Lesson Name – Machine Elements

Project Name - Design and Analysis of Hydraulic Crane

by

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MOTIVATION

Firstly, many occupational accidents can take place in factories since incorrect lifting and faulty machines. I've even encountered the like this problem in the workplace where I did an internship. In this project, I designed and analyzed a crane which can lift a heavy load with 900kg.The crane operates hydraulically there is which is connected to the vertical column and boom for lifting up and down the mass. Characteristic behaviors are analyzed on the static loads and calculated by handing with the given values but Hydraulic Crane has a dynamic structure. Therefore, we used the Simscape Multibody Toolbox in order to solve the problem dynamically. Dynamic analysis is made firm in Matlab.

PURPOSE

The purpose of this study is to demonstrate the static and dynamic solutions that is be different. That’s why I used the Matlab Simulink and Solidworks.

When we understand how the project is done, we will be able to analyze more accurately a moving system like robot arm etc.

In this project, Solidworks and Matlab Simulink were used respectively.

THEORETICAL STUDY

Design Analysis the Parts:

(A) Design of Vertical Column:

Vertical column is modeled as a strut or short compression member thus it is exposed to a compressive stress and this stress is the sum of simple stress component and bending components.

$$\sigma_c = \frac{P}{A} + \frac{Mc}{I}$$

$$= \frac{P}{A} + \frac{P e c A}{I A}$$

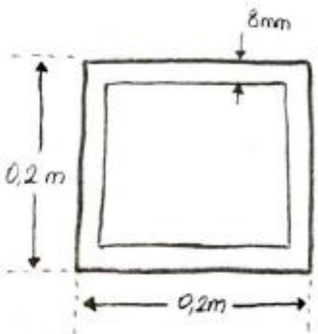
where,

c = distance between axis of column

e = eccentricity of the load

Therefore, $c = \frac{0{,}2m}{2} = 0{,}1m$ & $e = L_4 = 2000\ mm = 2m$

· The column cross-section subjected to compression stress is;



→ The area of cross-section;

$$A = \left[(0.2 \times 0.2) - (0{,}184 \times 0{,}184)\right]\ m^2 = 6{,}144 \times 10^{-3}\ m^2$$

$$(0{,}2 - 0{,}016) = 0{,}184$$

→ moment of Inertia;

$$I_{xx} = \frac{0{,}2^4}{12} - \frac{(0{,}2-0{,}016)^4}{12} = 3{,}781 \times 10^{-5}\ m^4$$

$$I_{yy} = \frac{0{,}2^4}{12} - \frac{(0{,}2-0{,}016)^4}{12} = 3{,}781 \times 10^{-5}\ m^4$$

Note: Buckling always occur about the axis having minimum radius of gyration ($\sqrt{I/A}$ = radius of gyration) or least moment of inertia, therefore in our case buckling occur along horizontal direction (I_{xx})

$$\sigma_c = \frac{P}{A} + \frac{Mc}{I} = \frac{P}{C} + \frac{PecA}{IA}$$

$$P = (900\ kg).(9{,}81\ m/s^2) = 8829\ N$$

$$P_{vertical,c} = 8829.\cos 0 = 8829\ N$$

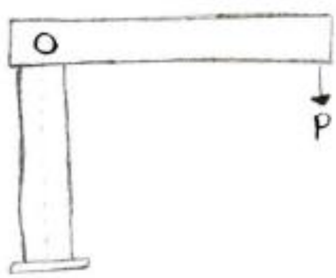
so,

$$\sigma_c = \frac{8829\ N}{6{,}144\times10^{-3}\ m^2} + \frac{(8829\ N)(2m)(0{,}1m)}{3{,}781\times10^{-5}\ m^4} = 48{,}139\ mPa$$

Now that the factor of safety is equal to 2,4 ; $\frac{S_y}{n} = \frac{235\ mPa}{2{,}4} = 97{,}916$ mPa is maximum allowable stress. Minimum yield Strength is 235 mPa for St-37. Therefore, the vertical column is designed safety.

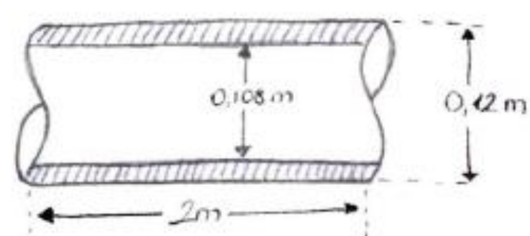
Ⓑ Design of Boom:

The boom is modeled as simply supported beam and it is subjected to a bending stress due to bending moment developed at the fixed and where it is pinned with the vertical column.



$\sigma_B = \frac{M}{Z}$ where; σ_B = Bending Stress
M = Bending moment
Z = Section modulus

Since the boom is hollow rectangular cross-section, the area of boom to which the effect of load P induces the stress is;



$$A = 2m.(0{,}12 - 0{,}108)m = 0{,}024\ m^2$$

• Moment of Inertia;

$$I_{xx} = \frac{2.(0,12^3 - 0,108^3)}{12}\ m^4 = 7,8048 \times 10^{-5}\ m^4$$

→ distance from neutral axis extreme fiber is $\frac{0,12}{2} = 0,06\,m = c$

→ Section modulus (Z);

$$Z = \frac{I}{c} = \frac{7,8048 \times 10^{-5}\ m^4}{0,06\ m} = 1,3008 \times 10^{-3}\,m^3$$

→ Bending moment (m);

$M = P \times L$ and L is the length of boom

$= (8829\ N) \times (2\,m) = 17658\ Nm$

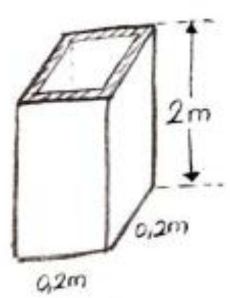
$$\sigma_B = \frac{M}{Z} = \frac{17658\ Nm}{1,3008 \times 10^{-3}\ m^3} = 13,575\ mPa$$

Allowable Bending Stress for St-37 with factor of safety, $n = 2,4$ is

$$\sigma_B = \frac{S_y}{n} = \frac{235\ mPa}{2,4} = 97,9166\ mPa$$

Since the allowable bending stress is greater than the induced bending stress due to load applied, then the boom is designed safe.

(c) Mass of Vertical Column:



Volume Vertical Column = $\forall = A\ h$

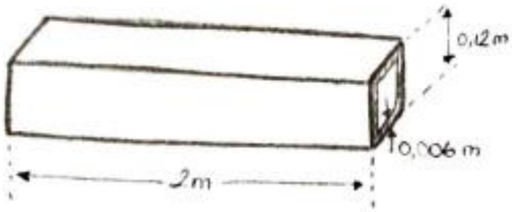
$(6,144 \times 10^{-3} m^2)\ (2m) = 12,288 \times 10^{-3}\ m^2$

since the material for the vertical column is made from St-37 the mass density of St-37 is 7800 kg/m³.

$m = \rho . \forall \Rightarrow (7800\ kg/m^3)\ (12,288 \times 10^{-3}\ m^2)$

$= 95,846\ kg$

(D) Mass of Boom.



The Volume of the Boom $\Rightarrow \forall = A\,L$

$$\rightarrow (0{,}12^2 - 0{,}108^2)\,m^2 \times (2\,m)$$

$$\forall \rightarrow 5{,}472 \times 10^{-3}\,m^3$$

$$m = \rho \cdot \forall \Rightarrow (7800\ kg/m^3)\,(5{,}472 \times 10^{-3}\,m^3)$$

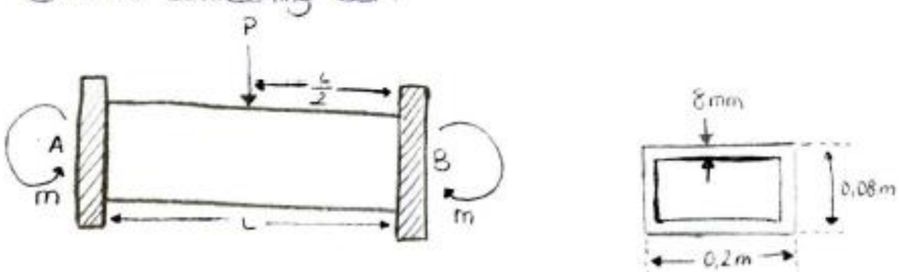
$$= 42{,}682\ kg$$

The mass of other components like hydraulic piston, pin, gripper of the mass all in one are estimated to be 25 kg. Additionally the design is proposed to lift a load of 900 kg and the total mass applied on the base plate is;

$$m_T = (900 + 25 + 42{,}682 + 95{,}846)\ kg = 1064{,}028\ kg$$

$$P_T = m_T\, g = (1064{,}028\ kg)\,(9{,}81\ m/s^2) = 10438{,}114\,N$$

(E) Center Connecting Bar.



$$P = \frac{P_T}{2} = 5219{,}057\ N \qquad A = (0{,}2 \times 0{,}08)\,m^2 - (0{,}184 \times 0{,}064)\,m^2 = 4{,}224 \times 10^{-3}\,m^2$$

$$I = \frac{(0{,}2)(0{,}08)^3}{12} - \frac{(0{,}184)\,(0{,}064)^3}{12} = 4{,}513 \times 10^{-6}\ m^4$$

$$c = \frac{0{,}08}{2} = 0{,}04\ m$$

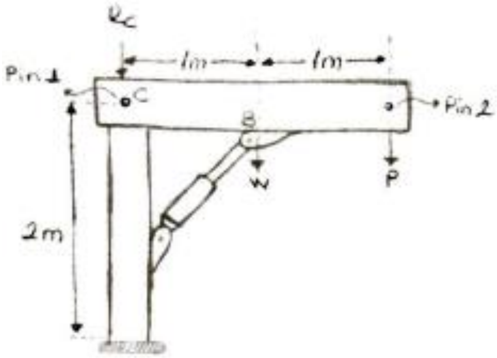
$$M_{A,B} = P_T\,\frac{L}{4} = (10438{,}114\ N)\cdot\left(\frac{1{,}3\,m}{4}\right) = 3392{,}387\ Nm$$

$$\sigma_B = \frac{Mc}{I} = \frac{(3392{,}387\ Nm)\cdot(0{,}04\,m)}{4{,}513 \times 10^{-6}\,m^4} = 30{,}067\ MPa \ \text{(induced stress)}$$

Allowable Bending stress is $\sigma = \frac{S_y}{n}$ so $\frac{235}{2{,}4}$ MPa $= 97{,}916$ MPa

Therefore, the bar is designed safety.

(F) Design of Pins.



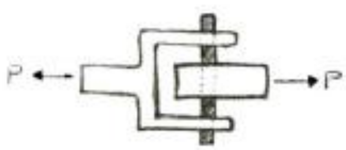
The pin that connect the vertical column and the boom is subjected to shearing force.

$$\sum M_B = 0$$

$$1\,R_C = 1 . P \Rightarrow R_C = P$$

$$\text{so } R_C = 8829\ \text{N} \ (\text{Shear Force on Pin } 1)$$

Since the pin is subjected to high tensile and shearing stress, the material for the pin should be ductile Material selected St-37. Minimum Yield Strength of St-37 is 235 MPa.



$$\tau_{max} = \frac{\sigma_n}{2 . n} = \frac{235\ \text{MPa}}{2(2,4)} = 48,958\ \text{MPa}$$

and then use 48 MPa

Thus, the pin will be subjected to double shear and then the pin is designed as follow;

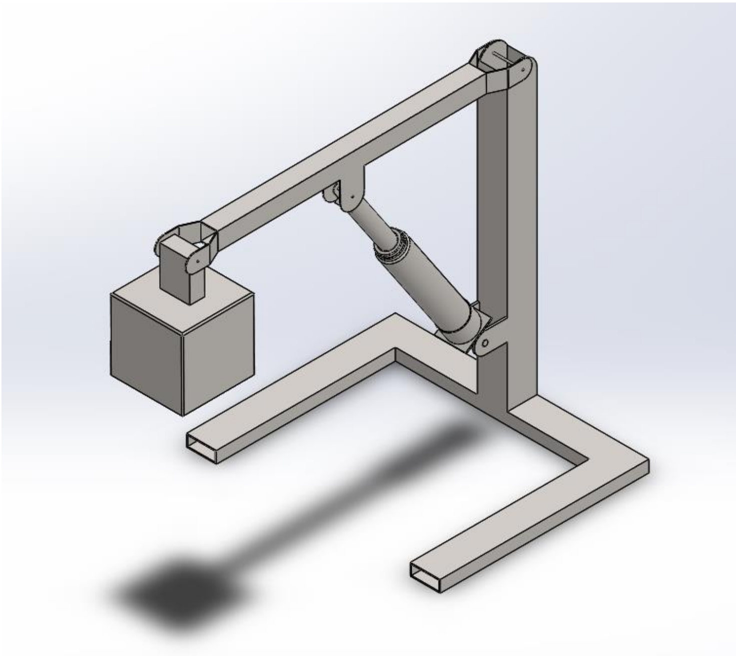
$$\tau = \frac{P}{2A} \Rightarrow A = \frac{P}{2\tau} \Rightarrow \frac{\pi D^2}{4} = \frac{P}{2\tau}$$

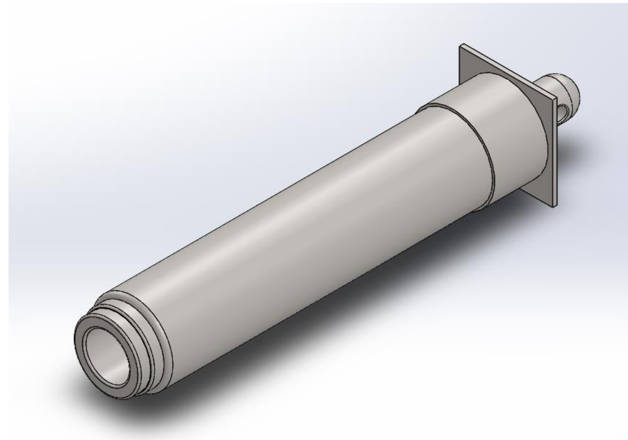
$$\text{so} \quad D = \sqrt{\frac{2P}{\pi . \tau}}$$

$$D = \sqrt{\frac{2\,(8829\ \text{N})}{\pi . (48 \times 10^6\ \text{MPa})}} = 0,01082\ \text{m} = 10,82\ \text{mm}$$

so we can use $12\ \text{mm} = d$.

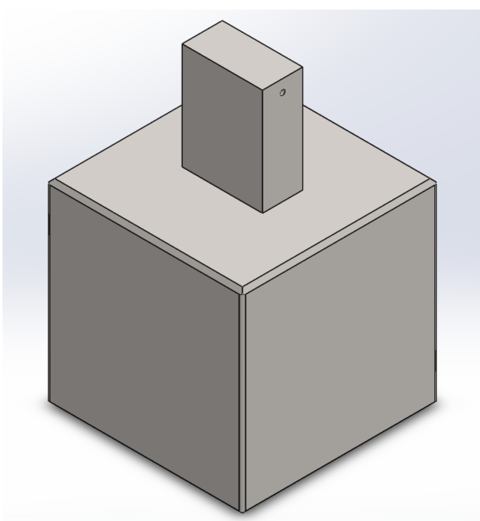
. Now that, P and Rc are equal to eachother, therefore diameter of pins must be equal.

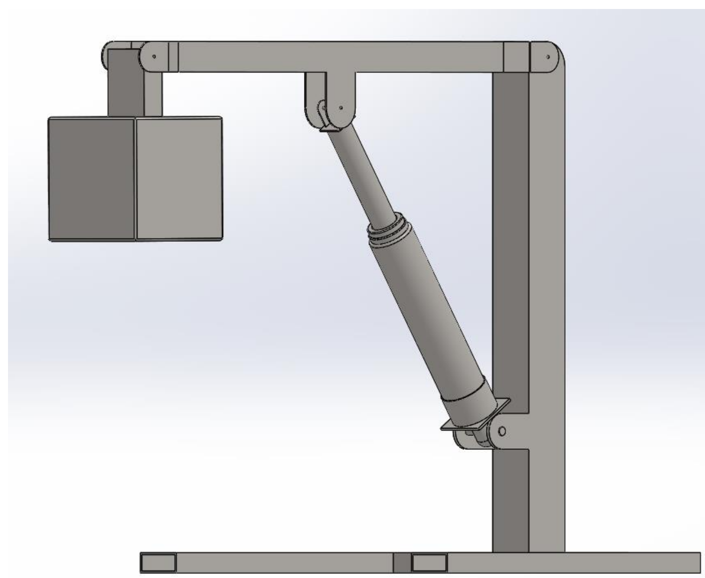




Bottom Piston Housing

Master View





Mass

Figure 1- Figure of Parts and Assembly

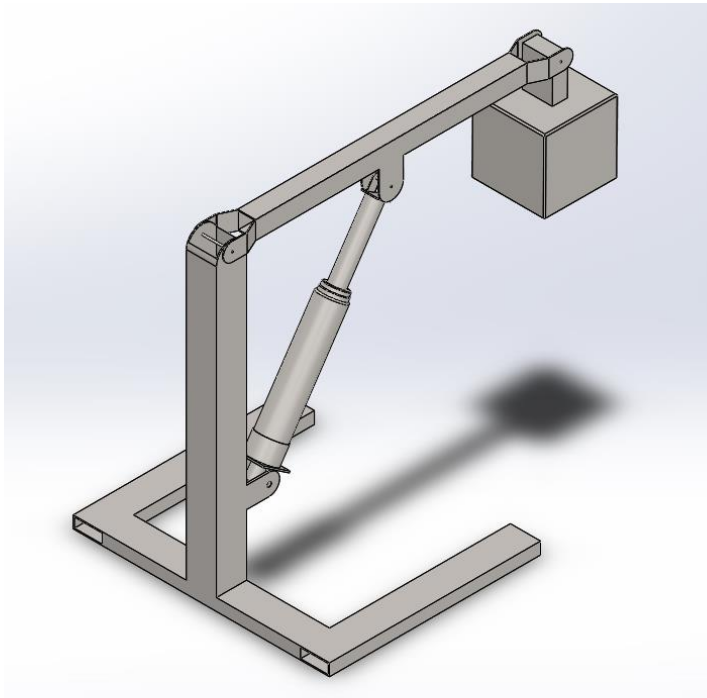


Figure 2

As it is seen in the Figure 2, interiors of the base plate are empty in order to the weight prevent the occurrence of very heavy.

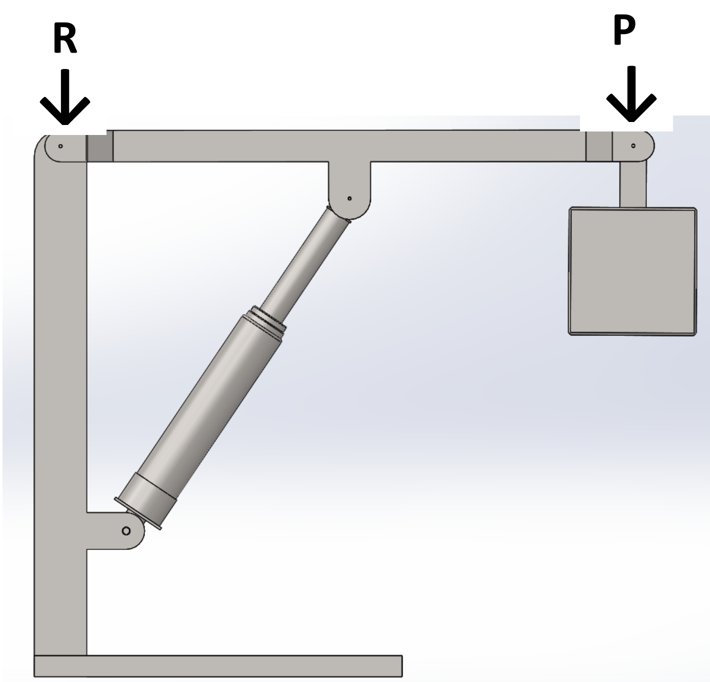


Figure 3

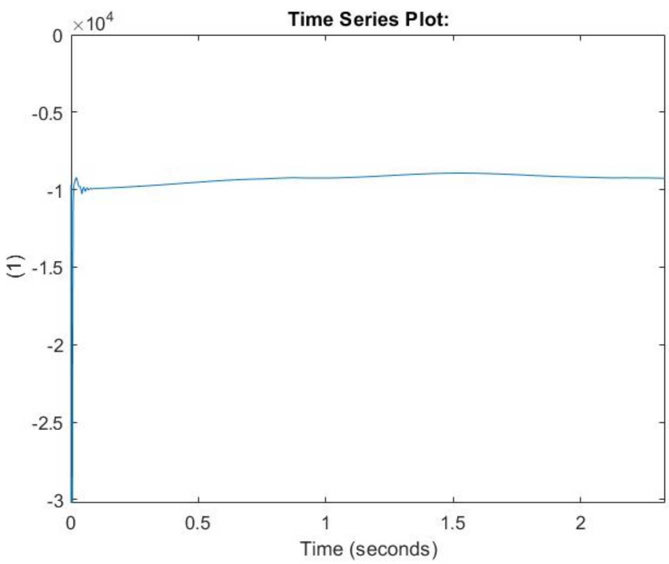
Now that the total moment is equal to zero, R and W must be same. We assume that the weight of the piston is negligible.

R: Shear Force on Pin
P: Weight of the mass

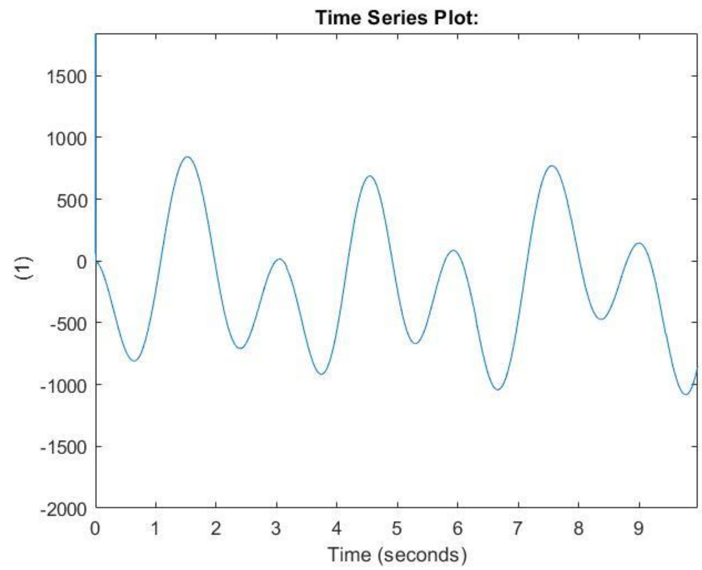
SIMULATION AND RESULTS

<u>Diameter Calculation in terms of Force on the Revolute Joint:</u>

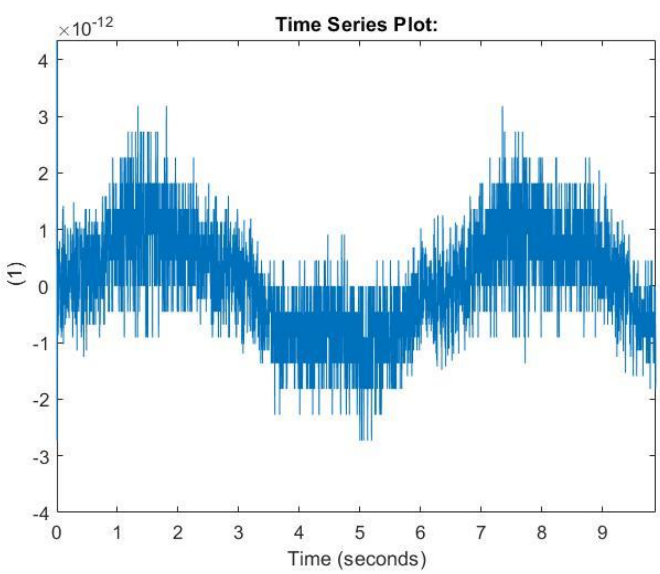
```
function D = fcn(a,b,c)

sigma_y = 235e+6;        %Minimum Yield Strength
n = 2.4;                 %Factor of Safety
Tau_max = sigma_y/(2*n);
F_eq = sqrt(a^2 + b^2 + c^2);
D = sqrt((2*F_eq)/(pi*Tau_max))*1000;

end
```



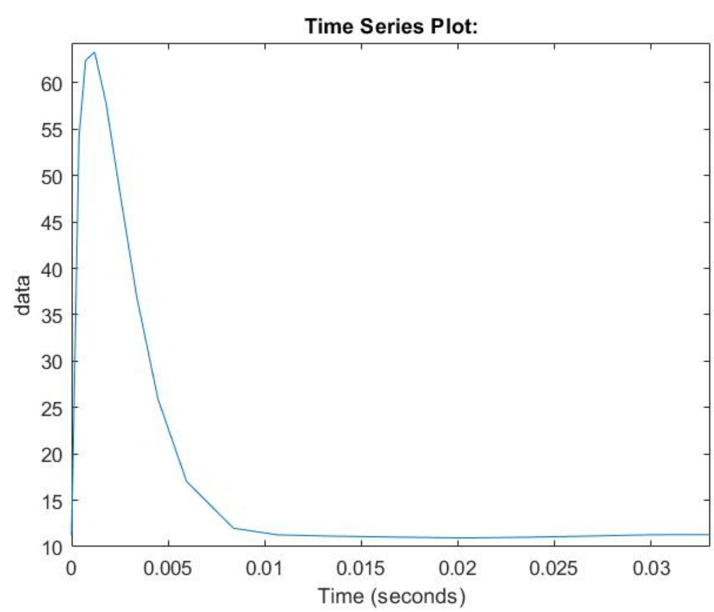
Force on the x-axis at the revolute joint



Force on the y-axis at the revolute joint



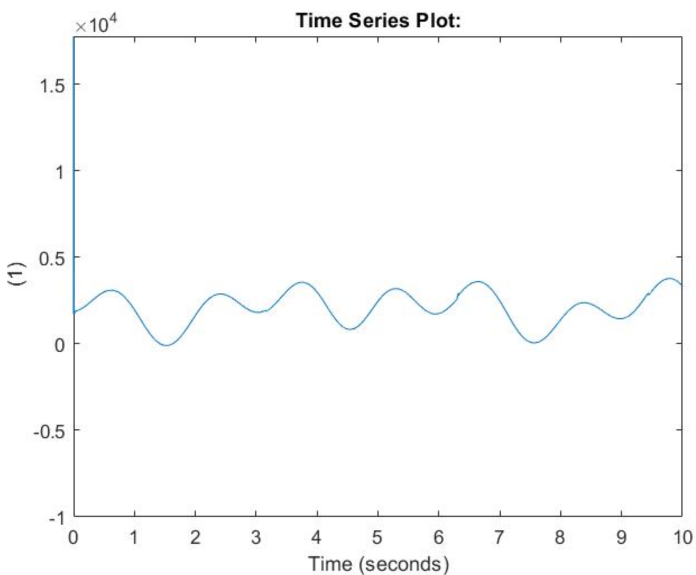
Force on the z-axis at the revolute joint



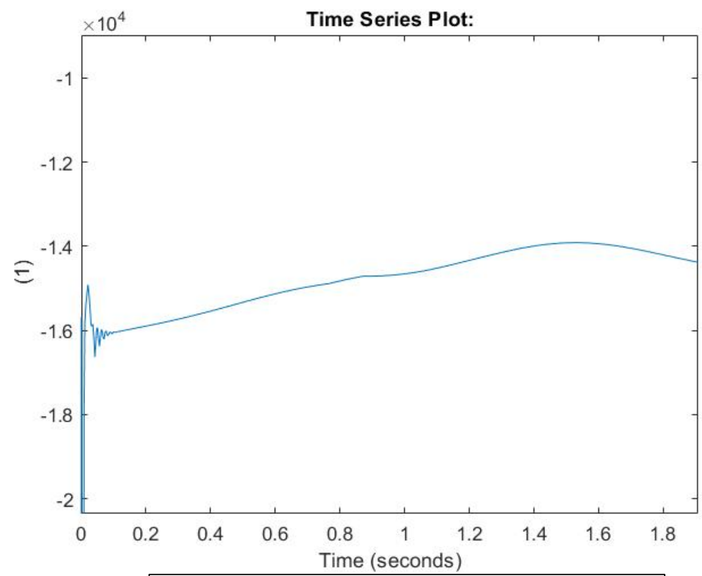
Diameter in terms of the Force acting (Simout)

Diameter Calculation in terms of Moment on the Revolute Joint:

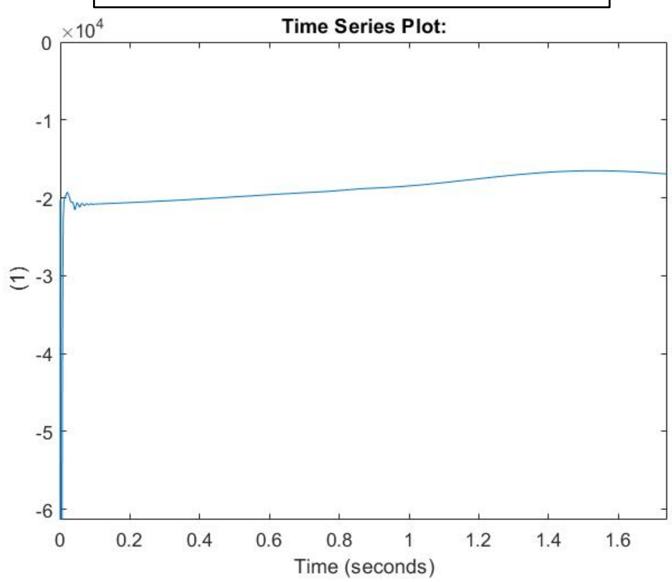
```
function d = fcn(T_x,T_y,T_z)

T = abs(T_z);
n = 2.4;                %Factor of Safety
sigma_y = 235e+6;       %Minimum Yield Strength
Tau_max = sigma_y / n ;
d = 1000*(((16*T)/(pi*Tau_max))^(1/3));

end
```



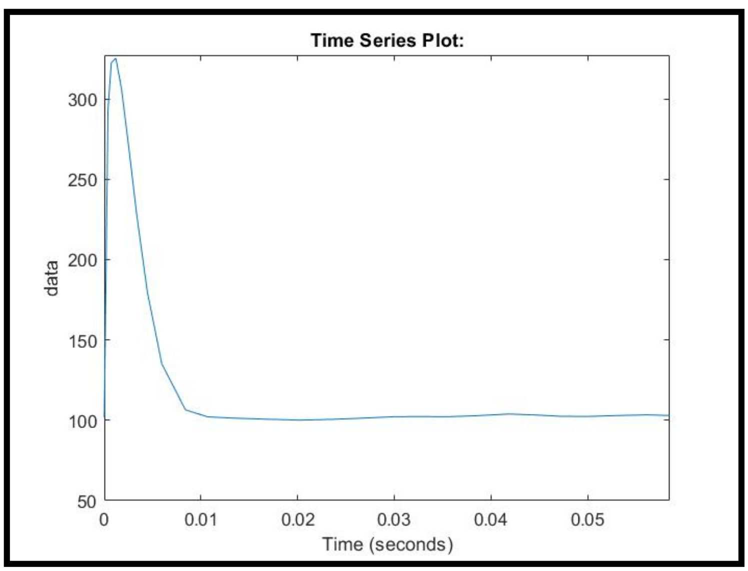
Moment on the x-axis at the revolute



Moment on the y-axis at the revolute



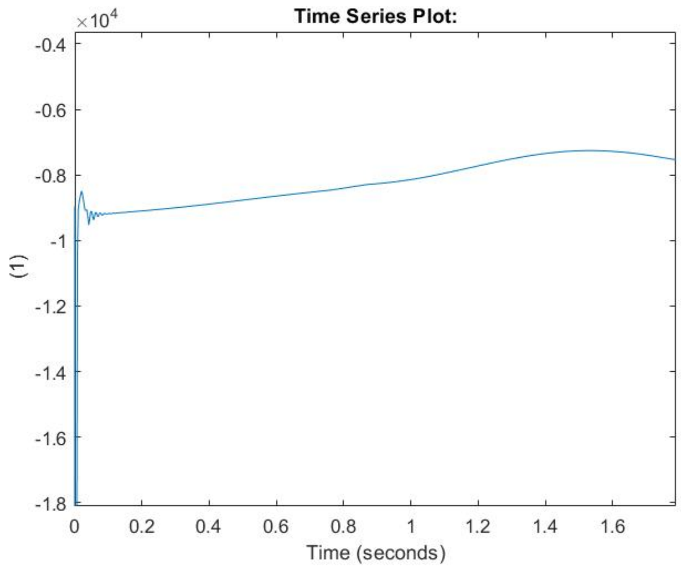
Moment on the z-axis at the revolute



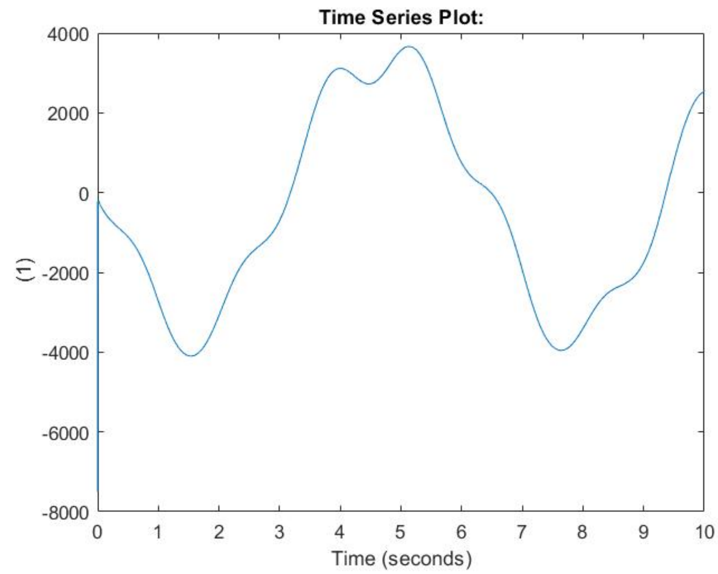
Diameter in terms of the Torque acting (Simout1)

Diameter Calculation in terms of Force on the Mass Joint:

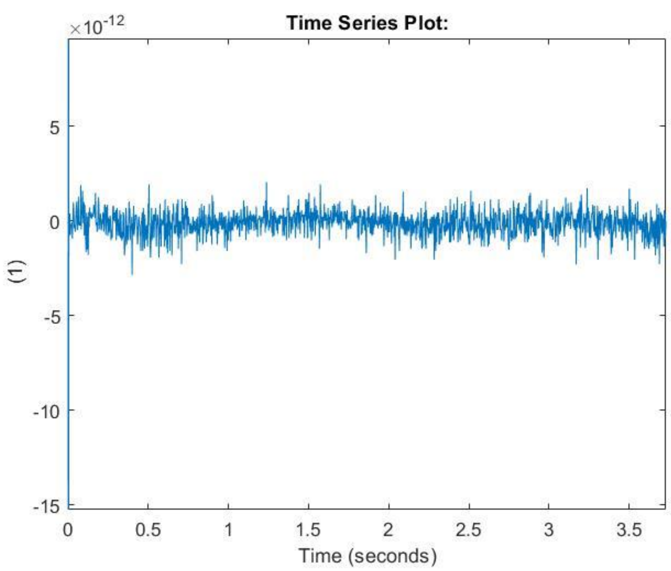
```
function dia = fcn(x,y,z)

sigma_y = 235e+6;        %Minimum Yield Strength
n = 2.4;                 %Factor of Safety
Tau_max = sigma_y/(2*n);
P_mg = sqrt(x^2 + y^2 + z^2);
dia = sqrt((2*P_mg)/(pi*Tau_max))*1000;

end
```



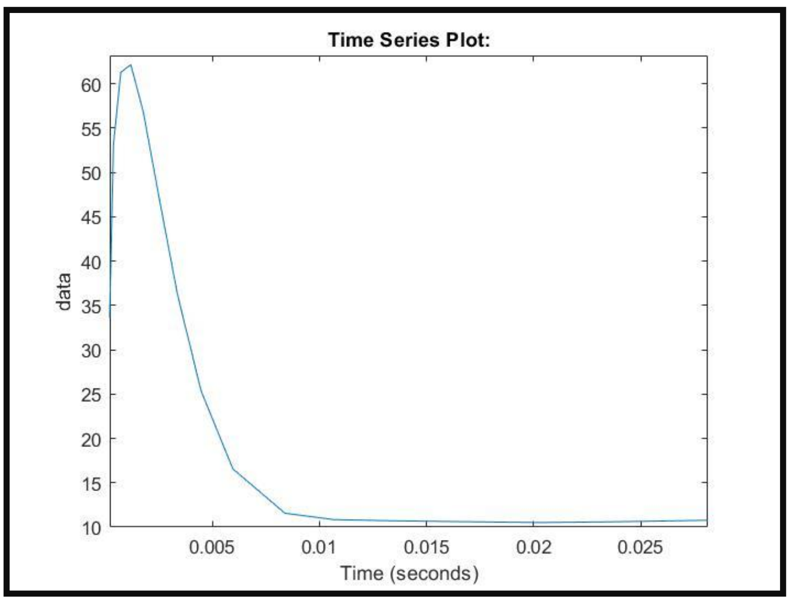
Force on the x-axis at the revolute joint



Force on the y-axis at the revolute joint



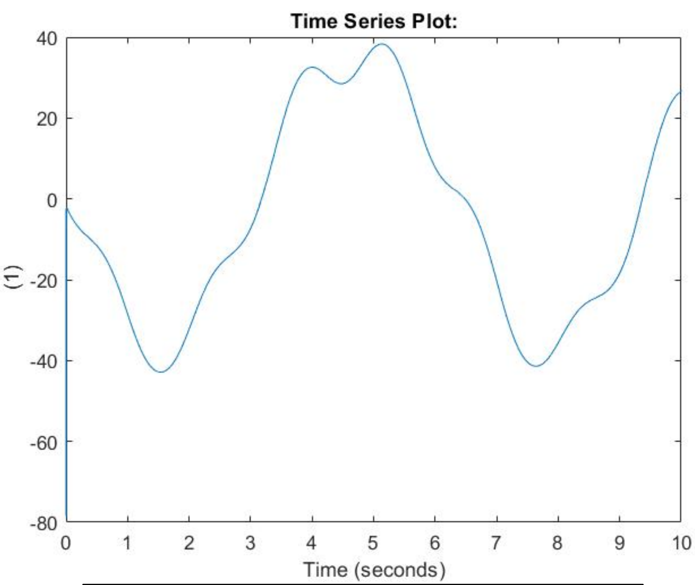
Force on the z-axis at the revolute joint



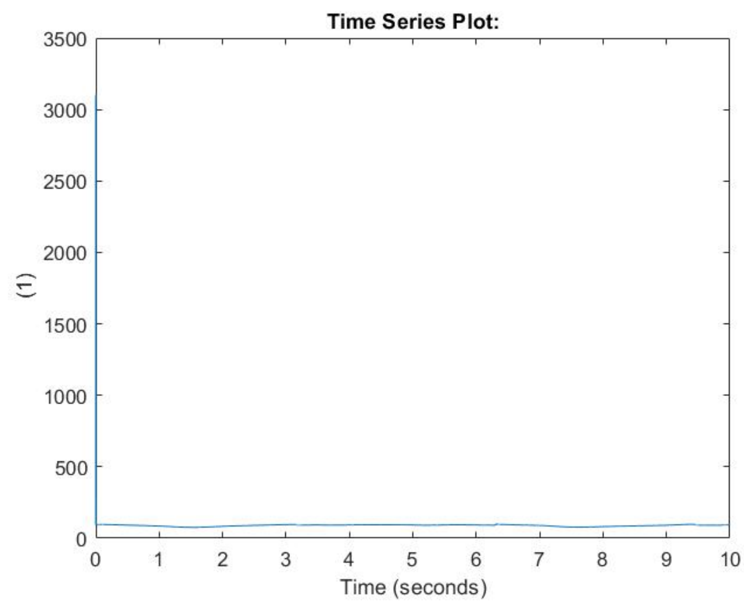
Diameter in terms of the Force acting (Simout2)

Diameter Calculation in terms of Moment on the Mass Joint:

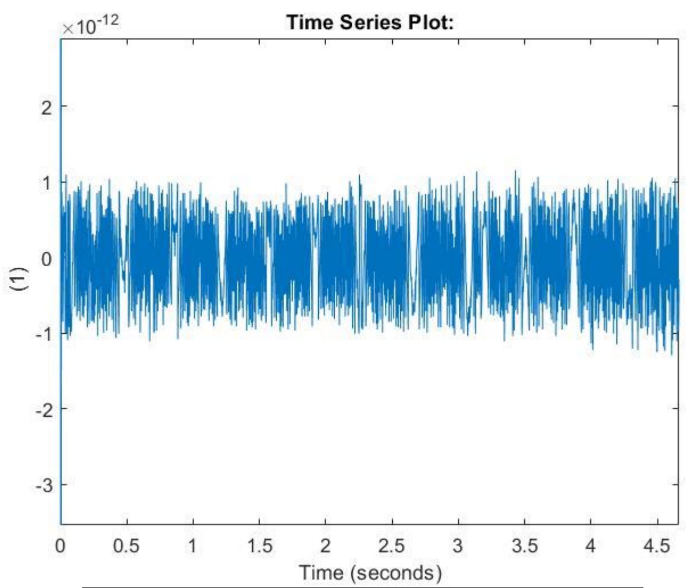
```
function b = fcn(Mx,My,Mz)

T = abs(My);
n = 2.4;                %Factor of Safety
sigma_y = 235e+6;       %Minimum Yield Strength
Tau_max = sigma_y / n ;
b = 1000*(((16*T)/(pi*Tau_max))^(1/3));

end
```



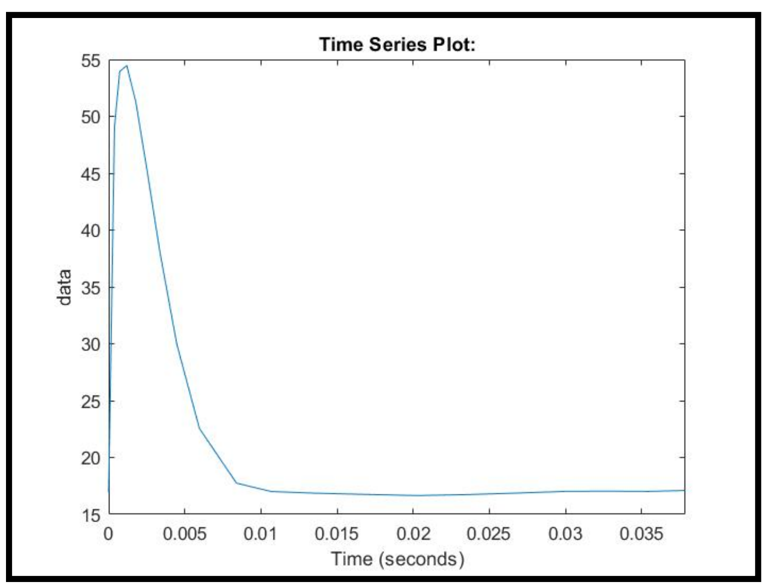
Moment on the x-axis at the revolute joint



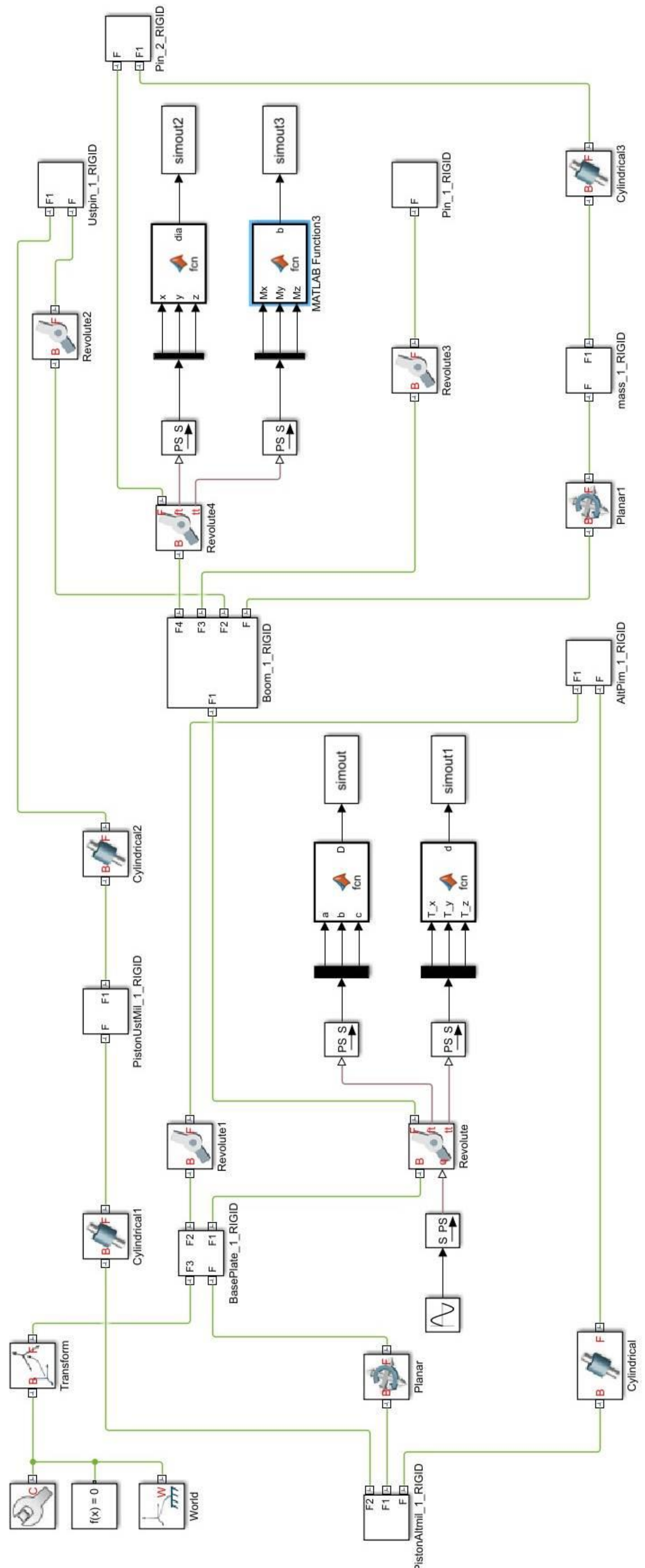
Moment on the y-axis at the revolute joint



Moment on the z-axis at the revolute joint



Diameter in terms of the Torque acting (Simout3)


Pin_2_RIGID
Ustpin_1_RIGID
simout2
simout3
MATLAB Function3
Pin_1_RIGID
Cylindrical3
Revolute2
Revolute3
mass_1_RIGID
Revolute4
Planar1
Boom_1_RIGID
AltPim_1_RIGID
simout
simout1
Cylindrical2
PistonUstMil_1_RIGID
Revolute1
Revolute
Cylindrical1
BasePlate_1_RIGID
Transform
Planar
Cylindrical
f(x) = 0
World
PistonAltmil_1_RIGID

After the Revolute joint that is the acting to boom is finding, I chose the second order filter at the Input Handling in the S-PS Converter Block since the second order filter provides the first and second derivatives. Then, I logically made a decision amplitude of the Sine Wave Block. Before the Matlab Function is constituted, PS-S Converter Block and Demultiplexer are added. The Simulink-PS Converter block converts the input Simulink® signal into a physical signal and now that the 3-axis *(x,y,z)*, Demux is used. Calculations that is the required are done and the dynamic analyze is scanned by using To Workspace and repeated this steps. I also decided how can be points to take into consideration of the material selection. Some points are the strength, hardness, ductility and like these parameters. When we consider these parameters, I decided St-37 since it has the high Yield Strength, easier machinability with respect to other same class materials, cost etc…

REFERENCES

[1] Design and Development of Portable Crane in Production Workshop: Case Study in BISHOPTU AUTOMOTIVE INDUSTRY, Ethiopia