ISSN: 2278-0181
Vol. 3 Issue 1, January - 2014

Analysis of Knuckle Joint of 30C8 Steel for Automobile Application

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ABSTRACT

The aim of the present paper is to study calculate the stresses in Knuckle joint using analytical method. Further study in this direction can made by using various directions of the pin and the capacity to withstand load. It is also to be noted that instead of mild steel pin we can also use high strength high modulus steel pin that can further enhance the capacity to withstand higher loads. The shape of the knuckle joint can be changed for improved properties. One can carry out the analysis by changing the shape in the part in the knuckle joint in order to conserve materials and energy. The knuckle joint is proposed to develop in the present study is for an applied force of 25 KN. The diameter of the pin is proposed to be around 23 mm. The material of the knuckle joint is considered as mild steel grade 30C8, in order to do the stress analysis; mesh was developed for the knuckle joint. The mesh consists of 64229 nodes and 4310 elements. 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 23mm diameter can withstand a load of 25 kN if we use a factor of safety of 2. Further optimization of the diameter of pin, it depicts that a pin of 12 mm is enough to withstand a load of 25 kN. however if we use a pin of 25 mm the range of pulling load can be enhance to even 80 kN.

Keywords

Ansys software, 30C8 steel, Stress analysis, knuckle joint pin, tensile failure, Chemical Composition: C: 0.25-0.35, Mn: 0.60 – 0.90, Si: 0.10 – 0.35 P: 0.030, S: 0.035

Introduction

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. Such parts create positive interference with the jointing parts and thus prevent any relative motion and thus help transmit the force. One should remember that the rivets in a riveted joint had exactly the same role as it prevents the slipping of one plate over the other (in lap joint) and moving away of one plate from other (in butt joint). The rivets provided positive interference against the relative motion of the plate. Knuckle joint is another promising joint to join rods and carry axial force. It is named so because of its freedom to move or rotate around the pin which joins two rods. 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 parts. This joint cannot sustain compressive force because of possible rotation about the pin (1,2).

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. 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. This type of joint allows a small angular movement of one rod relative to another. The joint can be easily connected and disconnected. Knuckle joint is found in valve rods, braced girders, links of suspension chains, elevator chains, etc. The figure of a knuckle joint is shown in Fig. 1. The knuckle joint assembly consists of following major components:

1. Single eye.

2. Double eye or fork.

3. Knuckle pin.

4. Collar. and

5. Tapper pin.

The end of one of the rods is forged in the form of a fork while the end of the other rod has an eye, which can be inserted in the jaws of the fork. A cylindrical pin is passed through the holes in the forks and the eye. The pin is secured in position by a taper pin, split pin or a thin nut screwed up to shoulder on the end of the pin. The ends of the rods are made octagonal for good hard grip. A knuckle joint is used to connect the two rods which are under the tensile load, when there is requirement of small amount of flexibility or angular moment is necessary. There is always axial or linear line of action of load.

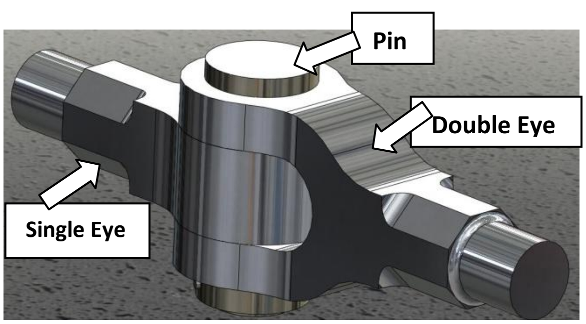

Fig. 1 Photograph of a knuckle joint

At one end of the rod the single eye is formed and double eye is formed at the other end of the rod. Both, single and double eye are connected by a pin inserted through eye. The pin has a head at one end and at other end there is a taper pin or split pin. For gripping purpose the ends of the rod are of octagonal forms. Now, when the two eyes are pulled apart, the pin holds them together. The solid rod portion of the joint in this case is much stronger than the portion through which the pin passes. The knuckle joint is used in many applications such as wiper, tie rod joint of roof truss, tension link in bridge structure, link of roller chain, in tractor, elevators chains, valve rods, wire line tool-string etc.

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.

The failure mechanism of knuckle joint has been studied by several investigators (3-6). Jones (3) has reported that shear failure due to torsional loading is the normal failure mechanism in many engineering components. Pantazopoulos et.al (4) have studied the failure of a knuckle joint of a universal coupling system. It was mentioned that torsional overload of the knuckle joint is the major cause of failure. However, in many cases it was reported that wear of material due to severe friction leading to delamination wear (5,6).

In general, the materials used for making knuckle joint is 30C8 Steel and the chemical compositions are C-0.25-0.35, Mn- 0.60 – 0.90, Si- 0.10 – 0.35, P- 0.030, S- 0.035

(a) Dimensions or Various Parts of the Knuckle Joint

The dimension of various parts of the knuckle joint is fixed by empirical relation as given below. It may be noted that all the parts should be made of the same material and in the present case 30C8 steel is used for calculation. The design data is obtained from many sources (7-9)

If d is the diameter of rod, then 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

Calculations

METHOD OF FAILURE OF KNUCKLE JOINT

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.

σt, and σc = Permissible stresses for the joint material in tension ,shear and crushing respectively.

In determining the strength of the joint for the various method of failure, it is assumed that

1. There is no stress concentration,

2. The load is uniformly distributed over each part of the joint.

1. Failure of the solid rod in tension

Since the rods are subjected to direct tensile load, therefore tensile strength of the rod,

(π /4)*d2*σt

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

P= (π/4)*d2*σt

From this equation, diameter of the rod (d) is obtained.

2. Failure of the knuckle pin in shear

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

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

And the shear strength of the pin

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

Equating this to the load p acting on the rod, we have

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

From this equation ,diameter of the knuckle pin (d1) is obtained .This assumes that is no slack and clearance between the pin and the fork and hence there is no bending of the pin .But in actual practice, the knuckle pin is loose in forks in order to permit angular movement of one with respect to the order, therefore the pin is subjected to bending in addition to shearing .By making the diameter of knuckle pin equal to the diameter of the rod(d=d1) a margin of strength is provided to allow for the bending of the pin. In case, the stress due to bending is taken into account, it is assumed that the load on the pin is uniform distributed along the middle portion means the eye end and varies uniformly over the forks as shown .thus in the forks, a load p/2 acts through a distance of t1/3 from the inner edge and the bending moment will be maximum at the centre of the pin. The value of maximum bending moment is given by

$M = [(p/2) \{(t1/3) + (t/2)\}] – [(p/2)*(t/4)]$

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

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

And section modulus

Z = (π/32) *(d1)3

Since maximum bending (tensile) stress,

σt = M/Z

From this expression, the value of d1 may be obtained.

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

The single eye or rod end may tear off due to the tensile load. We know that area resisting tearing

= (d2 –d1) t

Tearing strength of single eye or rod end

= (d2 –d1) t *σt

Equating this to the load p we have

p = (d2 –d1) t *σt

From this equation, the induced tensile stress for the single eye or rod end may be checked .In case the induced tensile stress is more than the allowable working stress, then increase the outer diameter of the eye.

4. 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

= (d2 –d1) t

Shearing strength of single eye or rod end

= (d2 –d1) t *

Equating this to the load p, we have

p= (d2 –d1) t *

From this equation, the induced shear stress for the single eye or rod end may be checked

5. 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

=d1*t

Crushing strength of single eye or rod end

=d1 *t* σc

Equating this to the load p, we have

p = d1 *t* σc

From this equation, this induced crushing stress for the single eye or pin may be checked .In case the induced crushing stress in more than the allowable working stress, then increasing the thickness of the single eye (t)

6. Failure of the forked end in tension

The forked end or double eye may fail in tension due to the tensile load. We knows that area resisting tearing

= (d2 –d1) *2t1

Tearing strength of the forked end

= (d2 –d1) *2t1 * σt

Equating this to the load, we have

p = (d2 –d1) *2t1 * σt

From this equation, the induced tensile stress for the forked end may be checked

7. 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

= (d2 –d1) *2t1

Shearing strength of the forked end

= (d2 –d1) *2t1 *

Equating this to the load, we have

p = (d2 –d1) *2t1 *

From this equation, this induced shear stress for the forked end may be checked. In case, the induced shear stress is more than the allowable working stress, then thickness of the fork (t1) is increased.

8. 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

= d1 * 2t1

Crushing strength of the forked end

= d1 * 2t1 * σc

Equating this to the load, we have

p = d1 * 2t1 * σc

From this equation, the induced crushing stress for the forked end may be checked.

Analytical calculation of knuckle joint

To design the knuckle joint firstly we choose the diameter of the rod (pin) which can bear the applied stress at a safe range. A load of 25KN is applied. The yield stress of the mild steel is 200MPa. Taking the factor of safety 3, the allowable yield stress is 66.67 MPa and the allowable shear stress is 55 MPa and crushing stress is 83 MPa.

a. Diameter of the rod

We know that

P= (π/4)*d2*σt

$25*10^3$=(π/4)*d2*66.67

D=21.8mm

Let take d =23mm

Then failure of solid rod in tension

P= (π/4)*d2*σt

$25*10^3$=(π/4)*(23*10-3)2*σt

σt =60.17MPa

b. Outer diameter of eye d2=2*d d2=46mm

c. Diameter of knuckle pin head and collar d3=1.5*d d3=34.5mm

d. Thickness of single eye or rod end t=1.25*d t=28.75mm

e. Thickness of fork end t1=0.75*d t1=17.25mm

f. Thickness of pin head t2=0.5*d t2=11.5mm

g. Diameter of the pin d=d1

This is valid when there is no slake and clearance b/w pin and fork end and hence there is no bending of the pin but in actual practice the knuckle pin is loose in forks and order to permit angular movement of one with respect to the order, therefore the pin is subjected to bending in addition to shearing.

In case the stress due to bending is taken into account maximum bending moment will be maximum at the center of the pin.

M = [(p/2) {(t1/3) + (t/2)}] – [(p/2)*(t/4)]

M = (p/2) [(t1/3) + (t/2)-(t/4)]

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

M= (25*103)/2*[(17.25/3)+(28.75/4)]*10-3

M= 161.718 N-m And Z= (π/32)*(d1)3

Since, maximum bending tensile stress σt = M/Z

σt =(161.718*32)/(π*(d1)3)

d1=29.12mm=29.5mm.

3. Failure of the single eye in tension

p = (d2 –d1) t *σt

25*103= (46-29.5)*10-3*28.75 σt =52.70MPa.

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

4. Failure of single eye in shearing

p = (d2 –d1) t * =52.70Mpa.

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

5. Failure of single eye in crushing

p = d1 *t* σc   σc= (25*103)/ (29.5*29*10-6) σc=29.22Mpa

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

6 Failure of fork end intension

p = (d2 –d1) *2t1 * σt σt= 42.08Mpa

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

7. Failure of fork end due to shear stress

p = (d2 –d1) *2t1 * σt =42.08Mpa

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

8. Failure of forked end in crushing stress p = d1 * 2t1 * σc

σc=(25000)/(2*18*29.5*10-6) σc= 23.54Mpa

(It is less than allowable crushing stress so it is safe)

Methodology

MODEL USING ANSYS (FINITE ELEMENT ANALYSIS)

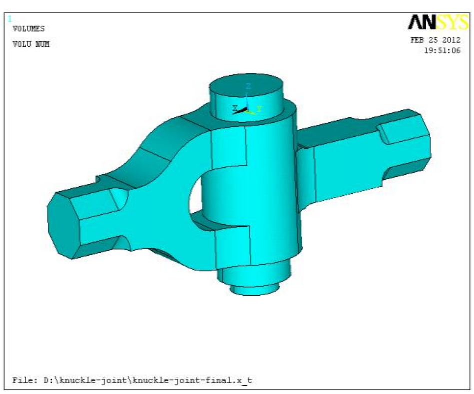

Fig 2 CAD model of Knuckle Joint made by ANSYS Software

The stress analysis, displacement diagram and the strain energy are obtained using ANSYS, commercial FEA software (10)

At the first instance we have prepared the CAD model of knuckle joint using ANSYS software as shown in Fig. 2. In the knuckle joint there are three basic components namely: eye-end, fork-end and the pin, the dimensions of the pin essentially depends upon the load applied in the joints. While preparing the CAD model, we have assigned the proper dimensions and the material properties of the above mention parts. The material selected in the present investigation is mild steel having 0.2% carbon. The pin is having 23 mm diameter and the thickness of the fork-end is 15 mm. Following material properties have been considered for the FEM analysis:

(a). Young's modulus: $2x10^5$ MPa

(b). Poisson's ratio=0.3

(C)Yield strength=200 MPa

MESHING OF KNUCKLE JOINT USING ANSYS & LOADING

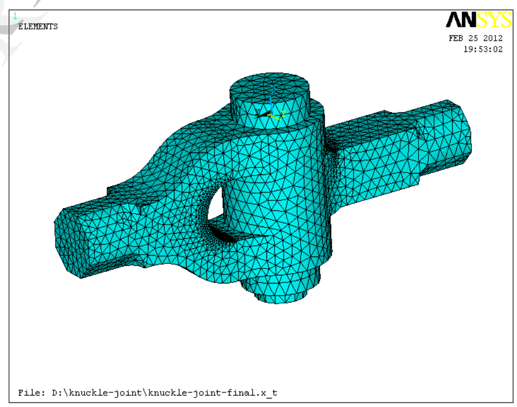

Fig. 3 Meshing of Knuckle Joint made by ANSYS

The basic need for ANSYS analysis is to divide the whole section into many 4 Nodded tetrahedral elements. This will enables us to analyze the stress and strain of the components and various points of the said components. In the present case number of nodes was 64229 and the numbers of elements were 4310. A typical drawing of the meshing of the knuckle joint is shown in Figure 3. An axial load of 25 kN has been applied at the end of the joint.

ISSN: 2278-0181
Vol. 3 Issue 1, January - 2014

RESULT AND DISCUSSION

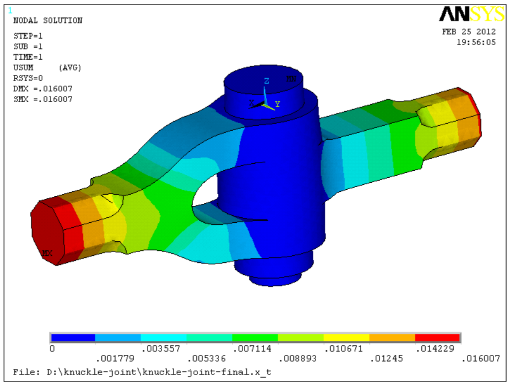

Fig. 4 Displacement diagram of Knuckle joint obtained from ANSYS software

In actual application, knuckle joint is a portion where it experiences maximum stress. The material selection depends upon the fact that it could withstand the stresses develop in the joint and also could deform elastically during its operation. ANSYS software has a unique module which able to measure the amount of deformation i.e. change in length of the joints Fig. 4 shows a typical diagram depicting the elongations in each part of the knuckle joint. It may be mentioned that maximum elongation experienced at the two ends of the joints and minimum elongation occurred around the pin section. The above results show that the elongation at the two ends of the joints is around 0.016007mm and the minimum elongation which is found around the pin is .001mm. It may be mentioned at this juncture that elongation experienced by the components are less and can be use safely for the application.

STRESS ANALYSIS OF KNUCKLE JOINT USING ANSYS

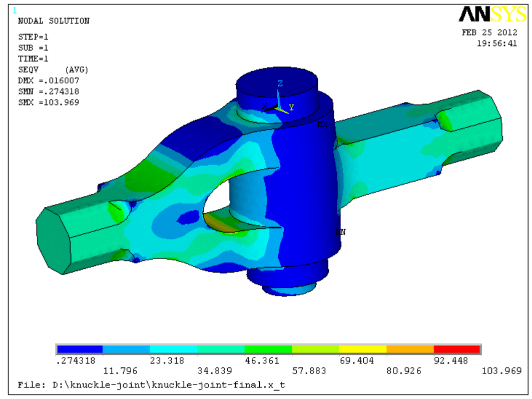

Fig. 5 The stress analysis obtained from ANSYS software

When a load of 25 KN is applied in the system, the ANSYS analysis depicts that the maximum stress experience in the pin is 103.969MPa. It is evident from the literature that mild steel having 0.1-0.2% carbon shows a maximum yield strength of 200 MPa this clearly states from the analysis that pin of 23 mm diameter can sustain an applied load of 25 KN if we consider a maximum allowable yield stress of 200 MPa the calculation show that a pin of 12.6 mm can able to sustain the pulling load of 25 KN such kind of analysis able to provide information about the stress experiences by the components of knuckle joint.

The ANSYS analysis indicated the maximum stress experiences at the interface between the pin, eye-end and the fork end.

STRAIN DENSITY ENERGY OF KNUCKLE JOINT

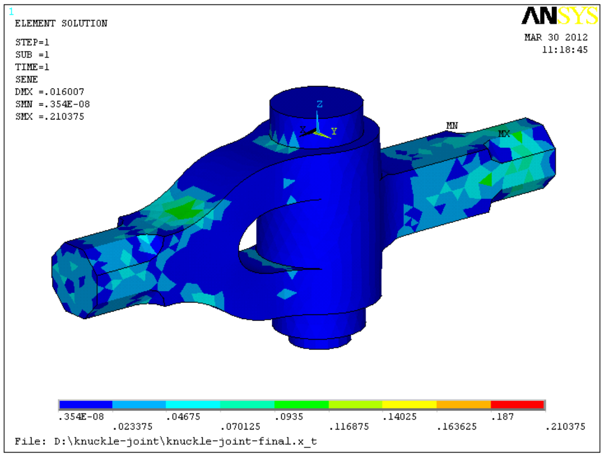

Fig. 6 Strain Energy Diagram of Knuckle Joint

A knuckle joint is a component which is joining the load and the energy provider. For example, when a tractor is pulling a water tanker, a knuckle is used to join both the parts. In this case the tractor is an energy provider and the energy is consumed in pulling the load and the energy is transferred through the knuckle. The energy consumed in the joint is shown in the Fig.6. It may be noted that the maximum energy consumed at the end portion of the knuckle.

Conclusions

1. The knuckle joint proposed to develop in the present study is for an applied force of 25 KN. The diameter of the pin is proposed to be around 23 mm. The material of the knuckle joint is considered as mild steel grade 30c8.

2. Based on the above, a CAD model was developed using ANSYS, commercial FEA software.

3. In order to carry out the stress analysis, mesh was developed for the knuckle joint. The mesh consists of 64229 nodes and 4310 elements.

4: ANSYS software was run and the stress contour, displacement contour, strain energy contour were obtained.

5. Based on the ANSYS analysis, it shows that a pin of 23mm diameter can withstand a load of 25 kN if a factor of safety of 2 is used. Further optimization of the diameter of pin, it depicts that a pin of 12 mm is enough to withstand a load of 25 kN. however if we use a pin of 25 mm the range of pulling load can be enhance to even 80 kN.

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