Analysis of Powered Roller Conveyor using FEA

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Abstract— In the industry, for movement of material from point to point, material handling equipment is necessary. Material handling equipment transports, transfers, moves the material to the desired destination. Roller conveyor is the material handling equipment used to transfer the material over a fixed path. Components of powered roller conveyors are shaft/Axle, rollers, frame, chain, selection of bearing etc. It designed on the basis of load carring capacity, distance to be moved, velocity etc. In this project the present powered roller conveyor, the rollers deflect to the design loading. The excessive deflection and its corresponding reactions are transmitted to the bearings. This work is undertaken to do the analysis and optimization of powered roller conveyor using FEA. In this project stress analysis of existing system is carried out analytically and by using FEA. The geometrical modifications suggested to avoid its deflection. In the present work, an attempt is made also of the reduction in weight of existing roller conveyor by optimizing the critical parts of (e.g. inner & outer diameter of roller, thickness of roller) conveyor without affecting functional performance. Optimization gives optimum design for same loading condition with large weight reduction.

Key words: Power Roller, Catia, Solid Work, ANSYS

I. INTRODUCTION

A roller is an assembly of three major components:

1) Shaft /Spindle/Axle
2) Tube / Hollow Shaft
3) Bearings.

The length of a roller is defined in terms of the "between frame widths" or "BF". The best method for ordering additional or replacement rollers is to always specify the between frames dimension (BF). This will ensure a proper fit for rollers and conveyor frames. If end-user does not know the BF dimension, simply have this person measure between the frames of the specified unit. However, there are times when getting a between frames dimension is difficult. In this instance, it is very important to use the proper terminology to select a roller size. The only dimension acceptable in determining roller length when the BF is not known is the "end-of-bearing" (EOB) measurement. The importance here cannot be overstated. Since conveyor/roller manufacturers vary the length of the roller tube in relation to the manner in which the bearing is inserted and depending on the individual bearing being used countless dimensions are possible.

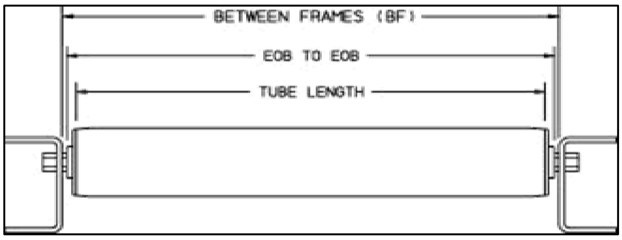

Fig. 1: Roller Component

II. PROBLEM DEFINITION

In discussion with Om Sai Enterprise, in this industry machining of automotive components ranging from 200 kg to 2000 kg carried out. The present powered roller conveyor carring capacity of 2500 kg.

In discussion with industry we came across the exact conditions of the problem. When moving weight on roller assembly is changed to & above 2000Kg. following points are observed.

1) A deflection in shaft is observed at load 2000 kg which is below of prescribed capacity of roller conveyor assembly.
2) Thereby crushing the bearings and stopping the movement of the roller, which is affecting production.

This work is under taken for analysis and optimization of powered roller conveyor using FEA for solving problem.

From last one year, rejection percentage is 5 to 6 % due to failure of roller assembly in powered roller conveyors. So design & analysis of the roller assembly that should with stand up to the total weight more than 2000Kg without failure in running conditions is the desired goal.

III. SPECIFICATION OF CURRENT POWER ROLLER CONVEYOR

1) Length of one roller conveyor: 2250 mm
2) Width of roller conveyor: 666 mm
3) Outer Roller dia.: 60 mm
4) Inner Roller dia.: 44 mm
5) Roller thickness: 08 mm
6) 6. Roller length: 603 mm

IV. SCOPE OF WORK

Currently said industry is using powered roller conveyor. As discussed before, there is problem of excessive deflection of present conveyor system. After study of present conveyor system, the present analysis focuses on below points.

1) Study existing roller conveyor system and study the design of shaft/axle.
2) Study the design of roller.
3) Selection of bearing.
4) Modified the design of shaft/axle and roller.
5) To carry out FEA analysis of present and proposed modified design for same loading condition.

6) Weight reduction of roller and compare the weight reduction between present roller conveyor system and proposed roller conveyor system.
7) Optimization of material and recommendation of optimized new solution.

V. ANALYTICAL DESIGN CALCULATION FOR SHAFT ANALYTICAL DESIGN CALCULATION FOR PRESENT SHAFT

A. *Material – MS –Grade C50*

E= 2.00×10^5 Mpa, ρ= 7800 Kg/m3, Syt = 1050Mpa
Considering uniformly distributed load & FOS =2
Allowable Stress (σall) = Syt / Fs = 1050/2=525Mpa

1) Maximum Stress Calculation for given condition

W= 2000/4= 500 kg (Load act on 4 rollers at a time)
d = Outer diameter of shaft = 20 mm
L = Length of shaft = 666 mm
y = Distance from neutral axis = 20/2 = 10
a = Distance of bearing from shaft end =45 mm
l= Distance between two bearing center =576 mm
Considering beam with uniformly distributed load,

2) Maximum Moment (Mmax) :-

Mmax =Wa = 5000×45
Mmax = 225000 N-mm

a) Moment of Inertia (I)
= π (d^4) / 64 = π (20^4) /64
I = 7850 mm4

3) Maximum bending stress (σb)

(σb) = Mmax × y/ I = 225000 × 10/ 7850
(σb) = 286.62 Mpa

4) Checking Factor of Safety for design-

Fs = σall / σb = 525/286.62
Fs = 1.8316

As calculated Fs is smaller than assumed Fs, Selected design can be considered as not safe.

5) Maximum Deflection (Ymax)

(Ymax) = wa^2 (2a+3l)/6EI
= (5000×45^2) * (2×45 + 3×576) / (6 ×7850×2.00×10^5)
= 2000000× (1818)/ (6×1.57× 10^9)
ymax = 1.9541 mm

a) Weight of Shafts
= cross-section area*width * mass density* number of shafts
= π $(0.010)^2$ *(0.666*7800*18)
= 29.3610 Kg

VI. ANALYTICAL DESIGN CALCULATION FOR PROPOSED SHAFT PROPOSED DESIGN OF SHAFT

A. *Material – MS –Grade C50*

E=2.00×10^5Mpa, ρ=7800Kg/m^3, Syt =1050Mpa considering uniformly distributed load & FOS = 2 Allowable Stress (σall) = Syt / Fs = 1050/2=525Mpa

1) Maximum Stress Calculation for given condition

W= 2000/4= 500 kg (Load act on 4 rollers at a time)
d = Outer diameter of shaft = 20 mm
L = Length of shaft = 666 mm
y = Distance from neutral axis = 20/2 = 10
a = Distance of bearing from shaft end =20 mm
l= Distance between two bearing center =626 mm
Considering beam with uniformly distributed load,

2) Maximum Moment (Mmax)

Mmax =W× a = 5000 × 20
Mmax = 100000 N-mm

a) Moment of Inertia (I)
= π (d^4) / 64 = π (20^4) /64
I = 7850 mm^4

3) Maximum bending stress (σb)

(σb) = Mmax × y/ I = 100000 * 10/ 7850
(σb) = 127.39Mpa

4) Checking Factor of Safety for design

Fs = σall / σb = 525/127.39
Fs = 4.1213

As calculated Fs is greater than assumed Fs, selected material & design can be considered as safe.

5) Maximum Deflection (Ymax)

(Ymax) = wa^2 (2a+3l)/6EI
= (5000×20^2) * (2×20 + 3×626) / (6 ×7850×2.00×10^5)
= 200000 × (1918) / (6×1.57×109)
ymax = 0.4072 mm

As compared to length 626 mm deflection of 0.4072 mm is very small. Hence selected design can be considered as safe.

VII. ANALYTICAL DESIGN CALCULATION FOR PRESENT ROLLER

A. *Material – MS C20*

E= 2.00×10^5 Mpa, ρ= 7860 Kg/m^3, Syt = 520 Mpa
Considering uniformly distributed load & FOS =2
Allowable stress (σall) = Syt / Fs = 520/2=260 Mpa

1) Maximum stress calculation for given condition

W= 2000/4= 500kg (Load act on 4 rollers at a time)
D1= Outer diameter of roller = 60 mm
D2 = Inner diameter of roller = 44 mm
w = Width of roller = 603 mm
y = Distance from neutral axis = 60/2= 30
Considering beam with uniformly distributed load,

2) Maximum Moment (Mmax)

(Mmax) = W×L^2/8
= (5000×603 x 603) /8
Mmax = 227255 Nmm

a) Moment of Inertia (I)
= π ($D1^4$ - $D2^4$)/64 = π (60^4 – 44^4)/64
I = 451959 mm4

3) Maximum bending stress (σb)

(σb) = Mmax × y/ I
= 227255 × 30/ 451959
(σb) = 15.08 Mpa

4) Checking Factor of Safety for design

Fs = σall / σb = 260/15.08
Fs = 17.2413

As calculated Fs is greater than assumed Fs, selected material & design can be considered as safe. Hence weight reduction can be possible.

5) Maximum Deflection (ymax)

= 5×W×L^3/384EI
= (5×5000*603^3) / (384*2.00*10^5* 451959)
ymax = 0.158 mm

As compared to length 603 mm deflection of 0.158 mm is very negligible. Hence selected roller can be considered as safe.

a) Weight of Rollers
= cross-section area× width ×mass density× number of rollers
= π (60^2 - 44^2) ×603×7860×18/4 =111.4385 Kg

VIII. ANALYTICAL DESIGN CALCULATION FOR PROPOSED ROLLER

A. Need of Optimization

As factor of safety of rollers is very high there is scope of weight reduction in this component.

- Selection of critical parameter
- Roller outer diameter
- Roller inner diameter
- Roller thickness

B. Material – MS

E= 2.00×105 Mpa, ρ= 7860 Kg/m3, Syt = 520 Mpa
Considering uniformly distributed load & FOS =2
Allowable stress (σ_{all}) = Syt / Fs = 520/2=260 Mpa

1) Maximum stress calculation for given condition

W= 2000/4= 500kg (Load act on 4 rollers at a time)
D1= Outer diameter of roller = 60 mm
D2 = Inner diameter of roller = 50 mm
w = Width of roller = 603 mm
y = Distance from neutral axis = 60/2 = 30
Considering beam with uniformly distributed load,

2) Maximum Moment (Mmax)

((Mmax) = $W \times L^2/8$ = (5000×603 *603)/8
Mmax = 227255 Nmm

a) Moment of Inertia (I)

= $\pi (D1^4 - D2^4)/64 = \pi (60^4 - 50^4)/64$
I = 329209 mm4

3) Maximum bending stress (σb)

(σb) = Mmax × y/ I
= 227255 × 30/ 329209 (σb) = 20.7091 Mpa

4) Checking factor of safety for design

Fs = σall / σb
= 260/20.70 Fs = 12.555

As calculated Fs is greater than assumed Fs, selected material & design can be considered as safe.

5) Maximum Deflection (ymax)

= $5 \times W \times L^3/384EI$ = (5×5000×6033) / ($384 \times 2.00 \times 10^5 \times$ 329209)
ymax = 0.217 mm

As compared to length 603 mm deflection of 0.217 mm is very negligible. Hence selected roller can be considered as safe.

a) Weight of rollers

= cross-section area ×width ×mass density× number of rollers = $\pi (60^2 - 50^2)$ ×603×7860×18/4
=73.67 Kg

IX. RESULT OF SHAFT BY FEA

A. Present Shaft Result by FEA

Total Deformation-Total deformation is 2.0804 mm.

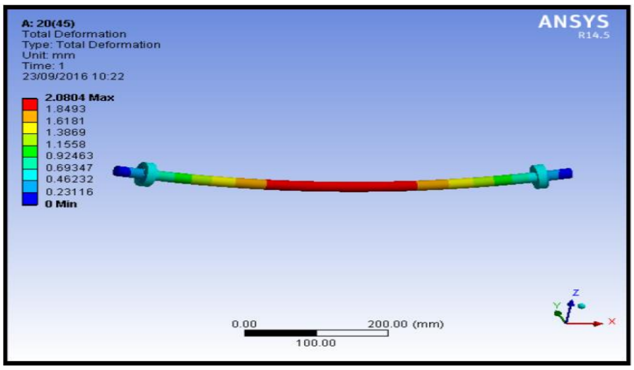

Fig. 2: Total deformation of shaft in mm.

1) Von Mises (Yield) Stress

Von mises stress defines the maximum yielding stress at the particular location which useful before manufacturing in actual practice. The stress is 296.1Mpa.

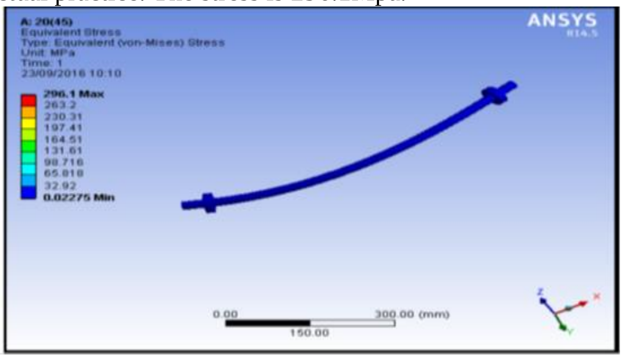

Fig. 3: Equivalent stress of shaft, Mpa

B. Proposed shaft result by FEA

Total Deformation-Total deformation is 0.3720 mm.

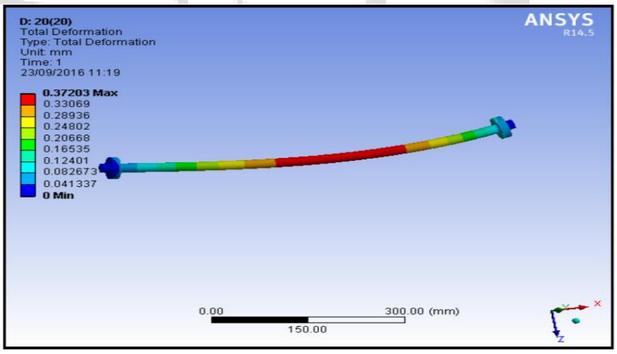

Fig. 4: Total deformation of shaft in mm.

1) Von Mises (Yield) Stress

Von mises stress defines the maximum yielding stress at the particular location which useful before manufacturing in actual practice. The stress is 136.34Mpa.

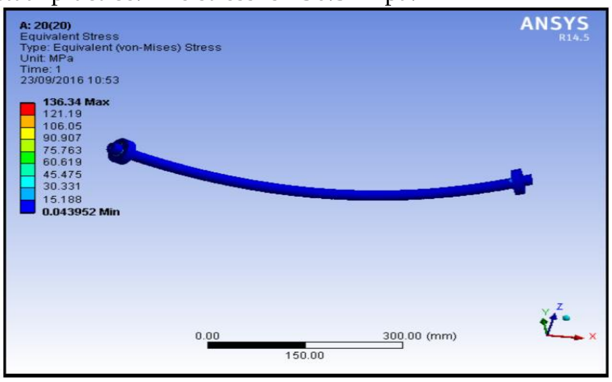

Fig. 5: Equivalent stress of shaft, Mpa

C. Present roller result by FEA

Total Deformation:-Total deformation is 0.1450 mm.

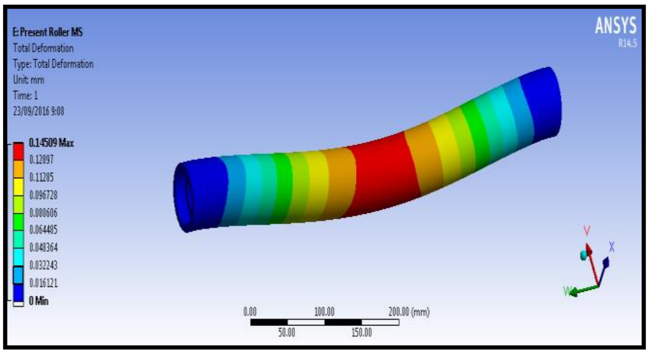

Fig. 6: Total deformation of roller in mm

1) Von Mises (Yield) Stress

Von mises stress defines the maximum yielding stress at the particular location which useful before manufacturing in actual practice. The stress is 13.889 Mpa

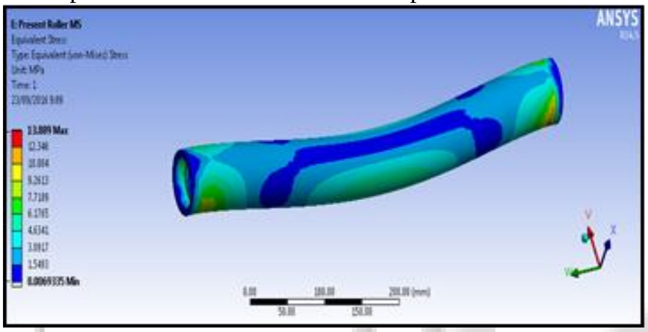

Fig. 7: Equivalent stress of roller, Mpa

D. Proposed roller result by FEA

Total Deformation-Total deformation is 0.2085 mm.

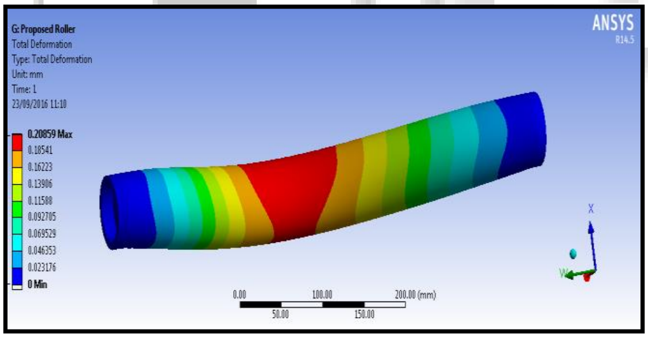

Fig. 8: Total deformation of roller in mm.

1) Von Mises (Yield) Stress

Von mises stress defines the maximum yielding stress at the particular location which useful before manufacturing in actual practice. The stress is 21.641Mpa.

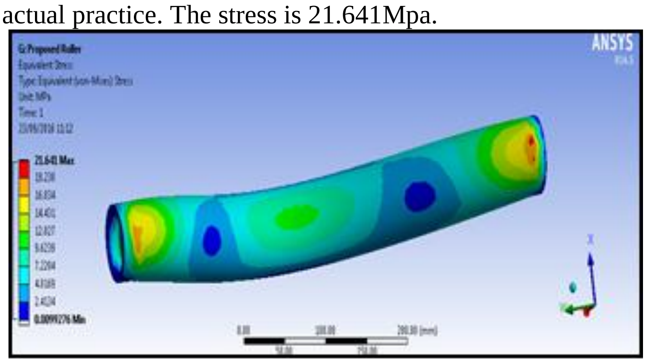

Fig. 9: Equivalent stress of roller, Mpa

Description	Unit	Present Shaft	Proposed Shaft
Maximum bending stress (σb)	Mpa	286.62	127.39
Checking factor of safety for design Fs		1.8316	4.1213
Deflection (Y Max)	mm	1.9541	0.4072

Table 1: Results- Effect of proposed design compared with present design of shaft

Existing powered roller conveyor fails above 2000 kg. But due to redesign of shaft design, load carring capacity increases from 2000 kg to 3000 kg.

Deflection is reduced by 85 %. The exact reduction is from 1.9541 mm to 0.4028mm. This leads to good quality of product and increased life of roller. Now failure percentage mostly reduced.

Description	Unit	Present Roller	Proposed Roller
Maximum Bending Stress (σb)	Mpa	15.08	20.709
Deflection (Y Max)	mm	0.158	0.217
Weight of Shaft W	Kg	111.43	73.67

Table 2: Present roller design calculation

Present roller design calculation shows the factor of safety as much greater than the requirement and thus there is a scope for weight reduction. We have tried for optimizing weight of conveyor assembly by reducing diameter of the roller because roller is the crucial part of the conveyor assembly and its weight is more as compared to other components. Here an attempt is made to optimize weight of conveyor assembly by reducing thickness of roller. Critical parameters which reduce the weight are roller outer diameter, roller inner diameter and roller thickness.

Sr. No.	Name of Component	Weight (Kg)	
		Present Design	Proposed Design
1	Rollers	111.43	73.67
2	Shafts	29.36	29.36
3	Bearing	03.90	03.90
	Total Weight	144.69	106.93

Table 3: Results- Effect of proposed roller design compared with present design for total weight of roller assembly.

For present design of roller assembly weight is 144.69 kg and for proposed design of roller assembly weight is106.93 Kg.

X. Conclusion

In this study, analysis and optimization of powered roller conveyor using FEA is done. Analytical calculations are carried out to determine Shaft diameter and roller diameter. It is verified by making FEA model in CATIA V5 and analysis is carried out in Ansys. Closely matching results are obtained between analytical and ANSYS.

Both geometrical and material optimization is carried out. In geometrical optimization, variation in inner and outer diameter of roller and its stress analysis is done. The stress is within the permissible limit. Material optimization is done using Johnsons method and objective is to reduce weight and cost without influencing functional performance.

Existing powered roller conveyor deflects above 2000 kg. But due to redesign of shaft, load carrying capacity increases from 2000 kg to 3000 kg. Deflection is reduced by 85 %. i.e. from 1.9541 mm to 0.4028mm. It also leads to good quality of product and increased life of roller. Also, failure percentage is mostly reduced.

For present design of roller assembly weight is 144.69 kg and for proposed design of roller assembly, the weight is 106.93 Kg. Thus, 26.10 % of weight reduction is achieved due to optimized design i.e. a total reduction in weight by 37.76 Kg. Thus, in the optimized design, weight of the roller is reduced, weight carrying capacity is increased. Due to reduction of weight, the cost of the roller is also reduced.

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