Design and Modification of Prototype Vibratory Conveyor Hopper

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Abstract:
Design and development of Vibratory Conveyor Hopper is done in this paper and it is design according to customer requirement and purpose of use. The performance of New Designed Hopper with vibrating conveyor was observed with various parameters like Bearing, Flywheel, Shaft, Belts etc. on this Prototype which can used as a Benchmark for the further research and development. The assembly of different parts has been done and the conveyor is tested for various loads, Torque, Stresses and Deformations. Also, the suitable diameters for bearings and Flywheels has been Calculated for this Specific Design.

Keywords — Prototypes, Conveyor, Stress Concentration, Torque, Load, Stress.

I. INTRODUCTION

High strength, cured, fiberglass reinforced plastic material was first introduced for use of vibratory conveyor springs in 1954. Since then, fiberglass springs have been used successfully in a wide range of vibratory conveyors, feeders and screening equipment, providing infinite fatigue life and corrosion free service [6].

These springs are made of 3M’s Scotch ply brand reinforced plastic, a unique material in which the epoxy resins are reinforced with continuous, linearly aligned, non-woven filaments. This allows the plies to be laid up so that the reinforcement is oriented to the greater strength and stiffness in the direction in which it is most needed [1].

Now a day’s Vibratory Conveyor are used widely so the Application point of view the Modification of Conveyor may led to the optimization of Cost and Compactness of the conveyor may help in the view of space and Mobility of the Machine. So, our approach is to optimize the Cost as well as Material used for conveyor by calculating some of the loads and critical diameters used in this conveyor [5].

A. Process of Prototyping

Prototyping is an attractive idea for complicated and large systems for which there is no manual process or existing system to help determine the requirements. The prototypes are usually not complete systems and many of the details are not built in the prototype. The goal is to provide a system with overall functionality. Fig. 1 shows the layout of prototyping [3].

![Fig. 1. Process of Prototyping]

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B. Need For Prototyping

1. Prototype model should be used when the desired system needs to have a lot of interaction with the end users.
2. Typically, online systems, web interfaces have a very high amount of interaction with end users, are best suited for prototype model. It might take a while for a system to be built that allows ease of use and needs minimal training for the end user [8].

II. DESIGN CALCULATION OF PROTOTYPE COMPONENTS

Design requirement:
Rotating speed of motor= 1440 rpm
Rotating speed at output= 290 to 310 rpm
Assuming 3 Hp induction motor= 2.23 KW
Service = 24 hour per day.
Material conveyer = Sugar

A. Selection of Belt

Correction factor (Fa) =1.4
1) Design power = Fa X Transmitted power
   = 1.4 X 2.238
   =3.1332 KW
2) From design data book cross section of belt selected is “A” section.
3) Pitch diameter of smaller and bigger pulley from reference is selected as
   Smaller pulley diameter (d) = 85mm
   Speed ratio = \( \frac{1440}{290} \)
   Larger pulley diameter (D) = (d) x Speed ratio
   = 85 X 4.96
   =422 mm ≈ 420 mm
4) Pitch length of belt (L) =
   \( 2C + \left( \frac{\pi}{2} \right) \left( \frac{D+d}{4C} \right) \)
   = 2 X 600 + \left( \frac{\pi}{2} \right) (420 + 85)
   = 2040 mm
5) Correct centredistance:
   \( L_b = 2C + \left( \frac{\pi}{2} \right) \left( D + d \right) \left( \frac{(D-d)^2}{4C} \right) \)

2050 = 2C + (\( \frac{\pi}{2} \) (420 + 85)) + (\( \frac{(420-85)^2}{4C} \))
2050C = 2C^2 + 793.25C +28056.25
2C^2 – 1256.75 +28056.25 = 0
C = 605.19 mm ≈ 610 mm
6) Correction factor for belt pitch length (Fc) from design data book = 1.04
7) Correction factor for arc of contact (F_d):
   \( \alpha_s = 180 - \sin^{-1} \left( \frac{D-d}{2C} \right) \)
   \( \alpha_s = 180 - \sin^{-1} \left( \frac{420-85}{2 \times 610} \right) \)
   \( \alpha_s = 164.06 \approx 163° \)
Correlation factor for arc of contact (F_d) = 0.96
8) Power rating of single v-belt:
   \( P_r = 1.17 + 0.17 \)
9) No of belts required:
   No of belt required = \( \frac{P \times F_d}{P_r \times F_d} \)
   = \( \frac{2.236 \times 0.96}{1.17 \times 0.17} \)
   = 1.60 ≈ 2 belts

B. Design of Flywheel Assumptions

Material = FG260
Motor power = 2.236kw, Stroke = 30mm, Speed ratio = 4.96
Max fluctuation of Speed = 280 to 300 rpm
Width of Flywheel (b) = 2 x Depth of Flywheel (t)

1) Coefficient of Fluctuation Calculation (Cs): The difference between maximum fluctuations of speed to mean speed is called coefficient of fluctuation of speed (Cs).
   \( W_{max} = \) Maximum speed during cycle
   \( W_{min} = \) Minimum speed during cycle
   \( W_{mean} = \) Mean speed
   \( W_{mean} = \frac{(W_{max} + W_{min})}{2} \)
   Assume Cs =0.0350
2) Mass Moment of Interia (I) Calculation: Kinetic energy of system (Ke) calculation:
   Work done, \( W = 6000 \times 0.030 \times 0.15 \)
   = 27 N-m
   Thus, energy absorbed is 27 N-m
   Work done per cycle = \( 2 \pi \times 27 \times 4.96 \)
   = 841.44 N-m
Ke = W – (WC X 0.08)
= 841.44 – (27 X 0.08)
= 839.28 N-m

Therefore, mass moment of inertia is
I = Ke / (Cs X Wmean²)
= 839.28 / (0.0350 X 295²)
I = 0.27 Kg.m²

For flywheel rim, [4]
Mass moment of inertia (Ir) = C X I
= 0.9 X 0.27
= 0.24Kg.m²

We know that,
Ir = Mr X Rm²
= (2π Rm X b X t)XRm² X ρ
0.243 = (2π X 0.210 X 2 X t²)X0.2102 X 7200

Therefore, b = 20 mm
b = 2 x 20
b = 40 mm

C. Design of Shaft
Material for shaft = EN 8
Material property:
Syt = 465 N/mm²
Sut = 700 N/mm²

Kb = Combined shock and fatigue factor
applied to bending moment = 2.0
Ke = Combined shock and fatigue factor
applied torsional moment = 1.5

Coefficient of friction (μ) = 0.30

Design of shaft on strength basis [2]
According to ASME code
τmax = 0.30 Syt
= 0.30 X 465
= 139.5 N/mm²

τmax = 0.18 Sut
= 0.18 X 700
τmax = 126 N/mm²

Select whichever is minimum. If keyways are present reduce above value 25 per cent.
τmax = 0.75 X 126
τmax = 94.5 N/mm²

Torque developed by motor:
P = \( \frac{2πNT}{60} \)

2236 = \( \frac{(2π X 1440 X T)}{60} \)
T = 14.82 Nm

Torque transferred to flywheel is:
Tf = T X velocity ratio
Tf = 14.82 x 4.96
Tf = 73.50 Nm

Load acting on flywheel arc are:
Tf = (T1 - T2) X Rf

Solving this equation, we get,
T1 = 636.31 N
T2 = 289.39 N

Design of shaft for fatigue loading: [10]

\[ \sigma_e = K_a K_b K_c K_d S_e \]

\[ S_e = K_a K_b K_c K_d S_u \]

\[ S_u = (0.5 \times 700) \]

\[ = 350 \times \frac{N}{mm²} \]

\[ = \frac{1400 MPa}{mm²} \]

\[ S_e = Endurance Strength in N/mm² \]

K_a = Surface finish factor
K_b = Size factor
K_c = Reliability factor
K_d = 0.77

a = 272 & b = -0.995 as shaft is forged.
K_b = 0.85; \quad 7.5 < d \leq 50

K_c = 1;
K_d = \frac{1}{K_f}
K_f = 1 + q(K_c - 1)

q = Notch sensitivity factor = 0.55
K_f = Stress concentration factor = 2
K_f = 1.55
K_d = \frac{1}{K_f} = 0.716
\[ S_e = K_dK_pK_cK_dS_e \\
= 0.77 \times 0.85 \times 0.897 \times 0.716 \times 350 \\
= \frac{77}{N} \left( \frac{32 \times M_b \max}{\pi d^3 s^3} \right) = \frac{S_e}{f_s} \\
(32 \times 29700/\pi d^3) = (77/2) \]

\[ d_s = 33.33 \text{mm} \]
\[ d_s \approx 50 \text{mm} \]

D. Selection of Bearing

Pedestal bearing = 2211 k, H-311 double row self-aligning ball bearing [7].

At the centre of shaft = 22218 k, H318 double row spherical roller bearing

<table>
<thead>
<tr>
<th>Sr. No.</th>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Belt design power</td>
<td>3.13kw</td>
</tr>
<tr>
<td>2</td>
<td>Belt cross section</td>
<td>A-Type</td>
</tr>
<tr>
<td>3</td>
<td>Smaller pulley diameter</td>
<td>85mm</td>
</tr>
<tr>
<td>4</td>
<td>Larger pulley diameter</td>
<td>420mm</td>
</tr>
<tr>
<td>5</td>
<td>Belt pitch length</td>
<td>2050mm</td>
</tr>
<tr>
<td>6</td>
<td>centre distance</td>
<td>610mm</td>
</tr>
<tr>
<td>7</td>
<td>No of belts</td>
<td>2nos</td>
</tr>
<tr>
<td>8</td>
<td>Mass moment of inertia</td>
<td>0.27kg.m²</td>
</tr>
<tr>
<td>9</td>
<td>Self-align ball bearing</td>
<td>2211K,H311</td>
</tr>
<tr>
<td>10</td>
<td>Spherical roller bearing</td>
<td>22218K,H318</td>
</tr>
<tr>
<td>11</td>
<td>Shaft diameter</td>
<td>50mm</td>
</tr>
</tbody>
</table>

III. MANUFACTURING OF PROTOTYPE

Manufacturing is the production of useful components using labour, machine tools. We built prototype in which raw material is transformed into finished good. To build this prototype we manufactured some components in company like eccentric shaft, flywheel etc. Also some components brought out from outside such as bearings, belt etc. [9].
IV. CONCLUSIONS

A prototype was built on the basis of measurements taken from actual hopper system so as to carry out different tests on the prototype which are required to be conducted on the actual system such as alignment test. Alignment test was proved positive which eliminates various other problems like lateral motion of hopper, failure of main bearings. Alignment Test performed on Prototype shows better stability of the vibrating conveyor, when material used as supporting element of different material. Due to some inherent properties of Composite material, it has a capability to store certain amount of energy which is released and stored repetitively during the cycle of operation. This reduced the power consumption of the system allowing us to a motor with lower capacity.

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REFERENCES

3. G.L.Hinueber, “Reinforced plastic springs for vibratory conveyors”.