MATERIAL HANDLING SYSTEM – A CASE STUDY FOR SMALL SCALE INDUSTRY

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ABSTRACT

To protract in the today's globalization world manufactures have to develop innovative expertise with help of available resources so as to swell efficiency of the Organization. This paper discuss the Design and Development of an material handling system for a Pipe Industry.

***Keywords*: Cycle Time, Gantry Crane, Movable Support, Productivity, Small Scale Industry, Storage Rack**

I. INTRODUCTION

Material handling is the movement, protection, storage and control of materials and products throughout manufacturing, warehousing, distribution, consumption and disposal.In small scale industries the material handling is generally done manually. Regarding pipes handling is very difficult. The precaution is taken that there is no damage to the pipes. Whenever, the specific pipe is required for the manufacturing of product then that pipes is carried by labors up to the cutting machine. Due to the manual material handling a lots of time is consummated. So the cycle time of production increases which decreases productivity of the plant, also increases cost per unit product.

II. PROBLEM STATEMENT

The problem faced by Selected Pipe Industry in material handling is time and labor consumption. We had notice that around 3 to 4 hours are required to unloading the truck as well as 6 labors are required for manual transfer of pipes. As no of worker engaged in transferring the raw material towards cutting machine is large and also the time for transferring is more. It results into more working cost. To minimize this cost & time we are going to Design the Material Handling system.

III. METHODOLOGY TO ADDRESS PROBLEM

3.1 Industrial Survey of Manufacturing Plant

The survey consists of total consumption per month of raw material in the plant. Also getting information about the various types of pipes required for manufacturing, Depending upon the consumption of various types of pipes, the position of that pipes in the rack is decided.

3.2 Deciding Parameters of Production Design

Depending upon the human comfort and quick accesses of regularly required raw materials the design of rack and MHS is based. Maximum amount of raw material is stored into the rack.

3.3 Design of Actual System

By Calculating the load, moment which are acting on the system we had find out the actual dimension on the system

IV. INDUSTRIAL SURVEY

When raw material comes to plant by truck, two labors are required to unload the truck. These two people travels the distance of 15 m to store the pipe in storage rack whose position is fixed near the wall. To unload the Truck of 3 ton two people requires 3 hrs.When the pipe of particular size is required for manufacturing, two labors are required to carry the pipes from storage rack to machine through the distance of 15 m. After that two more labors are required for perfect aligning of pipes with pipe cutting machine. Total 1 hour is required to carry the pipes from storage rack up to the So the total 6 labors are required to unloading the pipes from the truck up to its loading to the machine.6 labors are required for the unloading and loading of the pipes they requires 4 hours to unload the 3 ton pipes per day. The salary of each worker is 200 /-daily that is half of the shift is going to waste on the material handling that is 600Rs are waste on material handling.The plant is semi automated only 14 workers are working in the plant. If out of these 14 workers, 6 workers are busy for loading and unloading. Then it also affects on production of the plant. so maximum 2 workers will be forwards the loading and unloading section. So it is desired to implement low budget material handling system in SSI.

V. DESIGN PROCEDURE FOR MATERIAL HANDLING SYSTEM

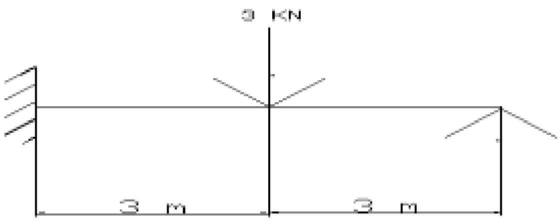

Fig. 1

The diagram shows the force acting on the single rack. We had total 7 racks and at the top, one gantry crane.

Total load $= (7 \times 3000\ N)$ + wt of material handling system

$= (7 \times 3000) + 2000N$ = =23,000N

The moment at the end of single rack is given as {Moment = $(\frac{3}{16}) \times W \times L$}[5]

where, W=3000N, L=6m

Moment of the each cantilever at the end of each rack, $=(\frac{3}{16}) \times W \times L$ $=(\frac{3}{16}) \times 3000 \times 6 = 3375$Nm

Total moment due to 7 rack is given as $7 \times 3375 = 23{,}625$Nm

Now, There is a moment due to material handling system.

When the load is at extreme condition and it is given as, moment = 3000×6=18,000Nm

Due to the weight of MHS the moment is given as,

wt. of material handling system =2000N

It is acted at the centre of span, & it is given as $=2000 \times 3$ =6000Nm

Total moment acting on the column is the summation of above mentioned equations

Total moment $=23{,}625 + 18{,}000 + 6000$=47625

Total moment =48,000Nm

VI. DESIGN OF COLUMN (TRIPOD SUPPORT) (MATERIAL =YST 310 GRADE)[10]

Consider the minimizing the weight instead of using single column .We use 3 independent column such as follows

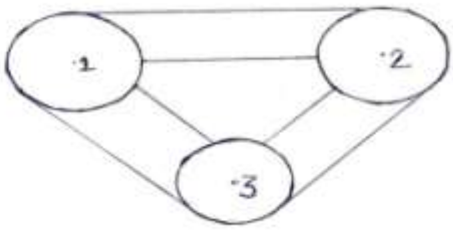

Fig.2

Column 1 ,2 125mm dia , Column 3 100mm dia

Cross section area of 3 triangle:- $18.5 + 22.8 + 22.8 = 64.1\ cm^2 = 64.1 \times 10^2\ mm^2$

Direct stress assuming f.o.s. of 1.5, axial stress , $f_a = \frac{23000 \times 1.5}{64.1 \times 10^2} = 5.38\ N/mm^2$

For bending Stress , $Z = \frac{bh^2}{24} = \frac{500 \times 353 \times 353}{24} = 2.59 \times 10^6\ mm^3$

$f_b = \frac{M}{z} = \frac{48000 \times 10^3 \times 1.5}{2.59} = 27.79\ N/mm^2$

For Slenderness ratio , $K = 0.236 \times h = 0.236 \times 353 = 83.308\ mm$

$\frac{l}{k} = \frac{3}{0.0833} = 35.95$, by interpolation, the allowable axial stress is given as,

$30 \rightarrow 178,\quad 40 \rightarrow 169$ For $35.95 \rightarrow 172.645$

$f_a = 172.645\ N/mm^2,\ \ f_b = 205\ N/mm^2$

Now for the check of safer design, $\frac{f_a}{F_a} + \frac{f_b}{F_b} < 1,\ \ \frac{5.38}{172.645} + \frac{27.79}{205} = 0.16 < 1,$ Design is safe.

VII. DESIGN OF SECTION (RECTANGULAR)[10][5]

choosing the cross section of (80×40)

Therefore properties of section are,

thickness =4mm, Z_{xx}=$16.20 cm^2 = 16.20 \times 10^3 mm^3$

Moment will be the the same, i.e. M=2250000 Nm

The induced stresses are calculated as, $\sigma_{max} = \frac{M}{z}$ = 138.88N/mm^2

Therefore the induced stresses are less than allowable stresses. Therefore design is safe. By selecting the cross section 80×40 for the rectangular section.

VIII. WELD DESIGN 1[6][8]

Welding design between circular section and rectangular section

Considering fillet weld

We provide circular angle at a distance of 1m. internal therefore load on each angle is 3000/6 = 500N.

By taking factor of safety as 2

Therefore ,total load $500 \times 2 = 1000N$

Primary stresses are given as, $\tau_1 = \frac{P}{A}$

by selecting the circular section of 15mm diameter

$\tau_1 = \frac{1000}{\pi \times 15 \times t} = 21.22/t$ N/mm^2

Now the moment of inertia of weld is given as

$I_{xx} = \pi \times t \times r^3 = \pi \times t \times 7.53 = 1325.35 \times t \quad mm^4$

Now bending stress is given as

$\sigma_b = \frac{M_b \times Y}{I_{xx}} = \frac{1000 \times 137.5 \times 7.5}{1325.35 \times t} = 778.09 / t$ N/mm^2

From the maximum shear stress theory

$\tau = \sqrt{(\sigma_b / 2)^2 + \tau_1{}^2} = \sqrt{((778.09/2 \times t)^2 + (21.22 / t)^2)} = \sqrt{\frac{151356.072}{t^2} + \frac{450.28}{t^2}} = 389.62/t$

N/mm^2

For fillet welds, Permissible shear stress =(95 N/mm^2)[4]

From equation 1

$95 = \frac{389.62}{t}$, t = 4.1 mm ,t = h cos450, h = $\frac{4.1}{cos45}$, h = 5.8 =6 mm

IX. WELD DESIGN 2 [6][8]

design between bush and rectangular lever[8]

The cross section of lever are 80×40

A = $2\,(80t + 40t)$

=240t

Primary shear stress

$\tau_1 = P/A = 3000/240t = 12.5/t$

$I_{xx} = t\,[\frac{bd^2}{2} + \frac{d^3}{6}] = t \times 74.666 \times 103$ mm^4

$\sigma_b = \frac{M_b \times Y}{I_{xx}} = \frac{3000 \times 3000 \times 20}{t \times 74.66 \times 103} = \frac{2.41 \times 10^3}{t}$

Now From the maximum shear stress theory

$\tau = \sqrt{(\sigma_b / 2)^2 + \tau_1{}^2} = \sqrt{(2.41 \times 10^3/2 \times t)2 + (12.5 / t)2} = 1205.06 / t$

Now, (τ =95 N/mm^2)[4] $95 = 1205.06 / t$, t = 12.68 mm, t = h cos450, h = 12.68/cos45,h = 17.94 mm

X. DESIGN OF PIN

By selecting the material as steel of C45[6],σ_{ut} = 600 N/mm2,τu = 300 N/mm^2

by using factor of safety 2

The pin is in shear and the load of 3000 N is acted on it

Shearing area =$(\pi / 4) \times d_1{}^2$

d_1 = diameter of pin, $p = (\pi / 4) \times d_1{}^2 \times \tau$

assuming the diameter of pin as 20mm, $3000 = \pi / 4 \times 20 \times \tau$, $\tau = 9.54$ N/mm^2

which is less than allowable stress there for design is safe

pin diameter = 20 mm, clearance = 3 mm on both side, internal diameter of bush + circular part

$= 20 + 2 \times 3$=26

Considering $d_2 = 1.5\ d_1$, d_2 = outer diameter of circular part, d_2 = 40 mm, d_1= 26 mm

XI. DESIGN OF CIRCULAR BUSH

By using theory of knuckle joint[9]

1.design for tension , $P = (d_2 - d_1) \times t \times \sigma_t$, $3000 = (40 - 26) \times 100 \times \sigma_t$, $\sigma_t = 2.14$ N/mm^2

induced stress is less than allowable stress

2. In shearing, $P = (d_2 - d_1) \times t \times \tau$, $3000 = (40 - 26) \times 100 \times \tau$, $\tau = 2.14$ N/mm^2

3.design in crushing, $P = d_1 \times t \times \sigma_c$, $3000 = 26 \times 100 \times \sigma_c$, $\sigma_c = 1.15$ N/mm^2

XII. DESIGN FOR CIRCULAR SUPPORT ON COLOUMN

1.For tension, $P = (d_2 - d_1) \times t \times \tau$, $3000 = 44 \times 60 \times \tau$, $\tau = 1.13$ N/mm^2,

2 .for shearing, $P = (d_2 - d_1) \times t \times \tau$, $3000 = 60 \times 600 \times \tau$, $\tau = 1.13$ N/mm^2

3 .for crushing$P = d_1 \times t \times \sigma_c$, $3000 = 60 \times 600 \times \sigma_c$, $\sigma_c = 0.18$ N/mm^2, design is safe

XII. DESIGN OF MOVABLE SUPPORT

As this is the case of propped cantilever the reaction of movable support

R=$\frac{5}{16} \times W$=$\frac{5}{16} \times 3000$=937.5N≈1000N

By taking the factor of safety as 1.5, The reaction at movable support is R=1.5×1000,R=1500N

By keeping the distance of Rest point 0.5M from column, Total Moment=15000$\times 0.5$ =750NM

R=1500N , M=750NM

XIV. DESIGN OF COLUMN

By selecting section 15mm inner diameter, 3.20mm thickness, Properties of section

A_{cs}=1.82$\times 10^2$ mm^2, Z=0.70$\times 10^3$ mm^3, Radius of gyration (k)=0.65cm

$f_a = \frac{1500}{1.82 \times 10^2}$=8.24N/mm^2

$f_b = \frac{M}{Z} = \frac{750}{0.70 \times 10^3}$=1.071N/mm^2

Now, $\frac{L}{K} = \frac{2000}{6.5}$=307.69

Now allowable direct stress for this ection

300→13, 350→10, By interpolation for 307.69=12.53

Check for design

$\frac{f_a}{F_a} + \frac{f_b}{F_b} < 1$

$= \frac{8.24}{12.53} + \frac{1.071}{205}$=0.66

Hence design is safe

By keeping the distance between column and screw as 0.25 M

The load on the screw is calculated as per theory of lever

W$\times l_1 = p \times l_2$

1500$\times 0.5$=p$\times 0.25$

P=3000N

Design of Screw and Bolt (Material Bolt→C.I), (Material Nut→hardened steel on C.I)[9]

Nominal Dia(d_1)=40mm, Minor Dia(d_c)=33mm

D_0=40mm, Pitch= 7mm, The screw has triple start thread

μ=tan Ø =0.15, Ø=11.30^0

Lead=no. of start $\times$ pitch

=3×27=21mm, A_c=855mm^2

d=$\frac{d_0+d_c}{02}$=$\frac{40+33}{2}$, D=36.5mm

Direct Stress =$\frac{W}{A_c}$=$\frac{3000}{855}$, σ_c=3.50N/mm^2

Tan α = $\frac{lead}{\pi \times d}$ = $\frac{21}{\pi \times 36.5}$, α=.02092

α=11.820, T=p$\times \frac{d}{2}$=W tan(α+Ø)$\frac{d}{2}$, T=3000 tan (11.82+11.30)$\frac{36.5}{2}$, T=23,375N-mm

For calculation of shear stress,

T=$\frac{\pi}{16} \times \tau \times d_c{}^3$, 23375=$\frac{\pi}{16} \times \tau \times 33^3$, τ =3.33N/mm^2

According to the maximum shear stress theory

τ_{max}=$\frac{1}{2}\sqrt{\sigma_c{}^2 + 4\tau^2}$ =$\frac{1}{2}\sqrt{3.50^2 + 4 \times 3.31^2}$= 3.74 N/mm^2

Ideal torque (T_0)=$W tan\alpha \times \frac{d}{2}$, T_0 = 11457 Nmm

Efficiency (η) =$\frac{T_0}{T}$=$\frac{11457}{23375}$=0.49, =49%

It is self locking screw

Dimension of nut

d_0=40.5 mm

Fig. 3

R_f = 3000 – 1500 = 1500 N

Design of fulcrum pin ,Given bearing pressure for the material 25 N/mm^2

d_p = dia of pin , l_p = length of pin

Bearing area = $d_p \times l_p$ ($l_p = 1.25 d_p$)

Load on fulcrum pin = bearing area $\times$ bearing pressure = 1.25$d_p^2 \times P_b$, 1500 = 31.25 $d_p{}^2$, d_p = 6.92 ≈8mm

checking pin for shearing, 1500 = $2 \times \frac{\pi}{4} \times d_p^2 \times \tau$, 1500 =$2 \times \frac{\pi}{4} \times 8^2 \times \tau$, τ = 14.92 N/mm^2

The value of induced stress is less than permissible value i.e. τ = 500N/mm^2,Hence design is safe

Thickness of bush = 2 mm

Total dia of hole = 12 + 2 $\times$ 2 = 16 mm, Moment = 750 Nm

We know that section modulus, Z = $\frac{1}{6} \times t \times b^2 \quad (b = 4t)$, Z = 2.67 t^3, $\sigma_b = 70\ N/mm^2$

$\sigma_b = \frac{M}{Z} = \frac{750}{2.67 \times t^3}$, 70 = $\frac{750}{2.67 \times t^3}$

t= 5 mm , b= 4t = 20 mm

shear stress induced in the lever

$\tau = \frac{3000}{20 \times 5} = \frac{3000}{100}$ = 30N/mm^2

Hence the Design is Safe.

14.1 Design of Material Handling Crane (Gantry Girder)[7]

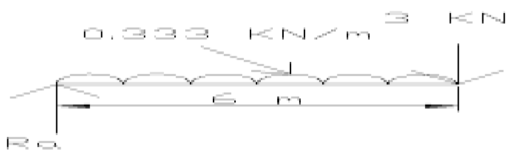

Fig. 4

Reaction of column = $(0.333 \times 6) + 3$

= 5 KN

Reaction on each wheel is $\frac{5}{2} = 2.5\ KN$

By taking section of side view of crane

Fig. 5

Bending moment is given as, B.M.$_1$ = $\frac{W \times l^2}{8} = \frac{0.25 \times 0.4^2}{8}$ =$5 \times 10^{-3}\ KNm$

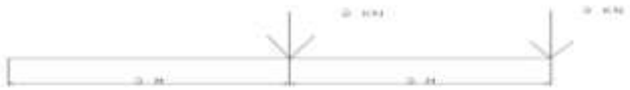

Fig. 6

B.M.$_2$ =6+18 = 24 KNm,B.M.$_2$> B.M.$_1$, Design is based on B.M.$_2$

Now calculating section modulus , for steel $\sigma_{bc} = 165\ N/mm$

Z=$\frac{M}{\sigma_{bc}}$=$\frac{24 \times 10^6}{165}$=145.45 $\times\ 10^3\ mm^3$

Depending upon this section modulus ,Selecting the I section ISMB 200 from steel table ,Properties of ISMB 200

1. Sectional area (a) = 32.33 cm^2
2. Thickness of web (t_w) = 5.7 mm
3. Thickness of flange (t_f) = 10. Mm
4. Moment of inertia (I_{xx}) = 235.4 $\times\ 10^4\ mm^4$

5. Moment of inertia (I_{yy}) = 150 $\times 10^4\, mm^4$

6. Width of flange = 100 mm

For selecting the 'C' channel , we have to consider width of the flange and keeping the 25 mm clearance on both side. By selecting the ISMC 150 channel

From steel table ,Properties of IMSC 150

1. Area of cross Section (a) = 2088 cm^2
2. Thickness of web (t_w) = 5.4 mm
3. Thickness of flange (t_f) = 9mm
4. Moment of inertia (I_{xx}) = 779.4 $\times 10^4\, mm^4$
5. Moment of inertia (I_{yy}) = 102.3 $\times 10^4\, mm^4$
6. Width of flange = 2.22 mm

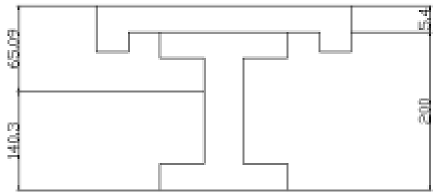

Fig. 7

To find out $\bar{Y}$ of built up section , $\bar{Y} = \frac{a_1 y_1 + a_2 y_2}{a_1 + a_2}$, $\bar{Y} = \frac{32.33 \times 105.4 + 2088 \times 2.7}{32.33 + 20.88}$, $\bar{Y} = 65.09mm$

Now moment of inertia about X-X axis , of total built up section

$I_{xx} = I_{xx} + a_1 h_1^2 + a_2 h_2^2 + I_{xx}$ = 2235.4 $\times 10^4 + 32.33 \times 40.31^2 + 102 \times 10^4 + 2088 \times 42.89^2$

=31.44 $\times 10^6\, mm^4$

$I_{yy} = I_{yy1} + I_{yy2}$ = 150 $\times 10^4 + 779.4 \times 10^4$ =9.294 $\times 10^6\, mm^4$

$\sigma_{bc} = \frac{M_x \bar{y}}{I_{xx}} = \frac{24 \times 10^6 \times 140.3}{31.44}$ = 1.1$\times$107.099 N/mm^2 =117.8089 N/mm^2. The design is safe

XV. CONCLUSION

By implementing the material handling system shown in Fig.8 we are minimized the cycle time of production and also reduced the no. of labors from 6 to 2 which indirectly increases productivity as well as profit of the organization. We are saving 66% manpower required for material handling and saving 89% of cost of material handling.

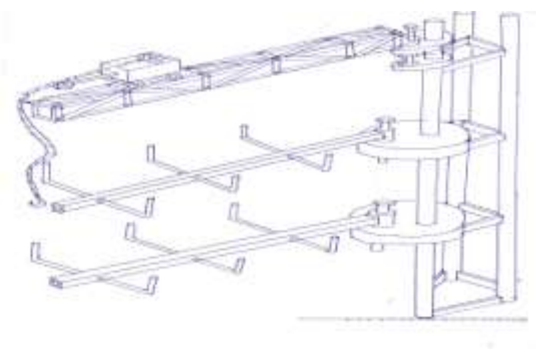

Fig.8 Material Handling System

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