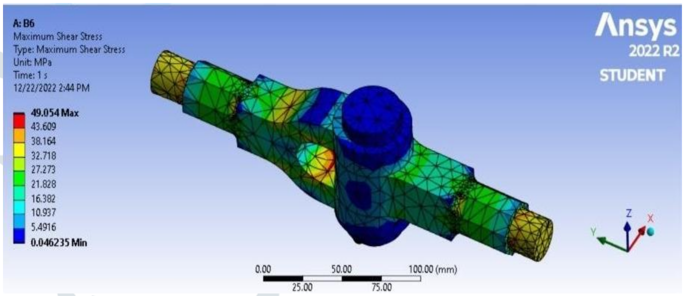
StainlessSteel



PLAINCARBONSTEEL(30C8):





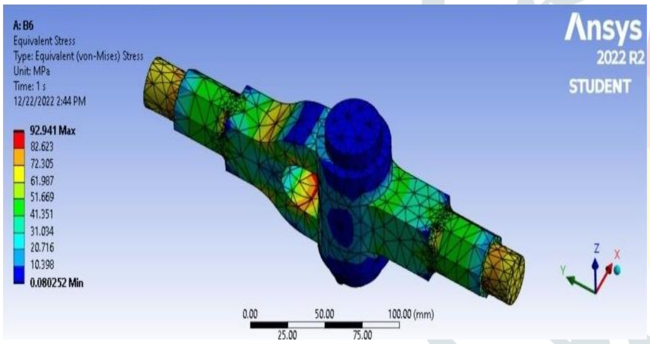


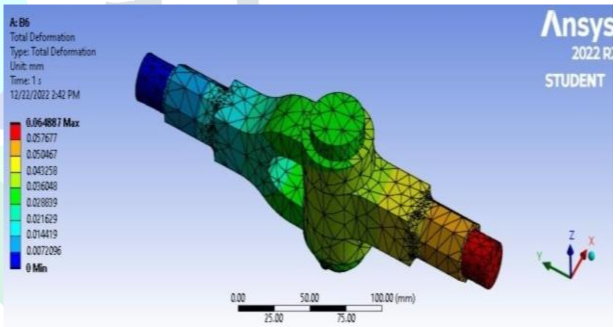
.EquivalentStressofStainlessSteel

totalelongationofplaincarbonsteel(30c8)

MaximumShearStressofStainlessSteel

EquivalentStressofPLAINCARBONSTEEL(30C8)

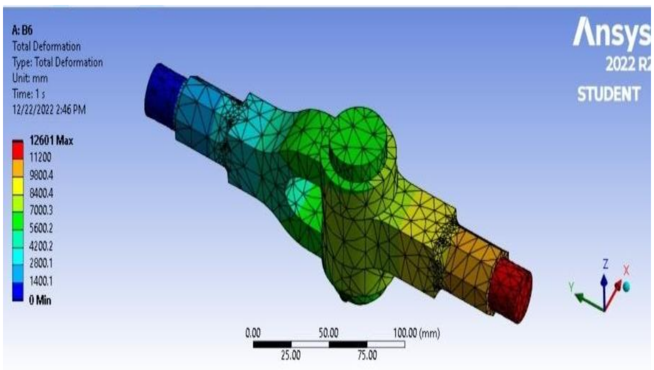




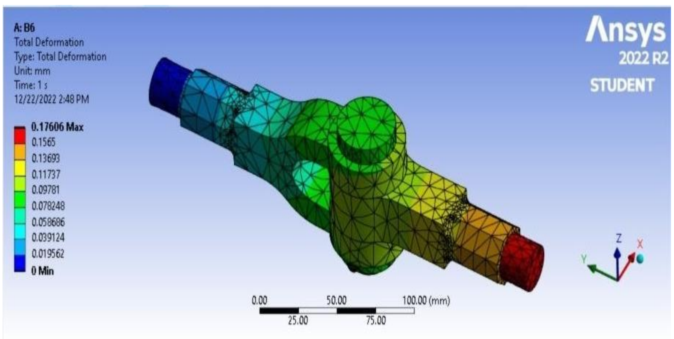
TotalElongationofStainlessSteel

maximumshearstressofplaincarbonsteel(30c8)

GRAYCASTIRON

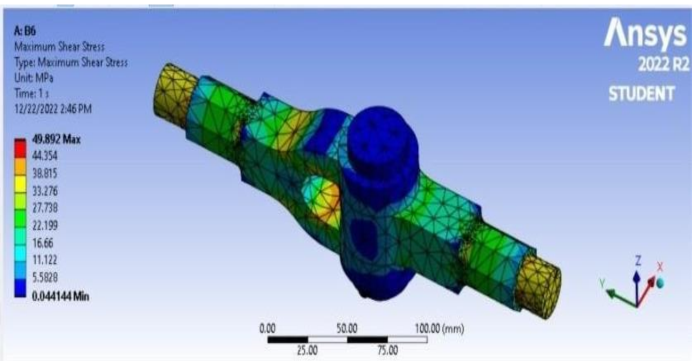


AluminiumAlloy

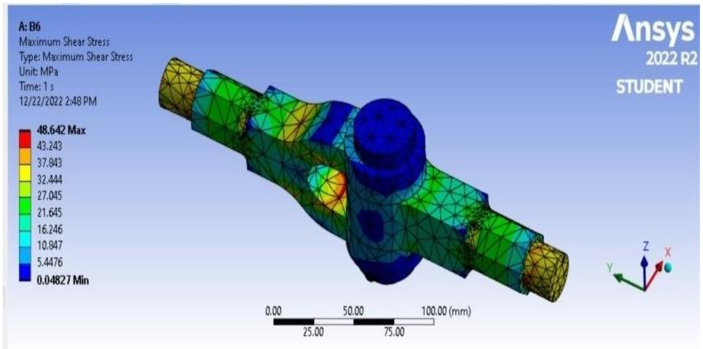


TotalElongationofGRAYCASTIRON

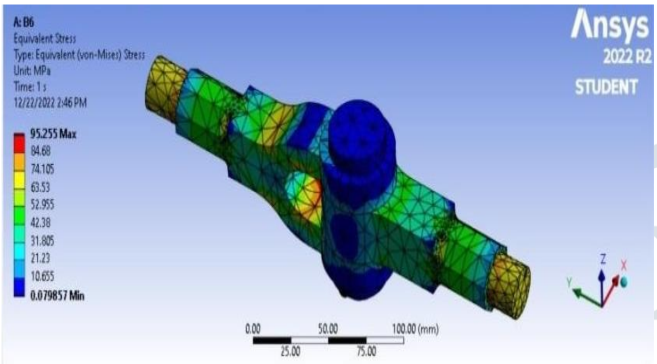
TotalElongationofAluminiumAlloy



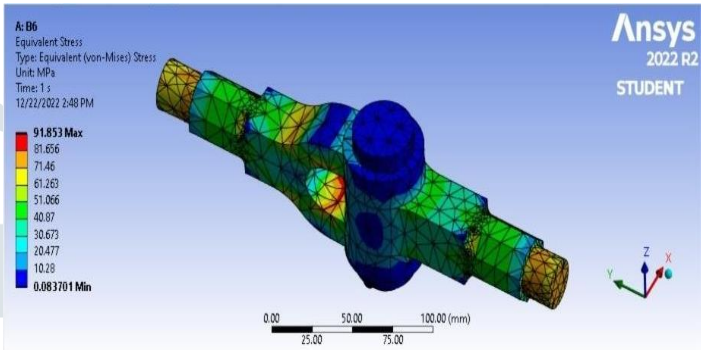
.MaximumShear StressofGRAYCASTIRON



Maximum ShearStress ofAluminiumAlloy



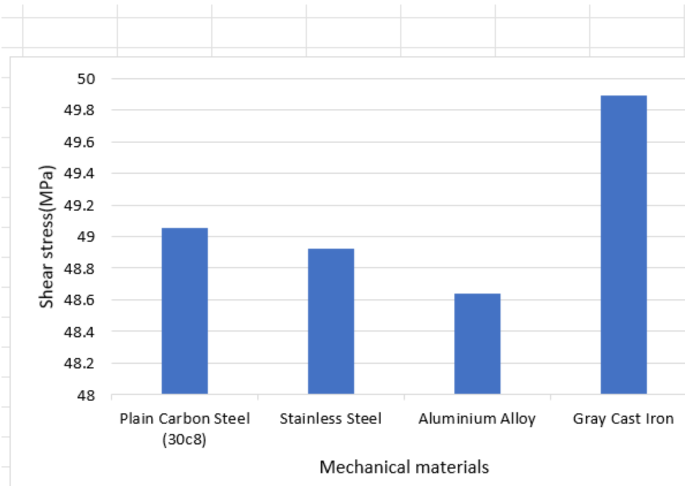
EquivalentStressofGRAYCASTIRON



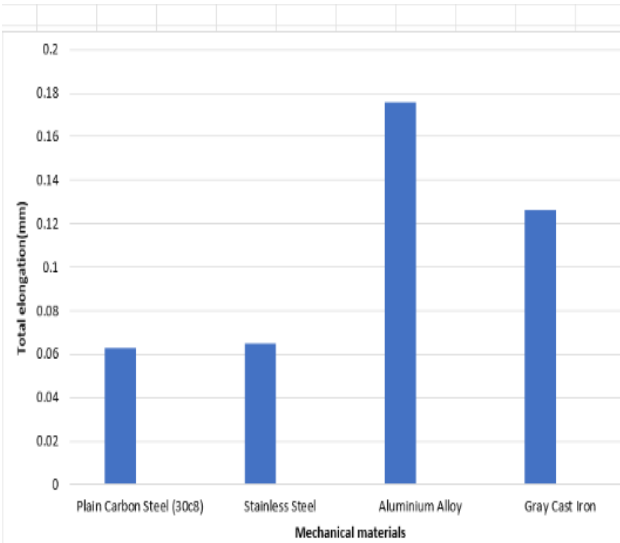
EquivalentStressofAluminiumAlloy

5.1 RESULT

MechanicalParameters	Plaincarbon steel(30C8)	StainlessSteel	Aluminiumalloy	Gray CastIron
Totalelongation(mm)	0.0626	0.06488	0.17606	0.126
Shearstress(MPa)	49.054	48.921	48.642	49.892
Equivalentstress(MPa)	92.941	92.586	91.853	95.255



Shear stress in different materials

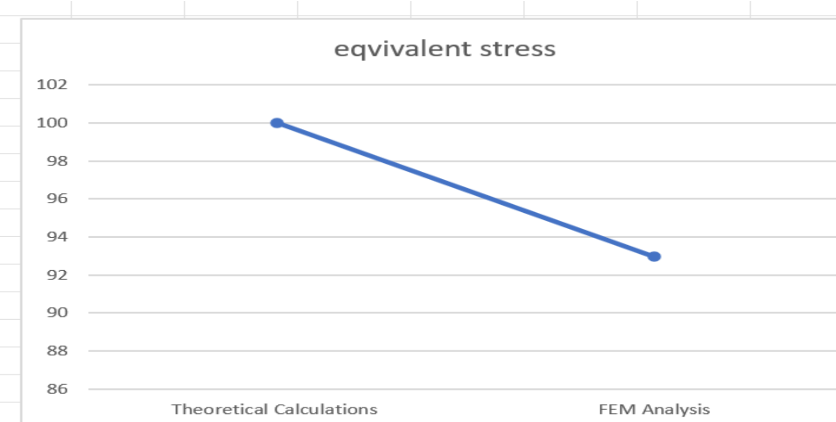


Total elongation in different material

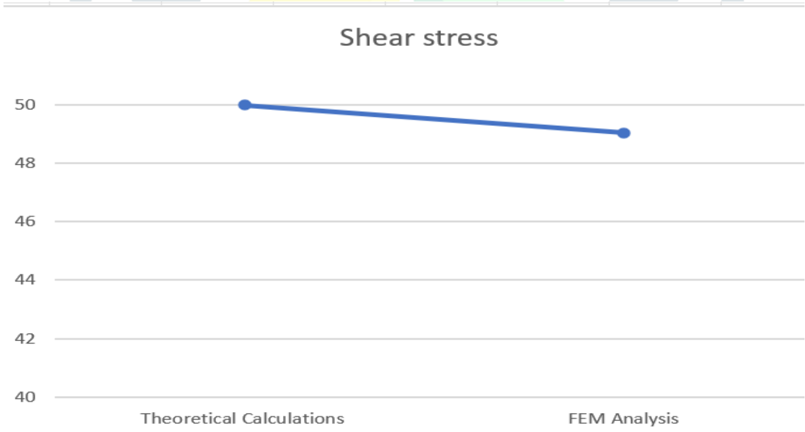
COMPARISON

Table5.2Analysis ofResults

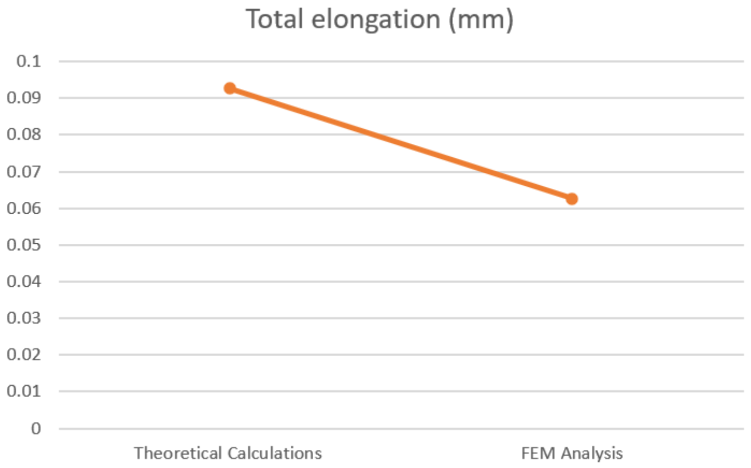
Element	Theoretical calculations	FEAanalysis
Totalelongation	0.0926	0.0626
Shearstress	50	49.054
Equivalentstress	100	92.941



Equivalennt stress



Shear stress



Total elongation

CONCLUSIONS

Static structural analysis and shape optimization of knuckle joint has been performed using ANSYS 2022R2 and CATIA V5R21 and the results have been compared for different materials. By applying the tensile load on one end of the knuckle joint of 35MPa magnitude and constraining the knuckle pin. The value of maximum stresses found to be 95.255 MPa in grey cast iron and the minimum valueof stresses is found to be 91.853 MPa in Aluminium alloy.

The maximum value of deformation is found to be 0.17606 mm in Aluminium alloy and the minimum value of deformation is found to be 0.06267 mm in Plain carbon steel (30C8). The maximum value of Shear stress is found to be 49.892 MPa in grey cast iron and the minimum value of Shear stress is found to be 48.642 MPa in Aluminium alloy.

Shape optimization gives the optimal masses of different material used in analysis. With the help of shape optimization light weight yet rigid design for knuckle joint is developed which helps in reducing the mass of knuckle joint. This work is a design byanalysis study for a knuckle joint. It does not prove that, this is the way that the structural element of knuckle joint have to be designed. But it shows actually there areefficient design tools which help the designer. The procedures like kinematic simulation, design by analysis and shape optimization using FEM drive the designerin an integrated CAE environment.

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