GANTRY LOADER FOR MILLING MACHINE (GLMM)

A project submitted by

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1.Problem statement:

It is required to design a gantry loader system for CNC milling machine for picking the work piece from a specified location and place it on the milling machine table/vice. The weight carrying capacity of the loader should be minimum 3 kg. A vertical lift of 1500 mm, horizontal travel of 1000 mm is essential in the working envelope. It is required to design the mechanism for picking the work piece from the pickup location and transfer it to the work table. The design should clearly provide the mechanism implemented, design calculations for strength/dimensions for each link/component and justify selection of any off-the-shelf component like bolts, bearings, bushes, motors, couplings etc. Also you can select the gripper/end effectors of our choice but we need to clearly specify its features and limitations.

2. Overview:

The Gantry Loader – Milling Machine is a semi-automatic system responsible for load and unloads the job at a specific height using the combinations of control systems and sliding mechanism. GLMM provides the near accuracy solution for pick and place in bi-axial applications.

3. Working:

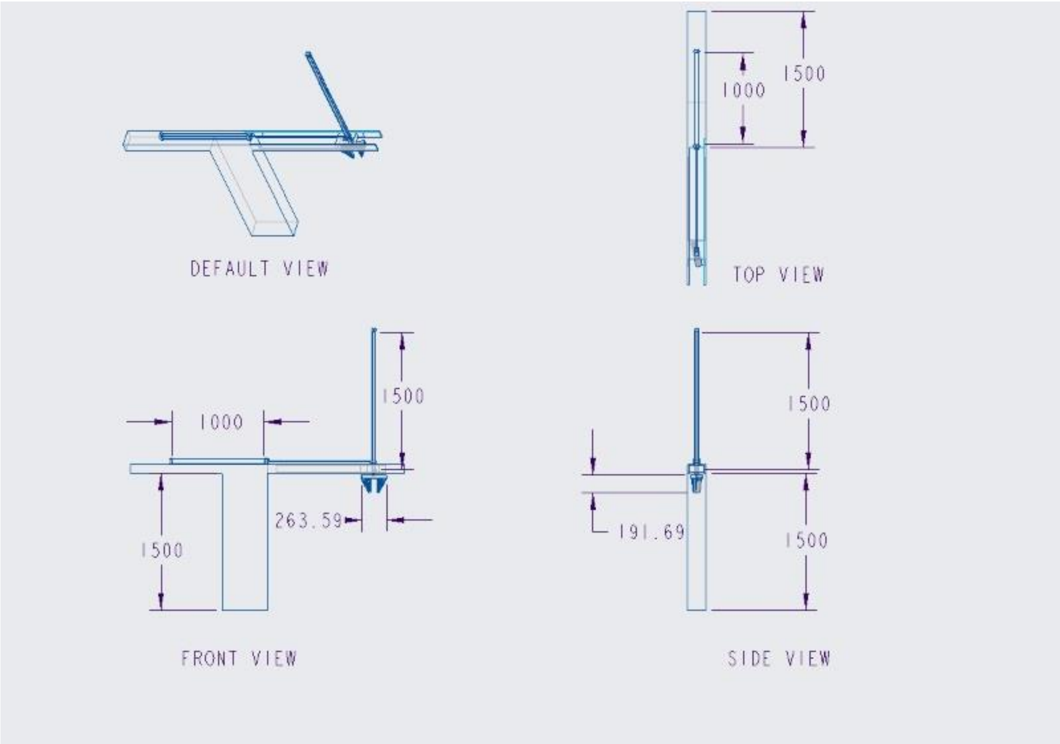
- ❖ GLMM mainly involves a body frame which supports the sliding plate.
- ❖ The movement of the system will be in two axes (horizontal and vertical).
- ❖ The mechanism used to achieve the required motion is pneumatic actuators.
- ❖ The body frame holds the sliding plate at a vertical distance of 1500mm and also the sliding plate in turn holds the pneumatic actuator to pick up the load at certain distance.
- ❖ The translation motion of the sliding plate between loading and unloading of the work part is achieved by another pneumatic actuator which is horizontally mounted at a required distance to cover the basic motion of 1000mm.

❖ These two single rod pneumatic actuators are responsible for picking up the work part and sliding across to the required position.
❖ The vertically mounted pneumatic cylinder is attached to the adaptive gripper to hold the work part, which is separately controlled by motor.

4. Material Selection:

❖ Material of the sliding plate that carries actuator along with gripper in unloaded condition and actuator along with gripper holding job of minimum 3kg weight in loaded condition, requires reasonable ductile property to make an axial movement by withholding the entire load. Mild steel is a desired material for this application since it has good ductility to withstand tension.
❖ The frame of the GLMM that carries the actuator system along with sliding plate of respectable amount of weight, should also able to withstand over all tension and bending. Since mild steel is good with withstanding tension by having reasonable yield strength and its nature of machinability and weldability makes it suitable for body frame of GLMM.

5. Orthographic View:



6. Catalogues used in design calculations:

- We have selected NC9 & PA-2 pneumatic actuators and DHAS adaptive grippers for design calculations.
- The required catalogue for calculations is providing below for reference.

Catalogue data for Pneumatic Cylinders:

Push and Pull Force in Pounds

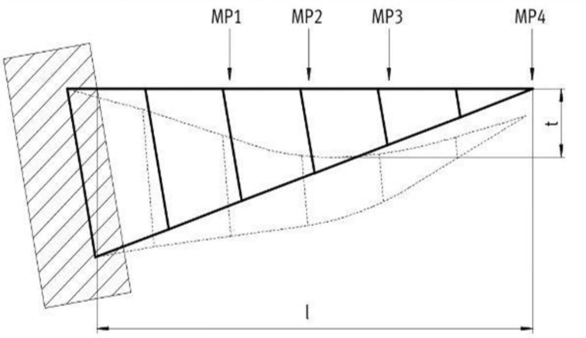
Bore Ø	Rod Ø	Operating Direction	Piston Area (inches^2)	Operating Pressure in psi		
				60	80	100
1 1/2	5/8	Push	1.767	106	141	177
		Pull	1.460	88	117	146
2	5/8	Push	3.142	189	251	314
		Pull	2.835	170	227	284
	1	Push	3.142	189	251	–
		Pull	2.357	141	189	–
2 1/2	5/8	Push	4.909	295	393	491
		Pull	4.602	276	368	460
	1	Push	4.909	295	393	491
		Pull	4.124	247	330	412
3 1/4	1	Push	8.296	498	664	830
		Pull	7.511	451	601	751
	1 3/8	Push	8.296	498	664	830
		Pull	6.811	409	545	681
	1 3/4	Push	8.296	498	664	830
		Pull	5.891	353	471	589
4	1	Push	12.566	754	1005	1257
		Pull	11.781	707	942	1178
	1 3/8	Push	12.566	754	1005	1257
		Pull	11.081	665	886	1108
	1 3/4	Push	12.566	754	1005	–
		Pull	10.161	610	813	–

Table A – NC9 or PA-2 Series Cylinder with Rod Lock Weights in Pounds

Bore Ø	Rod Ø	Single Rod Cylinders Basic Weight - Zero Stroke		Add Per Inch of Stroke	Double Rod Cylinders Basic Weight - Zero Stroke		Add Per Inch of Stroke
		MX0, MX1, MX2, MX3, MF1, MF2, MS4	MP1, MPU3, MS2, MT1, MT2, MT4		MDX0, MDX1, MDX3, MDF1, MDS4	MDS2, MDT1, MDT4	
1 1/2	5/8	6.0	6.6	0.30	6.5	7.1	0.60
2	5/8	10.0	10.4	0.50	11.7	12.1	1.00
	1	10.5	11.0	0.65	12.5	13.0	1.30
2 1/2	5/8	13.6	14.3	0.60	16.0	16.7	1.20
	1	14.1	14.6	0.75	16.6	17.1	1.50
3 1/4	1	24.6	25.6	0.80	30.1	31.1	1.60
	1 3/8	25.1	26.1	1.00	30.6	31.6	2.00
	1 3/4	25.9	26.9	1.25	32.2	33.2	2.50
4	1	38.7	43.7	1.00	45.7	50.7	2.00
	1 3/8	39.2	44.2	1.20	46.2	51.2	2.50
	1 3/4	40.0	45.0	1.50	47.8	52.8	3.00
5	1	56.3	63.3	1.10	65.3	72.3	2.20
	1 3/8	56.8	63.8	1.30	65.8	72.8	2.60
	1 3/4	57.6	64.6	1.55	67.4	74.4	3.10
6	1 3/8	104.8	113.8	1.50	116.8	125.8	3.00
	1 3/4	105.6	114.6	1.75	118.5	127.5	3.50
	2	106.3	115.3	2.00	120.0	129.0	4.00
8	1 3/8	158.4	163.4	2.00	172.4	177.4	4.00
	1 3/4	159.2	164.2	2.25	174.1	179.1	4.50
	2 1/2	161.7	166.7	3.00	179.1	184.1	6.00

Catalogue data for gripper:

Indentation depth t as a function of gripping force F_G (per gripper finger) at 23°C



MP1 Measuring point 1
MP2 Measuring point 2
MP3 Measuring point 3
MP4 Measuring point 4
l Total length
t Indentation depth

Workpieces are best gripped in the middle of the gripper finger (MP2).

The values may differ under other ambient conditions (additional information on request).

Size	l [mm]	MP1 [mm]	MP2 [mm]	MP3 [mm]	MP4 [mm]
60	50	15	25	35	50
80	80	30	40	50	80
120	115	47.5	57.5	67.5	115

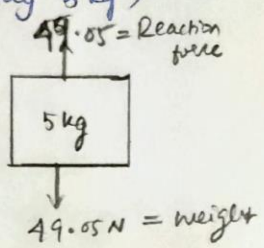
Size	Indentation depth at MP2 [mm]
60	12
80	20
120	30

7. Design Calculations:

① Pneumatic Actuator selection for loading operation ;

Required Conditions as,

- stroke(d) = 1500 mm (i.e. approx. 60 inches)
- Force required to lift the load (say 5 kg)



$\Rightarrow m = 5\ kg$

$d = 1500\ mm$

Consider, $\sum F_{net} = 0$

$\therefore W = m \times g = 5 \times 9.81 = 49.05 \simeq 50\ N$

$\therefore$ 50N of reaction force acts in opposite direction.

~~Now, 50N reaction force to 5~~

Now to overcome 50 N reaction force, we can assumption at 55 N of force to lift the job.

- From the catalogue,

to achieve 55N, we can select the standard push & pull of 106 pounds & 88 pounds

(i) 471.5 N and 391.4 N.

in accordance with above force values,

pressure = 60 psi = 4.13685 bar

Bore ϕ = $1 \times \frac{1}{2}$ inches = 38.1 mm

Rod ϕ = 5/8 inches = 15.875 mm

Piston area in push = 1.767 inches^2 = 1139.9 mm^2

Piston area in pull = 1.460 inches^2 = 941.93 mm^2

- Cylinder weight Calculation:

Based on bore and rod diameter, from Catalogue,

Basic weight of Cylinder at zero stroke = 6.0 pounds

pounds per inch of Stroke } = 0.30 pounds/inch

∴ Total Stroke length × per inch of Stroke } = 60 × 0.30

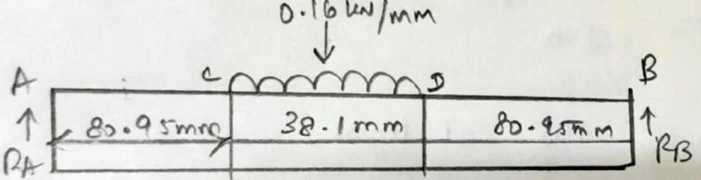
= 18 pounds.

∴ Total weight of cylinder } = 6 + 18

= 24 pounds

i.e) = 11 kgs.

(2) Structural Analysis of Sliding Plate under load :



from the figure above,

lets Consider $\Sigma F_V = 0$

$\Rightarrow R_A - 6.096 + R_B = 0$

$R_A + R_B = 6.096N \rightarrow (1)$

Also,

lets consider, $\Sigma M_A = 0$

$\Rightarrow (R_A \times 200) - (6.096 \times 100) = 0$

$(R_A \times 200) = 609.6$

$\boxed{R_A = 3.048 kN}$

(2)

Now substitute R_A in (1),

$\Rightarrow R_B = 6.096 - 3.048$

$\boxed{R_B = 3.048 \text{ kN}}$

For bending moment,

lets take,

Moment at D, $M_D = R_B \times 80.95$

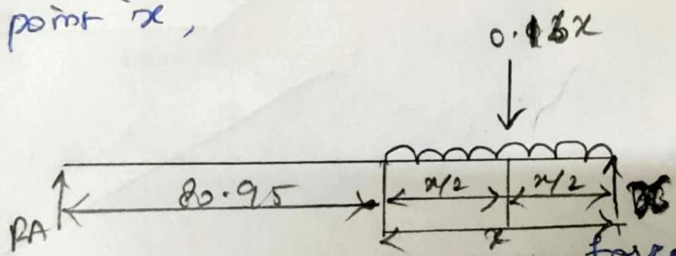
$= 3.048 \times 80.95$

$M_D = 246.7$ kN-mm

Moment at C, $M_C = R_A \times 80.95$

$M_C = 246.7$ kN-mm.

Now to find the maximum bending moment at the point 'x',



from the above diagram, ~~moment~~ forces at x,

$R_A - 0.16x = 0$

$3.048 - 0.16x = 0$

$x = \frac{3.048}{0.16}$

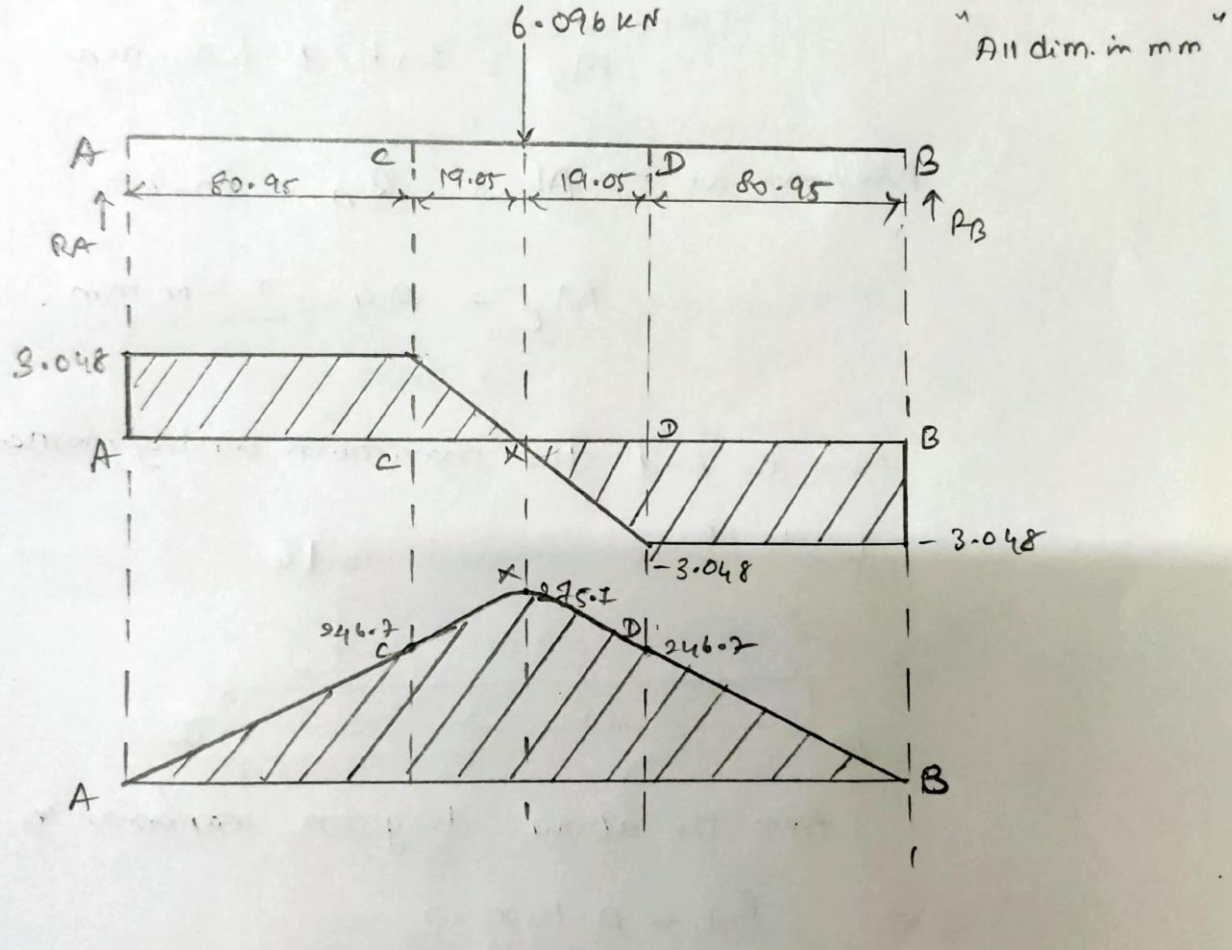
$x = 19.05$

Moment at $x \Rightarrow (R_A \times 100) - (3.048 \times 9.52)$

$= (3.048 \times 100) - (3.048 \times 9.52)$

$(\text{Max. B.M}) = 275.78 \text{ KN.mm}$

SFD and BMD :-



③

In the sliding plate,

$$\text{working stress} = \frac{F}{A}$$

$$= \frac{(16 \times 9.81)}{4000} \qquad \left[\because F = P \times g \right.$$
$$A = b \times t = 200 \times 20\,mm$$

$$= \frac{156.96\,N}{4\,m}$$

$$= 39.24\,N/m^2$$

(or)

$$= 0.03924\,N/mm^2.$$

$$\left.\begin{array}{l}\text{Yield strength of}\\ \text{Mild steel}\end{array}\right\} = 370\,N/mm^2$$

(or)

$$37 \times 10^7\,N/m^2$$

Also, lets assume factor of safety = 2,

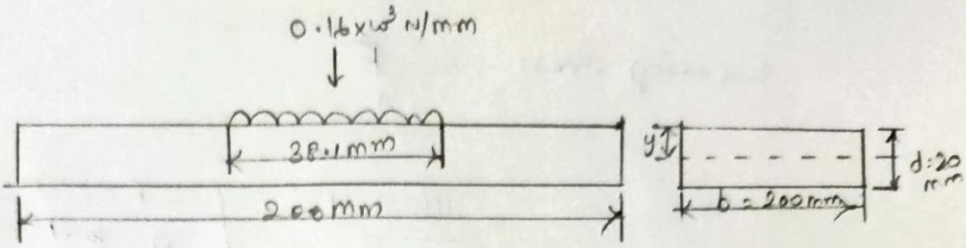
$$\text{then, the } \left.\begin{array}{l}\text{max. working}\\ \text{Stress}\end{array}\right\} = \frac{370}{2}$$

$$= 185\,N/mm^2$$

$\therefore$ Max. working stress < Yield strength

$\therefore$ The design is safe.

Bending stress :-



flexural equation, $\frac{M}{I} = \frac{F}{Y} = \frac{E}{R}$

Here M (Bending moment) $= \frac{WL}{4}$

$$= \frac{6096 \times 200}{4}$$

$$M = 30.4 \times 10^4 \text{ N-mm}$$

and, $I = \frac{bd^3}{12} = \frac{200 \times (20)^3}{12} = 13.33 \times 10^4 \text{ mm}^4.$

from,

$$\frac{M}{I} = \frac{F}{Y}$$

here, F is bending stress

$$F = \frac{30.4 \times 10^4 \times 10}{13.3 \times 10^4}$$

$\therefore F = 22.86 \text{ N/mm}^2.$

Here, bending stress < yield strength

$\therefore$ The sliding plate return to its original state after removal of load.

(4)

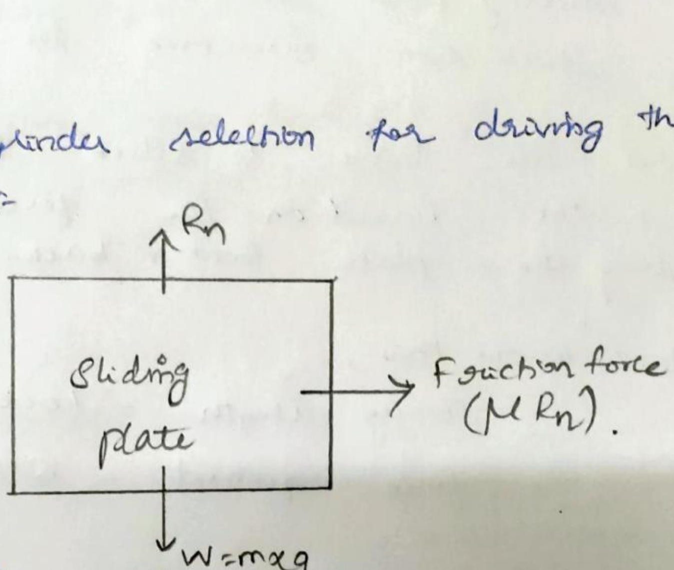
Maximum deflection $y = \dfrac{WL^3}{48EI}$

$$= \frac{6096 \times (200)^3}{48 \times 2.1 \times 10^5 \times 1.3 \times 10^8}$$

$$= 3.72 \times 10^{-4} \text{ mm}$$

$= 0.000372$ mm deflection takes place under elastic region.

(4) Pneumatic cylinder selection for driving the sliding plate :-



Here to find weight,

mass = Vol. × Density

$= (200 \times 200 \times 20)^3 \times (7.840 \times 10^{-6})$

mass (of sliding plate) $= 6.24$ kg.

Total Weight = (mass of sliding plate + mass of cylinder with loading) × 9.81

$= (6.24 + 16) \times 9.81$

$= 218$ N

Frictional force involved,

$$F_f = \mu \times R_N$$
$$= 0.6 \times 218 \quad (\because 0.6 \text{ is } \mu \text{ for dynamic})$$
$$= 130.8 \text{ N}$$

$\therefore$ The total force required to move the sliding plate front and back will be 218N, which can overcome the frictional force of 130N.

Now, we have to select the pneumatic actuator based on the force required to drag the plate front or back.

We know that,

Stroke length = 1000 mm

force required = 218N (W≅22 kg)

from the catalogue,

we can select the standard push and pull of 189 pounds and 170 pounds.

It gives, pressure = 60 psi = 4.13685 bar

Bore ϕ = 2 inches = 50.8 mm

Rod ϕ = 5/8 inches = 15.875 mm

Piston area in push = 3.142 inches^2 = 2027 mm^2

piston area in pull = 2.835 inches^2 = 1829.1 mm^2

Pneumatic cylinder mass calculation :- (5)

Based on the bore and rod diameter of the Cylinder, from catalogue,

Basic weight of Cylinder under Zero stroke } = 10-0 pounds

Basic weight of cylinder under inches per stroke } = 0-50 pounds/inch

Total stroke length × pounds per inch of stroke } = 60 × 0.50

= 30 pounds.

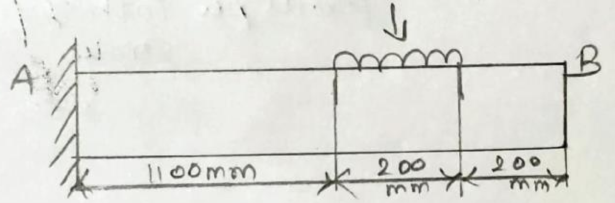
∴ Total weight of cylinder } = 10+30

= 40 pounds (or)

= 18.1437 kg.

(5) Since the load acting on both the sides of the frame of the structure, there will be a bending stress that leads to bending on both sides of frame. Therefore inorder to analyze the horizontal frame supported by vertical beam, the entire structure-member is divided into 2 halfs as cantilever beams to analyze the maximum bending stress on each halfs.

1st half (to the right) :-



lets $\sum F_y = 0$

$\Rightarrow R_A - 74(kN) = 0$

$R_A = 74$ kN

let, Bending moment at B = 0

B.M at D = 0 (since no force)

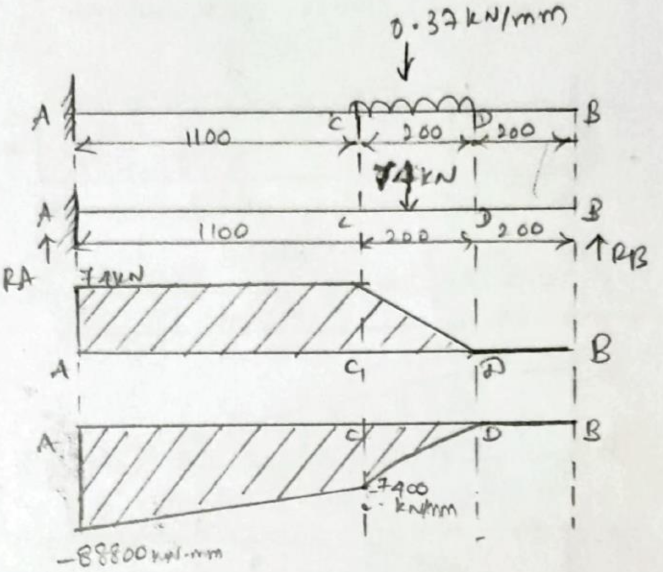
B.M at C = $-(74 \times 100)$

$= -7400$ kN-mm

B.M at A = $-(74 \times 1200)$

$= -88800$ kN mm

SFD & BMD :



"All dim. in mm"

for bending stress (F)

we can use $\frac{M}{I} = \frac{F}{Y}$

Here $I = \frac{bd^3}{12}$

$= \frac{200 \times (20)^3}{12} = 13.3 \times 10^4 \ mm^4$

Then

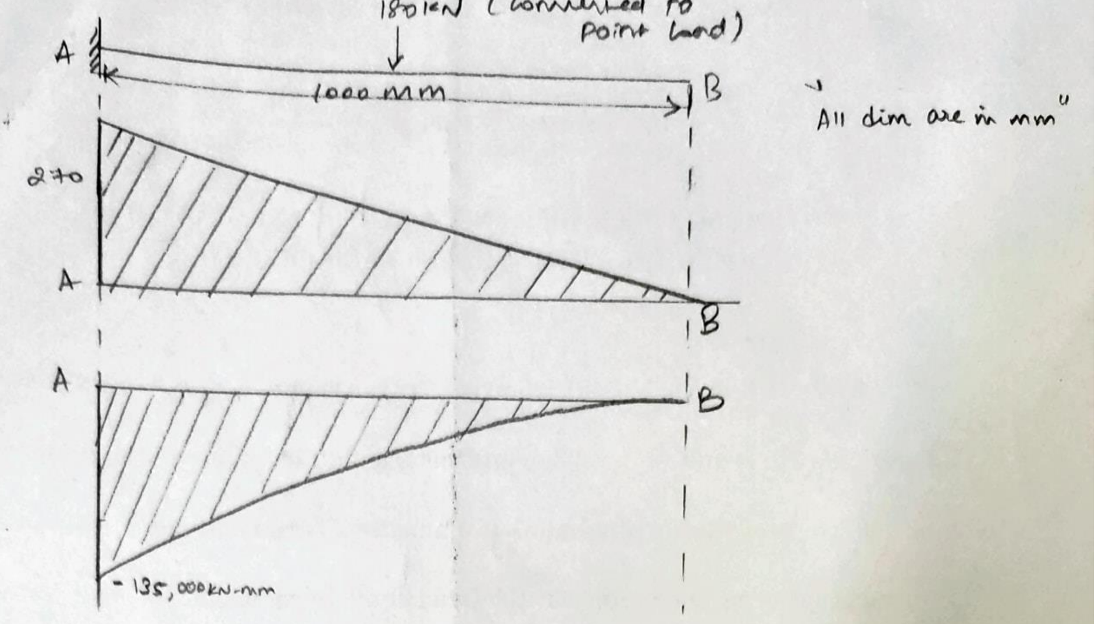
$F = \frac{M \times Y}{I} = \frac{88800 \times 10^3 \times 10}{13.3 \times 10^4}$

$F = 6.67 \times 10^3 \ N/mm^2$

$\therefore \ 6.67 \times 10^3 < 250 \ Mpa$ (yield strength of Mild steel)

$\therefore$ The bending on right side of frame due to loading is under elastic condition.

2nd half of frame (to the left)



Total length of span (beam) = 1000 mm (from support)

Total distance covered by the cylinder on the top (horizontal mounted) = 1500 mm.

Since, we are considering as a cantilever beam, we are taking the span length of 1000 mm and the weight distribution of 1500 mm in that 1000mm length..

from the figure, we know that $R_B = 0$,

consider $\Sigma F_y = 0$,

$\therefore \Rightarrow R_A - 270 = 0 \Rightarrow R_A = 270\ kN$.

Bending moment at $B = 0$

and Bending moment at $A = -(270 \times 500)$

$= -135,000\ kN\text{-}mm$.

Also

$I = \frac{bd^3}{12} = 13.3 \times 10^4\ mm^4$ (from previous beam)

$\frac{M}{I} = F/y \Rightarrow F = (135,000 \times 10^3 \times 10)/13.3 \times 10^4$

$\Rightarrow F = 10150.37\ N/mm^2$

This bending stress (F) is less than yield strength of mild steel (250 Mpa) $\therefore$ Design is safe.

⑥ Gripper Analysis :-

from the catalogues we take Jaw length = 80 mm

i.e. ≅ 3 inches

Mass of Job to be lifted = 5 kg (assumption)

Mass in pounds = 11.023 pounds

≃ 11 pounds

Since it is a adaptive gripper, it doesnot depend on friction, hence it is a encompassing gripper type. ∴ $\mu = 1$.

Also, Jaw style factor = 1
(encompassing gripper)

Also, part $G_s = \frac{11}{9.75}$

$G_s = 1.13$ (because gravity plays a part during lifting)

(i) Gripping force required (F_g) = part weight × (1 + part G_s) × Jaw style factor

= 11 × (1 + 1.13) × 1

(F_g) = 23.43 pounds

(ii) Jaw torque = Jaw length × Grip force

= 3 × 23.43 pounds

= 70.29 in pounds

(iii) part torque (Horizontal) } = Jaw length × part weight × Acceleration

$= 3 \times 11 \times 1.13 = 37.29$ in pounds

part torque (vertical) } = Jaw length × part weight × (1 + Acceleration).

$= 3 \times 11 \times 2.13 = 70.29$ in pounds

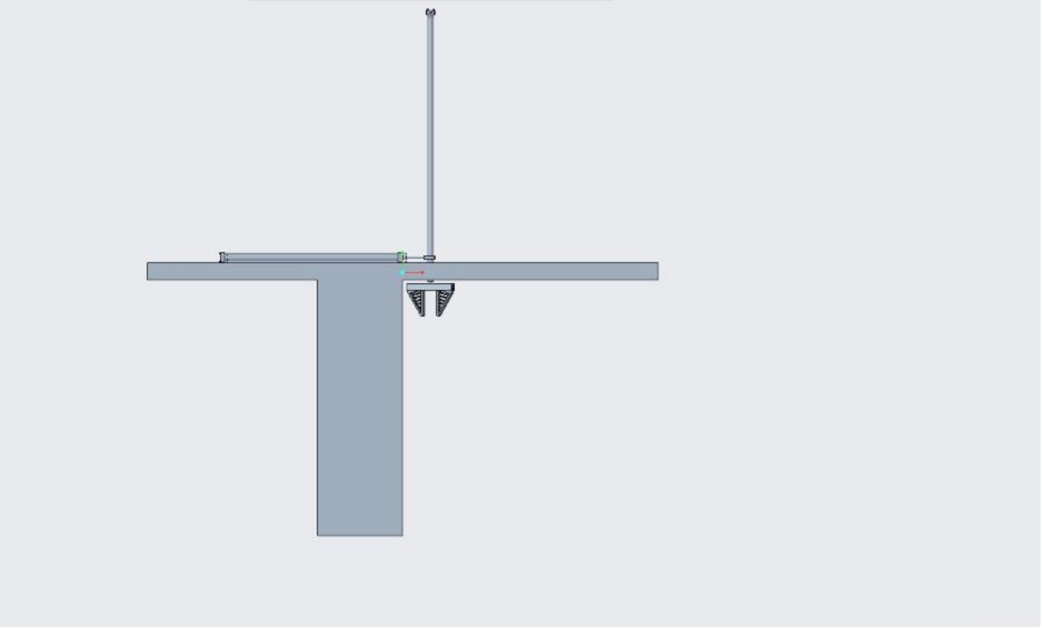
(iv) Total torque = Jaw torque + part torque

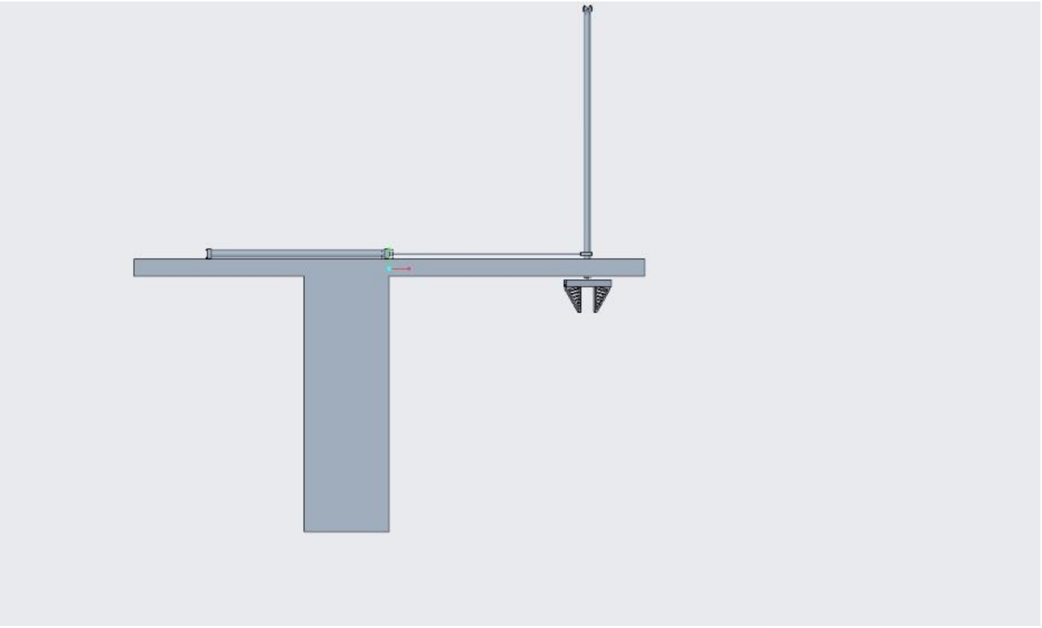
$= 70.29 + (37.29 + 70.29)$

$= 177.87$ in pounds $= (20$ N-m$)$

(v) specification ⇒ 23.43 ≃ 24 pounds of gripper force and 177.87 ≃ 178 in pounds of torque.

8. CAD Design:





9. Conclusion:

Gantry Loader for Milling Machine utilizes the combination of control systems with end effectors and precise dimensional properties of parts to transfer the work piece from pick-up location, till it unloads at the work table. Among various approachable solutions, we have selected a concept that emphasizes the minimal cost cutting factor along with reasonable automation in the work environment.