Design Analysis and Performance Evaluation of Floating Fulcrum Pump for Auto-Control Lubrication System

HETAL CHAUHAN[1], P.S. BAJAJ[2]

[1]M-TECH(DESIGN) Student, Department of Mechanical Engineering
Shri Sant Gadgebaba College of Engineering & Technology, Bhusawal, Maharashtra, India, 425203.
[2]Assistant Professor, Dept. of Mechanical Engineering,
Shri Sant Gadgebaba College of Engineering & Technology, Bhusawal, Maharashtra, India, 425203.

ABSTRACT: *A centralized lubrication system is a system that delivers controlled amounts of lubricant to multiple locations on a machine while the machine is operating. The advantages of this new technology are clear with respect to occupational safety, ALS offers numerous advantages over water-mixed metal working fluids. A major advantage is the substantially better compatibility concerning skin care and lesser pollution. Heart of the ALS system is the pump. The conventional pumps used are of fixed displacement type, thus the volume flow cannot be controlled hence they are not useful for the ALS. The variable displacement pumps like the bent axis piston pump and variable displacement pump are extremely costly for ALS system. Thus there is a need of an variable displacement pump through application of stroke variation mechanism in form of floating fulcrum design and optimize the output of piston pump and to integrate it with the automated lubrication system at relatively low cost to reduce initial cost, operational cost and at the same time achieve precise lubricant control for effective lubrication in modern machinery. The Proposed auto control variable displacement pump is an innovative kinematic link base stroke changing mechanism that is controlled using a automatic control link mechanism that can precisely vary the stroke of the mechanism and thus the pumping unit volume flow rate can be controlled. Project work will include the floating fulcrum linkage development through mathematical modeling. Synthesis design and analysis using ADAMS software. Design and analysis of components using ANSYS workbench 16.0. Optimization of flow parameters for pump through appropriate design of experiment using Taguchi Method by application of Minitab software. The strength analysis of the kinematic linkage parts will be done using ANSYS workbench 16.0, whereas the actual model that will be developed will be tested to determine the performance characteristics of the pump and there by determine the maximum and minimum volume flow rate of the system, volumetric efficiency and precision of flow control.*

***Keywords**: ALS, Kinematic link pump, Automatic control*

INTRODUCTION

An **Automatic Lubrication System** (**ALS**), often referred to as a **centralized lubrication system**, is a system that delivers controlled amounts of lubricant to multiple locations on a machine while the machine is operating. Even though these systems are usually fully automated, a system that requires a manual pump or button activation is still identified as a centralized lubrication system. The system can be classified into two different categories that can share a lot of the same components.

Oil systems: Oil systems primary use is for stationary manufacturing equipment such as CNC milling.

Grease systems: Grease system primary use is on mobile units such as trucks, mining or construction equipment.

Reason for an automatic lubrication system:

Automatic lubrication system is designed to apply lubricant in small, measured amounts over short, frequent time intervals. Time and human resource constraints and sometimes the physical location on machine often makes it impractical to manually lubricate the points. As a result, production cycles, machine availability, and manpower availability dictate the intervals at which machinery is lubricated which is not optimal for the point requiring lubrication. Auto lube systems are installed on machinery to address this problem.

Benefits:

Auto lube systems have many advantages over traditional methods of manual lubrication:

1. All critical components are lubricated, regardless of location or ease of access.
2. Lubrication occurs while the machinery is in operation causing the lubricant to be equally distributed within the bearing and increasing the machine's availability.
3. Proper lubrication of critical components ensures safe operation of the machinery.
4. Less wear on the components means extended component life, fewer breakdowns, reduced downtime, reduced replacement costs and reduced maintenance costs.
5. Measured lubrication amounts means no wasted lubricant.

6. Safety- no climbing around machinery or inaccessible areas (gases, exhaust, confined spaces, etc.).
7. Lower energy consumption due to less friction.
8. Increased overall productivity resulting from increase in machine availability and reduction in downtime due to breakdowns or general maintenance.
9. In this system lubrication the engine parts are lubricated under pressure feed.

OBJECTIVES

1. Design and kinematic synthesis of Kinematic of stroke adjuster mechanism to give zero to maximum stroke displacement of output link displacement and point to point control of the displacement using stroke variation control by control motor.
2. Design & analysis of components of the variable stroke mechanism and the output displacement link for pump application using ANSYS work bench 16.0.
3. Design and selection of and single cylinder piston pump to which the variable stroke linkage will be applied to.

DESIGN METHODOLOGY

In our attempt to design a special purpose machine we have adopted a very a very careful approach, the total design work has been divided into two parts mainly;

- System design
- Mechanical design

System design mainly concerns with the various physical constraints and ergonomics, space requirements, arrangement of various components on the main frame of machine on of controls position of these controls ease of maintenance scope of further improvement; height of m/c from ground etc.

In Mechanical design the components are categories in two parts.

- Design parts
- Parts to be purchased

For design parts detail design is done and dimensions thus obtained are compared to next highest dimension which are readily available in market this simplifies the assembly as well as post production servicing work. The various tolerances on work pieces are specified in the manufacturing drawings. The process charts are prepared & passed on to the manufacturing stage. The parts are to be purchased directly are specified & selected from standard catalogues.

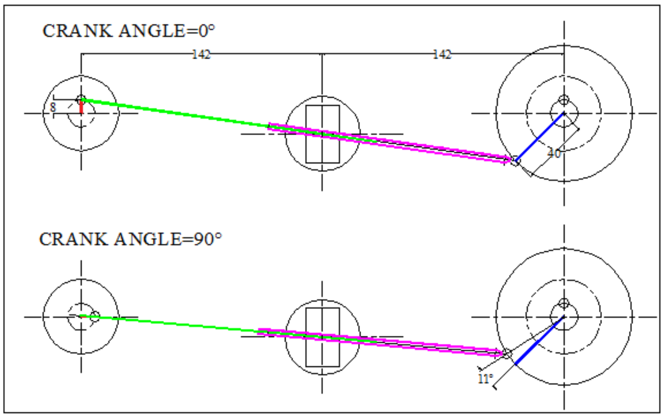
SYSTEM DESIGN

Kinematic Overlay method: Theoretical Design of kinematic linkage- The Overlay method is a graphical trial and error procedure for the synthesis of mechanisms to generate a sequence of specified positions of the output link. The overlay method is extremely versatile, easiest and quickest of all methods to use and widely applicable in the design of mechanism. The synthesis procedure using overlay method involves three simple steps:

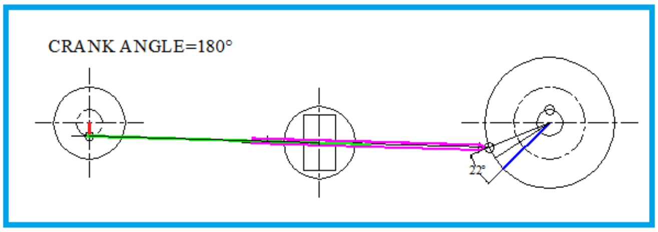
1. Assume any suitable output link length and layout an output link sector depicting the sequence of the specified positions of the output link in one cycle.
2. Select a convenient coupler link length, normally about 75-125% of the output link's length with centers at each moving end of the output link and the coupler length as a radius, draw coupler loci circles.
3. Finally, draw the input link sector showing all the corresponding input link positions on a piece of transparency called the overlay. Fit the overlay such that all the moving ends of the input link lie on their respective coupler loci circles. A large variety of solutions can be obtained if the overlay sector is drawn for different input link lengths.

The overlay method is simple and quite accurate for all practical purposes. Different alternate mechanism configurations can be obtained from which the designer can visually check and select the most desirable configuration, satisfying the geometric proportions, dead center, and limit positions and transmission angle characteristics, etc. The overlay method is also applicable to the slider crank and inverted slider crank mechanism. The method yields a mechanism whose generated output has no precision points, but whose deviation is kept within a narrow tolerance band throughout the range of operations.

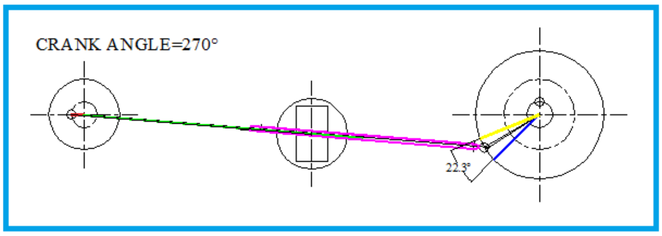
The system design comprises of development of the mechanism so that the given concept can perform the desired operation. The mechanism is basically an inversion of four bar kinematic linkage, hence the mechanism is suitably designed using Grashoff's law and the final outcome is shown in the figure below:



Above figure shows the kinematic linkage design using 2D AutoCAD where in the locus of each end point of the linkage are used to the response of the output link for given control link position. Here it is clear for the change in angle of crank from 0 degree to 90 degree the angle of oscillation changes from 0 degree to 11 degree.



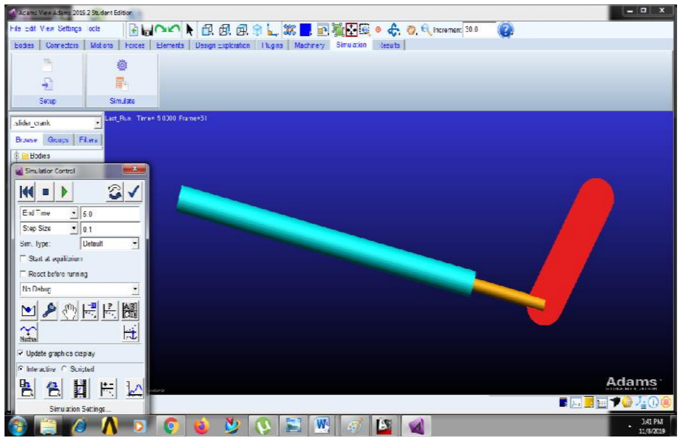
Here it is clear for the change in angle of crank from 0 degree to 180 degree the angle of oscillation changes from 0 degree to 21 degree.



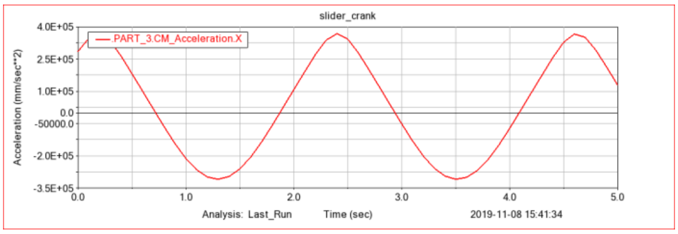
Here it is clear for the change in angle of crank from 0 degree to 180 degree the angle of oscillation changes from 0 degree to 22.3 degree.

Thus the kinematic analysis of the linkage suggest that the maximum angle of oscillation is 22.3 degree for the given configuration of linkage and it can be varies from 0 to 22.3 degree by varying the position of the fulcrum.

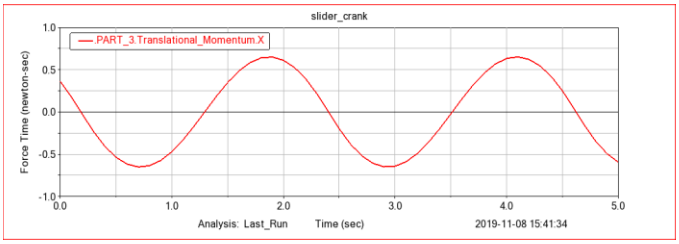
Kinematic synthesis of the linkage using ADAMS software:



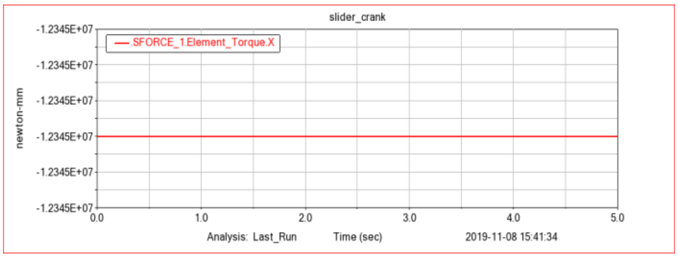
The above figure shows the graphic user interface of the analysis of the slider crank mechanism where in the simulation of the link will be done to get the velocity and acceleration plots of the mechanism:



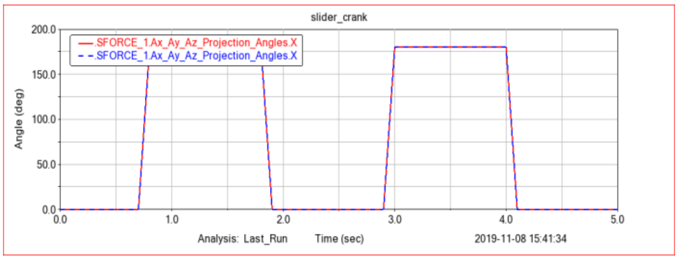
The plot above gives the maximum accelration of the link 3 ie the kinematic link that impart the cam the required motion for pumping action. The maximum acceleration is found to be $3.5E^5$ which is 0.35 m/sec2.



The maximum translational momentum of the link limited to 0.7 N/sec indication lesser inertial action which will help to control the linkage better because less the inertia of the link better will be the accuracy of positioning and thus better control on the flow the pump.



The maximum torque is 1.2345 E^7 N-mm acting at the output link which will generate the pumping force.



The output from the linakge is given after 180 degree travel of the linkage thus there is no phase shift and pumping stroke of suction will be take place during 0 to 180 degree of the input crank and the discharge stroke will take place from the motion of 180 to 360 degree of the input crank.

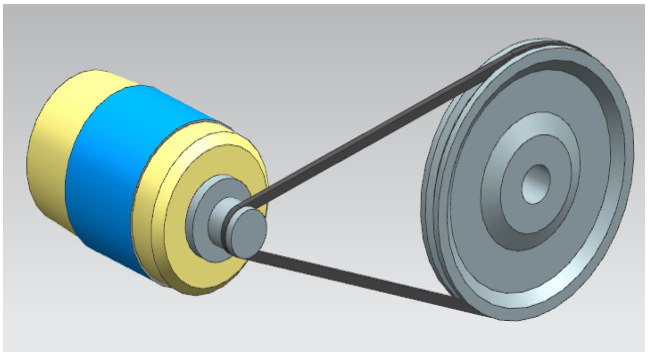
Results of Adams analysis of the slider crank linkage:

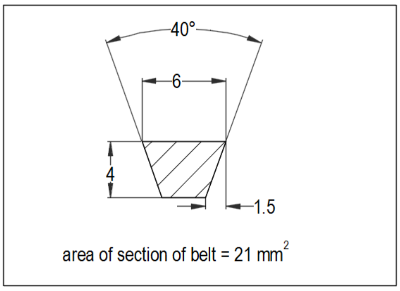
1. The maximum acceleration of the output link is $3.5E^5$ mm/sec^2 which is 0.35 m/sec2 .
2. The maximum translational momentum of the link limited to 0.7 N/sec indication lesser inertial action which will help to control the linkage better.
3. The maximum torque is 1.2345 E^7 N-mm acting at the output link which will generate the pumping force.
4. There is no phase shift and pumping stroke.

DESIGN AND ANALYSIS OF PARTS OF THE PUMP LINKAGE

Design of Belt Drive

Power is transmitted from the motor shaft to the input shaft of drive by means of an open belt drive,
Motor pulley diameter = 20 mm
IP shaft pulley diameter = 100 mm
Reduction ratio = 5
IP shaft speed = 1000 rpm (Design Speed)
Tmotor = 0.48 N-m
Torque at IP shaft = 0.48 Nm
Coefficient of friction = 0.23
Maximum allowable stress in belt = 7 Mpa

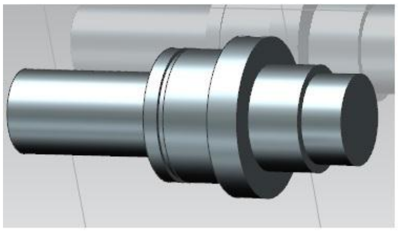
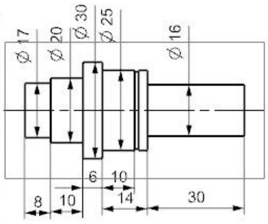




Area of belt = 21 mm^2
Maximum allowable tension in belt = Maximum allowable stress x Area
=7 x 21= 147 N
Center distance = 200
$\alpha = 180 - \sin^{-1}(D\text{-}d) / 2C$
$\alpha = 180 - \sin^{-1}(110\text{-}20)/2\text{x}200$
$\alpha = 136^0$
$\alpha = 2.37^c$
Now,
$e^{\mu\alpha/\sin(\theta/2)} = e^{0.2 \times 2.37 \sin(40/2)} = 4$
width (b_2) at base is given by,
$b_2 = 6 - 2\,(4 \tan 20) = 3.1$
Now mass of belt /m length = 0.23 kg/m
V = ΠDN / (60 x 1000) = 4.188m/sec
$T_c = mV^2$
Tc = 4.034 N
T_1 = Maximum tension in belt - Tc
T_1= 147- 4.034 = 142.9 N
$T_1 / T_2 = e^{\mu\alpha/\sin(\theta/2)} = 4$
T_2 = 35.74 N
Power transmitting capacity = (T1 - T2) v = (142.9 - 35.4) x 4.81/1000 = 21.5 Kw
Thus the belt can safely transmit power of 0.05 Kw

Tdesign = P x 60 / (2 π 9500) = 0.05
Reduction ratio of pulley drive = 100 / 20 = 0.252 N-m

Design of Input Pinion Shaft

DESIGNATION	ULTIMATE TENSILE STRENGTHN/mm^2	YEILD STRENGTH N/mm^2
1. EN24	800	680

As per ASME code

$\Rightarrow fs_{max} = 104\ N/mm^2$

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft

$$Te = \frac{\Pi}{16}\ fs\ d$$

$$\Rightarrow \quad fs_{act} = \frac{16 \times 0.48 \times 10^3}{\Pi \times 16^3}$$

$fs_{act} = 0.596 N/mm^2$

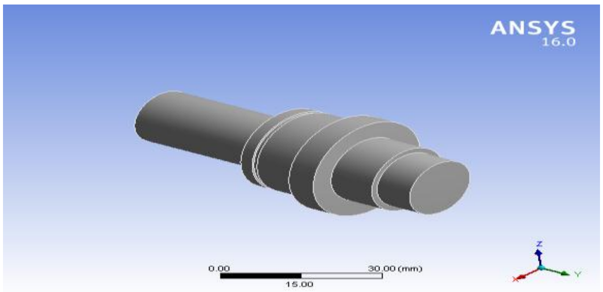
As; $fs_{act} < fs_{all}$

Input Pinion shaft is safe under torsional load.

Analysis of Input Pinion Shaft

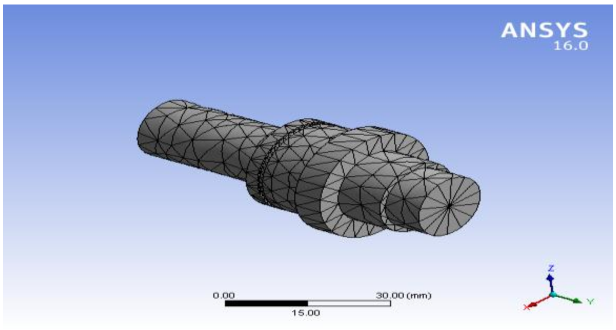
Geometry

Geometry was developed using Unigraphix Nx-8 software and the step file was used as input to Ansys.



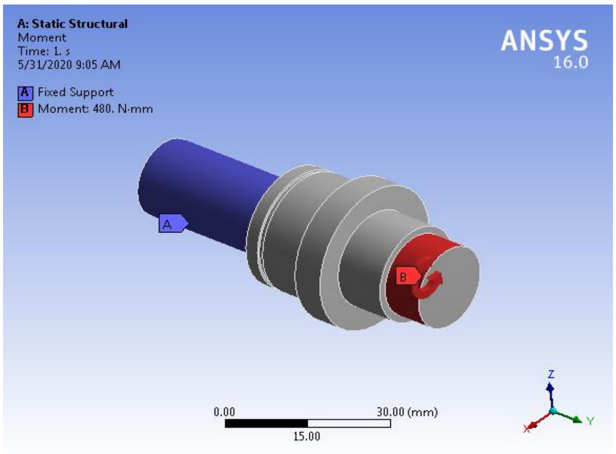
Meshing

Meshing was done using Ansys free mesher and mesh details are as follows:



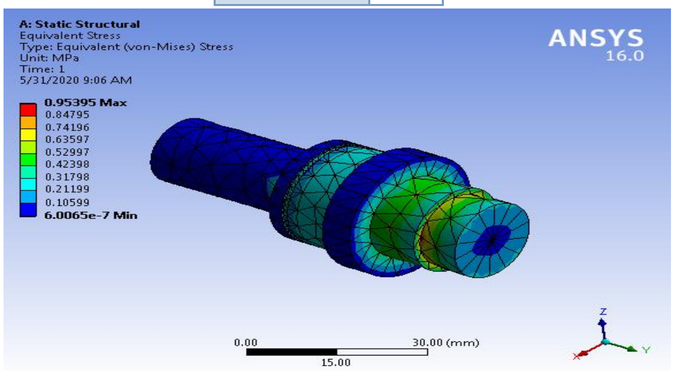
Boundary Conditions

The boundary conditions and loading conditions were defined as below



Von-mises Stresses

Statistics	
Nodes	4563
Elements	2591
Mesh Metric	None



The maximum Von-misses stresses in the part are 0.95MPa which is far below the allowable value 104MPa hence the part is safe under given loading conditions.

Selection of Ball Bearing for Input Pinion Shaft

Series 60 (Light Duty Series)

ISI No	Bearing of basic design No (SKF)	d	D1	D	D_2	B	Basic capacity	
125AC05	6005	25	21	47	42	12	11900	6550

$P = X\,F_r + Y\,F_a$

Neglecting self weight of carrier and gear assembly

For our application $F_a = 0$

$\Rightarrow P = X\,F_r$

Where, $F_r = Ra = (T1 + T2) = (142.9 + 35.74) = 178.64$

As; $F_r < e \Rightarrow X=1$

$\Rightarrow$ **P = 178.64 N**

Calculation dynamic load capacity of bearing,

$L = \left(\frac{C}{P}\right)^p$ where, p = 3 for ball bearings

Machine used for 8hr of service per day;

$L_H = 4000 - 8000$hr

But; $L = \frac{60\,n\,L_H}{10^6}$

$L = 60 \times 1000 \times 4000/10^6 = 240$ mrev

Now; $600 = \frac{(C)^3}{(178.64)^3}$

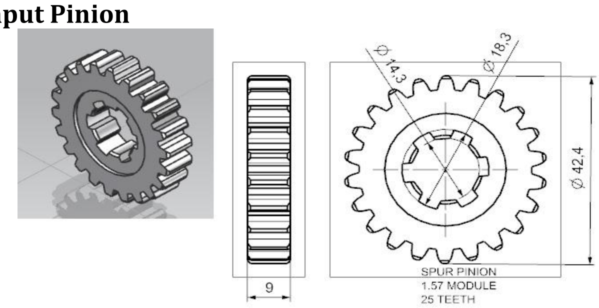
$\Rightarrow$**C = 1110.15 N**

$\Rightarrow$As the required dynamic capacity of bearing is less than the rated dynamic capacity of bearing

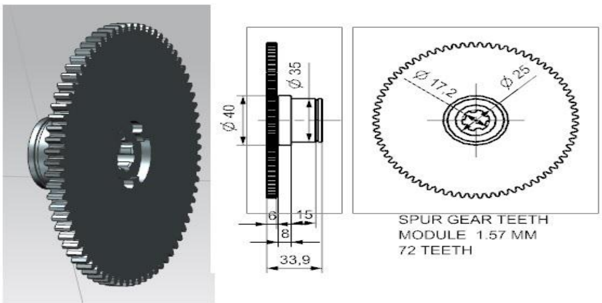
$\Rightarrow$ Bearing selected 6005 zz is safe.

Design of Spur Gear Pair for Drive

Input Pinion



Input Gear



Maximum load = Maximum torque / Radius of gear

Maximum torque = 0.48N-m

No of teeth on gear = 72

No of teeth on Pinion = 25

Module = 1.57 mm

Radius of pinion by geometry = (25 x 1.57) / 2= 19.7 mm

Maximum load = $T/r = 0.48 \times 10^3/19.7 = 24.36$N

b = 10 m

Material of spur gear and pinion = Alloy steel

S_{ult} pinion = S_{ult} gear = 700 N/mm^2

Service factor (Cs) = 1.5

The gear and pinion arrangement where as pinion has 25 teeth and gear has 72 teeth share the entire tooth load considering pinion is weaker.

$\Rightarrow$Pt = (W x Cs) = 36.5 N

P_{eff} = 36.5N (as Cv =1 due to low speed of operation)

P_{eff} = 36.5N --------(A)

Lewis Strength equation

$WT = S_{bym}$

where,

$Y = 0.484 - \frac{2.86}{Z}$

$\Rightarrow y_g = 0.484 - 2.86 = \frac{0.3696}{25}$

$\Rightarrow S_{yg} = 258.72$

$W_T = (S_{yp}) \times b \times m$

$= 258.72 \times 10m \times m$

W_T= 2587.2 m^2----------(B)

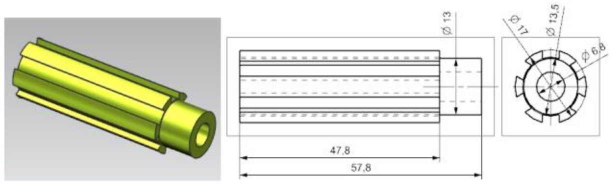
Equation (A) & (B)

2587.2 m^2 = 36.5

$\Rightarrow$**m = 0.12 mm**

Selecting standard module = 1.57mm for ease of construction as we go for single stage gear box making size compact achieving maximum strength and proper mesh.

Design of Input Spline Shaft



DESIGNATION	ULTIMATE TENSILE STRENGTHN/mm^2	YEILD STRENGTH N/mm^2
EN24	800	680

As per ASME code

$\Rightarrow fs_{max} = 104$ N/mm

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft

T_{design}= 0.480 x(72/25) =1.3824

$T_e = \frac{\Pi}{16} fs\ d^3$

$\Rightarrow \quad fs_{act} = \frac{16 \times 1.3824 \times 10^3}{\Pi \times 13^3}$

$fb_{act} = 3.07/mm^2$

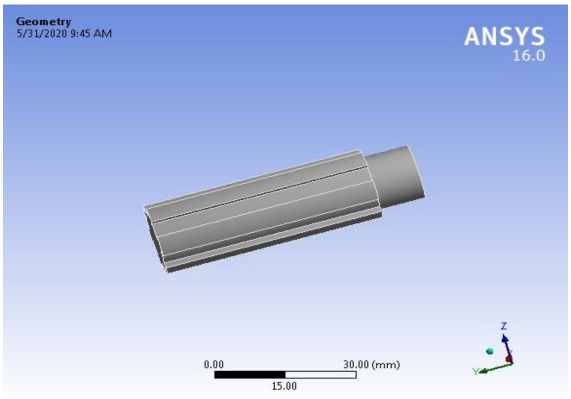
As; $fs_{act} < fs_{all}$

Input spline shaft is safe under torsional load.

Analysis of Input Spline Shaft

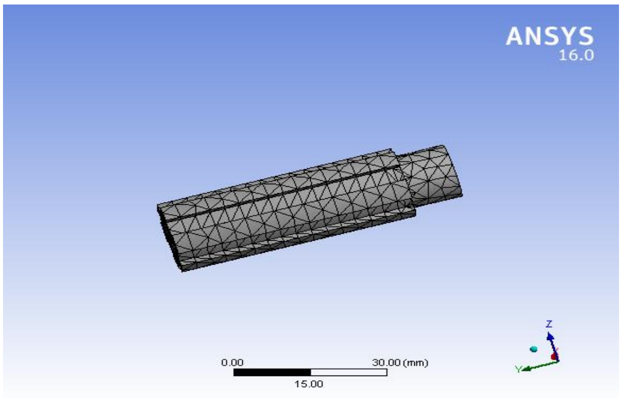
Geometry

Geometry was developed using Unigraphix Nx-8 software and the step file was used as input to Ansys



Meshing

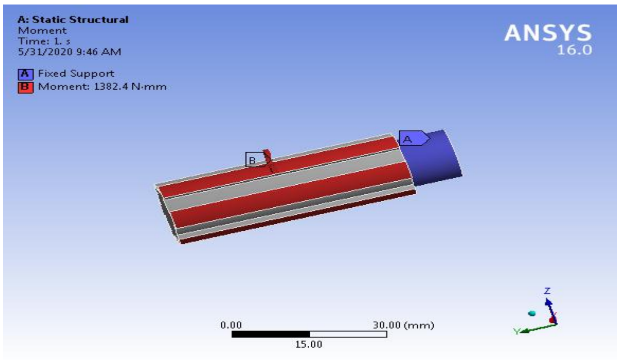
Meshing was done using Ansys free mesher and mesh details are as follows



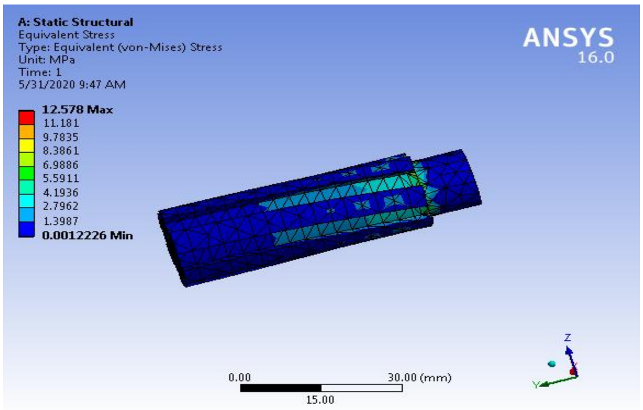
Statistics	
Nodes	4259
Elements	2224
Mesh Metric	None

Boundary Conditions

The boundary conditions and loading conditions were defined as below

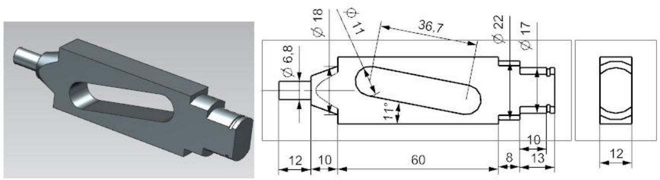


Von-mises Stresses



The maximum Von-misses stresses in the part are 12.578 MPa which is far below the allowable value 104 MPa hence the part is safe under given loading conditions.

Design of Slotted Crank Link



DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm²	YEILD STRENGTH N/mm²
C45	600	480

$\Rightarrow$ fs allowable $= 81\ N/mm^2$

As per ASME code

$\Rightarrow$ $fs_{max} = 104\ N/mm^2$

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft T_{design} = 0.480 x(72/25) =1.3824

$Te = \frac{\Pi}{16}\ fs\ d^3$

$$\Rightarrow \quad fs_{act} = \frac{16 \times 1.3824 \times 10^3}{\Pi \times 6.8^3}$$

$fs_{act} = 21.51/mm^2$

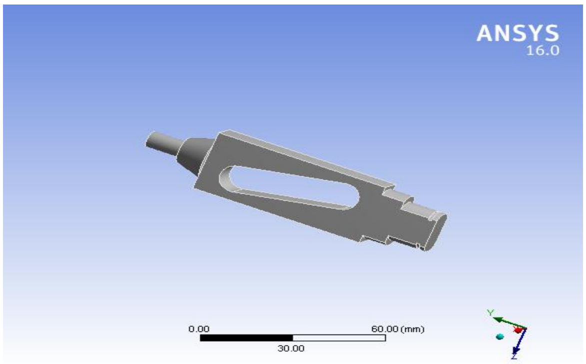
As; $fs_{act} < fs_{all}$

Slotted crank link is safe under torsional load.

Analysis of Slotted Crank Link

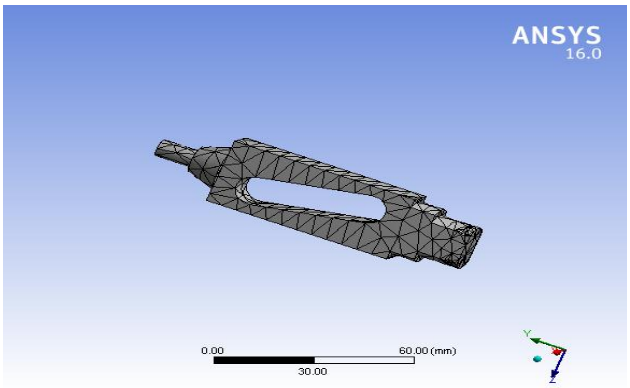
Geometry

Geometry was developed using Unigraphix Nx-8 software and the step file was used as input to Ansys



Meshing

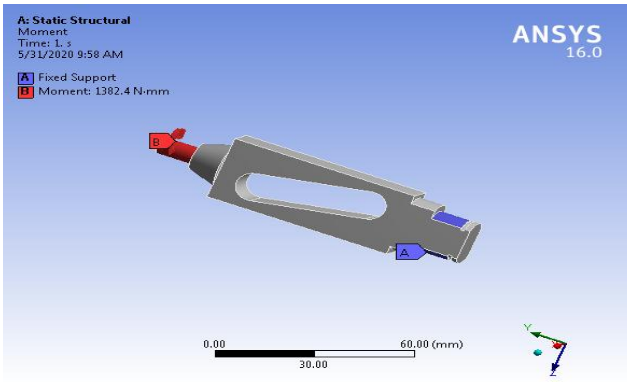
Meshing was done using Ansys free mesher and mesh details are as follows



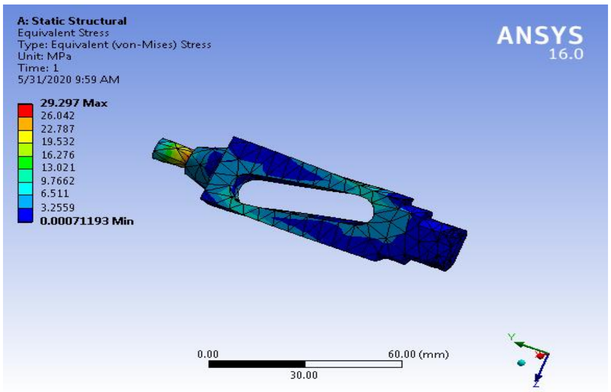
Statistics	
Nodes	2078
Elements	1019
Mesh Metric	None

Boundary Conditions

The boundary conditions and loading conditions were defined as below

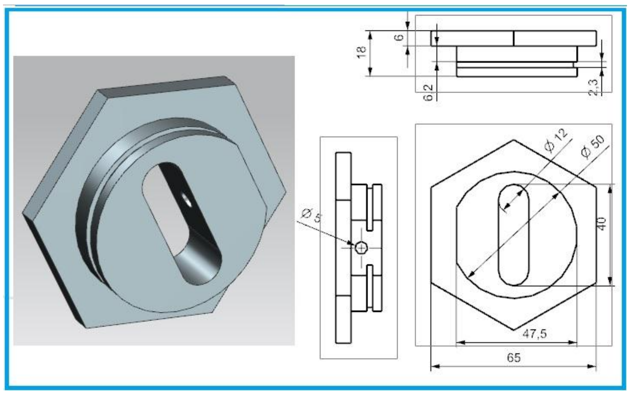


Von-mises Stresses



The maximum Von-misses stresses in the part are 29.27MPa which is far below the allowable value 81MPa hence the part is safe under given loading conditions.

Design of Crank



DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
C45	600	480

$\Rightarrow$ fs allowable = $81N/mm^2$

As per ASME code

$\Rightarrow$ $fs_{max} = 104\ N/mm^2$

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for torsional shear failure of shaft

$T_{design} = 0.480 \times (72/25) = 1.3824$

$$Te = \frac{\Pi \, fs \, d}{16}$$

$$\Rightarrow \quad fs_{act} = \frac{16 \times 1.3824 \times 10^3}{\Pi \times 50^3}$$

$fb_{act} = 0.054/mm^2$

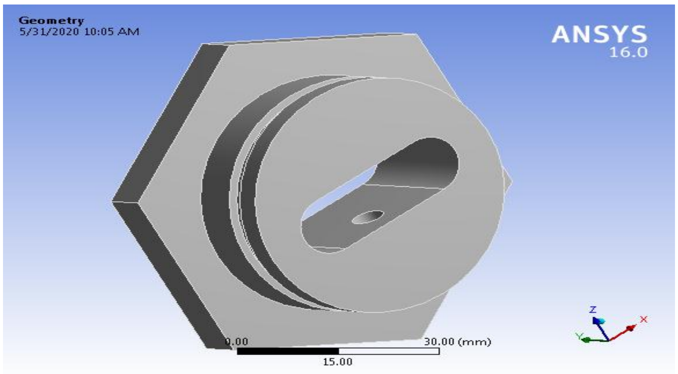
As; $fs_{act} < fs_{all}$

Crank is safe under torsional load.

Analysis of Crank

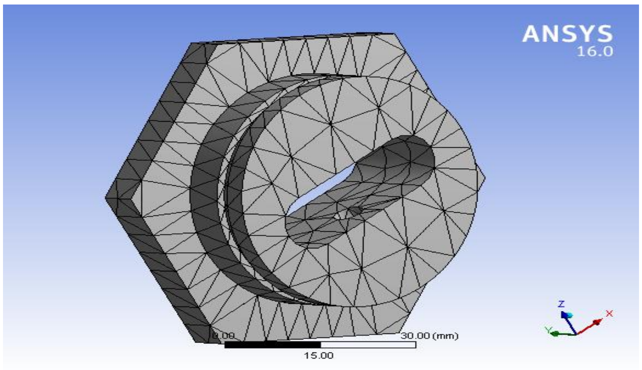
Geometry

Geometry was developed using Unigraphix Nx-8 software and the step file was used as input to Ansys



Meshing

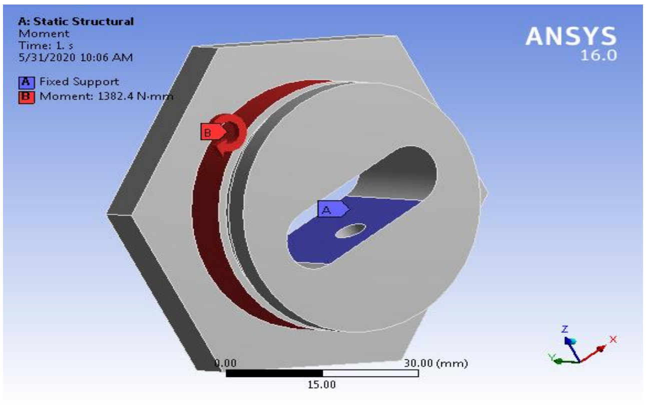
Meshing was done using Ansys free mesher and mesh details are as follows



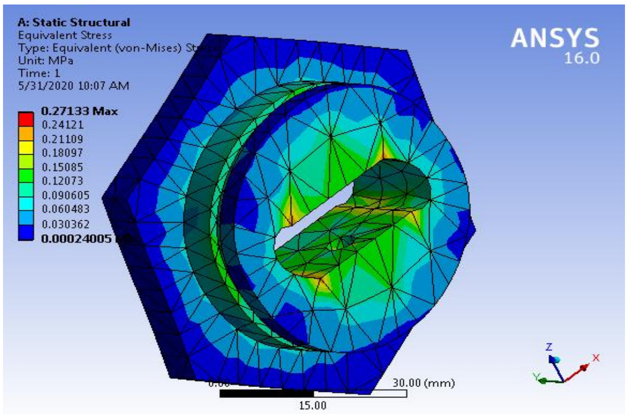
Statistics	
Nodes	3545
Elements	1902
Mesh Metric	None

Boundary Conditions

The boundary conditions and loading conditions were defined as below

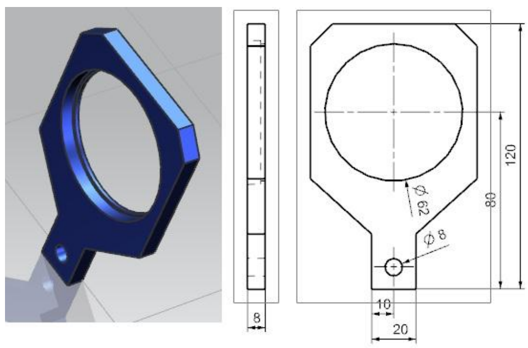


Von-mises Stresses



The maximum Von-misses stresses in the part are 0.271 MPa which is far below the allowable value 81 MPa hence the part is safe under given loading conditions.

Design of Connecting Rod



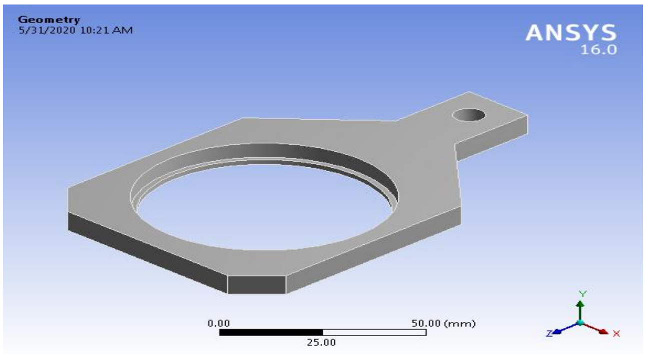
DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm^2	YEILD STRENGTH N/mm^2
2. C45	600	480

⇒ ft allowable = 120 N/mm² (FOS =4)
As per ASME code
⇒ fs_{max} = 104 N/mm
This is the allowable value of shear stress that can be induced in the shaft material for safe operation.
Check for Tensile failure of connecting rod
T_{design} = 0.480 x (72/25) = 1.3824
Crank Force = T_{design} / eccentricity =1382.4 / 5 = 276.5
⇒ Tensile stress = Force / Area
⇒ ft_{act} =276.5 / (20x6)
ft_{act} = 2.304/mm²
As; $ft_{act} < ft_{all}$
Connecting Rod is safe under tensile load.

Analysis of Connecting Rod

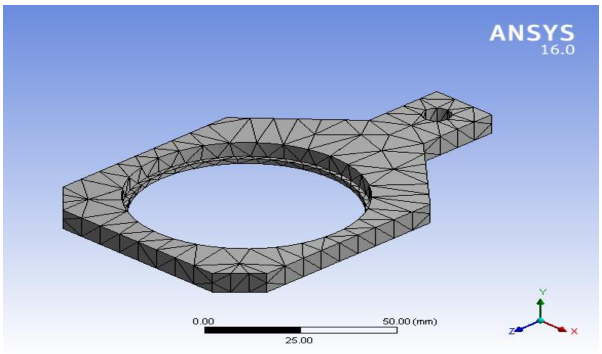
Geometry

Geometry was developed using Unigraphix Nx-8 software and the step file was used as input to Ansys



Meshing

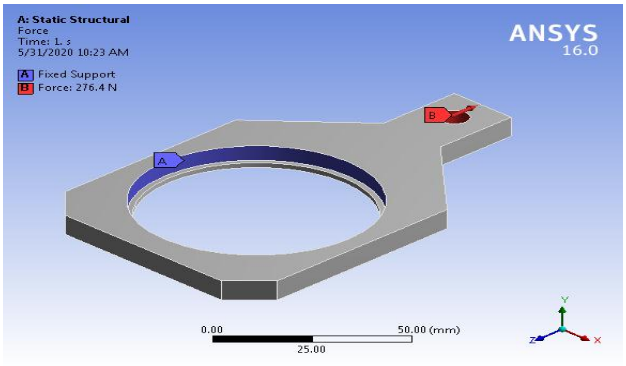
Meshing was done using Ansys free mesher and mesh details are as follows



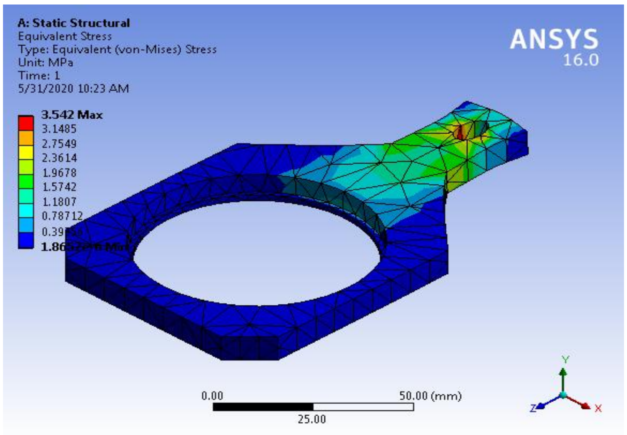
Statistics	
Nodes	1977
Elements	945
Mesh Metric	None

Boundary Conditions

The boundary conditions and loading conditions were defined as below

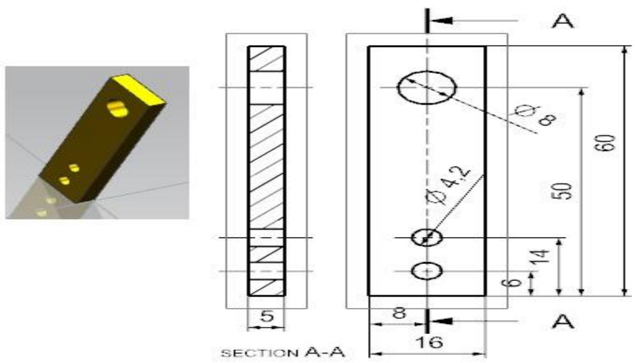


Von-mises Stresses



The maximum Von-misses stresses in the part are 3.52MPa which is far below the allowable value 120 MPa hence the part is safe under given loading conditions.

Design of Connecting Link



DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm²	YEILD STRENGTH N/mm²
3. BRASS	400	280

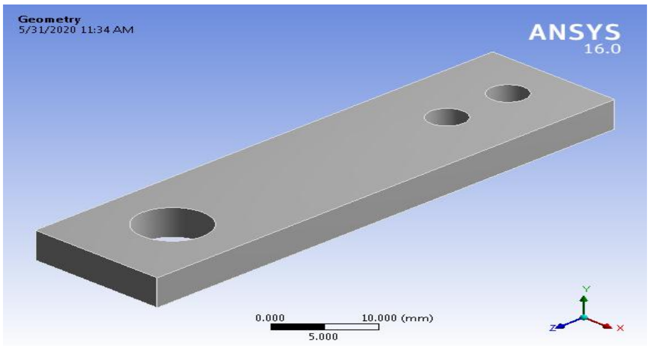
⇒ f_t allowable =70 N/mm² (FOS =4)
As per ASME code
⇒ $f_{s\,max}$ = 104 N/mm²
This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for Tensile failure of connecting rod
T_{design} = 0.480 x (72/25) = 1.3824
Crank Force = Tdesign / eccentricity =1382.4/5 = 276.5
⇒ Tensile stress = Force / Area
⇒ $f_{t\,act}$ = 276.5 / (16x5)
$f_{t\,act}$ = 3.456/mm²
As; $f_{t\,act} < f_{t\,all}$
Connecting link is safe under tensile load.

Analysis of Connecting Link

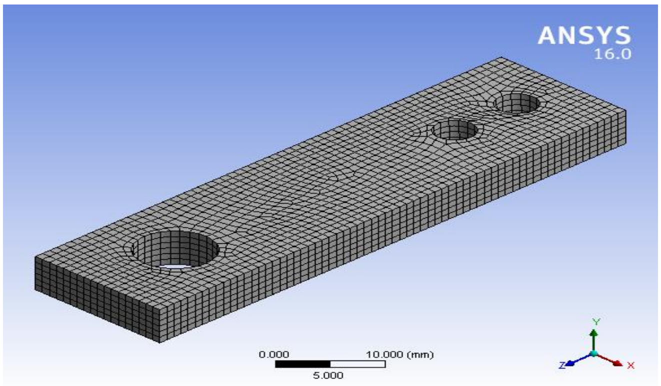
Geometry

Geometry was developed using Unigraphix Nx-8 software and the step file was used as input to Ansys



Meshing

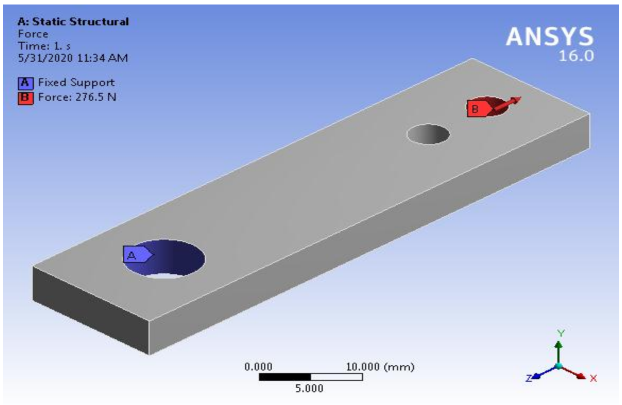
Meshing was done using Ansys free mesher and mesh details are as follows



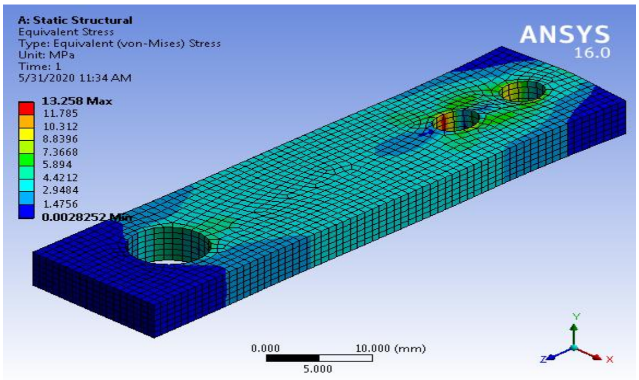
Statistics	
Nodes	1977
Elements	945
Mesh Metric	None

Boundary Conditions

The boundary conditions and loading conditions were defined as below

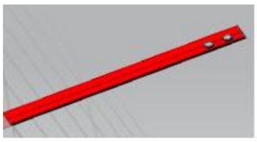


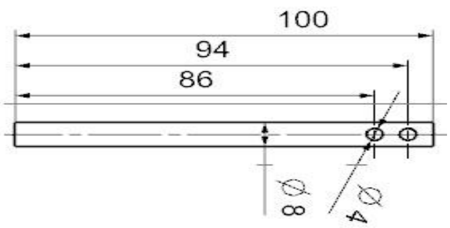
Von-mises Stresses :



The maximum Von-misses stresses in the part are 13.25 MPa which is far below the allowable value 70 MPa hence the part is safe under given loading conditions.

Design of Piston Rod





DESIGNATION	ULTIMATE TENSILE STRENGTH N/mm²	YEILD STRENGTH N/mm²
4. C45	600	480

⇒ f_t allowable =120 N/mm² (FOS =4)
As per ASME code
⇒ $f_{s\,max}$ = 104 N/mm²

This is the allowable value of shear stress that can be induced in the shaft material for safe operation.

Check for Compression failure of piston rod

T_{design} = 0.480 x (72/25) = 1.3824

Crank Force = Tdesign / eccentricity =1382.4/5 =276.5

$\Rightarrow$ Compressive stress = Force / Area

$\Rightarrow$ ft_{act} =276.5 / (Πx 8 ^{2} /4)

ft_{act} = 5.50/mm^2

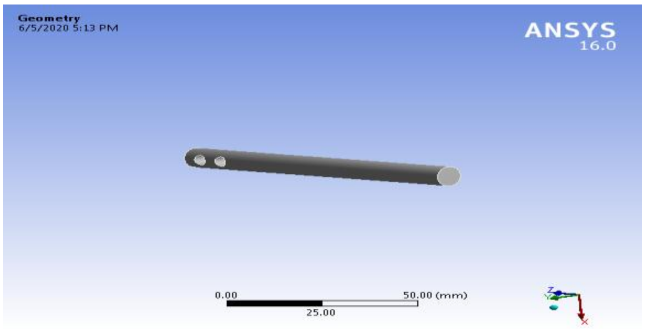
As; $ft_{act} < ft_{all}$

Piston rod is safe under tensile load.

Analysis of Piston Rod

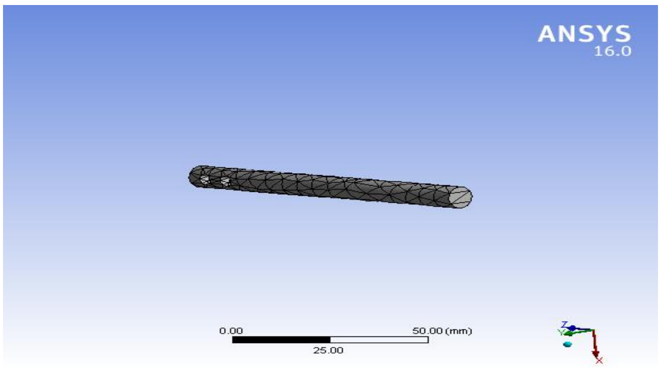
Geometry

Geometry was developed using Unigraphix Nx-8 software and the step file was used as input to Ansys



Meshing

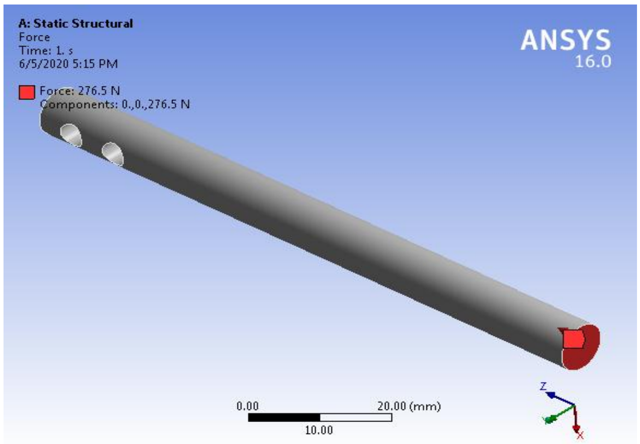
Meshing was done using Ansys free mesher and mesh details are as follows



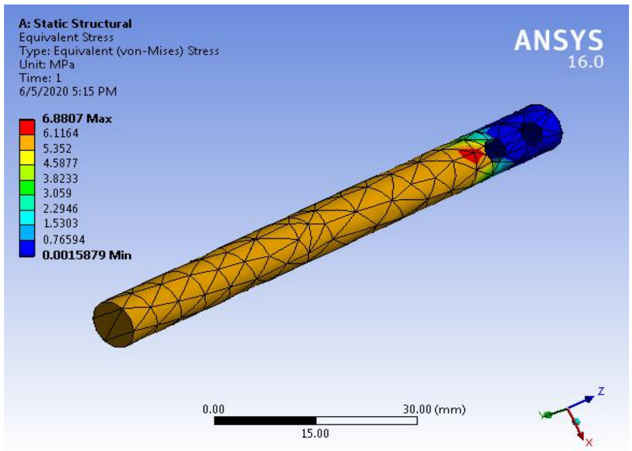
Statistics	
Nodes	1977
Elements	945
Mesh Metric	None

Boundary Conditions

The boundary conditions and loading conditions were defined as below



Von-mises Stresses



The maximum Von-misses stresses in the part are 6.67 MPa which is far below the allowable value 120 MPa hence the part is safe under given loading conditions.

CONCLUSIONS

1. The sizing, design analysis critical components of Floating fulcrum pump is successfully done and the dimensions of the components have being determined.
2. Kinematic linkage design using graphical kinematic overlay method and the maximum angle of oscillation was found to be 22.3 degrees.
3. Adams software was used for kinematic analysis and minimal acceleration of the output link was found and also the resultant momentum was found to be low resulting in lower inertia forces thereby better linkage control.
4. Structural analysis of the components of the kinematic linkage was done and parts were found to be safe by both theoretical as well as analytical method.

REFERENCES

[1] Shengrong GUO, Jinhua CHEN, Yueliang LU, Yan WANG, Hongkang DONG "Hydraulic piston pump in civil aircraft: Current status, future directions and critical technologies" Chinese Journal of Aeronautics, 2nd Jan 2019.

[2] Pawankumar J. Lohar, Dr. S. S. Gawade "Analysis of Gerotor in a Lubricating Oil Pump" International Journal of Advance Research in Science and Engineering, Volume No.07, Special Issue No.04, April 2018.

[3] Alessandro Corvaglia, Massimo Rundo "Comparison of 0D and 3D Hydraulic Models for Axial Piston Pumps" 73rd Conference of the Italian Thermal Machines Engineering Association (ATI 2018), 12-14 September 2018, Pisa, Italy.

[4] F. Fornarelli, A. Lippolis, P. Oresta, A. Posa "Investigation of a pressure compensated vane pump" 73rd Conference of the Italian Thermal Machines Engineering Association (ATI 2018), 12-14 September 2018, Pisa, Italy.

[5] Massimo Borghi, Barbara Zardin, Sergio Materi, Pierluigi Argentino "Fluid-dynamic analysis of an in-line water piston pump" 73rd Conference of the Italian Thermal Machines Engineering Association (ATI 2018), 12-14 September 2018, Pisa, Italy.

[6] Khalid Nafees and Aas Mohammad 2017 "Dimensional synthesis of a planar five-bar mechanism for motion between two extreme positions", Australian Journal of Mechanical Engineering, DOI 10.1080/14484846.2017.1335456, 6th Jun 2017.

[7] Zhonghai MA, Shaoping WANG, Jian SHI, Tongyang LI, Xingjian WANG "Fault diagnosis of an intelligent hydraulic pump based on a nonlinear unknown input observer", Chinese Journal of Aeronautics, 17th march 2017.