INTERNATIONAL JOURNAL OF RESEARCH IN MECHANICAL, MECHATRONICS AND AUTOMOBILE ENGINEERING(IJRMMAE)
ISSN: 2454-1435 (Print) | 2454-1443 (online)
Volume 4 Issue 3, October – Dec. 2018 - www.ijrmmae.in – Pages 84-96

DESIGN AND ANALYSIS OF KNUCKLE JOINT USING FEA SOFTWARE

S.Ramamoorthy, K.Sakthivel, K.Sivasubramaniyan, D.Manikandan
Assistant Professor
Department Of Mechanical Engineering, A.R Engineering College,Villupuram
Email:Ramamoorthy2610@gmail.com

ABSTRACT:

Knuckle joint is a joint between two parts allowing movement in one plane only. It is a kind of hinged joint between two rods, often like a ball and socket joint. Knuckle joints are most common in steering and drive train applications where it needs to move something but also need to allow for offset angles. The main aim of this project work is to study and calculate the maximum stresses induced in Knuckle joint, The load carrying capacity is considerably very low in mild steel pin. So we can also use high strength, high modulus molybdenum pin that can augment the capacity to withstand higher loads. The knuckle joint is proposed to develop in this study for an applied force of 150 KN. The diameter of the pin is planned, expected to be 24 mm. The material of the knuckle joint is considered as molybdenum material, in order to do the stress analysis the knuckle joint is designed using CATIA software and the mesh was developed for the same using ANSYS software. Based on the analytical calculations and ANSYS result, it shows that a pin of 24mm diameter can resist a load of 150 KN.

I) INTRODUCTION:

A knuckle joint is used when two or more rods subjected to tensile and compressive forces are fastened together such that their axes are not in alignment but meet in a point. Knuckle joint is a joint between two parts allowing movement in one plane only. It is a kind of hinged joint between two rods, often like a ball and socket joint. There are many situations where two parts of machines are required to be restrained, for example two rods may be joined coaxially and when these rods are pulled apart they should not separate i.e. should not have relative motion and continue to transmit force. Similarly if a cylindrical part is fitted on another cylinder (the internal surface of one contacting the external surface of the other) then there should be no slip along the circle of contact. Such situations of no slip or no displacements are achieved through placing a third part or two parts at the jointing regions. A knuckle joint is understood to be a hinged joint in which projection in one part enters the recess of the other part and two are held together by passing a pin through coaxial holes in two cylindrical parts should not have relative motion and continue to transmit force.

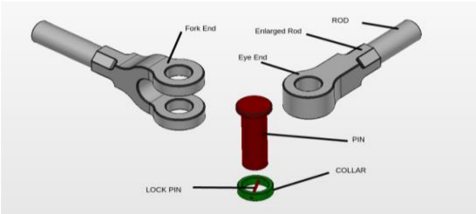

Figure 1: Knuckle Joint Diagram

A knuckle joint is used to connect two rods which are under the action of tensile forces, when a small amount of flexibility or angular moment is necessary. It is basically a tensile joint. However, if the joint is guided, it may support a compressible load. This joint can be readily disconnected for adjustments or repairs.

- The knuckle joint assembly consists of following major Parts:

1. Single eye.
2. Double eye or fork.
3. Knuckle pin.
4. Collar.
5. Tapper pin.

- A knuckle joint may be failed on the following three modes
    1. Shear failure of pin (single shear).
    2. Crushing of pin against rod.
    3. Tensile failure of flat end bar.

II)PROBLEM STATEMENT:

- To select the best suitable material for Knuckle joint which can withstand a tensile force of 150KN
- To analyze the stresses using ANSYS software.

III)DESIGN PROCEDURE:

The dimensions of various parts of the knuckle joint are fixed by empirical relations as given below.

If d is the diameter of rod,

Diameter of pin,	d1	= d
Outer diameter of eye,	d2	= 2d
Diameter of knuckle pin head and collar,	d3	= 1.5d
Thickness of single eye or rod end,	t	= 1.25d
Thickness of fork,	t1	= 0.75d
Thickness of pin head,	t2	= 0.5d

Let,

P= Tensile load acting on the rod,

d= Diameter of the rod,

d1= Diameter of the pin,

d2= Outer diameter of eye,

t= Thickness of single eye,

t1= Thickness of fork.

The rods are subjected to direct tensile load, Therefore tensile strength of the rod,

Assume P=150 KN

$$P = (\pi/4) \times d^2 x\ \sigma_t$$

$$150000 = (\pi/4)\ d^2 x\ 324$$

d=24mm

Diameter of the rod (d) = 24mm
Diameter of pin, d_1 = d= 24mm
Outer diameter of eye, d_2 = 2d= 48mm
Diameter of knuckle pin head and collar, d_3 = 1.5d= 36mm
Thickness of single eye or rod end, t = 1.25d= 30mm
Thickness of fork, t_1 = 0.75d= 18mm
Thickness of pin head, t_2 = 0.5d= 12mm

The selected material is molybdenum ,the following material properties have been considered for the Finite Element analysis.

PROPERTIES	METRIC
Tensile strength (annealed)	324 MPa
Shear strength	500 MPa

INTERNATIONAL JOURNAL OF RESEARCH IN MECHANICAL, MECHATRONICS AND AUTOMOBILE ENGINEERING(IJRMMAE)
ISSN: 2454-1435 (Print) | 2454-1443 (online)
Volume 4 Issue 3, October – Dec. 2018 - www.ijrmmae.in – Pages 84-96

Compressive yield strength	400 MPa
Modulus of elasticity	330 GPa
Poisson's ratio (calculated)	0.38
Yield strength	550 MPa
Density	10.3 g/cm³
Melting point	2625°C

Table 1: Properties Of Molybdenum Material

a)Failure of the knuckle pin in shear:

Since the pin is in double shear, therefore cross-sectional area of the pin under shearing

$= 2 \times (\pi/4) \times (d1)^2$

And the shear strength of the pin,

$= 2 \times (\pi/4) \times (d1)^2 \times s$

Equating this to the load (P) acting on the rod, we have

$P = 2 \times (\pi/4) \times (d1)^2 \times s$

$150000 = 2 \times (\pi/4) \times (24)^2 \times s$

S=165.87MPa

It is less than allowable shear stress so it is safe.

The value of maximum bending moment is given by:

$M = (P/2) [(t1/3) + (t/4)]$

M = 1012.5 KN-mm

And section modulus,

$Z = (\pi/32) \times (d1$

Since maximum bending (tensile) stress,

$\sigma t = M/Z$

d1 = 32 mm

b)Failure of the single eye or rod end in shearing:

The single eye or rod end may fail in shearing due to tensile load. We know that area resisting shearing

$= (d_2 - d_1)\ t$

∴ Shearing strength of single eye or rod end

$= (d_2 - d_1)\ t \times \tau_s$

Equating this to the load (P), we have

$$P = (d_2 - d_1)\, t \times \tau_s$$

$$150000 = (48\text{-}32) \text{ x } 30 \times \tau_s$$

$$\boldsymbol{\tau_s = 312.5 \text{ MPa}}$$

Since, it is less than allowable shear stress so it is safe

c)Failure of the single eye or rod end in crushing:

The single eye or pin may fail in crushing due to the tensile load. We know that area resisting crushing

$$= d_1 \times t$$

∴ Crushing strength of single eye or rod end

$$= d_1 \times t \times \sigma_c$$

Equating this to the load (P), we have

$$P = d_1 \times t \times \sigma_c$$

$$150000 = 24 \text{x} 30 \text{x} \sigma_c$$

$$\boldsymbol{\sigma_c = 208 \text{ MPa}}$$

Since, it is less than allowable crushing stress so it is safe.

d)Failure of the forked end in tension:

The forked end may fail in tension due to the tensile load, We know that area resisting tearing

$$= (d_2 - d_1) \times 2t_1$$

∴ Tearing strength of the forked end

$$= (d_2 - d_1) \times 2t_1 \times \sigma_t$$

Equating this to the load (P), we have

$$P = (d_2 - d_1) \times 2t_1 \times \sigma_t$$

$$150000 = 2(48\text{-}32)18 \text{x} \sigma_t$$

$$\boldsymbol{\sigma_t = 260.4 \text{ MPa}}$$

Since, it is less than allowable tensile stress so it is safe

e)Failure of the forked end in shear:

The forked end may fail in shearing due to the tensile load. We know that area resisting shearing

$$= (d_2 - d_1) \times 2t_1$$

INTERNATIONAL JOURNAL OF RESEARCH IN MECHANICAL, MECHATRONICS AND AUTOMOBILE ENGINEERING(IJRMMAE)
ISSN: 2454-1435 (Print) | 2454-1443 (online)
Volume 4 Issue 3, October – Dec. 2018 - www.ijrmmae.in – Pages 84-96

∴ Shearing strength of the forked end

$$= (d_2 - d_1) \times 2t_1 \times \tau_s$$

Equating this to the load (P), we have

$$P = (d_2 - d_1) \times 2t_1 \times \tau_s$$

$$150000=2(48-24)18x\ \tau_s$$

$$\boldsymbol{\tau_s = 260.4\ \mathrm{MPa}}$$

Since, it is less than allowable shear stress so it is safe

f)Failure of the forked end in crushing:

The forked end or pin may fail in crushing due to the tensile load. We know that area resisting crushing

$$= d_1 \times 2\ t_1$$

∴ Crushing strength of the forked end

$$= d_1 \times 2\ t_1 \times \sigma_c$$

Equating this to the load (P), we have

$$P = d_1 \times 2\ t_1 \times \sigma_c$$

$$150000=24x2x18x\sigma_c$$

$$\boldsymbol{\sigma_c = 173.6\mathrm{MPa}}$$

Since, it is less than allowable crushing stress so it is safe.

IV)DESIGN AND ANALYSIS OF KNUCKLE JOINT:

CATIA is a very powerful 3D modelling software, it is dominating in automotive, aerospace and aeronautics. With CATIA, you can design aircraft, cars and other complex products. This software is aimed for professionals and engineers, especially for large enterprises. CATIA has a great set of tools, which can even be designed just for your company's design needs. In this project the knuckle joint model is developed by using CATIA Software. A typical design of the knuckle joint is shown in Figure 2.

INTERNATIONAL JOURNAL OF RESEARCH IN MECHANICAL, MECHATRONICS AND AUTOMOBILE ENGINEERING(IJRMMAE)
ISSN: 2454-1435 (Print) | 2454-1443 (online)
Volume 4 Issue 3, October – Dec. 2018 - www.ijrmmae.in – Pages 84-96

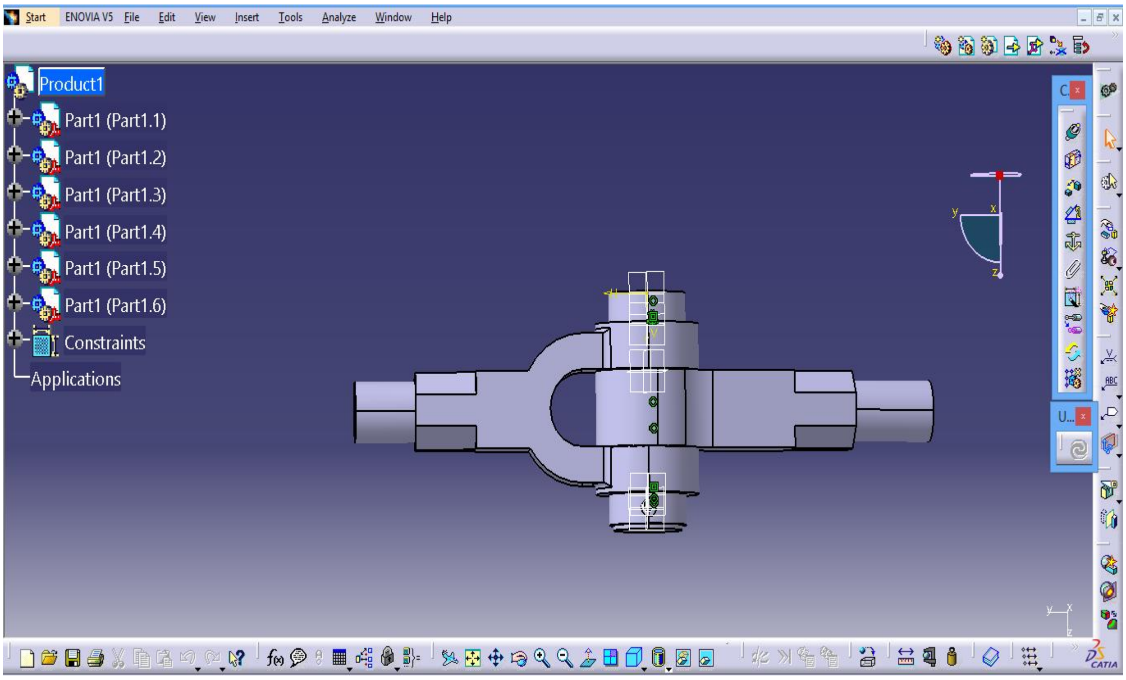

Figure 2:Assembled View Of Knuckle Joint

V)ANALYSIS SOFTWARE:

ANSYS is one of the best analysis and simulation software used to simulate engineering solutions. ANSYS simulates 3d models or structures or machine parts designs to stress, strength, temperature distribution, thermal conductivity, elasticity, fluid flow, air flow etc. Most of the analysis performed in an ANSYS APDL and simulations into ANSYS Workbench, which are one of the main products. It is easier and flexible software used to analysis and simulation of engineering designs and used by millions of engineers all over the world.

ANSYS is one of the best Computer Aided Engineering (CAE) and general-purpose Finite Element Analysis software that is widely used in engineering applications to solve the different engineering problems by using a simulation and analysis of engineering designs. In this project the FEM Analysis is performed on the knuckle joint using ANSYS14.0. A typical drawing of the meshing of theknuckle joint is shown in Figure 3 and the Figure 4 represents the boundary conditions applied to the knuckle joint

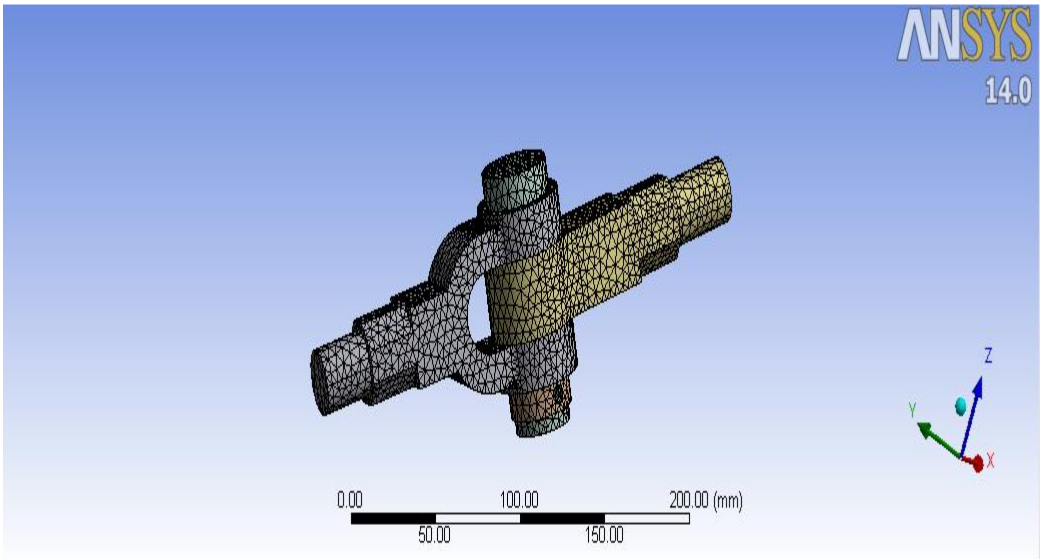

Figure 3: Meshing Of Knuckle Joint

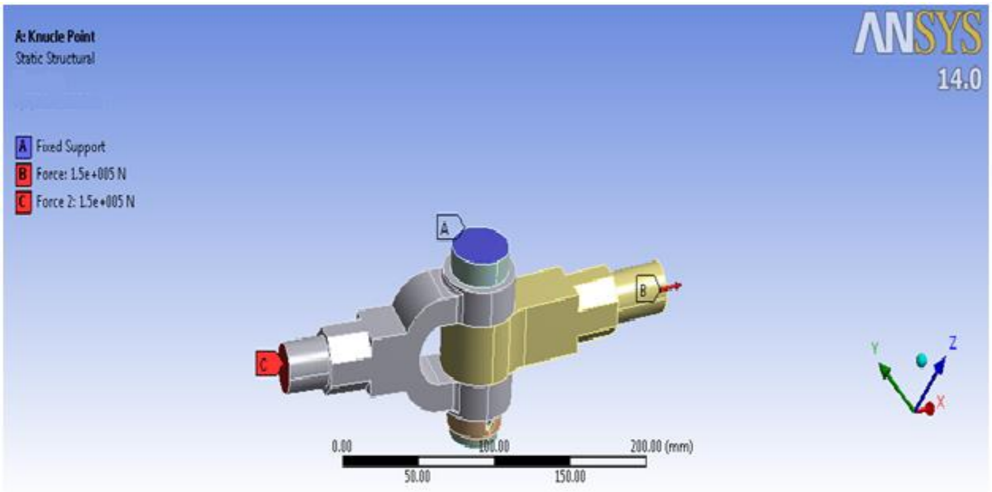

Figure 4: Boundary Conditions applied to knuckle joint

INTERNATIONAL JOURNAL OF RESEARCH IN MECHANICAL, MECHATRONICS AND AUTOMOBILE ENGINEERING(IJRMMAE)
ISSN: 2454-1435 (Print) | 2454-1443 (online)
Volume 4 Issue 3, October – Dec. 2018 - www.ijrmmae.in – Pages 84-96

a) STRESS ANALYSIS OF KNUCKLE JOINT:

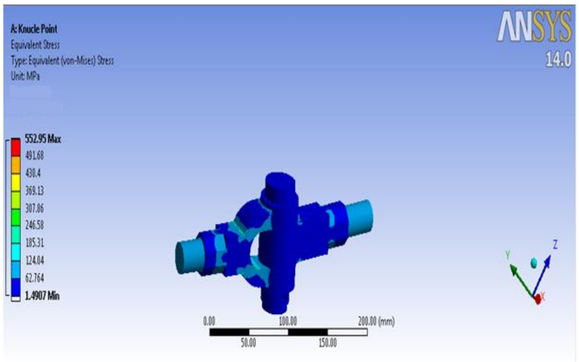

Figure 5 :Von-Misses stress without deformation

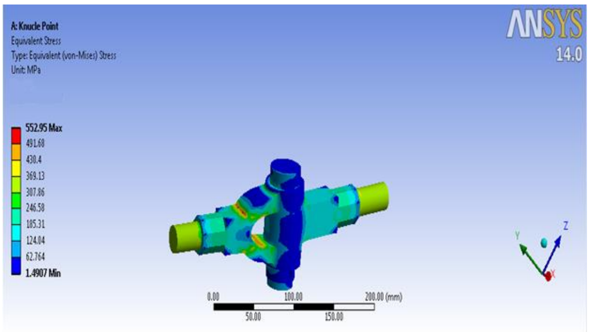

Figure 6: Von-Misses stress with deformation

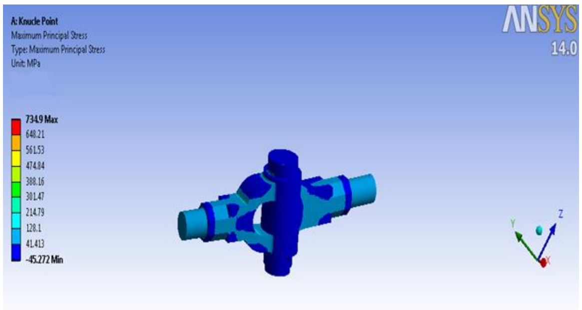

Figure 7: Maximum principal stresses without deformation

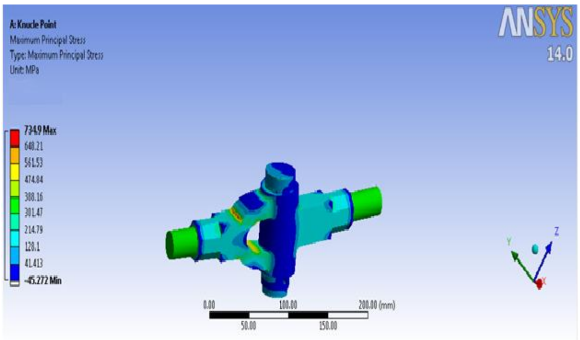

Figure 8: Maximum principal stresses with deformation

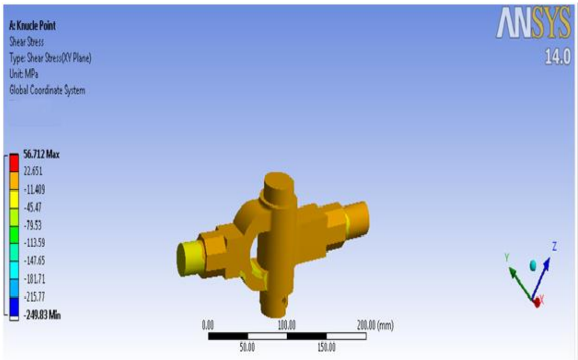

Figure 9: Shear stress without deformation

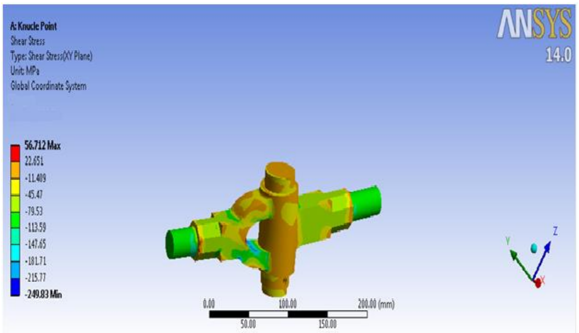

Figure 10: Shear stress with deformation

INTERNATIONAL JOURNAL OF RESEARCH IN MECHANICAL, MECHATRONICS AND AUTOMOBILE ENGINEERING(IJRMMAE)
ISSN: 2454-1435 (Print) | 2454-1443 (online)
Volume 4 Issue 3, October – Dec. 2018 - www.ijrmmae.in – Pages 84-96

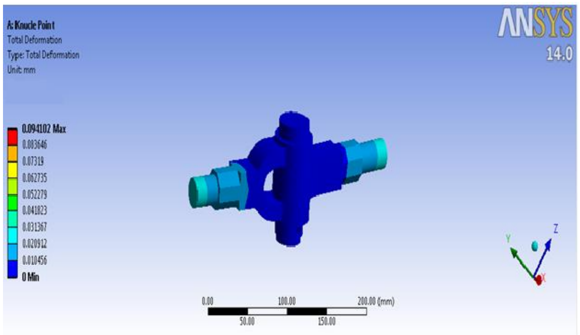

Figure 11: Deflection without deformation

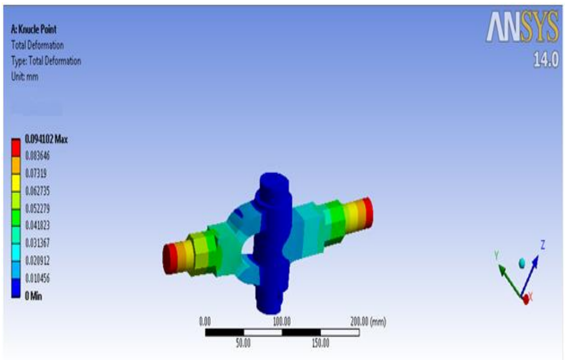

Figure 12: Deflection with deformation

VI) RESULTS AND DISCUSSIONS:

S.NO	Types	Max	Min
1	Shear stress(xy plane)	567.12Mpa	-249.83Mpa
2	Total deformation	0.09410 mm	0.00 mm
3	Maximum principal stress	734.9Mpa	-45.272Mpa
4	Von misses stress	552.95Mpa	1.4907Mpa

Table 2: Stresses And Deformation Results Of Molybdenum

VII)THEORETICAL CALCULATION:

Normal stress in x-x direction:

$$\sigma = \frac{P}{A}$$

P = 150000 N

A = 803.84 mm^2

$$= \frac{150000}{803.84} = 186.604 N/mm^2$$

Maximum principal stress:

τ = 550 x 0.3

τ = 165 N/mm^2

$$\sigma_{max} = \frac{\sigma}{2} + \frac{1}{2}\sqrt{(\sigma)^2 + 4\tau^2}$$

$$= \frac{186.6}{2} + \frac{1}{2}\sqrt{(186.6)^2 + 4 \times 165^2}$$

$$\sigma_{max} = 262.80 \quad {}^{N}/_{mm^2}$$

$$\sigma_{max} = 262.80 \quad {}^{N}/_{mm^2} < \sigma_{yt} = 275 \quad {}^{N}/_{mm^2}$$

Minimum principal stress:

$$\sigma_{min} = \frac{\sigma}{2} - \frac{1}{2}\sqrt{(\sigma)^2 + 4\tau^2}$$

$$\sigma_{min} = \frac{186.6}{2} - \frac{1}{2}\sqrt{(186.6)^2 + 4 \times 165^2}$$

$$\sigma_{min} = -76.2 \quad {}^{N}/_{mm^2}$$

Maximum shear stress:

$$\tau_{max} = \frac{1}{2}\sqrt{(\sigma)^2 + 4(\tau)^\wedge 2}$$

$$= \frac{1}{2}\sqrt{(186.6)^2 + 4(165)^\wedge 2}$$

$$\tau_{max} = 169.5 \quad {}^{N}/_{mm^2} < \sigma_{yt} = 275 \quad {}^{N}/_{mm^2}$$

Von misses Stress:

$$\sigma_{von\ mises} = \sigma_{max} + \sigma_{min} - 2\sigma_{max}\sigma_{min} < \left[\frac{\sigma_{yt}}{2}\right]^2$$

$$\sigma_{von\ mises} = 186.6 - 76.2 + 2 \times 186.6 \times 76.6 < \left[\frac{\sigma_{yt}}{2}\right]^2$$

$$= 28697.5 \quad {}^{\mathrm{N}}/_{\mathrm{mm}^2} < \left[\frac{550}{2}\right]^2 \quad {}^{\mathrm{N}}/_{\mathrm{mm}^2}$$

INTERNATIONAL JOURNAL OF RESEARCH IN MECHANICAL, MECHATRONICS AND AUTOMOBILE ENGINEERING(IJRMMAE)
ISSN: 2454-1435 (Print) | 2454-1443 (online)
Volume 4 Issue 3, October – Dec. 2018 - www.ijrmmae.in – Pages 84-96

$=169.4 < 275 \quad {}^{N}/_{mm^2}$

VIII)COMPARISON OF RESULTS:

Stress	Theoretical $^{N}/_{mm^2}$	Allowable Limit $^{N}/_{mm^2}$
Maximum principle stress	262.8	275
Maximum shear stress	169.5	550
Von mises	169.4	275

Table 3: Comparison Of Results

CONCLUSION

The knuckle joint proposed to develop in the present study is for an applied force of 150 KN. The diameter of the pin is proposed to be around 24 mm. The material of the knuckle joint is considered as molybdenum. Based on the above, a CAD model was developed using CATIA. In order to carry out the stress analysis, mesh was developed for the knuckle joint. ANSYS software was run and the stress contour, displacement contour, strain energy contour were obtained.

Based on the ANSYS analysis, it shows that a pin of 24 mm diameter can resist a load of 150kN if a factor of safety 2 is used.

REFERENCES

[1] J.K. Gupta, R.S. Khurmi. A textbook of machine design.

[2] Jones DRH, Engineering Materials 3 – materials failure analysis.

[3] Blake, Alexander (1985). Design of Mechanical joints.

[4] Bhandari, V.B. (2001). Introduction to machine design.

[5] Joseph E. Shigley, Charles R. Mischke, Thomas H. Brown, (2004). Standard handbook of machine design(3rded).

[6] Analysis of Knuckle Joint of 30C8 Steel for Automobile Application -Sourav Das, Vishvendra Bartaria2, Prashant Pandey: International Journal of Engineering Research & Technology (IJERT) Vol. 3 Issue 1, January – 2014

[7] Material Optimization for Knuckle Joint by Using FEA Somase Anil R1, Dhole Sumit M2, Bhavsar Urvesh P3, Shinde Tejas P4, Bhavsar Pritesh C5: International Research Journal of Engineering and Technology (IRJET) e-ISSN: 2395-0056 Volume: 06 Issue: 10 | Oct 2019 www.irjet.net