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Improvement in the Design of a Gear Drive Bucket Elevator

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ABSTRACT
This paper presents improvement in the design of a gear drive bucket elevator by introducing a ratchet mechanism which holds the chain links and buckets in position when there is a failure in order to prevent them from damage and also addresses the difficulty in joining the chain links together. The main function of the ratchet mechanism is to ensure that the chain is prevented from dropping to the bottom when there is a failure. The ratchet, basically, is made up of toothed wheel (sprocket/spur gear), a pawl/lever, and a spring. The calculations involve the design of the beam, pawl and the spring. As the ratchet mechanism will help prevent the chain together with the bucket from dropping to the bottom whenever there is failure, it will help reduce downtime leading to an increase in production. The mechanism will also prevent damage to buckets and chain links leading to reduction in maintenance cost.

Keywords: Ratchet mechanism, chain links, failure, maximum shear force, bending moment

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I. INTRODUCTION

The raw materials used in the production of cement are; clinker, limestone and gypsum. A crane is used to scoop the raw materials from the barges into a hopper and then transported via a conveyer belt to a shed and stored temporally. The manufacturing of cement generates large quantities of dust which are potentially environmentally damaging. Cement dust causes lung function impairment, pneumoconiosis and carcinoma of the lungs, stomach and colon. The cement particles are prevented from escaping into the atmosphere by the use of a dust plant. A bucket elevator is needed to lift the cement from the outlet of the mill through a height of 32.5 m into a centrifugal separator where the finer cement particles are separated from the coarse ones. The finer ones are then pumped into the silos to be stored via a screw pump. Bucket elevators are basically designed to move flowing powder or bulk solids vertically through a certain height. Bucket elevators use an endless chain or belt and have a series of buckets attached to them. The bulk material is spread into an inlet hopper. The buckets then dig into the material and convey it up and over the head sprocket/pulley and then throw the material out via a discharge throat. Bucket elevators are usually not self-feeding, and are fed at a controlled rate. The buckets are usually where the chain or belt path is vertically in a single plane. The buckets are returned down to the tail sprocket/pulley at the bottom. Research has shown that in spite of intelligent material selection on the part of engineers, or how careful engineers design machine component, failure still occurs.

According to Budynas and Nisbett (2011), mechanical failure can mean a part has separated into two or more pieces; has become permanently distorted, thus ruining its geometry; has had its reliability downgraded; or has had its function compromised. But Collins, (1993) defined mechanical failure as “any change in the size, shape or material properties of a structure, machine or machine part that renders it incapable of performing its intended function. Wear occurs as a result of abrasion, erosion, adhesive wear and surface fatigue but in context with the bucket elevator, wear of the chain is as a result of erosion which is the progressive loss of material from a surface by the mechanical action of fluids/particles on a surface. Solid impingement erosion is the type of erosion that causes failure of the chain. Fracture occurs when the forces tending to propagate the crack are greater than the force tending to arrest the crack Budinski and Budinski, (1999). Belt conveyors are used to transport materials as well as to drive the rollers. Khurmi and Gupta,(2005).

They represent the primary means of intermediate haulage in most of today's underground and surface mines, and have gained acceptance for main-line haulage. Developments in overland belt systems have extended their influence to surface facilities. Although belt conveyors are made of many important parts, none is more economically important than the belt conveyor itself, which in most cases will represent a substantial part of the initial cost. The tensioning device maintains a constant tension in the slack strand despite changing chain elongation caused by wear or thermal expansion and contraction Parmley, (1985).

Whenever the chain links of the bucket elevator fails it takes the technicians a minimum of three (3) days to fix it. This is because when one of the chain links fails, all the chains together with the buckets drop to the bottom of the bucket elevator. Chain block is then used to pull the chain link up through a height of between 15 m to 25 m depending on where it failed in order to join it. In addition to this, some of the links and buckets get damaged and they need to be replaced with new ones. This results in increased down-time and goes a long way to affect production time. Moreover, since the damaged buckets and links need to be replaced with new ones, maintenance cost is increased.

In view of this, the paper seeks to address this problem by introducing a ratchet mechanism which will hold the chain links and buckets in position when there is a failure in order to prevent them from damage and also address the difficulty in joining the chain links together.

II. MATERIALS AND METHODS

Proposed Design and Design Calculations

Figure 1 shows the sectional view of a chain bucket elevator. The proposed design incorporates a ratchet mechanism which restricts the chains with the buckets mounted on it to move only in the counter clockwise direction. The chains never fail when the buckets are empty but fail when it is filled with cement. In view of this, whenever the chain fails, it will descend to the bottom because of the force of gravity acting on it. In descending to the bottom, the motion of the chain tends to be in the clockwise direction and because the ratchet mechanism incorporated restricts the chain to move only in the counter clockwise direction, the chain is held in position and is prevented from dropping to the bottom of the elevator. The proposed design is as shown in the Figure 2.

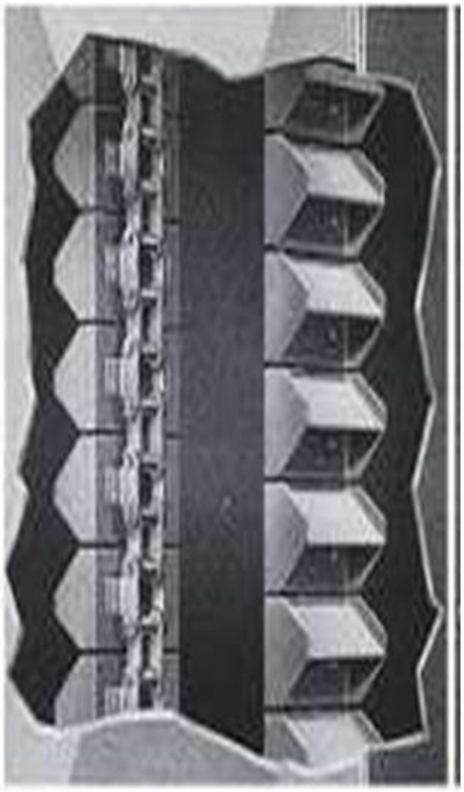

Fig.1 Sectional View of a Chain Bucket Elevator

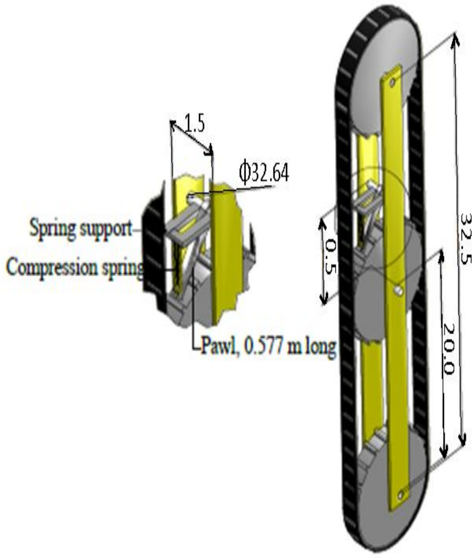

Fig. 2 Proposed Design with a Ratchet Mechanism

Design of the Beam

The beam is the horizontal member on which the pawl is mounted. It has fixed support as shown in the Figure 3.

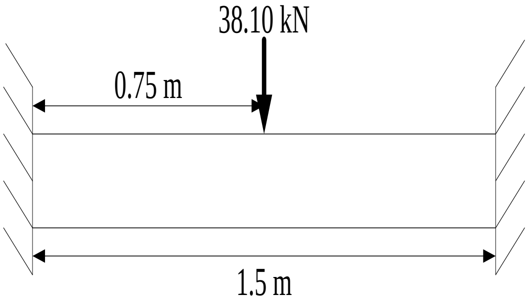

Fig. 3 Beam Support

Data collected from the plant on the existing bucket elevator;
Length of bucket elevator = 1.5 m
Weight of bucket fully loaded with cement = 36 kg
Mass of chain link per unit length = 88 kg/m
Height of bucket elevator = 32.5 m
The length of bucket elevator should be equal to the length of the beam for easy assembling.
∴ Length of beam = 1.5 m
The chains always fail within the range of 15 m to 25 m, finding the average of these two ranges gives 20 m
But the number of buckets at a height of 20 m = 59 buckets
∴ For 59 buckets, the Mass = 36 kg × 59 = 2124 kg
Mass per unit length of chain = 88 kg/m
Since the height is 20 m,
⟹ Mass of chain = 88 × 20 = 1760 kg
Total mass = mass of chain +mass of buckets = 1760 + 2124 = 3884 kg
Total weight = mg = 3884 × 9.81 = 38.10 kN

The free-body diagram of the beam is as shown in Figure 4.

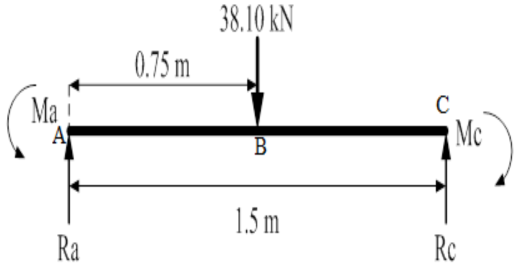

Fig.4 Free-Body Diagram

From Budynas and Nisbett (2011), for a beam with fixed support and with the load applied at the midpoint, from standard derivations,

$R_A = R_C = F/2$ (1)

$= 38.10/2 = 18.95$ kN

Also,

$M_A = M_C = (FL)/8$ (2)

$= (38.10 \times 1.5)/8 = 7.14$ kNm

Also, the Shear Force and Bending Moment for portion AB from standard derivation are given as;

$V_{AB} = F/2$ (3)

$= 38.10/2 = 18.95$ kN

$M_{AB} = \{F(4X - L)\}/8$ (4)

When X = 0 m; $M_{AB} = \{38.10[(4\times 0) - 1.5]\}/8 = -7.14$ kNm

When X = 0.75 m; $M_{AB} = \{38.10[(4\times 0.75) - 1.5]\}/8 = 7.14$ kNm

The Shear Force and Bending Moment for portion BC are also given as;

$V_{BC} = -F/2$ (5)

$= -38.10/2 = -18.95$ kN

$M_{BC} = \{F(3L - 4X)\}/8$ (6)

When X = 0.75 m; $M_{BC} = \{38.10[(3\times 1.5) - (4\times 0.75)]\}/8 = 7.14$ kNm

When X = 1.5 m; $M_{BC} = \{38.10[(3\times 1.5) - (4\times 1.5)]\}/8 = -7.14$ kNm

The Shear Force and Bending Moment diagram are as shown in figure 5.

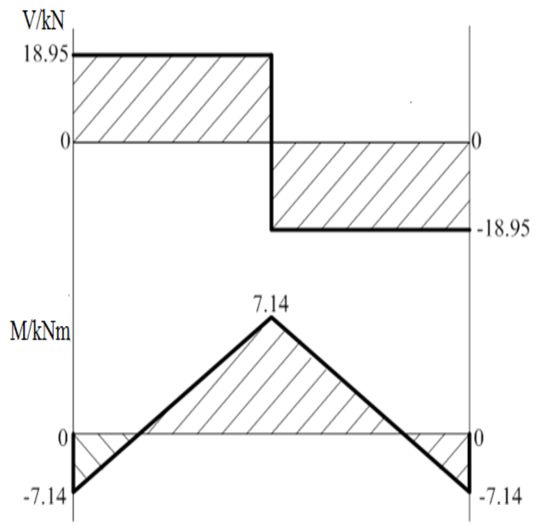

Fig.5 Shear Force and Bending Moment Diagrams

From Fig. 5, the maximum Shear Force and the maximum Bending Moment are 18.95 kN and 7.14 kN respectively.

The beam is subjected to two main types of stresses and these are: Bending and shear stresses.

From Budynas and Nisbett (2011), the maximum bending stress (σ_b) of a cylindrical beam is given as $\sigma_b = (32M)/(\pi d^3)$ (7)

Where M is the maximum bending moment

$\Rightarrow \sigma_b = (32\times 7.14 \times 10^3) / (\pi \times d^3) = 72727.443/d^3$

Also the maximum shear stress of a cylindrical beam is given as,

$\tau_{max} = (16V)/(3\pi d^2)$ (8)

Where V is the maximum shear force

$\Rightarrow \tau_{max} = (16\times18.95 \times 10^3) / (3\pi d^2) = 32170.519/d^2$

Using the distortion energy theory, the Von Mises Stress, ∇ according to Budynas and Nisbett, 2011 is given as;

$\nabla = S_y / n$ (9)

Where, S_y = Yield strength of the material and n= Factor of safety, taken as 5

According to Rajput (2010), the yield strength of AISI 1030 annealed steel is 430 MPa

$\Rightarrow \nabla = 430/5 = 86$ MPa

The Von Mises Stress, ∇, from Budynas and Nisbett, 2011is also given as;

$\nabla = \{ \sigma_b^2 + 3\,\tau_{max}^2\}^{0.5}$ (10)

$\Rightarrow 86\times10^6 = \{(72727.443/d^3)^2 + 3(32170.519/d^2)^2\}^{0.5}$

$7.396 \times 10^{15} = 5289280965/d^6 + 3104826878/d^4$

Multiplying through by d^6 / 3104826878 gives,

$2.382\times10^6\, d^6 = 1.704 + d^2$

$d^2\,\{2.382\times10^{\,6} d^4 - 1\} = 1.704$

Either $d^2 = 1.704$ or $2.382\times10^{\,6} d^4 - 1 = 1.704$

When, $d^2 = 1.704$; d = 1.305 m

Also, when, $2.382 \times 10^{\,6} d^4 - 1 = 1.704$; d= 0.03264 m = 32.64 mm

From the two diameters calculated, the diameter of the beam is taken as 32.64 mm because 1.305 m is too large.

Design of the Pawl

The pawl is the member that fits into the notch of the sprocket in order to lock the sprocket. It is rectangular in shape and from data collected,

The face width of sprocket = 50 mm

The tooth thickness of sprocket = 60 mm

Since the pawl is to fit into the notch of the sprocket,

The width of pawl should be = the face width of sprocket teeth

∴ The width of pawl = 50 mm

Also, the height of pawl = the tooth thickness of the sprocket teeth

∴ The height of pawl = 60 mm

From the proposed design, Figure 6 can be deduced and therefore length of the pawl, AC can be calculated.

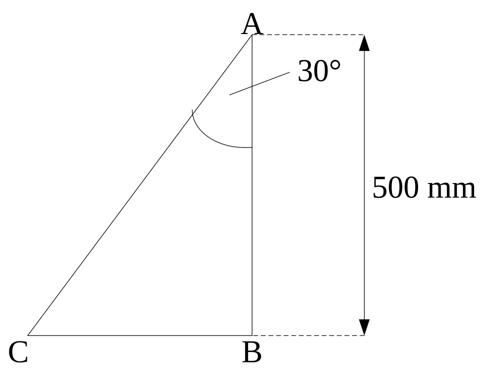

Fig.6 Pawl Length

From Figure 6,

The length of pawl, AC = 500 / cos30° = 577. 35 mm

Since the force acting on the pawl is 38.10 kN which is a compressive, the compressive stress is given by, $\nabla_c = -F/A$

$= -38.10/ (0.577 \times 0.05) = -1.3321$ MPa

From Euler formula, the critical load for a member fixed at one end and free at the other end is given by,

$P_c = (\pi^2 E\, I) / (4L^2) = (\pi^2 \times 210 \times 10^9 \times 0.05 \times 0.06^3) / (4 \times 12 \times 0.577^2)$

= 140.07 kN

Therefore the critical compressive stress is;

$\nabla_{cr} = 140.07 / (0.577 \times 0.05) = -4.855$ MPa

Since $\nabla_c < \nabla_{cr}$

$\Longrightarrow$The pawl will not fail when it is being acted on by the 38.10 kN compressive force.

Therefore the dimensions for the pawl are:

L = 577 mm
W = 50 mm
H = 60 mm

Design of the Spring

In the design of the spring, because of space considerations, the total number of turns and the overall length of the spring are specified.

Total number of turns N_t= 20 turns

Overall length, L_o= 300 mm

Also, the type of end of the spring is squared and ground because springs with squared and ground ends offer better transfer of loads.

Diameter of spring wire is chosen to be 2 mm so that the spring can withstand the load acting on it.

Table1 Dimensional Characteristics of Compression Spring (N_a = Number of active coil)

	Type of Spring Ends			
Term	**Plain**	**Plain and ground**	**Squared or closed**	**Squared and ground**
End coils N_e	0	1	2	2
Total coils N_t	N_a	$N_a + 1$	$N_a + 2$	$N_a + 2$
Free length L_o	$pN_a + d$	$p(N_a + 1)$	$pN_a + 3d$	$pN_a + 2d$
Solid length L_s	$d(N_t + 1)$	$d\ N_t$	$d(N_t + 1)$	$d\ N_t$
Pitch P	$(L_o - d)/N_a$	$L_o/(N_a + 1)$	$(L_o - 3d)/N_a$	$(L_o - 2d)/N_a$

(Source: Budynas and Nisbett, 2011)

From Table 1,
$N_t = N_a + 2$ ∴Number of active turns,
$N_a = N_t - 2 = 20 - 2 = 18$ turns
From the table, Solid Length, $L_s = dN_t = 2 \times 20 = 40$ mm
and Pitch, $P = (L_o - 2d)/N_a = (300 - 2\times2)/18 = 16.44$
But also, from Budynas and Nisbett, 2011,
L_o = deflection (y) + L_s (11)
$\Longrightarrow$ y = 300 − 40 = 260 mm
For stability of spring,
$L_o \leq 5.26D$

$\Longrightarrow L_o \leq 5.26(100) \leq 526$ mm
Since $L_o(300) < 526$ mm, It implies that spring is stable.
From Budynas and Nisbett, 2011, the outside diameter (OD), and inside diameter (ID) of the spring are given as; Outside diameter, OD = D + (12)
= (100 + 2) mm =102 mm
And inside diameter, ID = D − d (13) = (100 − 2) mm = 98 mm

III. RESULTS AND DISCUSSION

The proposed design in Figure.2 introduces a ratchet mechanism which will hold the chain link and bucket in position in order to address the difficulty in joining the chain links together which restricts the chains with the buckets mounted on it to move only in the counter clockwise direction. The design calculations involve calculations on the design of the beam, pawl and spring. Figure 5, shows the Shear Force and Bending Moment diagrams and the maximum values are 18.95 kN and 7.14 kNm respectively. From the two diameters calculated, the diameter of beam chosen is 32.64 mm because 1.305 m is too large. The compressive stress on the pawl is 1.3321 MPa. The critical compressive stress is 4.855 MPa which is greater than the compressive stress. Therefore a force of (38.10 kN) will not cause the pawl to fail in compression.

IV. CONCLUSION

A ratchet mechanism which consists of a toothed wheel which is the rotating member, a pawl which serves as a lock and a spring which holds the pawl against the teeth of the toothed wheel have been introduced into the already existing gear drive bucket elevator and this will help prevent the chain together with the bucket from dropping to the bottom whenever there is failure. This will help reduce down time which will increase production. The mechanism will prevent damage to buckets and chain links. This will, therefore, reduce maintenance cost.

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