Design for Production of Single Flight Screw Worm from White Cast Iron Using Sand Casting Technique

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Abstract
The design for production of single flight screw worm via sand casting using white cast iron was done. It takes into consideration the design aspect of engineering components coupled with its working drawings and principles, in this case, the screw worm and its application. The design entailed designed for shaft diameter, volume of material transfer per one revolution, factor of safety for live loading was designed, depth of the groove (both upper width and lower width) of the grooving and speed of conveyor shaft pulley. The design limitation was equally stated and selections of the materials for the production of shaft were equally undertaken where mild steel shaft diameter was calculated to be 77 mm through design analysis which eventually determined the internal diameter of the screw worm conveyor. The pulverizing (grinding) machine system on which the conveyor would be employed would be powered by a prime mover (electric motor, petrol or diesel engine) of between 1 kw to 1.5 kw rating. The modest-choice of prime rating was based on considerations of cost, availability, and transportation weight savings of the power unit

Keywords: design, production, flight screwworm, shaft, live loading, grinding machine, mild steel slicer

INTRODUCTION
Screw worms conveyors normally fail in a grinding machine due to continuous rubbing of screw surface due to particles from pulverizing grains; the constant rubbing of surface between the worm and worm housing; and the load from material for grinding. The design considerations to alleviate failure are a single worm conveyor with the ribs of its screws of trapezoidal cross section between divergent flanks to minimize wear of the housing surface has been selected and the trapezoidal shape of the ribs is so chosen such that they widen outwardly, provides forces in operation of the conveyor which are in part turned inwardly, thereby reducing the outward pressure with which the pulverizes material is directed against thinner surface of the housing. Grains and desirable powder play major role in poultry keeping and also in the staple food chain of majority, not only in Nigerians and indeed Africans. The above fact demanded for an efficient, easily available, cheap and strong screw conveyor system in pulverizes which could be powered by electric motor or fossil fuel prime movers. Of the grains, millet, guinea corn, beans, corn that forms the staple food of Nigerians, corn (being the hardest when dry) has been selected in this design consideration. There are different types of pitches such as single flight standard pitch, single flight half pitch, single flight variable pitch, single flight short pitch, single tapered flight standard pitch, double flight standard pitch, double flight standard pitch, ribbon flight, single cut-flight, and standard pitch with paddles. Oyetunji (2009) worked on Development of Small Injection Moulding Machine for Forming Small Plastic Articles for Small-Scale Industries but little or no work had been reported on design for production of single flight screw worm from white cast iron using sand casting technique. The objectives of the design are to determine an internal diameter of the conveyor hole that receives the shaft that bears the stresses of conveyance and grinding without failure and hence; the determine the diameter and type of shaft on which the conveyor will be keyed after the conveyor has been transitionally mounted

EXPERIMENTAL PROCEDURES
Boundary Conditions
Of the grains, millet, guinea corn, beans, corn, etc that constitutes the staple food of Nigerians, corn, being the hardest when dry, was selected in this design consideration, and design of single flight screw worm.

DESIGN OF SINGLE FLIGHT SCREW WORM
Design Analysis
The concept is based on initialization of the screw conveyor in the grain pulverizes (grinding machine). For the design of this work, therefore, 1.5 KW rated prime mover has been specifically selected.
Speed of Conveyor
The belt and pulley system as shown in the design schematic layout drawing as shown in Fig. 1, had been used to derive suitable torque which would be available for pulverizing the grains at the cracker unit, via a 150 mm; 300 mm pulley ratio, at 900 mm distance between pulley centers. Therefore, the diameter of motor pulley, \(D_1\) being 150 mm and the diameter conveyor shaft pulley, \(D_2\) was 300 mm, the distance between the two pulley centers of the system was put at the 900 mm.

Speed of \(D_1\) being 150 mm and the pulley ratio = 900 mm

\[ \text{Speed of conveyor shaft pulley} = \frac{D_1}{D_2 \times 1500} \text{ r.p.m} \]

\[ = \frac{150}{300 \times 1500} \]

\[ = 750 \text{ r.p.m} \]

As can be seen in Fig. 1 (See Appendix) (the schematic layout drawing), the shaft bearing the conveyor was subjected to combined bending and torsional stresses as the system was operated. This was as shown in the system force diagram, Fig. 2, configured from the system.

**Design Calculations**

**Bending Stresses**
The weight of the grains in the hopper when full and the weight of the conveyor were taken to act centrally on the shaft (within the length of the conveyor). The resultant pull of the pulley belt also constituted loading on the conveyor shaft. These force with the reaction of the roller bearings \(R_a\) and \(R_b\) determined the bending moment on the system conveyor shaft (Khurmi and Gupta, 2006).

**Torsional Stress**

As shown in Fig. 3, there would be to turning during grinding which must be overcome by the torque from the motor, belt and pulley drive system. Then, torsional stresses would be set up in this wise. For maximum capacity of the conveyor shaft to cope, the worst scenario of the teeth cracker coming together and having locked was assumes for this condition and possibility of misuse or poor handling, the factor of safety, F.S. was upgraded 1½ (one and half) times in the design calculations to cater for the above stated exigencies. Therefore the shaft bearing the conveyor could be of diameter and material capable of bearing the above stresses without failure.

**Determination of Forces on System**
The forces on the conveyor shaft are determined as \(F_s\), \(W\), \(R_a\) and \(R_b\) as well as pulverizer torque \(T_p\) were determined as b

\[ T_p = \frac{\pi \mu D_1 V}{2} \text{ Watts} \]

\[ \mu = 0.1 \]

\[ D_1 = 150 \text{ mm} \]

\[ V = 0.7 \text{ m/s} \]

\[ T_p = 5.5 \text{ Watts} \]

\[ \frac{T_1}{T_2} = \frac{D_1}{D_2} \]

\[ T_1 = T_2 e^{\mu D_1} \]

\[ \mu = 0.1 \]

\[ D_1 = 150 \text{ mm} \]

\[ D_2 = 300 \text{ mm} \]

\[ T_1 = 5.5 e^{0.1 \times 150} \text{ Watts} \]

\[ T_2 = 5.5 e^{0.1 \times 300} \text{ Watts} \]

From equation 5

**Figure 2:** System Force Diagram

These stresses were as analyzed in sections 1 and 2.

**Figure 3:** Bending moment acting on the conveyor shaft.
\[ 1.5 \times 1000 = T_1 \left(1 - \frac{1}{\mu^2}\right) \left(\frac{2 \pi r}{60} \times \frac{15}{2} \times 1500\right) \]
\[ 15000 = T_1 \left(1 - \frac{1}{\mu^2}\right) \times 706.86 \]
\[ T_1 \left(1 - \frac{1}{\mu^2}\right) = \frac{1500 \times 60}{706.86} \]

Taking \( \mu \) to be 0.3 as coefficient of friction between the rubberized belt and pulley
\[ \mu = 0.3 \times 2.975 = 0.893 \]
\[ \therefore \ T_1 = \frac{1500 \times 60}{706.86 \left(1 - \frac{1}{\mu^2}\right)} \]
\[ = \frac{90000}{706.86 \times 0.6} \]
\[ = 212.2N \]

From equation 5,
\[ T_2 = \frac{T_1}{\mu^2} = \frac{212.2}{2.44} \]
\[ = 87N \]
\[ \therefore \text{Resultant shear force on shaft at pulley point} = T_1 + T_2 \]
\[ = 212.2 + 87 \]
\[ = 299.2N = T_2 \text{, resultant shear force on pulverezer shaft.} \]

Referring to Fig. 4,
\[ P = \frac{P_0 \omega}{2} \]
\[ \text{where } P = \text{Power of motor} \]
\[ T_2 = \frac{2m \times 1.50C}{60} \]
\[ = \frac{P \times \frac{D_2}{2}}{\omega} \]
\[ = \text{Torque at conveyor shaft} \]
\[ = \frac{1500 \times 60^2 \times 300^2}{2 \times \pi \times 1500 \times 150} = 19.1N.m \]

Volume and Weight of Hopperful of Dry Grains

\begin{align*}
\text{Volume of Hopper} &= V \text{ol ABCDI = EFCHI} \\
&= \frac{1}{3} \left( (60 \times 46 \times 53.5) - (10 \times 15 \times 13.4) \right) \\
&= \frac{1}{3} \times [147660 - 2010] \\
&= \frac{1}{3} \times 145,650 \\
&= 48,550 \text{cm}^3
\end{align*}

The SG (specific gravity) of corn at 0% moisture was taken to be 1.5 (http://google/corn density); and estimated mass of material for casting the worm screw conveyor was 6kg.

\[ \text{Therefore mass of hopperful for corn} = \frac{4880}{1.5 \times 100} = 32.57 \text{kg} \]

\[ \text{Mass of corn and conveyor} = (32.57 + 6) \text{kg} = 38.57 \text{kg} \]

\[ \text{Weight of corn and conveyor} = 38.37 \times 9.81 = 37.641N \]

Comparing similar triangles from fig. 5,
\[ \frac{\Delta \text{DOI}}{\text{HII}} = \frac{\text{DI}}{\text{HI}} \]
\[ \text{i.e. } \frac{30}{\text{HI}} = \frac{46 + \text{HI}}{\text{HI}} \]

Referring to shaft loading in fig.2,
\[ W + P_2 - R_A - R_B = 0 \] ..........................(11)
\[ \text{i.e.} \quad 376.41 + 299.2 - R_A - R_B = 0 \quad (12) \]

Where 
\[ R_A \text{ and } R_B \text{ are conveyor shaft bearings as reactions.} \]

Also bending moment, \( M \) is
\[ N = (376.41 \times 0.79) + (299.2 \times 0.68) - (R_B \times 1.68) - (R_A \times 10) = 0 \]
\[ \text{i.e.} N = 297.93 + 205.56 - 166R_B - 36R_A = 0 \] ..........................(13)

\[ R_B = \frac{108.73}{0.36} - \frac{0.168}{0.36}R_A \]
\[ = 302.02 - 0.467R_A \]

\[ \text{Subst. } R_B \text{ in equation } (13) \]
\[ 376.41 + 299.2 - R_A - (302.02 - 0.464R_A) = 0 \]
\[ 375.58 - 0.533R_A = 0 \]
\[ \therefore R_A = \frac{375.58}{0.533} \]
\[ = 704.65N \]

\[ \text{Subst. } R_A \text{ in equation } (13) \]
\[ \text{therefore,} \quad 376.41 + 299.2 - 704.65 - R_B = 0 \]
\[ \therefore R_B = -31.1N \]

To determine the bending moment, \( N, R_A, \text{ and } R_B \text{ are substituted in equation } (13) \)
\[ i.e. N = 108.73 - (.168 \times 704.65) - (.36 \times -31.1) = 0 \]
\[ 108.73 - 110.38 + 11.2 = 0 \]
\[ M = 1.6N \]

**DETERMINATION OF SHAFT DIAMETER**

By Guest theory, (John and Stephen, 1958)
\[ T_e = \frac{4T_x}{3} \] ..........................(14)
where \( T_x \) is equivalent torque which would produce maximum stress.

Also
\[ T_x = \sqrt{M^2 + T^2} \] ..........................(15)
\[ \bar{T} = \text{Maximum permissible shear stress,} \text{torque} = \frac{1}{2} \sqrt{\sigma^2 + 4\tau^2} \] ..........................(16)

\[ Zt = \text{polar moment of inertia } (M^2) = \frac{\pi d^4}{16} \]

For mild steel loading, normal factor of safety (F.S) for live loading 8<F.S<20
For this project this is modified by 1.5 factors to cater for possible misuse, jamming or locking in place of the cracker plates teeth.
Therefore F.S for this design was
\[ 20 \times 1.5 \]
\[ \therefore F.S = 30, \]
\[ \frac{\text{Ultimate Direct Stress}}{\text{Design stress}} \]
\[ \text{For direct Bending stress } \sigma \text{ if UDS in } 160\text{MN/m}^2 \]

Then
\[ 30 = \frac{160\text{mn}}{\text{Design stress or}} \]

**Design Direct stress \( \sigma \)**
\[ \frac{160\text{MN/m}^2}{30} = 5.33\text{MN/m}^2 \]

Also

For shear stress, if ultimate shear stress is 45N/m²

Then for FS of 30,
\[ 30 = \frac{45}{\text{Design shear stress}} \]

**Design shear stress**
\[ \text{Design shear stress } \tau = \frac{45}{30} \text{MN/m}^2 \]
\[ = 1.5\text{MN/m}^2 \]

But

For Maximum permissible shear stress \( \bar{T} \), by Guest's theory
\[ \bar{T} = r + \frac{1}{2} \sqrt{\sigma^2 + 4\tau^2} \] ..........................(19)
\[ = r + \frac{1}{2} \sqrt{5.33^2 + 4(1 - 5)^2} \]
\[ \bar{T} = r + \frac{1}{2} \sqrt{28.41 + (4 \times 2.25)} \]
\[ \bar{T} = r + \frac{1}{2} \sqrt{30.41 + 9} \]
\[ \bar{T} = r + \frac{1}{2} \sqrt{37.41} \]
\[ \bar{T} = \frac{1}{2} \times 6.12\text{MN/m}^2 \]
\[ \bar{T} = 3.06\text{MN/m}^2 \]

Subst. in equation 19 above
\[ 29.16 \times 10^3 = 3.06 \times Zt \]
\[ Zt = \frac{29.16 \times 10^3}{3.06} \]
\[ \pi d^3 = 29.16 \times 10^3 \times 3.06 \]
\[ d^3 = 29.16 \times 10^3 \times 3.06 \]
\[ d = \frac{29.16 \times 10^3}{3.06 \times \pi} \]
### Volume of Material Transfer per One Revolution

\[
d = \sqrt{\frac{29.16 \times 10^2 \times 16}{3.06 \times \pi}} = \frac{466.560}{9.6132735} = \frac{544.44126311}{76.882mm}
\]

Hence volume \( V = \pi DE \times \frac{1}{2} (L + M) \times H \) (cross sectional Area)

Considering figure 6

Cross sectional Area \( A = \frac{1}{2} (L + M) \times H \)  \( \text{(21)} \)

Volume \( V = \pi DE \times \frac{1}{2} (L + M) \times H \)  \( \text{(22)} \)

\[
V = \frac{3\pi}{2} \left[ D - \frac{2H}{3} \right] (L + M) \quad \text{(23)}
\]

\( V \) = Volume of material transfer per one revolution

\( H \) = Depth of the groove

\( D \) = Maximum external diameter of the screw

\( L \) = Upper width of the groove

\( M \) = Lower width of the groove

\[
V = \frac{3\pi}{2} \left[ 122 - \frac{2\times12}{3} \right] [47.6 + 39.04]
\]

\(
= 18.85[122 - 8][86.84] \\
= 18.85[114][86.84] \\
= 189610.476mm^3
\)

Hence \( V = 189.6m^3 \)

### Primemover and Selection

The pulverizing (grinding) machine system on which the conveyor would be employed would be powered by a prime mover (electric motor, petrol or diesel engine) of between 1 kw to 1.5 kw rating. The modest-choice of prime rating was based on considerations of cost, availability, and transportation weight savings of the power unit. This consideration was considered important because, the type of pulverizers (grinding machines) for which these worm conveyors are designed would mostly acquire or owned and operated by low and medium citizens.

### CONCLUSIONS

The following conclusions were drawn from the work: The design entailed designed for shaft diameter, volume of material transfer per one revolution, factor of safety for live loading was designed, depth of the groove (both upper width and lower width) of the grooving), and speed of conveyor shaft pulley. The mild steel shaft diameter was calculated to be 77 mm through design analysis which eventually determined the internal diameter of the screw worm conveyor.

### REFERENCES


www.patent genius.com/double flight screw worm (2010)


www.foundry solutions and Design.com (September, 2011)

www.continental screws.com (September, 2011).

http: Google search/corn density /Specific gravity (September, 2011).

APPENDIX