DESIGN AND ANALYSIS OF CANTILEVER TYPE ROLLER CONVEYOR SYSTEM

[1]Jayesh Gaikwad, [2]Kunal Mhaske, [3]Nairu Rathod, [4]Vaibhav Tanpure, [*1]Sandeep Raut

[1]Under Graduate Scholar, [*1]Asistant Professor
[1]Mechanical Engineering Department,
[1]ABMSP's APCOER, Pune, India

Abstract: Now days a very common type of roller conveyor system used which is roller is supported in C channel at both ends, this behave like a simply supported beam. And the area/region below the roller conveyor system is large in value and it's not utilities in proper manners, and also designing a roller conveyor system in new format is cantilever type roller convey system. In cantilever type roller conveyor system the roller is supported in only on one of the end and another end is free. That cantilever roller we can use as a tilting mechanism. As a smaller length of the cantilever roller that resulting the increasing strength of the cantilever roller. And doing small changes in the roller and fixing mechanism the system can be fully automized, with the simplicity in construction and working. Also resulting in a elimination of manpower. We are here avoiding the interference between worker and the system for better safety purpose.

***Keyword* – Roller conveyor, mild steel roller, cantilever type tilting mechanism.**

I. INTRODUCTION

In general we are using different material for the different components. For roller rod, main rod and frame mild steel is used. But for the rollers there's various variety of material available in the market. In metal category steel, mild steel, aluminium, etc. And in non-metal category nylon, Teflon, etc. material available. If we used non-metal instead of metal the weight reduction take place, metal to metal contact is avoided resulting in no friction between semi-finished parts and conveyor system, no scratches produce on surface of the semi-finish parts, optimization take place, less magnitude of force acting on the cantilever roller rod, highest factor for the safety.

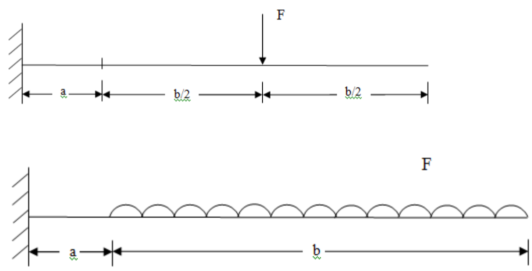
II. PROBLEM STATEMENT

After completing forming operation of metal sheet is moved from one place to another place by means of manpower. Minimum weight of one side panel sheet is 30 Kg and maximum weight of a sheet is 80-100 Kg. Minimum two labors are required for the material handling. The sheets are very heavy and have sharp edges. So, it is harmful for labors. Because of more man power the productivity of industry get decreases. More space is required for the storing and sorting this sheets. Hence we design and analysis the roller conveyor material handling system for side panels of heavy duty vehicles as well as we are trying to reduce manpower, increases productivity, make some place free to use for other purpose and reduce the overall cost of system.

III. DESIGN AND CALCULATION

3.3.1 Design Calculations For Roller Rod:

Load acting on individual roller
No. of roller = 20
Weight of one panel = 100 kg
F = (100×9.81) / (20) = 49.05N



a = 50 mm
b/2 = 320 mm
F = 49.06 N

Maximum bending moment is at point (A)
M_b = 15696 N-mm
M_a= 18148.5 N-mm

Mild steel property
Syc = 324 Mpa
M.P. = 1210°C
Considering F.O.S. = 04
$\sigma/y = M/I = E/R$
y = d/2
M = Maximum bending moment
d = 13.16 mm
ρ/A = 1.58 kg/m

σc = Syc/F.O.S. = 324/4 = 81 Mpa
$\sigma = \sigma c$
$I = (\pi/64)*(d^4)$
$d^3 = 32M/\sigma\ \pi = (32\times18148.5)/(81\pi) = 2282.2$
As per standard and design safety purpose selecting ⦰**16mm**

3.3.2 Design Calculations For Roller Rod:

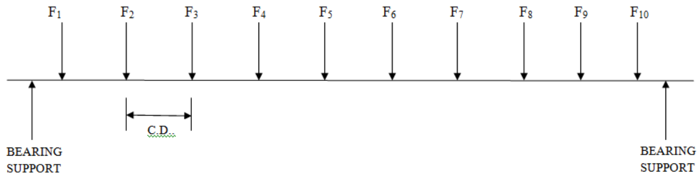
Torsional moment = F × r

Length of road = 6000mm

C.D. between two successive roller = 600mm

End point to support distance = 150mm

(Considering simply supporting beam)



As we know = $\theta_e = \theta_1+\theta_2+\theta_3+.......\theta_{10}$

$\theta_e = (T_eL_e)/(G_eJ_e) = (T_1L_1)/(G_1J_1)+......+(T_{10}L_{10})/(G_{10}J_{10})$

But values of L, G and J are same

$T_e= T_1+T_2+T_3+......+T_{10}$

$T_1= T_2= .... = T_{10}$

$T_e= 10 \times T$

Total load acting on beam = 49.05 × 10 = 490.5 N

Rc+Rd = 490.5

∴Rc/2 = Rd/2 = 245.25 N

$M_1 = M_{10} = 36787.5$ N

$M_2 = M_9= 154507.5$ N

$M_3 = M_8 = 242797.5$ N

$M_4 = M_7= 301657.5$ N

$M_5 = M_6$ =331087.5 N (Max.)

Maximum B.M. at 5/6 which is 331087.5 N-mm rod in also supported in a torsional moment

$T= F \times r = T_2= T_3 = ...T_{10}$

$T_1= 49.05\times 370=18148.5$ N-mm

But,

$T=T_1\times 10$

$T_1 = 181485$ N-mm

$T_e = \sqrt{[(M^2)+(T^2)]}$

$M_e = (1/2)\times(M+\sqrt{[(M^2)+(T^2)]}$

$\therefore M_e = d^3\times\sigma \times (\pi/32)$

$T_e= (\pi/16) \times \tau\times d^3$

$T_e = \sqrt{[(331087.5)^2+(181485)^2]}$

= 377565.54 N-mm

$= (1/2)\times(331087.5+\sqrt{[(331087.5)^2+(181455)^2]})$

M_e=354326.52 N-mm

τ= 115 Mpa $\therefore 377565.54=d^3\times115\times(\pi/16)$ ∴d≈25.57 mm

σ=162Mpa $\therefore 354326.52= d^3\times162\times(\pi/32)$ ∴d≈28.13 mm

As per standard and design safety purpose selecting diameter = **30 mm**

3.3.3 Design Calculations For M.S. Roller:

M.S. Roller (selected as per the roller rod diameter)   42×4 mm

∴ID=34.4 mm   OD= 42.4 mm

W=640 mm = b   y = (42.4/2)=21.2 mm

Load=49.05 N

$I=(\pi/64)\times (D^4 - d^4)=(\pi/64)\times [(42.4)^4 - (34.4)^4] = 89908.46\ mm^4$

$m=[(wl^2)/(8)]=[(49.05\times(640)^2)/(8)]=2511360$ N-mm

$\sigma b= (My/I)=[(2511360\times21.2)/(89908.46)]= 59.21$ Mpa

FOS= 4.41< 2 ..... Design is safe

3.3.4 Selection Of Bearing:

No of bearing required for main rod = 04

Distance between two bearings (CD) = 1900 mm

End point of bearing distance = 150 mm

(Pillow type bearing which is SAK30)

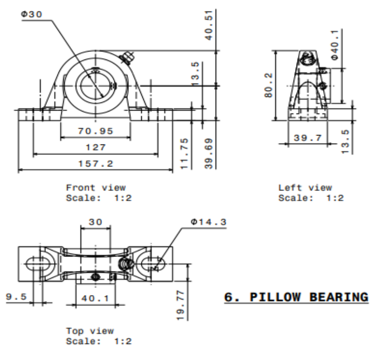
Because very low force is acting Selection of bearing is depends on the rod diameter as well as positioning of bearings. Hence we selected because very low force is acting.

No. Of bearing required for roller rod (We are using M.S. Roller) two bearing for each roller,

Total rollers are 20 i.e. Required bearing is 40 No.s

Selection of bearing is based on rod dimension's (Φ16 mm)

∴6202 zz bearing (single row deep groove) selected.



3.3.5 Selection of pneumatic actuator:

Weight of the side panel (deformed sheet) and whole system (roller arrangemnet) is 200 kg

F= 200×9.81=1962 N ≈ 2000 N = 2 KN

The extension and retraction of force is minimum as 2 KN.

Working pressure o the system= 8bar.

According to above consideration we selects PID series double acting cylinder having following characteristics cylinder.

(Considering mechanical advantage = 1 ∴ Force=Effort=2 KN)

Bore diameter= 63 mm   piston rod diameter = 20 mm

working pressure = 8bar   stroke length =125 mm

Force during extension=2494 N   force during retraction=2242 N

3.3.6 Selection Of Pulleys And Wire Rope:

As we know   Circumference = $2\pi r = \pi d$ Stroke length = 125 mm

Length = (length of arc)×[90(maximum tilting angle)/360

Length of arc = $(\pi d)/(4)= (\pi d\theta)/(4\times360)$=because tilting angle = $(\pi d\times90)/(4\times360)$

$125=(\pi d/4)$ D= 159.154≈160 mm

This pitch circle of grooves should be selected in the range of 160 mm

According ISO for load of 10 KN is 5200×12 which means outer diameter of pulley is =200 mm

groove pitch diameter =160 mm

12 represents the rope diameter

∴ **Selection is done for pulley and wire rope**

IV. ANLYSIS USING SUITABLE SOFTWARE (ANSYS)

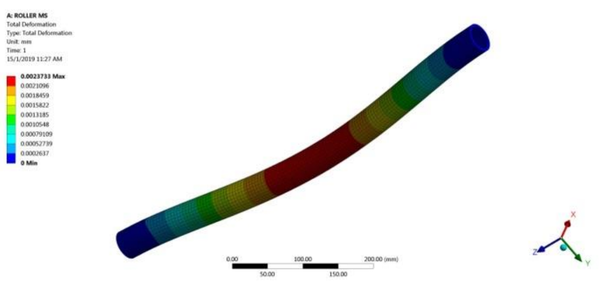


Fig. 1 deformation of M.S. roller

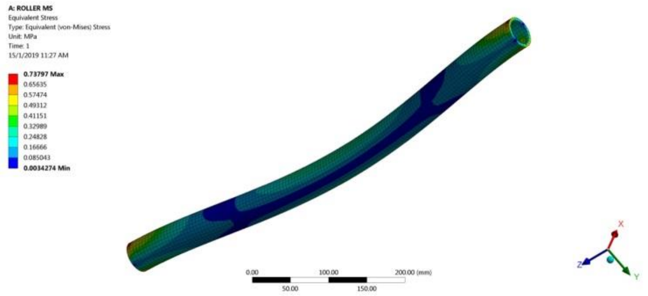


Fig. 2. Stresses in MS roller

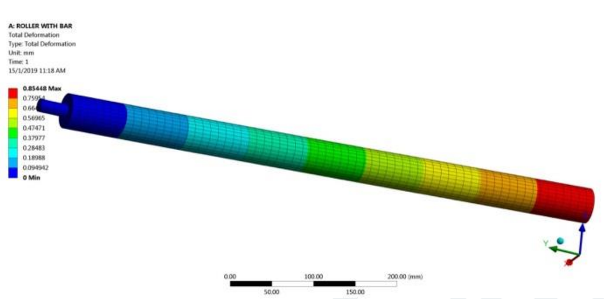


Fig. 3 deformation of M.S. roller with rod

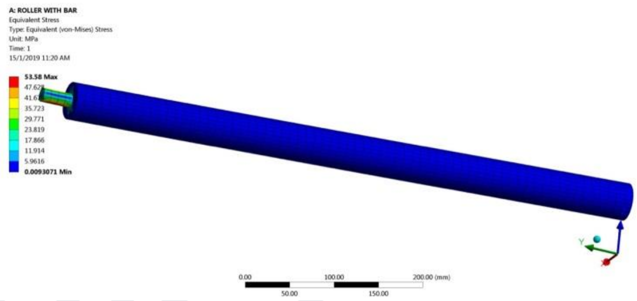


Fig. 4. Stresses in MS roller with rod

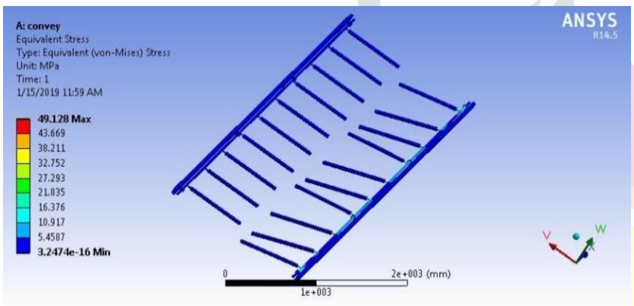


Fig. 3 Stresses in Whole assembly

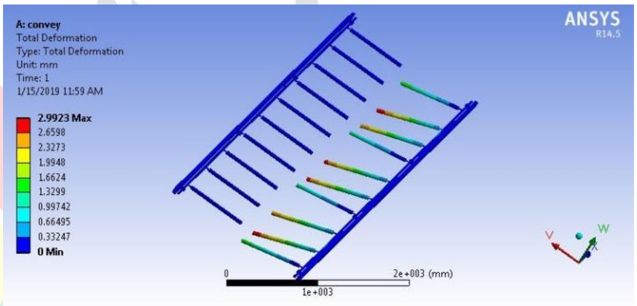


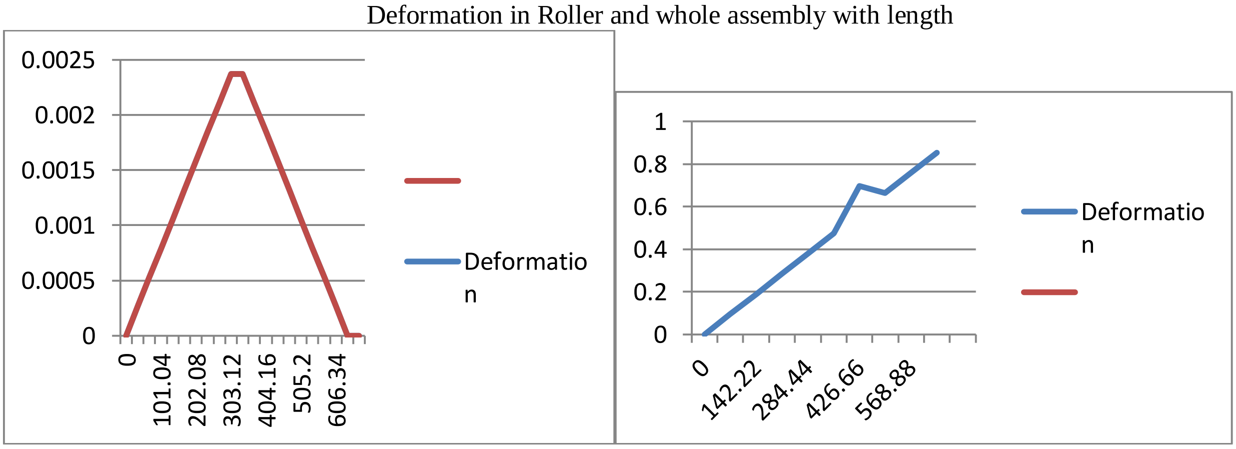
Fig. 4. deformation of Whole assembly

I. RESEARCH METHODOLOGY

1. Understanding the problem in proper manner
2. Study of traditional roller conveyor system
3. Selection of proper mechanism
4. Designing the system and modeling in CATIA software

V. RESULTS AND DISCUSSION

Deformation in Roller and whole assembly with length



VI. CONCLUSION

Selection of appropriate method for solving problem statement. After designing the system, the value deflection and stress are in the allowable limit. Values from the analysis software (ANSYS) are compared with analytical (manual) calculation, means validation of the calculations are done with analysis software. Because of FEA analysis can conclude factor of safety for the future references

VII. ACKNOWLEDGMENT

We would like to thank Mr. Arun Patel, and Mr. Sandeep Raut (Anantrao Pawar college of Engineering and Research)) for his continuous guidance throughout the research. We would also like to thank Anantrao Pawar College of Engineering and Research and all its faculty members for encouraging us to prepare this research paper.

REFERENCES

[1] Al D. Wieczorek ,b. Künne ‘Contribution to the development of supplementary guidelines of embodiment design for roller conveyors’ management and control of production logistics university of coimbra, portugal september 8-10, 2010.

[2] S.s. gaikwad, e.n. aitavade ‘static analysis of a roller of gravity roller conveyor for structural strength & weight’ international journal of advanced engineering technology.

[3] M. D. Jagtap, b. D. Gaikwad, p. M. Pawar ‘to study local behavior in roller conveyor chain strip’ journal of basic and applied engineering research print issn: 2350-0077; online issn: 2350-0255; volume 1, number 4; october, 2014 pp. 63-67.

[4] A b.kharage, b. Nelge, k. Dhumal ‘analysis and optimization of gravity roller conveyor using ansys’ international journal of engineering sciences & research technology april, 2015.

[5] G. S sinare, s. B zope ‘redesign of roller conveyor system for weight reduction’ international journal of modern trends in engineering and research (ijmter) volume 02, issue 10, [october – 2015].

[6] S. K. Nalgeshi ‘design and weight optimization of gravity roller conveyor’ international journal of innovations in engineering research and technology [ijiert] issn: 2394-3696 volume 3, issue 8, aug.-2016.