Design and Fabrication of an Excavator Used in Starch Industries

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Project Report 1988-89

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Certificate

Name .... Roll No. ....

This is the Bonafide Record of the Project titled

Design and Fabrication of an Excavator

Used in Starch Industries

Done by

Mr. ....

In partial fulfilment of the requirements for the Degree of Bachelor of Engineering in Mechanical Engineering Branch of Bharathiar University, Coimbatore-641 046 during the academic year 1988-89

....

Head of the Department

....

Staff in-charge

Submitted for the University Examination held on ....

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Internal Examiner

....

External Examiner

Dedicated to our
Beloved Parents

ACKNOWLEDGEMENT

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Coimbatore - 6.

TEAM MEMBERS.

1. SYNOPSIS

The starch industries in Salem and Dharmapuri districts in Tamil Nadu produce starch from TAPIOCA TUBERS. They use conventional methods for tapping the starch from the settling tanks and the starch is dried in the drying yards. The starch tapping and breaking into small pieces before spreading in the drying yards consume a lot of time and manpower and infact the labours become tired before finishing the process. The labour cost is more in this process.

In this project, we have combined both the tapping and breaking processes togather and tried to mechanise the whole operation. By mechanising the operation we have reduced the unnecessary manpower utilisation, labour cost and time consumption. This increases the capacity of the plant.

2. INTRODUCTION

The STARCH industries in Tamil Nadu use TAPIOCA TUBERS as raw material for starch production. The tapioca is cultivated in more than twelve districts in Tamil Nadu and the yield is more than the world average production per acre.

In the starch industries, the tapioca is first washed with water. And the black and white skins over the tubers are removed by hand. The skin removed tubers are crushed in crushers with water to get a tapioca - water crush. The water is separated from that crush using filters. The water will contain the starch particles and look like a milky liquid. The milky liquid is again filtered with a filter of 200 - 300 mesh size and filled in settling tanks to allow the starch particles to settle down. After sometime all the starch particles are settled under water due to gravity and forms a thick continuous layer of starch at the bottom of the tank. The thickness varies from 0.5 cm to 10 cms. Then the water is removed from the tank and now the starch is ready for drying.

The starch from the settling tank is tapped by hands and taken to the drying yards. In the drying yards it is broken into small pieces and spreaded on to dry up in sunlihgt. After drying, the starch is filled in bags, weighed and marketted. This is the process carried out in starch production.

The process of tapping the starch from the tanks is a time and labour consuming process and the old and conventional method of starch tapping using 'MURAMS' consumes a lot of

3. NEED FOR AN EXCAVATOR

The need of an excavator for starch industries may be realised with the following factors.

1) The reduction in time in the starch production decreases the factory overheads.

2) The reduction in manpower consumption considerably decreases the expenses in production.

3) This increases the capacity of the plant per day.

4) The drying time of starch in the drying yards may be reduced by suitably controlling the size of the starch pieces.

5) This elimates the tiresome work of tapping.

6) When centrifuges are used before drying artificially with driers in future, at that time also the excavators have to be used for tapping and easy handling of starch.

7) Since the size of starch in the drying yards is properly controlled, the drying will be even and the starch produced will be of uniform moisture content. This is a step towards the quality production.

8) Since starch is also a food product, excavator avoids the handling of starch with hands and the production becomes hygienic.

9) It is a step towards the technology upgradation in the industry.

manpower. This increases the factory overheads and inturn the production cost. Similarly the breaking of big starch cakes into pieces in the drying yards also consumes a lot of time and manpower and increases the time of production of starch which limits the capacity of the plant per day.

So, this project is aimed at the reduction of process time and manpower which reduces the factory overheads.

sing the processes. Here in this project, the 'EXCAVATOR' is designed to be used inside the tank to tap, break and fill the starch in the baskets, so that it can be easily taken to the drying yards. Mechanisation of the process considerably reduces the time in tapping and breaking and decreases the factory overheads. At the same time, since the time consumed is reduced very much, the starch may be dried twice per day which ultimately doubles the

capacity of the plant

5. SPECIFICATION OF THE MACHINE

Capacity of the excavator	=	19.84 Tonnes / hour.
Number of baskets tapped per minute	=	2
Overall height	=	1325 mm.
Overall length	=	1547 mm.
Overall width	=	730 mm.

Motor : 5 HP at 1440 rpm.

6. POWER REQUIREMENTS

6.1 Force Required for Lifting Starch in the Front Wedge

The angle of inclination of the wedge from the horizontal for lifting the above weight effectively upward beyond the limiting friction in the wedge must be less than the angle of repose (θ) which is given by the formula,

$$= 45° - \frac{\phi}{2} \quad \text{where } \phi = \text{frictional angle.}$$

Comparing rough wood on metals, we select the friction angle to be 29°

$$\therefore \text{Angle of repose } \theta = 45 - \frac{29}{2}$$

$$= 30.5$$

So, the wedge angle $= 30°$

Volume of starch to be lifted $= 8 \times 55 \times 31$

$= 13640 \text{ cm}^3$

It is found weight of 1 cm³ Starch $= 1.253$ *gms.*

$\therefore$ *Weight corresponding to 13640 cm³ Starch* $= 13540 \times 1.253$

$= 17090.9$ *gms.*

$= 17.091$ *kg.*

We know, external horizontal force required to lift 17.091 kg. $= \dfrac{W \sin(\theta+\phi)}{\cos\theta}$

$$= \frac{17.091 \times \sin (30 + 29)}{\cos 29}$$

$$= \underline{16.75 \text{ kgf.}}$$

Force required for cutting the starch

To find the cutting force required to cut 1 cm^2 area of starch, a specimen starch cake was subjected to similar conditions as in the machine. It is found that it requires 2.4 kg weight to cut a cross section of one square centimetre.

Cutting force required to cut 1 cm^2 area $= 2.4$ kg/cm^2

The machine has to cut the starch of height 8 cm on one side

On both sides it has to cut totally a height of 16 cms.

The power is calculated for cutting a unit length of starch.

$\therefore$ Force required, for cutting = cutting strength x area to be cut

$= 2.4 \times (16 \times 1)$

$= 38.4$ kg.

Power required for cutting and lifting in the wedge

Force required for lifting = 16.75 kgf.

Force required for cutting = 38.4 kgf.

Total force = 16.75 + 38.4

= 58.15 kgf.

But power required = Force x velocity

The machine is designed to move with a velocity of 6 m/minute so as to tap two baskets of starch in one minute.

$\therefore$ *power required* $= 55.15 \times 6/60$

$= 5.515$ *kgm/sec*

$= 54.1$ *Watts.*

With factor of safety n $= 3$

Horse power required $= \dfrac{54.1 \times 3}{736}$

$= 0.221$ *HP.*

6.2 Power required by the conveyor

Volume of starch coming above the conveyor at any istant = *height of starch to be cut x width of wedge x length of the conveyor*

$= 8 \times 55 \times 31$

$= 13640 cm^2$

Weight of starch coming above the conveyor = *Volume of starch x weight per* cm^3.

$= 13640 \times 1.253$

$= 17.091$ *kg.*

We know,

Force required to move the ablve weight up,

$$= \frac{W \sin(\theta + \phi)}{\cos\theta}$$

where,

= *angle of inclination of the conveyor with horizontal*

= *30 degrees.*

$= \text{Friction angle.}$

$= \tan^{-1}\mu$

Since the conveyors rollers rest on bearings, the friction in the ball bearing will be.

$= 0.001$

$\therefore \quad = \tan^{-1} 0.001$

$= 0.05729 \text{ degrees}$

$\therefore$ Force required $= \dfrac{17.091 \times \sin(30+0.05729)}{\cos 0.05729}$

$= 8.56 \text{ kgf.}$

This is the force just required to move the starch upwards

$\therefore$ power required to move the stach upwards with the velocity of 6 m/min.

$= 8.56 \times 6/60$

$= 0.856 \text{ kgm/sec.}$

$= 8.4 \text{ watts.}$

$= 0.015 \text{ HP}$

6.3 Power required to move the starch on the plates below the cutters.

Width of the plate after the conveyor $= 20$ cms

Length of the plate $= 55$ cms.

Volume of starch coming Over the plate $= 20 \times 55 \times 8$ (height of starch)

$= 8800 \text{ cm}^3$

Weight of starch $= 8800\ cm^3 \times 1.253\ gms/cm^3$

$= 11.026\ kg.$

Friction angle between and plate is expected to be 29 degrees.

Force required to lift the starch $= \dfrac{W \sin(\theta + \phi)}{\cos\theta}$

We know, $= 30$ degrees.

$\therefore$ Force required $= \dfrac{11.026 \times \sin(30 + 26)}{\cos 29}$

$= 10.805\ kgf.$

This starch is to be lifted with the velocity of 6m / min.

$\therefore$ Power required for lifting $= 10.805 \times \dfrac{6}{60}$

$= 1.08\ kg.m/sec.$

$= 0.014\ HP$

6.4 Power required by the reciprocating cutter

Height of starch $= 8$ cms

But the starch cake is inclined at an angle 30° from the horizontal.

$\therefore$ Actual travel of the cutter $= \dfrac{8}{\cos 30}$

$= 924$ cms

After crossing the circular cutter the continuous starch cake will be sliced longitudinally and each slice will be of 22 mm width. It will be enough to cut 1 cm width of the slice from the top to get the pieces separated from the slices. There will be 22 number of slices.

ie width of cut	= 22 cms
∴ Area to be cut	= 9.24 x 22
	= 203.3 cm^2
Force required for cutting or chapping	= Area x cutting strength
	= 203.3 x 2.4
	= 487.92 kgf.
Work done during one cutting	= Average force x distance travelled
Average force	$= \frac{487.92}{2}$
Work done	$= \frac{487.92}{2} \times 9.24$
	= 22.54 kgm
Speed of shaft	= 300 rpm
∴ HP needed to drive the cutter	$= \frac{\text{Work done x Speed}}{4500}$
	$= \frac{22.54 \times 300}{4500}$
	= 1.5 HP.

6.5 Power required by the elevator

The elevator designed in such a way that it can elevate the amount of starch tapped when the machine is moving with a velocity 6m / min.

Amount of Starch tapped per hour $= \dfrac{\text{Weight of Starch}}{\text{time}}$

$$= \frac{55 \times 8 \times 600 \times 60 \times 1.253}{1000 \times 1000}$$

ie capacity of the elevator = 19.84 ton / hr.

Velocity of the belt (V) = 1.47m/sec, choosen for high speed centrifugal discharge

We know,

power required by the elevator (N) $= \dfrac{QH}{367} \times 1.15 + k_2 K_3 V)$

Where K_2, K_3 = Factors

H = Height to which the load is elevated, m.

= 0.85 metres

K_2 = 1

For V buckets

K_3 = 1.1

∴ Power required $= \dfrac{19.84 \; 0.85}{367} = 1.15 + (1.1 \times 1.47)$

= 0.127 KW

= 0.173 HP

Capacity of the Buckets

We know,

$$\frac{i_o}{a} = \frac{Q}{3.6 \; V}$$

Where

a = Pitch of bucket, m

= 2.8 x height of the bucket, m.

= Bulk density of conveyed material ton/m^3

= 0.65

= bucket loading efficiency

= 0.75

$$\therefore \frac{i_o}{a} = \frac{19.84}{3.6x1.47x0.65x0.75}$$

$= 7.69$

$\therefore i_o = 7.69 \times (2.8 \times 0.05)$

$= 1076\ cm^3$

ie volume of each bucket should be 1076 cm^3.

Let A = Projection from the belt face

= a max x m

Where

a_{max} = Maximum dimension of the Starch pieces

= 20 mm

m = 4.25 for 80% of the material is of size a_{max}

= 90 mm

The bucket is sized in such a way that it will have a projection of 10 cms from the belt face and a height of 5 cms. The bucket is of 'V' type.

Minimum weight of the machine

Horizontal force against the movement of the machine = cutting force + Lifting force required in the wedge

$= 38.4 + 16.75$

$= 58.15$ kg

This horizontal force must be equated to the multiplication of the normal force ie, weight of the machine and the co-efficient of friction between the cast iron wheepl and the tank.

The rolling friction between the C.I. Wheel and the tank = 0.12 assume

We know,

Horizontal force against the movement $= \mu \times R_N$

Where R_N = Weight of the machine

$\therefore$ Weight of the machine $= \dfrac{\text{Horizontal force}}{\text{co-efficient of friction}}$

$= \dfrac{55.15}{0.12}$

$= 459$ kg.

Since body of the machine is the major weight constituting element, the thickness of the body plates may be accordingly choosen to fullfil the weight requirements the thickness of the body plate may be 8 mm.

6.6 Power required to move the machine

To move the machine forward, first of all the machine has to overcome the frictional force. Then some energy is spent in moving the machine with the required velocity of 6 metre per minute.

We know,

Frictional force $P = \mu \times R_n$

But $= 0.12$

Weight of the machine R_N	= *495 kg of the machine + Weight of Starch on the front wedge and conveyor + Weight of starch in the buckets + Weight of Starch in the basket.*
Weight of Starch in the front conveyor, wedge and plate	= *17.091 + 17.091 + 11.026*
	= *45.208 kg.*
Weight of Starch in the buckets of the elevator	= *Volume of one bucket x bulk density x Weight of Starch per cm^3 x no of buckets with Starch at any instant*
	= *1141 x 0.65 x 1.253 x 6*
	= *6.505 kg.*
Weight of Starch in the basket	= *Volume of the basket x bulk density x 1.253*
	= *41902.5 x 0.65 x 1.253*
	= *34.127 kg.*
∴ *Total weight of the machine*	= *459 + 45.208 + 6.505 + 34.127*
	= *544.5 kg.*
∴ *Force required to overcome the friction*	= *Total weight x co-efficient of friction*
	= *544.5 x 0.12*
	= *65.46 kgf.*

The velocity of the machine is so selected that the machine will fill two baskets.

To tap two baskets of Starch, the machine has to move with a velocity of 6m/min when the average thickness of starch is 3.5 to 5 centimetres.

We know,

Force required for movement = Mass of the body x Acceleration

Mass of the body $= \frac{544.5}{9.81}$

$= 55.5$ kg.

Also,

Under constant acceleration,

$$V^2 = u^2 + 2\ ax$$

Where

a = acceleration of the machine

x = distance travelled by the machine

= unit distance

u = Initial Velocity = 0

V = Velocity of the machine at a distance x from the Starting point.

The machine should attain the velocity of 6m/min within unit distance travel.

On rearranging,

$$2ax = V^2 - u^2$$

$$a = \frac{V^2 - u^2}{2\ ax}$$

$$= \frac{\left(\frac{6}{60}\right)^2 - 0}{2(0.01)}$$

$$= 0.5\ m/sec^2$$

$\therefore$ Force required for the movement of the machine $= 55.5 \times 0.5$

$= 27.75$ kgf.

Total force required by the machine for overcoming friction and for movement $= 65.46 + 27.75$

$= 93.21$ kgf.

ie torque required on the wheels $= 93.21 \times \frac{D}{2}$

Where D = diametre of the wheel

= 0.15 metre

$\therefore$ Torque $= 93.21 \times \frac{0.15}{2}$

$= 6.991$ kg.m

For 6m/min travel, the r.p.m of the wheel = DN

$6 = \quad \times 0.15 \times N$

$\therefore N = 12.73$ rpm

Power required by the machine $= \frac{2 \times \quad \times 12.73 \times 6.991}{4500}$

$= 0.124$ HP.

6.7 Power Required by the Rotating Cutter

The rotating cutter slices the continuous starch cake longitudinally and works with milling principle.

The cutting force = Shear resistance x Area

Resistance to shear = $2.4\ kg/cm^2$

Dia of the cutter(d) = 25 cm.

Depth of cut (a) = 8 cm.

Angle of contact of the cutter $= \sin \Psi_s = \dfrac{2\sqrt{a(d-a)}}{d}$

Reference : HMT Production Technology

Where Ψ_s = Angle of contact

$$\Psi_s = \sin^{-1} \frac{2\sqrt{8(25-8)}}{25}$$

= 68.89 Degrees.

Contact length of the cutter

$$= \frac{2\pi \frac{d}{2}}{360^\circ} \times 68.89^\circ$$

= 15 cm.

There are about 22 cutter plates of thickness 1.5 mm each. For calculation purposes, they are assumed to constitute a single cutter

Total width of cutters = 1.5 x 22

= 33 mm.

= 3.3 cm.

The mean chip thickness h_M is related to the dia of the cutter, depth of cut, revolution of the cutter per minute and the feed rate as

$$h_M = \frac{s}{n\ z} \sqrt{\frac{a}{d}}$$

Where s = Feed rate

= 6000 mm/min.

n = Speed of the cutter

=75 rpm.

z = Number of teeth

= 50

$$\therefore h_M = \frac{600}{75 \times 50} \sqrt{\frac{8}{25}}$$

= 0.09 cm.

When one tooth of the cutter passes over the starch it traces 15 x 3.3 = 49.5 cm^2 area and yields 49.5 x 0.09 = 4.455 cm^3 starch powder ie. chips.

Energy required by one tooth to remove 4.455 cm^3 = 2.4 x 49.5 x 0.09

= 10.69 kg - cm

= 0.1069 kg - m.

This is nothing but the torque required for the removal of 4.455 cm^3 starch.

$\therefore$ Torque required by all 50 teeth to cut = 0.1059 x 50

= 5.35 kg - m.

We know

$$HP = \frac{2\pi NT}{4500}$$

$$= \frac{2 \times \Pi \times 75 \times 5.35}{4500}$$

$$= 0.56 \ HP$$

6.8 Total Power of the Motor

$$\text{Power of the motor} = \frac{K_1 + K_2 + K_3 + K_4 + K_5 + K_6 + K_7}{\eta}$$

Where

K_1 = Power consumed in the wedge

= 0.221 HP.

K_2 = Power required by the conveyor

= 0.015 HP

K_3 = Power consumed for moving starch on the plates below cutters

= 0.014 HP.

K_4 = Power required by the rotating cutter

= 0.56 HP.

K_5 = Power required by the reciprocating cutter

= 1.5 HP.

K_6 = Power needed by the elevator

= 0.173 HP.

K_7 = Power utilised for the movement of the machine

= 0.124 HP

η = Efficiency of the machine.

There will be losses in the moving parts (No. of moving parts will be around 15 - 20). The power required by the wedge, conveyor and the guide plates have been calculated with limiting friction *condition. To impart motion, the power of the motor must be higher than the power required to overcome the limiting friction.*

So, the efficiency is assumed as 0.75

$$\therefore \text{The power of the motor} = \frac{0.221+0.015+0.014+0.56+1.5+0.173+0.124}{0.75}$$

$$= 3.5 \text{ HP.}$$

Giving an overload factor of 20 percent, the power of the motor

$$= 3.5 \times 1.2$$

$$= 4.2 \text{ HP.}$$

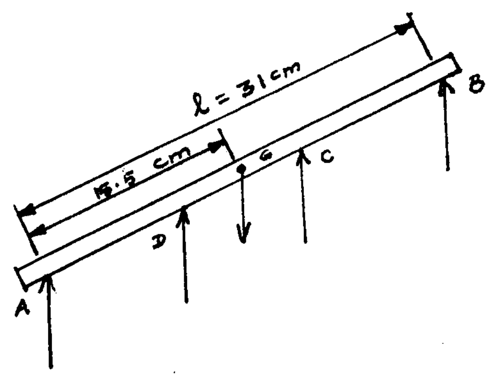
Therefore a motor of 5 HP at 1440 rpm is selected.

7. DESIGN OF EXCAVATOR COMPONENTS

7.1 Shaft design for conveyor

Self weight of the Shaft (2" - 55cm length)	= 8.470 kg.
Approximate weight of the conveyor belt acting on the roller	≃ 0.100 kg.

To find weight acting on single roller *(Starch weight)*



Weight of the starch	= W = 17.091 kg
Weight of the starch at A & B	= 17.091 x 15.5 / 31
	= 8.5444 Kg

These two rollers (A & B) itself will support the weight coming over the belt. Since the starch is acting as a single and contineous body and its centre of gravity lies at the centre.

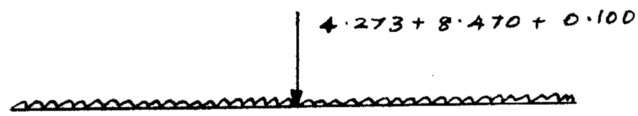
To compensate the belt sag and to addionally support the weight coming over, two idlers C & D are added. Because of this, the weight *is equally divided and supported by all the rollers.*

So weight coming over each roller is shown below.

Self weight of the shaft = 8.47 Kg

Approximate weight of the Conveyor belt = 0.100 Kg

Weight of the starch coming over single roller = 4.273 Kg



Total bending force $= W = 4.273 + 8.47 + 0.100$

Maximum bending moment $= Wl / 4$

$= (4.273 + 8.47 + 0.100) \times 28 / 4$

$= 89.94$ Kg-cm

To find maximum twisting moment

We know

Power required to convey 1 + .091 Kg starch up with a velocity 6 m/min is

H.P. = 0.015 H.P. : *With factor of safety*

We take H.P.= 1/4 H.P. = 0.25 H.P.

But $H.P. = 2 \pi N.Mt / 450000$

$Mt = 0.25 \times 450000 / 2 \times \pi \times 27.0$

$= 879$ Kg-cm.

Equivalent twisting moment + $= Mt_{eq}$

$= (K_b \cdot M_b)^2 + (K_t \cdot M_t)^2$

From P.S.G. D.D.B. Page 7.21

$K_b = 1.5$
$K_t = 1.0$ *Gradual loading revolving shaft*

$\therefore [M_t]_{eq} = \sqrt{(1.5 \times 89.94)^2 + (1 \times 875)^2}$

$= 885.3$ Kg-cm.

But $[M_t]_{eq} = (\pi / 16.) \cdot \tau \cdot d^3$

$d = \sqrt[3]{16 \times [M_t]_{eq} / \pi \times \tau}$

For C_{20} steel, $\tau = 275$ kg/cm^2

$d = \sqrt[3]{16 \times 885.3 / \pi \times 275}$

$= 2.5$ cm

By R_{20} series

$d = 2.50$ cm.

This diameter is applicable to the front roller of the conveyor.

Bearing design for the front roller of the conveyor

To calculate the load coming on a single bearing :

Weight of starch coming on the roller	=	4.273 Kg
Self weight of shaft	=	8.470 Kg
Weight of Belt	=	0.100 Kg
Total weight	=	4.273 + 8.470 + 0.100
	=	12.843 Kg.

This load is shared by two bearings on both ends

Load coming on single bearing = (12.843 / 2) = 6.4215 Kgf.

From PSG D.D.B Page no. 4.2

$= (X.\ F_r + Y.\ F_a)\ S$

S = 1.1 - 1.5 for rotary m/c with no impact we select $S = 1.4$

F_a = Axial load = 0

F_r = Radial load = 6.422 Kg

$F_a / F_r = (0 / 6.422) = 0$

From PSG D.D.B Page 4.4

Corresponding to deep groove ball bearings 'e' value ranges from 0.22 - 0.44

But e F_a / F_r ie e · 0

So $x = 1$, $y = 0$

$p = (1 \times 6.422 + 0 \times 0)\ 1.4 = 8.991$ Kgf.

From PSG DDB Page 4.2

$C = (L / L_{10})^{1/k}\ .P.$

L is revolution $= (27 \times 60 \times 16 \times 52 \times 5 / 10^6)$ m.r

Expecting a working hours of 16 hrs per week and for five years.

27 x 60 is regulation per second.

= 6.739 m.r.

$L_{10} = 1$ m.r.

$K = 3$ for ball bearings.

$\therefore \quad C = 6.739/1^{1/3} \times 8.991 = 16.98$ Kgf.

From PSG DDB Page 4.12

Here C is only 16.98 Kgf.

We select 60 series

To accomodate a 25 mm diameter requirement of the shaft we select

SKF 6005 bearing with 25 mm internal diameter which can accomodate a load of 780 kgf.

Dimensions are

$d = 25$ mm

$B = 12$ mm

$D = 47$ mm

$D_1 = 28$ mm

$D_2 = 44$ mm

To find the weight of starch in buckets

We know

Volume of one bucket	$= 1141\ cm^3$
Volume of starch occupies actually in one bucket neglecting the inter spaces	= Volume of bucket x bulk density of starch
	$= 1141 \times 0.65$
	$= 741.65\ cm^3$
But weight/cm^3 of starch	= 1.253 gms

∴ Weight of Starch in one bucket = 741.65 x 1.253

= 929.28 gms.

= 0.92928 kg.

Let at an instant 6 buckets have starch and 7 buckets are empty.

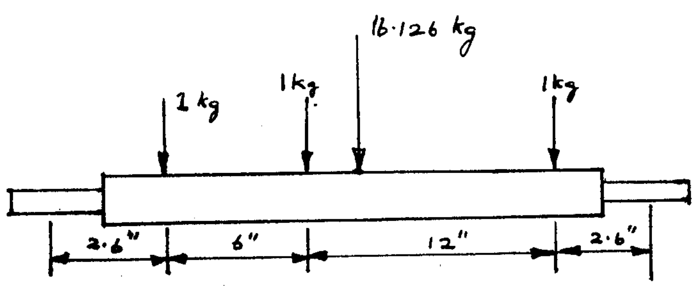
∴ Weight of Starch on 6 buckets = 0.92928 x 6

= 5.575 kg.

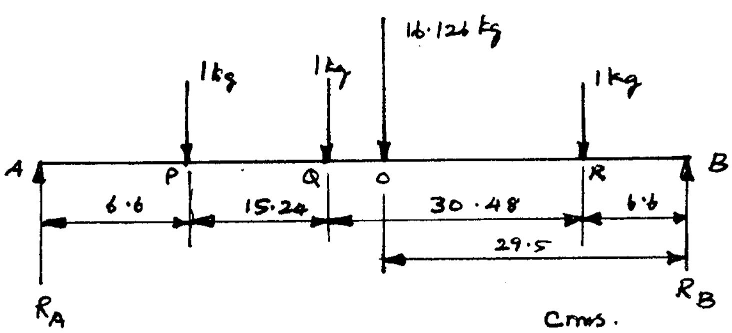
This weight of the Starch will be supported by Elevator shaft.

In addition to the above starch weight, the weight of the buckets, weight of belt and belt tension must be supported by the top shaft of the Elevator.

7.2 Shaft Design for Elevator



Here though the power requirement is less we design the shaft for 1 HP with factor of safety.



Taking moment about A,

$R_B \times 59 = 1 \times 52,32 + 16.126 \times 29.5 + 1 \times 21.84 + 1 \times 6.6$

$\therefore R_B = \dfrac{52.32 + 475.72 + 21.84 + 6.6}{59}$

$= 9.43$ kg,

We know the total load acting downwards

$= R_A + R_B$

$= 1+1 + 16.126 + 1 = R_A + 9.43$

$\therefore R_A = (19.126 - 9.43) = 9.696$ kg.

Bending moment at A = 0

BM at P $= R_A \times 6.6 = 9.696 \times 6.6 = 64$ kg - cm

BM at Q $= R_A \times 21.84 = 9.696 \times 21.84 = 211.76$ kg - cm

BM at O $= R_A \times 29.5 = 9.696 \times 29.5 = 286.03$ kg - cm

BM at R $= R_B \times 6.6 = 9.43 \times 6.6 = 62.24$ kg - cm

The max BM is occuring at the centre point '0'.

$\therefore$ BM $= 211.76\ kg^{-cm} = Mb.$

H.P. $= \dfrac{2\pi . N. Mt}{450000}$

$1 = \dfrac{2 \times \pi \times 246 \times M_t}{450000}$

$M_t = 291.13$ kg - cm

<u>From PSG DDB 7.21</u>

$[M_t]_{eq} = \sqrt{(K_b \cdot M_b)^2 + (K_t \cdot M_t)^2}$

$K_b = 1.5$ (Gradual loading - revolving shaft)

$$K_t = 1.0$$

$$[M_t]_{eq} = \sqrt{(1.5 \times 211.76)^2 + (1.0 \times 291.13)^2}$$

$$= 430.87 \text{ kg - cm}$$

$$\text{But } [M_t]_{eq} = \frac{\pi}{16} \cdot \tau \cdot d^3$$

$$\therefore \quad d = \sqrt[3]{\frac{M_t \times 16}{\pi \times \tau}}$$

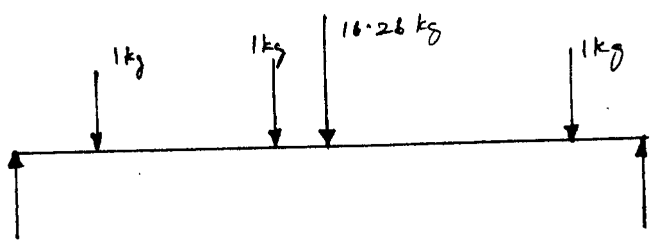
$$= \sqrt[3]{\frac{16 \times 430.7}{\pi \times 275}}$$

$$= 1.99 \text{ cm.}$$

From R_{20} series

$$d = \underline{2} \text{ cm.}$$

Elevator Shaft Bearing Design :



Total load on the shaft $= 16.126 + 1 + 1 + 1$

$= 19.126$ kg.

This load is supported by two bearings (assuming simply supported beam)

Load on each bearing $= \dfrac{19.126}{2} = 9.563$ kg.

= Radical load on bearing

= F_r

From PSG DDB page 4.2

Equivalent load P = $(X. F_r + Y. F_a) . S.$

Where F_r = Radial load

F_a = Axial load

Here $F_a = 0$, $F_r = 9.563$ kgf.

To Find X and Y

$$\frac{F_a}{F_r} = \frac{0}{9.563} = 0 \quad \text{ie} < e$$

We select 62 series ball bearing

For 62 series and $\frac{F_t}{F_r} < e$

$X = 1$ $Y = 0$

Service factor $S = 1.1 - 1.5$

(For Rotary m/c with no impact)

We select $S = 1.4$

$\therefore$ Equivalent load = $P = (1 \times 9.563 + 0)\ 1.4$

= 13.388 kgf.

From PSG DDB page 4.2

C = Dynamic capacity $= \left(\frac{L}{L_{10}}\right)^{1/K} . P .$

$$L = \frac{246 \times 60 \times 16 \times 52 \times 5}{10^6} \text{ m.r}$$

Expecting working hours of 16 hrs / week and for 5 years

$L = 61.40$ m.r.

$L_{10} = 1$ m.r.

$k = 3$ for ball bearings

$$\therefore \quad C = \left(\frac{61.40}{1}\right)^{\frac{1}{3}} \times 13.388$$

$$= \underline{52.81} \text{ kgf.}$$

From PSG DDB PAGE 4.12

For 20 mm diameter of the shaft and 52.81 kgf. Dynamic capacity we select the bearing of designation SKF 6004.

Dimensions

Internal diameter = d = 20 mm

Outer diameter = D = 42 mm

Width = B = 12 mm

RECIPROCATING CUTTER

7.3 Design of flywheel

Total travel by the cutter = 20 cm.

Working stroke length = 9.24 cm

This constitutes $\frac{4.6}{10}$ of a revolution.

$$\therefore \text{ Excess energy} = \Delta E = 19.46 \times \frac{4.6}{10}$$

$$= 8.95 \text{ kg - m.}$$

$$But\ \Delta E = I\ C_s\ W^2$$

$$= \frac{W}{g} R^2 \times C_3 \times W^2$$

$$C_S = 0.16 \text{ assume}$$

$$8.95 = \frac{W}{9.81} \times R^2 \times 0.16\left[\frac{2 \times \pi \times 250}{60}\right]^2$$

$$\therefore W R^2 = \frac{8.95 \times 9.81 \times 60^2}{0.16 \times (2 \pi \times 250)^2}$$

$$= 0.8006 \text{ kg - m}^2.$$

Space available will be enough to accomodate a flywheel of diameter approximately arround 15 cm.

$$\therefore W = \frac{0.8006 \times 10000}{12 \times 12}$$

$$= 55.59 \text{ kg.}$$

We accomodate two flywheels of compact size

$$So \quad W = \frac{55.59}{2} = 27.8 \text{ kg.}$$

But

$$W = \pi D t B p$$

$$ie \quad 27.8 = \pi \times 24 \times t \times B \times 0.0072$$

$$But\ \frac{B}{t} = 0.75 \text{ to } 2$$

$$We\ choose\ \frac{B}{t} = 0.75$$

$$B = 0.75\ t$$

$$27.8 = \pi \times 24 \times t \times 0.75t \times 0.0072$$

$$\therefore \quad t^2 = \frac{27.8}{\pi \times 24 \times 0.75 \times 0.0072}$$

$$t = 8.3 \text{ cm}$$

$$B = 6.1 \text{ cm}$$

Out side diameter of the flywheel $= 24 + \frac{t}{2} + \frac{t}{2}$

$$= 24 + 8.3$$

$$= 32.3 \text{ cm.}$$

Checking of Centrifugal stresses

$$\text{Stress} = \frac{3}{\cancel{R}4} \frac{\rho V^2}{g} + \frac{\pi^2 V^2 R}{2 n^2 gt}$$

$$= \frac{\rho V^2}{g} \quad \frac{3}{4} + \frac{\pi^2 R}{2n^2 t}$$

$$= \frac{0.00786 \times (\pi \times 24 \times \frac{250}{60})^2}{9.81} \times \quad \frac{3}{4} + \frac{\pi^2 \times 12}{2 \times 4^2 \times 2 \times 8.3}$$

$$= 76.939 \text{ kg/cm}^2$$

76.939 4000 kgf / cm^2 ultimate strength of steel. The design is safe.

7.4 Shaft Design

We know work done $= T_{max} \times \theta$

$$T_{max} = \frac{W.D.}{\theta} = \frac{19.46}{\theta}$$

Total travel by the cutter = 20 cms.

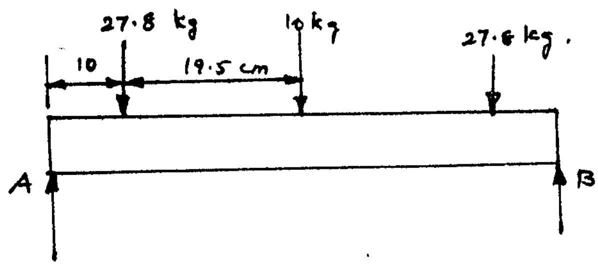
$$\therefore \quad \theta = \frac{360}{20} \times 9.24$$

$= 166.32$

$= 2.9$ radians

$$\therefore \; T_{max} = \frac{19.46}{2.9} = 6.7 \text{ kg - m.}$$

Assuming eccentric weight to be 10 kg. Let both, flywheels be placed near the walls. Let the distance beween each flywheel and the bearing centre be 10 cm.



Total weight acting downwards $= 27.8 + 27.8 + 10$

$= 65.59$ kgf.

$= R_A + R_B$

$$\therefore \; R_A = \frac{65.59}{2}$$

$= 32.8$ kgf.

B.M. due to Flywheel $= 32.8 \text{ kg} \times 0.10$

$= 3.28$ kg-m

B.M. due to eccentric $= 10 \times 0.295$

$= 2.95$ kg-m

Maximum Bending moment $= 3.28$ kg-m

$\therefore$ *Equivalent Torque* $= T^2 + M^2$

$= \quad 6.7^2 + 3.28^2$

$= 7.5 \ kg\text{-}m$

$= 750 \ kg\text{-}cm$

But

Torque $= \frac{\pi}{16} \cdot \tau \cdot d^3.$

$= 275 \ kg\text{-}cm$

$\therefore \ d = 3 \ \frac{750 \times 16}{\times 275}$

$= 2.4$ cm.

By R_{20} Series

$d = 25$ mm.

Bearings

We know

Load $P = X.F_r + Y.F_a \quad S$

$S = 1.6 - 4$ (for impact hammer mills)

$S = 2.5$ choosen.

$F_r = 32.8$ kgf.

$F_a = 0$

$\frac{F_a}{F_r} = 0.$ For $\frac{F_a}{F_r} = 0$, $X = 1$ and $Y = 0$

$\therefore P = 1 \times 32.8 + 0 \quad 2.5$

$= 82$ kgf.

Dynamic Capacity $C = \left(\frac{L}{L_{10}}\right)^{1/K} . P .$

$L_{10} = 1$ m.r. $K = 3$ for ball bearings

$$L = \frac{300 \times 60 \times 16 \times 52 \times 5}{10^6} \text{ m.r.}$$

$= 74.88$ m.r.

$$C = \left(\frac{L}{L_{10}}\right)^{1/K} . P .$$

$$= \left(\frac{74.88}{1}\right)^{1/3} \times 82$$

$= 345.6$ kgf.

From PSG DDB page 4.12

To accomodate 345.96 kgf. and 25 mm. dia

We select

SKF 6005 bearing, $B = 12$mm, $d = 25$ mm.

$D = 47$ mm, $D_1 = 28$ mm.

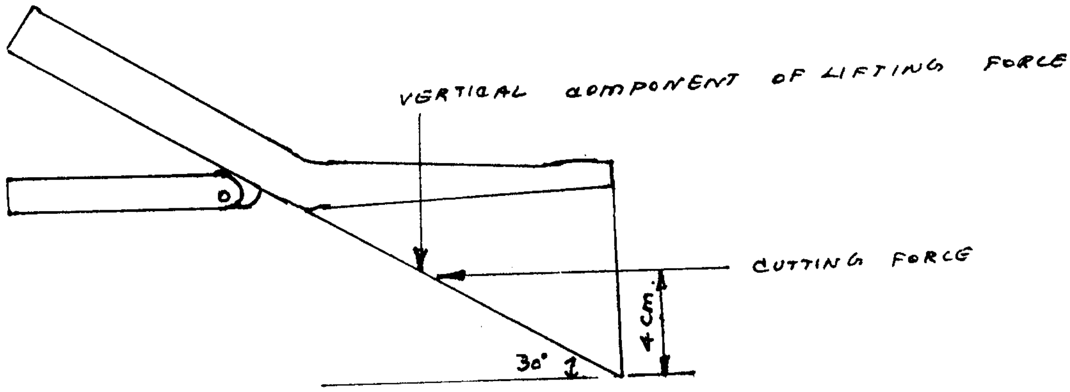
$D_2 = 44$ mm, $C = 780$ kgf.

7.5 Bake Roller Shaft Design for Conveyor

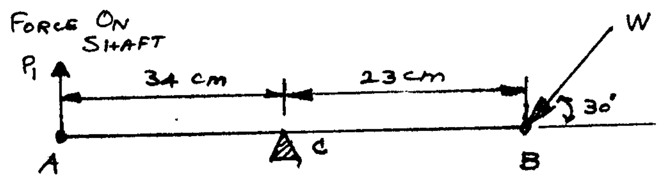
The cutting force and the wedge plates and the lifting forces act on the back roller shaft. This may be calculated as follows.

The forces acting on the shaft is calculated considering the cutting and lifting separately.

The above forces are transmitted to the shaft through the side housings of the conveyor and the housing acts as lever.



The uniform cutting force on one side is concentrated at the centre of the plate from the base 4 cms above.



We know

$$\frac{W}{P_1} = \frac{34}{23}$$

$$\therefore \quad P_1 = \frac{W \times 23}{34}$$

Total cutting force $= 38.4$ kgf.

Cutting force per side $= \frac{38.4}{2} = 19.2$ kgf.

Vertical component of cutting force $= 19.2 \times \cos 60$

$= 9.6$ kgf.

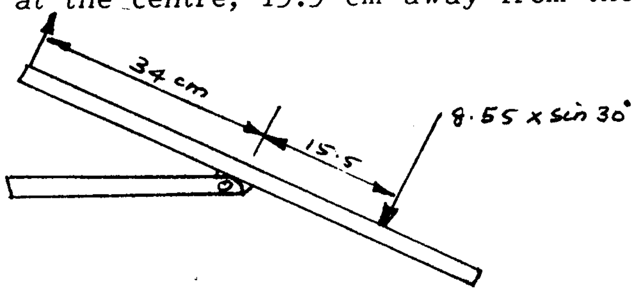
$$\therefore \quad P_1 = \frac{9.6 \times 23}{34} = 6.49 \text{ kgf.}$$

The lifting force also acts on the shaft. The weight of starch coming on the wedge before the conveyor = 17.091 kgf.

The weight is supported by the two housings weight coming on a housing $= \frac{17.091}{2}$

$= 8.55$ kgf.

Since Starch is a uniform structure, the weight is distributed over the length of the housing uniformly and it may be assumed to be concentrated at the centre, 15.5 cm away from the end.



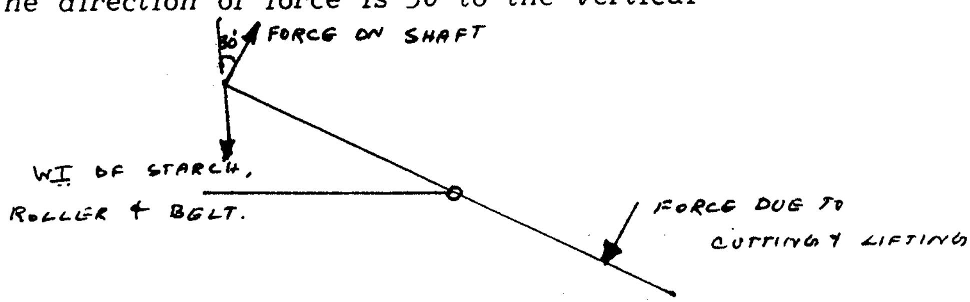
$\therefore P_2 \quad = \frac{4.275 \times 34}{15.5} = 9.377$ kgf.

$\therefore$ Total force acting on the shaft $= P_1 + P_2$

$= 9.377 + 6.49$

$= 15.87$ kgf.

The direction of force is 30° to the vertical



The vertical component of the above force is opposite to the force due to the weight of Starch Coming on the Shaft, weight of the roller and the belt weight.

$\therefore$ Net force acting on the shaft $= (15.87 \times \cos 30) - \dfrac{4.273 + 8.47 + 0.100}{2}$

$= 9.5$ kgf.

Reaction at the Fulcrum

Reaction at the Fulcrum = [Reaction due to leverage + Reaction due to Weight of Starch Coming over the Conveyor].

Reaction due to leverage = [Force acting at the Fulcrum due ot cutting force + Reaction due to lifting force]

We know,

Reaction at the fulcrum = Force acting + Force transmitted

$\therefore$ [For levers of first order]

$\therefore$ Reaction due to cutting force $= 9.6 + 6.49$

$= 16.09$ kgf.

Reaction due to lifting force $= 4.275 + 9.377$

$= 13.652$ kgf.

$\therefore$ Total reaction due to leverage $= 16.09 + 13.652$

$= 29.742$ kgf.

With factor of safety, the shaft may be expected to transmit 0.3 HP power.

We know,

$$H.P. = \frac{2\ \pi .N.\ Mt}{4500}$$

$$\therefore \quad M_t = \frac{0.3 \times 4500}{2\ \pi \times 27} = 7.957\ kg\text{-}m$$

$$= 795.7\ kg\text{-}m$$

$\therefore$ Equivalent twisting moment $= \sqrt{(K_b.M_b)^2 + (K + M_t)^2}$

$K_b = 1.5$ For Gradual loading and

$K_t = 1.0$ revolving shafts.

$$\therefore \quad [M_t]_{eq} = \sqrt{(1.5 \times 0.0570^2 + (1.0 \times 7.957)^2}$$

$$= 7.957\ Kg\text{-}m$$

$$= 795.7\ Kg\text{-}m$$

$$\text{But} \quad [M_t]_{eq} = \frac{\pi}{16} . \tau . d^3.$$

$$\tau = 275\ kg/cm^2.$$

$$\therefore \quad d = \sqrt[3]{\frac{795.7 \times 16}{275 \times \pi}}$$

$$= 2.45\ cm.$$

From R_{20} Series,

$$d = 2.5\ cm.$$

The fulcrum is at the end of the conveyor and the Starch over the conveyor is supported by the fulcrum and the back roller shaft.

Weight of the Starch over the conveyor $= 34 \times 55 \times 8 \times 1.253$

$= 18.744$ *kgf.*

Weight supported by the fulcrum $= \frac{18.744}{2}$

$= 9.372$ *kgf.*

$\therefore$ *Total reaction at fulcrum* $= 29.742 + 9.372$

$= 39.1$ *kgf.*

Shaft Design

Bending moment due to 9.5 kg force $= 9.5 \times 0.006$

$= 0.057$ *kg-m*

The power to be transmitted = Power required for running the conveyor + Power required for moving the starch below moving the starch below the cutter

$= 0.015 + 0.014$

$= 0.029$ *HP.*

Bearing Design

We know load $P = [X.f_r + Y.f_a]\ S.$

$S = 1.1 - 1.5$ *(for rotary machine with no impact)*

$= 1.4$

$F_r = \frac{38.4}{2} = 19.2$ *kgf.*

For $\frac{F_a}{F_r} = 0$ e, $X = 1$ *and* $Y = 0.$

$\therefore\ P = [\ 1 \times 19.2 + 0\ ]\ 1.4$

$= 26.88$ *kgf.*

Dynamic Capacity $= C = [\frac{L}{L_{10}}]^{1/k} .P.$

$P = 26.88$ *kgf.*

$K = 3$ *for ball bearings.*

$L_{10} = 1$ *m.r.*

$C = [\frac{6.73}{1}]^{\frac{1}{3}} \times 26.88$

$= 50.77$ *kgf.*

To accomodate 50.77 kgf. and 25 mm. diameter shaft we select

SKF 6005 bearing,

Dimensions :

$B = 12$ mm $d = 25$ mm.

$D = 47$ mm $D_1 = 28$ mm.

$D_2 = 44$ mm $C = 780$ kgf.

7.6 Design of Reverse Gears

Reverse gear is attached to the 20 rpm Shaft of the gear box. The output of the reverse gear shaft is attached to the machine wheel at the bottom. The wheel is rotating at 12.73 rpm. The 20 rpm is reduced in '3' stages, two in the reverse gears and one in the chain wheel.

Forward drive set

H.P. = 0.124 H.P. (Actual)

With factor of safety the gear is designed for the transmission of 0.4 H.P.

Speed of the pinion = 20 rpm

Reduction ratio = i = 1.125

Design Torque = $M_t = M_t$ k.k.d.

$$[M_t] = \frac{71620 \times 0.4}{20} = 1432.1 \text{ kg-cm}$$

$$[M_t] = 1432.4 \times 1.3$$

$$= 1862.1 \text{ kg-cm}$$

Design compressive stress $[\sigma_c] = C_B \cdot H_B \cdot K_{cl} \cdot Kgf / cm^2$

C_B = Coefficient depending on surface handness for cast iron,

= 20

H_B = 170 - 200 = 185 (assume) = Hardness.

K_{cl} = life factor for surface strength

$= 6 \surd \frac{10^7}{N}$

But $N = 60\ n.T.$

n = rpm = 20 rpm

T = life in hours = 10,000 hrs.

$\therefore N = 60 \times 20 \times 10000$

$= 12 \times 10^6$

$\therefore K_{cl} = [\frac{10^7}{12 \times 10^6}]^{1/6}$

$= 1.42.$

$\therefore [\sigma_c] = 20 \times 185 \times 1.42$

$= 5254$

We know

$i = 1.125$

$\therefore$ centre distance = a

$a > (i \pm 1)\ 3 \surd [0.74/\sigma_c]^2\ E\,[M_t]\,/ i\,\Psi$

For external gears +ve sign is assumed

$\Psi = 0.3$ (For open gears)

$E = 2E_1/E_2/E_1+E_2$

E_1 = Youngs modulus of pinion C_{15} steel

E_2 = Youngs modulus of wheel Cast Iron

$E_1 = 2.08 \times 10^6$ kgf/cm^2

$E_2 = 1 \times 10^6$ kgf/cm^2

$\therefore$ $E = 2 \times 2.08 \times 10^6 \times 1 \times 10^6/(1 + 2.08)10^6 = 1350649$

$\therefore$ $a = (1.125+1)\ 3\ \ (0.74/5254)^2 \times (1350649 \times 1862.1)/(1.125 \times 0.3)$

$= 11.2$ cm

By R_{20} series

$a = 11.2$ cm

Module based on the beam strength = m

$m = 1.26\ \ 3\ \ \ M_t\ /y \times (\ _b) x \Psi_m x Z_1$

Z_1 = Number of teeth on pinion

= 25 (assume)

Ψ_m = 10 General

y = Form factor

= 0.421 for 25 teeth

$[\sigma b]$ = Design bending stress

$= 1.4 \times kbl\ x\sigma_{-1}/n \times k\sigma$

σ_{-1} = Endurance limit stress in bending

$= 0.35\ \ _u + 1200$

But $\sigma_u = 4300$ kgf/cm^2 for C_{15} steel

$\therefore \sigma_{-1} = 0.35 \times 4300 + 1200$

$= 2705 \ kgf/cm^2$

k_{bl} = Life factor for bending

$= 9\sqrt{10^7/N}$

$N = 12 \times 10^6$

$\therefore k_{bl} = 9\sqrt{10^7/12 \times 10^6}$

$= 0.979$

n = factor of safety = 2.5 for C.I.

$K\sigma$ = Stress concentration factor

= 1.2 for C.1.

$\therefore [\sigma b] = 1.4 \times 0.979 \times 2705/2.5 \times 1.2$

$= 1235.8 \ kgf/cm^2$

$\therefore m = 1.26 \times \sqrt[3]{1862.1/(0.421 \times 1235.8 \times 10 \times 25}$

= 0.3 cm.

m = 3 mm.

Number of teeth on pinion = Z_1

$Z_1 = 2 \ a/m(i+1)$

$= 2 \times 11.2/0.3(1.125+1)$

= 35 teeth

Z_2 = Number of teeth on wheel

$= Z_1 \times i$

$= 35 \times 1.125$

= 39 teeth.

Pitch diameter $d_1 = m.Z_1 = 3x\ 35 = 105$ mm.

$d_2 = m.Z_2 = 3 \times 39 = 117$ mm.

We know

Face width $b = \Psi_m . m$

$= 10 \times 3$

$= 30$ mm.

7.7 Design of gear with reduction ratio i = 1.22

The third gear is linked with the second gear to have a reduction of 1.22

H.P = 0.4 H.P.

Speed of the pinion = 17.77 rpm

Gear reduction ratio = i = 1.22

Design Torque = $[M_t] = M_t.k.kd$

$k.kd = 1.3$

$M_t = 71620 \times 0.4/17.77$

$= 1612$ kgf-cm

$\therefore\ [M_t] = 1612 \times 1.3$

$= 2095.6$ kgf-cm

Design compression stress$[\sigma c] = C_B.H_B.K_{cl}$

C_B = Coefficient depending upon the Surface Hardness

= 23 (C.I.Wheel)

H_B = Hardness

= 250 (Assume)

K_{cl} = Life factor for Surface Strength

$= 6\sqrt{10^7/N}$

$N = 60 \times n \times T$

$n = 17.77$ rpm

T = Life = 10000 hrs.

$\therefore N = 60 \times 17.77 \times 10000$

$= 10662000$

$\therefore Kcl = 6\sqrt{10^7 / 10662000}$

$= 0.989$

$\therefore \sigma_c = 23 \times 250 \times 0.989$

$= 5686$ kgf/cm^2

The centre distance = $a > (i \pm 1)\ 3\sqrt{[0.74/(\sigma_c)]^2\ E\,[M_t]/i\,\Psi}$

$\Psi = 0.3$ (for open gears)

$i = 1.22$

We know,

$E = 2E_1E_2/E_1+E_2 = 1350649$ kgf/cm^2

$\therefore a = (1.22+1)\ 3\sqrt{[0.74/5656]^2 \times 1350649 \times 2095.6/0.3 \times 1.22}$

$= 11.3$ cm

By R_{20} series

$a = 12.5$ cm.

The module $m = 1.20\sqrt{3}\ [M_t]/y(\sigma b)\Psi_m.Z_1$

Z_1 = Number of teeth on pinion

$Z_1 = 44$

y = Form factor

$= 0.471$ for 44 teeth.

$\Psi_m = 10 = b/m$ general

$[\sigma b]$ = Design bending stress

$= 1.4 \times kbl \times \sigma_{-1}/n.k\sigma$

kbl = Life factor for bending

$= 9\sqrt{10^7/N}$

$= 9\sqrt{10^7/10662000}$

$= 0.992$

n = factor of safety

= 2.5 for C.I.

$K\sigma$ = Stress concentration factor

= 1.2 for C.I.

σ_u = ultimate tenstile stress

= 3300 kgf/cm^2 for Grade 25

$\therefore \sigma_{-1} = 0.45 \times 3300$

$= 1485\ kgf/cm^2$

$\therefore [\sigma b] = 1.4 \times 0.992 \times 1485/2.5 \times 1.2$

$= 687.5\ kgf/cm^2$

$\therefore m = 1.26 \sqrt[3]{2095.6/0.471 \times 687.5 \times 10 \times 39}$

= 0.255 cm.

m = 3 mm. Rounded off

Number of teeth on pinion $Z_1 = 39$

Number of teeth on wheel $Z_2 = Z_1 \times i$

$= 39 \times 1.22$

= 48 teeth.

Pitch diameter $d_1 = m.Z_1$

$= 3 \times 39$

$= 117$ mm.

$d_2 = m.Z_2$

$= 3 \times 48$

$= 144$ mm.

Face width $= b = \Psi_m \times m$

$= 10 \times 3$

$= 30$ mm.

For reverse drive S similar gear are designed and properly positioned for a central distance of 11.2 cm and the gear will contact both the gears of the gear box shaft and the output shaft of the reverse gear set, when properly positioned.

7.8 Design of spur gears for gear box

The spur gears have to transmit power required for the movement of the machine, conveyor and rotating cutters.

The horse power transmitted by the gears =1 Hp.

Reduction in gears = 4.

From PSG DDB 8.15

Design Torque $M_t = M_t.k.kd$

$k.kd = 1.3$ for symmetric scheme

M_t = Torque transmitted

$= 71620 \times HP/n$

n = Speed of the pinion.

$= 71620 \times 1 / 80 = 895.25$ kgf-cm.

$M_t = 895.25 \times 1.3$

$= 1163.8$ kgf-cm

From PSG DDB 8.16

Design compressive stress $= [\sigma_c] = C_R . HRC . Kcl . Kgf/cm^2$

Where C_R = coefficient depending on hardness

= 220 for C_{35} steel.

HRC = Surface Hardness.

= 69 for C_{20} steel.

Kcl = Life factor for surface strength.

$= 6 \sqrt{10^7/N}$

Where N = 60 n T.

T = Life in hours

$\therefore$ $N = 60 \times 80 \times 10000$

$= 48 \times 10^6$

$\therefore$ $K_{cl} = 0.77$

$\therefore$ $[\sigma_c] = 220 \times 69 \times 0.77$

$= 11688.6$ kgf/cm^2

From PSG DDB 8.13

Centre distance $a = (i-1) \; 3\sqrt{[0.74/\sigma_c]^2 . E[M_t]/i\Psi}$ for internal gears.

$\Psi = b/m = 1.0$ for low speed reducers.

E = Equivalent young's modulus.

$= 2E_1E_2/E_1+E_2$

E_1 = Young's modulus of pinion C_{35} steel.

= 2.060×10^6 kgf/cm^2

E_2 = Young's modulus of the wheel C_{15} steel.

= 2.080×10^6 kgf/cm^2

$\therefore$ $E = 2 \times 2.060 \times 10^6 \times 2.080 \times 10^6 / 2.060 \times 10^6 + 2.080 \times 10^6$

= 2069951.7 kg/cm^2

$\therefore$ $a = (4-1) \;\; 3 \;\; 0.74/11688.6 \;^2\; (2069951.7 \times 1163.8)/(4 \times 1)$

= 6.7 cm.

By R_{20} series

a = 10 cm.

From PSG DDB 8.13

Module $m = 1.26 \;\; 3 \;\; M_t/y \;\; b \; \Psi_m Z_1$

Where

Z_1 = No. of teeth on pinion

= 16 assume

Ψ_m = 10 general

y = Form factor

= 0.355 for 16 teeth.

$[\sigma b]$ = Design bending stress

= $1.4 \times kbl/n \times k\sigma \times \sigma_{-1}$ for rotation in one direction.

Where σ_{-1} = Endurance limit stress in bending.

= $0.35\sigma_u + 1200$

σ_u = Ranges from 5200-6200 for C_{35} steel.

= 5700 kgf/cm^2

$\therefore \quad \sigma_{-1} = 0.35 \times 5700 + 1200$

$= 3195 \ kgf/cm^2$

k_{bl} = Life factor for bending.

$= 9\sqrt{10^7/N}$

$= 0.84.$

n = Factor of safety for steel.

$= 2.5.$

K_σ = Stress concentration factor.

= 1.2 for steel.

$\therefore \quad [\sigma b] = 1.4 \times 0.84 \times 3195/2.5 \times 1.2$

$= 1252.4 \ kgf/cm^2$

$\therefore \quad m = 1.26 \ \sqrt[3]{1163.8/0.355 \times 1252.4 \times 10 \times 16}$

$= 0.32$ cm

$= 3.2$ mm

= 4 module Rounded off.

From PSG DDB 8.22

$Z_1 = 2a/m(i+1)$

$= 2 \times 10/0.4 \ (4+1)$

= 10 teeth.

Pitch dia of pinion $(d_1) = m.Z_1$

$= 4 \times 10$

$= 40$ mm.

Pitch dia of wheel $(d_2) = m.Z_2$

$= 4 \times 40$

$= 160$ mm.

From PSG DDB 8.14

$$\text{Face width} = b = \Psi_m . m$$
$$= 10 \times 4$$
$$= 40 \text{ mm.}$$

7.9 Design of clutch

The clutch is attached to the motor through 'V' belts. The motor runs at 1440 rpm. The clutch is a lever operated multiple plate friction clutch. It has 'V' grooves over its periphery. The clutch speed is 584 rpm after reduction in the pulleys.

Horse power to be transmited by the clutch = 3.5 HP with factor of safety.

From PSG DDB 7.89

$$\text{Transmitted Torque} = 71620 \times 3.5/584$$
$$= 429.2 \text{ kgf-cm.}$$

$$\text{Design Torque } [m_t] = K_w . m_t$$

Where

K_w = Factor based on working condition

$= K_1+K_2+K_3+K_4$

Where K_1 = Driver dynamic characteristic factor.

= 0.33 for belt transmissions and motor drive. PSG DDB 7.90.

K_2 = Driven shaft dynamic characteristic factor.

= 1.25 for driven machines where starting torque is greater than the nominal value. PSG DDB 7.91.

K_3 = Wear factor.

= 0.25 for 584 rpm.

K_4 = Frequency of operation factor.

= 1.8 Expecting 240 engagements in 8 hrs. PSG DDB 7.91.

∴ $K_w = 0.33 + 1.25 + 0.25 + 1.8$

$= 3.63.$

∴ Design Torque $M_t = 429.2 \times 3.63$

$= 1557.9$ kgf-cm.

From PSG DDB 7.89

Minimum number of friction surfaces $i_{min} = [M_t] / 2\pi . P_a . b . \mu\, r_m^2$

Where

P_a = Allowable pressure between plates.

$= k.P_b.$

K = Speed factor corresponding to speed at max radious.

= 0.85 for speed of 4.04m/sec at the max radious of 13.2 cm.

P_a = Basic pressure kgf/cm^2

= 2.5 for compressed asbestos on steel, dry running.

∴ $P_a = 0.85 \times 2.5$

$= 2.125.$

$b = r_{max} - r_{min}$

$r_{max} = 6.6$ cm.

$r_{min} = 4.6$ cm.

∴ $b = 2$ cm.

= Friction coefficient

= 0.45 for asbestos on steel.

r_m = mean radius.

$= r_{max} + r_{min} / 2$

$= 6.6 + 4.6/2$

$= 5.6$ cm.

$\therefore \quad i_{min} = 1557.9/2 \text{ x } \quad \text{x } 2.125 \text{ x } 2 \text{ x } 0.45 \text{ x } 5.6^2$

$= 4.134$

≈ 5 surfaces.

From PSG DDB 7.9

Actual pressure between plates $= \sigma = 1557.9/2\text{x} \quad \text{x}5\text{x}2\text{x}0.45\text{x}5.6^2$

$= 1.76$ kgf/cm^2

$P_a > \sigma$

The design is safe.

Axial force $Q = \pi . \sigma . [r_{max}^2 - r_{min}^2]$

$= \pi \text{ x } 1.76 \, [6.6^2 - 4.6^2]$

$= 123.85$ kgf.

Force at the end of the axial cam $Q' = Q/i_d$

Where

i_d = Number of operating levers

= 3 Because the clutch has 3 axial cams.

$\therefore \quad Q' = 123.85/3$

$= 41.3$ kgf.

Force at the end of the handle or force with which the clutch is operated $= Q'' = Q'a/l$

Where

a = Distance between the clutch shaft axis and the axis of the lever pin of the axial cam.

$= 7.3$ cm.

l = Length of the levers

$= 55$ cm. Totally

$\therefore \quad Q'' = 41.3 \times 7.3/55$

$= 5.48$ kgf.

7.10 Design of belt transmission for reciprocating cutter

The reciprocating cutter shaft also transmits the power for the elevator.

Total horse power to be transmitted = Power for reciprocating cutter + power for the elevator.

$= 1.5 + 0.173$

$= 1.673$ HP.

From PSG DDB 7.58

Design horse power $\doteq$ Rated HP x Service factor (F_a)/Length correction factor (F_c) x correction factor for angle of arc of contact (F_d)

$F_a = 1.0$ for light duty operation PSG DDB 7.69.

F_c is found using nominal pitch length of the belt.

Nominal picth length of the belt $L = 2c + \pi/2 \quad D+d \quad + (D-d)^2/4c$

Where

C = Centre distance between the shafts.

$= 967$ mm.

d = dia of the smaller pulley

$= 75$ mm.

Let n_1 = speed of smaller pulley.

= 300 rpm.

n_2 = speed of larger pulley.

= 80 rpm.

n = efficiency of the belt.

= 0.98 assume.

D = dia of the larger pulley.

$= d.n_1/n_2.m$

$= 75 \times 300/80.10.98$

= 275.6 mm.

$\therefore$ Nominal pitch length $L = 2 \times 967 + \pi/2\,[275.6 + 75] + (275.6-75)^2 / 4 \times 967$

= 2495 mm.

Length correction factor $F_c = 1.02$ for L = 2495 mm. PSG DDB 7.60

Angle of arc of contact $= 180-(D-d)/c \times 60^\circ$

$= 180 - (275.6-75)/967 \times 60$

$= 167.5^\circ$

For 167.5°, $F_d = 0.97$ PSG DDB 7.68.

$\therefore$ Design HP $= 1.673 \times 1/1.02 \times 0.97$

= 1.69 HP.

= 1.24 Kw

From PSG DDB 7.62

Equivalent pitch dia $d_e = d_p \times F_b$

Where F_b = Small dia factor

= 1.14 for D/d = 3.67

d_p = pitch dia of smaller puller

= 75 mm.

$\therefore$ d_e = 75 x 1.14

= 85.5 mm.

Velocity of the belt = x 0.075 x 300/60

= 1.18 m/sec.

To transmit the above design horse power we select section B belt.

Kilowatt rating of the belt for d_e = 85.5 mm and r = 1.18m/sec is

Kw = 0.42 K w

From PSG DDB 7.70

No. of belts = Design Kw x F_a/F_e x F_d x Kw

= 1.24 x 1/1.02 x 0.97 x 0.42

= 2.98

= 3 belts.

From PSG DDB 7.58

Nominal top width of belt = 17 mm.

Nominal thickness of belt = 11 mm.

Weight per metre = 0.189 kgf.

7.11 Belt transmission between motor and clutch

Power to be transmitted = 3.5 HP. with factor of safety.

Design horse power = Rated HP x F_a/F_c x F_d

Where

F_a = service factor

= 1.0 for light duty. PSG DDB 7.69

F_c = Length correction factor.

Nominal pitch length of the belt (L) = 2c + /2 D+d + $(D-d)^2/4c$

Where

c = centre distance between shafts

= 320 mm.

d = dia of the smaller pulley

= 75 mm.

Let, n_1 = speed of smaller pulley.

= 1440 rpm.

n_2 = speed of larger pulley

= 584 rpm

From PSG DDB 7.61

D = dia of larger pulley ie clutch

= $d.n_1/n_2$.

= 75 x 1440/584 x 0.98 assuming 0.98 efficiency.

= 181 mm.

$\therefore$ Nominal pitch length (L) = 2x320+ /2 181+75 +$(181-75)^2/4x320$

= 1050.9 mm.

For L = 1050.9 mm, F_c = 0.84. PSG DDB 7.59

F_d = correction factor for angle of arc of contact.

Angle of arc of contact = 180-(D-d)/c x 60°

= 180-(181-75)/320 x 60°

= 160°

For angle of act of contact 160°, $F_d = 0.95$ for V-V combination

$\therefore$ Design HP $= 3.5 \times 1/0.84 \times 0.95$

$= 4.38$ HP

$= 3.228\ K_w$

We select 'B' section belt

From PSG DDB 7.62

Equipment pitch dia $d_e = d_p \times F_b$

Where

$d_p = 75$ mm.

F_b = small dia factor

1.13 for $D/d = 2.41$

$\therefore$ $d_e = 75 \times 1.13$

$= 84.75$ mm.

From PSG DDB 7.64

The rating of the belt for $V = 5.65$ m/sec.

$K_w = 1.57\ k_w$.

From PSG DDB 7.70

No. of belts = Design $K_w \times F_a / K_w \times F_e \times F_d$

$= 3.228 \times 1/1.57 \times 0.84 \times 0.95$

$= 2.57$

$= 3$ belts.

The belt dimensions are,

From PSG DDB 7.58,

Nominal top width $= 17$ mm.

Nominal thickness = 11 mm.

Weight per metre = 0.189 kgf.

Design of worm and worm wheel

Horse power	=	3.5 H.P.
Speed of worm	=	584 rpm.
Reduction i	=	7.3
Centre distance	=	18 cm.

Reference : Machine design by R.K.Jain

The centre distance = c

$c/L_n = 1/2\pi[1/\sin\lambda + VR/\cos\lambda]$

Where

c = Centre distance between shafts

= 18 cm

L_n = Nominal lead

VR = 7.3

= Lead angle

But $V_R = \cot^3\lambda$

$\cot = \sqrt[3]{VR}$

= 1.939

$\therefore \lambda = 27°16'$

$\therefore 18/L_n = 1/2\pi[1/0.458 + 7.3/0.888]$

$L_n = 10.87$ cm.

Axial lead $= L_n/\cos\lambda$

$= 10.87/\cos 27°16'$

= 12.22 cm.

Since the gear reduction is 7.3. The number of threads may be 4

Axial pitch (P) = 12.22/4

= 3.05 cm.

Module (m) = P/π = $3.05/\pi$ = 0.97

Taking standard module = 9 mm.

Axial pitch = π x 0.9

= 2.82 cm.

Normal lead = L = Threads x axial pitch

= 4 x 2.82

= 11.28 cm.

Normal lead = L_n = 11.28 x Cos λ

11.28 x 0.888

= 10.01 cm.

∴ *Central distance*

$C/L_n = 1/2\pi \left[1/\text{Sin}\ \lambda + 7.3/\text{Cos}\ \lambda\right]$

$C = L_n/2\pi \left[1/\text{Sin}\ 27°16' + 7.3/\text{Cos}\ 27°16'\right]$

= 16.57 cm.

Number of teeth in gear n_g = i x No. of threads

= 7.3 x 4

= 29.2

= 29 teeth.

Pitch diameter of worm = d = $L/\pi \text{Tan}\lambda$

= $11.28/\pi \text{Tan}\ 27°16'$

= 6.96 cm.

Addendum diameter of worm $d_a = d + 2m$

$= 6.96 + 2 \times 0.9$

$= 8.76$ cm

Length of threaded portion $= \ell > (12.5 + 0.09n_g)m$

$\ell = (12.5 + 0.09 \times 29)\ 0.9$

135.9 mm.

Allowing about 25 mm for the feed marks of the cutter

$l = 160.9$ cm.

Pitch diameter of the worm wheel $= D = n_g.m.$

$= 29 \times 0.9 = 26.1$ cm

Addendum circle dia $= D_a = D + 2m$

$= 26.1 + 2 \times 0.9 = 27.9$ cm.

Outside diameter $D_0 = D_a + 1.5$

$= 29.4$ cm.

Rim width $< 0.67\ d_a$ for quadraple threads

$= 0.67 \times 8.76$

$= 5.87$ cm.

Its face angle $= 90°$

Then face width $= w = d \times \dfrac{\dfrac{\text{Face angle}}{2} \times \pi}{180}$

$$= \frac{6.96 \times 90/2 \times \pi}{180}$$

5.5 cm.

Now pitch line velocity of gear $= V = \pi D N_g/100$

Where N_g = Speed of the worm gear.

$= 80$ rpm.

$V = \; x \; 26.1 \; x \; 80/100$

$= 65.5$ m.p.m.

The tangential force on gear $= 3.5 \; x \; 4500/65.5$

$= 240.4$ kg.

Considering a service factor $= 2$

Tangential force $= 240.4 \; x \; 2$

$= 480.8$ kgf.

But Allowable Tangential load F_t by Lewis equation $= S.W.m.y.C_r$

Where,

S = Allowable compressive stress for bronze

$= 1000 \; kgf/cm^2$

W = Width of the gear

$= 5.5$ cm.

m = module $= 0.9$ $\quad$ y = Form factor

$y = \pi [0.175 - 0.841/n_g]$

$= \pi[0.175 - 0.841/29]$

$= \pi \; x \; 0.146 = 0.458$

$C_r = 366/366+v = 366/366+65.5$

$= 0.848$

Tangential force $F_t = 1000 \; x \; 5.5 \; x \; 0.9 \; x \; 0.458 \; x \; 0.848$

$= 1922$ kgf.

Since the allowable F_t is greater than actual F_t, the design is safe.

The torque acting on the gear shaft $= T_g$

T_g = H.P. $x \; 4500/2\pi \; x \; 80$

$= 3.5 \times 4500/2\pi \times 80$

$= 31.3$ kg-m.

$\therefore$ moment on the worm shaft $= T_w$

$= T_g/i.\eta$

η = efficiency

$T_w = 31.3/7.3 \times 0.89$ $\eta = 0.89$ (assumed)

$= 4.8$ kg-m

$= 480$ kg-cm

But Turning force on the worm (F_t) = Axial force on the worm wheel

$F_t = 2.T_w/d$

$= 2 \times 4.8/0.0696 = 137.7$ kg.

Similarly Axial force on the worm (F_a) = Turning force on the wheel

$F_a = 2 \times T_g/D$

$= 2 \times 31.3/0.261 = 239.8$ Kg.

Radial force on the worm $= F_r = F_a.\tan\phi$

Where ϕ = Pressure angle

$= 25°$ for lead angle upto $35°$

$\therefore$ $F_r = 239.8 \times \tan 25°$

$= 111.8$ kgf.

Distance between the bearings of the worm shaft may be taken equal to the pitch dia of the worm wheel.

$D = 26.1$ cm.

B.M. due to F_r in the vertical plane $= F_r.D/4$

$= 111.8 \times 26.1/4$

$= 729$ Kg-cm

B.M. due to in vertical

Plane $= F_a.d/4$

$= 239.8 \times 6.96/4$

$= 417.3$ kg-cm.

$\therefore$ Total B.M. in the vertical

Plane $= 729 + 417.3$

$= 1176.3$ kg-cm.

B.M. due to F_t in the horizontal plane

$= F_t.D / 4$

$= 137.9 \times 26.1/4$

$= 899.8$ Kg-cm.

Equivalent twisting moment $= \sqrt{(K_t.T_w)^2 + (K_b.m)^2}$

Where,

K_t = Combined shock and fatique factor applied to T_w

K_b = Combined shock and fatique factor applied to m

= 1.5 for gradual loading

$\therefore$ Equivalent twisting moment on worm shaft $= T_c$

$T_c = \sqrt{(1 \times 480)^2 + (1.5 \times 1480.9)^2}$

$= 2272.6$ Kg-cm.

But $T_c = /16 \times \tau_s \times d^2{}_w$

Where, τ_s = shear strength of the shaft

$= 550$ kgf/cm^2

d_w = shaft dia of the worm

$\therefore \quad d_w^3 = 2272.6 \times 16/ \quad \times 550$

$\therefore \quad d_w = 2.7$ cm

= 30 mm for standard bearings

Actual shear stress = $2272.6 \times 16/\pi \times 3^3$

= 428 kgf/cm^2.

Since the actual shear stress is less than the allowable shear stress, the design is safe.

Design of worm wheel shaft

Axial force on worm wheel = 137.9 kgf.

Radial force on worm wheel = 111.8 kgf

Turning force on the worm wheel = 239.8 kgf.

B.M. due to axial force will be in the vertical plane end

= 137.9 x 26.1/2

= 1799.6 Kg-cm

B.M. due to radial force will also be in the vertical plane and it is equal to Radial force x 1/4 (distance between the bearing of wheel)

Assuming a centre distance of 18 cm.

B.M. due to radial force = 111.8 x 18/4 = 503.1 kg-cm.

$\therefore$ Total B.M. in the vertical plane

= 1799.6 + 503.1

= 2302.7 Kg-cm.

B.M. due to turning force will be in the horizontal plane

$= \text{Turning force} \times \text{Centre distance}/4$

$= 239.8 \times 18/4$

$= 1079.1$ kg-cm.

$\therefore$ Resultant bending moment on

wheel shaft $= M_g = \sqrt{(2302.7)^2 + (1079.1)^2}$

$= 2543$ Kg-cm.

Twisting moment of the worm wheel = 3130 kg-cm.

Equivalent Twisting moment $= \sqrt{(K_t . T_g)^2 + (Kb.mg)^2}$

Where $K_t = 1$, $kb = 1.5$ for gradual loading

$\therefore$ $T_{eq} = \sqrt{(1 \times 3130)^2 + (1.5 \times 2543)^2}$

$= 4934$ Kg-cm.

But $T_{eq} = \pi/16 \times \tau \times d^3{}_g$.

$\therefore$ $D_g = \sqrt[3]{T_{eq} \times 16/\pi \times \tau}$

Where, $\tau = 550$ kg/cm^2

$\therefore$ $d_g = \sqrt[3]{4934 \times 16/\pi \times 550}$

$= 3.5$ cm.

Bearing for worm shaft

The axial force $F_a = 239.8$ Kgf.

The radial force $F_r = 111.8$ Kgf.

Equivalent load $P = (XF_r + YF_a)S$

S = Service factor = 1.3 for no impact

X = radial factor

Y = Thrust factor

$F_a/F_r > e$ corresponding to taper roller bearings

$X = 0.4$, $Y = 1.6$

$\therefore \quad P = [0.4 \times 111.8 - 1.6 \times 239.8] \times 1.3$

$= 556.9$ Kg.

Dynamic capacity

$C = [(L/L_{10})]^{1/k} \times P$

Where,

$L_{10} = 1$ m.r

L = required life of bearings

$= 584 \times 60 \times 16 \times 52 \times 5/10^6$

$= 145$ m.r.

$C = [145/1]^{1/3} \times 556.9$

$= 2930$ Kg.

We select SKF 32206A Taper roller bearing Dimensions :

d	=	30 mm
D	=	62 mm
D_1	=	37 mm
D_2	=	36 mm
D_3	=	52 mm
C	=	4380 Kg.

Bearing for worm wheel

The axial force F_a = 239.8 Kg.

The radial force F_r = 111.8 Kg.

Equivalent load $P = [X.F_r + Y.F_a]\ S$

S = Service factor = 1.3 for no impact

X = radial factor

Y = thrust factor

$F_a/F_r = 239.8/111.8 = 2.14$

For $F_a/F_r > e$

FROM PSG DDP 4.4

$X = 0.67$

$Y = 3.1$

$\therefore\ P = [0.67 \times 111.8 + 3.1 \times 239.8] \times 1.3$

$= 1063.7$ Kg.

Dynamic capacity $C = [L/L_{10}]^{1/k} \times P$

Where $L_{10} = 1$ m.r

$L = 80 \times 60 \times 16 \times 52 \times 5/10^6$

$= 19.96$

$C = [19.96/1]^{1/3} \times 1063.7$

$= 2885$ Kg.

FROM PSG DDB 4.21

Corresponding to 2885 Kg dynamic capacity and diameter 35 mm we select

SKF NU 2207 cylindrical roller bearing

DIMENSIONS :

d = 35 mm

D	=	72 mm
B	=	23 mm
D_1	=	42 mm
D_2	=	65 mm

8. CONSTRUCTION DETAILS

Body of the machine

Two mild steel plates of dimension 1250 x 710 x 8 mm are purchased and are cut at suitable places to give a definite shape to the machine and to facilitate the functining of various components of the machine. Holes are bored at suitable places on both the plates to accomodate the bossings for the shaft bearings. Then with the use of accurately dimensioned cross links, the plates are separated 600 mm from each other in the vertical position. Then two 'L' angles of size IS 7575 are bent similar to the bottom shape of the machine and are welded to the body plates at the bottom. 'L' angles are extended 21" from the body plate, at the back of the machine on which the motor, gear box and basket for starch collection are mounted. Suitable number of cross links, of suitable dimensions are welded at appropriate places so that a distance of 600 mm is maintained between the plates. In the front of the machine, a cross plate is welded to the body at the bottom to fix the front caster wheels. Similarly a cross plate is welded between the 'L' angle extensions at the back to mount the back wheel of the machine. Stay links are welded between the body plates and angle extensions. Now the body is ready for mounting various components.

Conveyor and wedge

The conveyor is mounted in front of the machine which has a wedge in the front which can be folded vertically. The side housings of the conveyor is mounted on the shaft which is fitted to the body bossings. The other end of the conveyor housings have little extensions between which the wedge can be folded vertically. The conveyor is supported by two idlers and it must be of bottom conveyor. The belt rollers are supported by the bearings at the end.

The front wedge is mounted in such a way that it has 30° inclination from the ground level. Mild steel plates of 3 mm thickness is bent to have a base of 550 mm width and side cutting edges of 100 mm height. The base and side cutting edges are suitably supported by mild steel strips of 8 mm thickness. The wedge is mounted between the conveyor housing extensions. The wedge surface and the conveyor belt surface must be in the same plane of inclination.

Milling cutter

Twenty two steel sheets of 1.5 mm thickness and 250 mm in dia which is having 50 teeth on its periphery is mounted on 1.25" shaft and each plate is separated 23 mm from the other using distance pieces. This shaft is mounted in the body 1 mm above the conveyor top surface and 125 mm beyond the centre axis of the conveyor back roller inside the body.

Reciprocating cutter and eccentric

Reciprocating cutter is fabricated as shown in the drawing and the bearings are mounted in such a way that the bearings roll on the guides provided on the body. The cutter has finger like cutting edges at the bottom and a chip breaker plate of width 20 mm is welded on the side with an inclination of 60° from the cutting edge surface. The cutter is guided in such a way that it reciprocates in between the elevator cover and the milling cutters with sufficient clearance.

The eccentric is mounted on the eccentric shaft which is supported by the bearings mounted inside the bossings. The eccentric connected to the reciprocating cutter.

Elevator

The elevator is mounted between the reciprocating cutter and the motor. Thirteen buckets ('V' shape) are bolted to the elevator belt and the belt is mounted on the elevator pulleys. The top shaft is supported by the bearings in the bossings which are bolted to the body. The bottom shaft is fitted into the rectangular bossing with siliding arrangement. A single idler is provided between the elevator pulleys, below the belt surface. Cotton belt must be used for the elevator. A funnel like sheet metal fabrication is mounted at the back of the body near the top pulley of the elevator so that the starch pieces are guided to the basket.

Motor, gear box and clutch

Motor and gear box are mounted at the back of the body, on the 'L' angle cross plates. The clutch is mounted on the worm shaft extension of the gear box. Drive pulleys are mounted on the gear box shaft extensions and power is transmitted to various shafts through belts. The coverings provided above the motor and gear box are rigid enough to withstand the weight of basket full of starch placed above the covering. The clutch, gear and steering levers are mounted at the back of the motor and gear box.

Body coverings

The body coverings is suitably bent to provide good appearance to the machine and are bolted to the body. All moving parts are covered well with sheet metal and the covering shapes must add the appearance of the machine.

9. COST AND SUITABILITY

9.1 Cost of the Machine

Sl. NO.	DESCRIPTION OF PARTS	MANUFACTURE/PURCHASE	COST (APPROX) Rs. P.
1.	Motor	Purchase	2300.00
2.	Clutch Assembly	Purchase	1162.00
3.	Clutch Pulley	Purchase	41.00
4.	Clutch Belts-V3	Purchase	120.00
5.	Gear Box	Fabrication	2570.00
6.	Sprockets 7-No.	Purchase	377.00
7.	Chains	Purchase	300.00
8.	Shafts	Purchase	235.00
9.	Front Rollers	Purchase and Machining	200.00
10.	Conveyor Belt	Purchase	98.00
11.	Conveyor Side Housing	Purchase and Machining	95.00
12.	Elevator Belt (Cotton)	Purchase	260.00
13.	Elevator Pulleys	Purchase	300.00
14.	Circular Cutter	Purchasing and Cutting	213.00
15.	Circular Cutter Spacings	Purchasing and Machining	37.00
16.	Reciprocator	Fabrication	370.00
17.	Receiprocating Cutter	Fabrication	87.00
18.	Cast Iron Wheels-2 No.	Purchase and Machining	110.00
19.	Caster Wheels - 2 No.	Purchase and Fabrication	170.00
20.	All Bearings	Purchase	768.00

Sl. NO.	DESCRIPTION OF PARTS	MANUFACTURE/PURCHASE	COST (APPROX) Rs. P.
21.	Body Frame	Purchase	161.00
22.	All Cross Angles	Purchase	124.00
23.	All Body Coverings	Purchase and Fabrication	320.00
24.	All Idlers	Purchase and Machining	140.00
25.	Clutch and Turning Linkages	Fabrication	36.00
26.	Front Wedge	Fabrication	67.00
27.	Base Plate Above Caster	Cutting and Welding	47.00
28.	Elevator Buckets	Fabrication	142.00
29.	Flywheel	Purchase and Machining	120.00
30.	All Bossings	Casting and Machining	90.00
31.	Plating Charges	Plating	100.00
32.	Labour Charge for Fabrication	Fabrication	1050.00
33.	Body of the Machine	Purchase and Machining	1100.00
		Total	13,310.00

Since the industry benefits more by using the excavator, this much investment is quite tolerable by the industry. Because the excavator usage eliminates the unnecessary time and manpower consumption there by increasing the quality and quantity. At the same time factory overheads decreases verymuch per year and the total decrease is expenses will be approximately 15-20 thousands per year.

9.2 REDUCTION IN LABOUR COST WHEN EXCAVATOR IS USED

The cost calculation is made with an assumption of working hours per day and a minimum wage of Rs.25 for male workers and Rs.8 for womens. The expenses are calculated for tapping breaking and spreading of 6 cm thickness of starch in one hour.

S. No.	OPERATION	CONVENTIONAL METHOD				EXCAVATOR TECHNOLOGY			
		Male / Female	Number of Labours	Hours of work Hrs.	Labour cost Rupees	Male / Female	Number of Labours	Hours of work Hrs.	Labour cost Rupees
1.	Tapping	Male	4	4	49.92	Male	1	1	3.12
2.	Lifting to trying yards	Male	3	4	37.44	Male	4	1	12.48
3.	Breaking	Female	3	3.5	10.50	-	0	0	00.00
4.	Spreading	Female	1	3	3.00	Male	2	1.5	9.36
					100.86				24.96

	Rs. P.
The cost of electric current to operate a 3 HP motor per hour	4.50
Interest to the investment in the excavator per day	5.60
Maintainerce charge per day approximate	2.50
	12.60

∴ Total expense when excavator is used = 24.96 + 12.60

= Rs.37.56

∴ Reduction is expenses over conventional method = 100.86 - 37.56

= Rs.63.30

If the industry runs for 10 months,

The reduction in expenses = 63.30 x 300

= Rs.18,990/=

So, excavator technology is most suitable for starch industries.

10. FURTHER DEVELOPMENTS

The following developments may be made in the machine in future.

1. *The overall dimensions may be reduced.*
2. *The control in steering may be improved with the use of a small steering gear box.*
3. *Beater type cutters may be tried in the machine.*
4. *To provide still better hygeinic operation, the elevator buckets, wedge and the parts where the starch comes in contact may be replaced by suitable plastics.*

13. CONCLUSION

This project thus povides a suitable method for saving a considerable labour cost and time in starch industries. The usage of excavators will definitely increase the capacity of the plant.

The excavator technology is the starting point for technology upgradation in starch industries which leads to better quantity and quality production of starch.

This project enables the starch industries to produce more with considerably lesser expense.