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Design and Manufacturing of Portable Harvester

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Abstract- *The project is intended to help small-scale grain growers to meet an increased demand for diverse, locally grown grains by designing a small scale harvester. To refine our prototype and final design, we worked closely with a three person review panel, made up of grain farmers and industrial designers. With this prototype, we hope to provide farmers nationwide with a way to harvest grains on small plots of land in cities and along the periphery of urban areas. Our primary intension was to reduce down the capacity as well as the cost of the machine. This was thought so as to make the machine affordable to the Indian farmers. According to the survey, the per capita land owned by Indian farmer is tiny so this makes it obvious that the existing machines are not laconic. The capacity was so reduced that it would cater the needs on the fields. The harvester design is kept simpler so that any discontinuity in working would not affect the work. The machine being small in size is very easy to transport.*

Keywords- *Small scale grain growers, Per capita land, Capacity, Cost, Laconic*

I. INTRODUCTION

As India has large agricultural sector, the main crops in lower rainfall regions are Jowar, Hybrid and Bajra. The machines are not available to cut upper part of these crops and harvest them simultaneously. Also the latest technologies involved in design and manufacturing have made them more costly to buy for common farmer. These systems available require manual cutting. This conventional method requires high labour efforts and cost. The Harvesters available now are large in size and capacity which is not feasible as far as Indian scenario is concerned. So by taking this problem into account we are designing and manufacturing a harvester fulfilling all the basic needs as expected from same but at lower cost and capacity. It may be considered as automation in the agricultural practices. Our main concern is to build such a system which would harvest the crop and enable the farmer to increase productivity. It also reduces the time required for total process.

II. PROBLEM DEFINITION

It's a machine which harvests upper part (Kanis) of crops like Jowar and Bajra. Now in conventional methods the crop is cut first manually and stored, then it is harvested in harvester. So we are designing and manufacturing a harvester doing the work as explained above. The machine designed by us can harvest the crop in single machine itself. The cost of present harvesters is not affordable for common farmers. The machines for such applications are not available as single unit.

III. OBJECTIVES

- A. To Design and manufacture a harvester in reduced size and capacity.
- B. To come up with a harvester which would be economic to farmers.
- C. To make the harvester portable which can be moved to places with fewer efforts.
- D. To make the harvester multipurpose so as to use it for other than above crops.
- E. To make the design simpler to reduce the maintenance cost.

IV. METHODOLOGY

A. Data Collection

The design of the harvester starts from the initial data collection. The initial data includes information about crop, in this height of crop, the period of growth, etc. Also the data about working of present harvesters, the driving mechanism given to them, the power required for them is collected. The data collection is done by market survey and doing visit to actual field.

B. Layout Finalization

The final layout of machine is done according to the data.

C. Detailed Design

The next part of process is to design various parts of system in detail which are-

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- 1) Crusher
- 2) Sieve
- 3) Blower
- 4) Power train

The design consists of all information about the parts. The design of conveying system is based on design of above parts.

D. Evaluation and Drafting

Once the components are designed, they are tested for their strength on the analysis software. Then they are assembled and the mechanisms are observed for their proper working and non-interrupted motion. The drafting and production drawing as per the codes and conduct of the sponsor are taken into account.

E. Assembly

The parts are designed and evaluated for assembly. The harvester is then assembled for final appearance.

F. On Field Trials

The combined harvester is taken to actual field conditions for testing. Testing will be carried out at start of February as it is the season for crops like Jowar, Bajra. All the components are tested for their proper functioning. If any problem arises, corrective action will be taken and the trial will again be taken to confirm proper functioning of all parts.

G. Expenditure

The key of this project is to be able to provide robust and workable design to the Indian farmers so that the project can be subsequently taken up for many more such machines in future. The estimated cost of this project is around Rs. 15000 to 18000.

V. INITIAL CONCEPT OF THE PROPOSED EQUIPMENT

It is mainly agriculture based project. It includes harvesting the cut crop. The work consists of designing and manufacturing of following components and their assembly-

A. Thresher

It is the most important part of harvester. It is made up of number of studs on its periphery. They are so arranged that it keeps a constant gap in between two adjacent bolts and in between two adjacent rows as well. In some threshers there are some cutting blades provided in order to smash down the Kanis into pieces so that it would help to reduce down the efforts to crush. The studs are fixed on a metal plate of certain width in a line. Such numbers of strips are attached to two metal rings at both ends to make a closed ring type structure. A net of metal rods is placed at bottom side of crusher ring which is curved. This net is having a gap in network so that it allows the grains to fall down through it. The casing is provided on this crusher with suitable clearance provided in between them, such that the Kanis gets trapped in this gap and gets shattered. The bolts strike the Kanis at high speed in order to separate grains off it. The casing is provided with an opening for inserting Kanis for threshing or crushing. The casing is bolted to frame.

B. Blowers

There are two blowers provided on machine. The work of picking up the husk to blow away is divided into two parts by these two blowers. These blowers are driven by the same shaft. This shaft also drives the crusher as well (being a common shaft running through the length). Blower consists of 4 closed structured metal plates. This structure helps to create negative pressure inside the blower which allows the husk to be lifted up. The duty of one of the two blowers is to lift the husk from coarse sieve and the rest has to lift the husk from fine sieve. As the grains come out of crushing chamber they are allowed to fall on a sieve. The 1st blower then lifts the husk at this point. Most of the husk is been lifted by this blower. After this the grain comes to fine sieve where the remaining husk is been lifted by second blower.

C. Sieve

The sieve serves the function of separating the husk from grains. There are two sieves provided for this. The two sieves are so attached to get single unit. The first sieve is coarse one. It separates the husk at maximum possible. The grain then gets over second

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sieve so that the remaining husk is separated. The two blowers are so placed over the sieve that they suck the husk from sieve. 1st blower is placed at the start of coarse sieve and the second blower at fine sieve at the end. The grains after passing through these two sieves come out of fine sieve and are now ready to store.

D. Support Frame

All components are rested on the support frame. It is a rectangular shaped structure where there are four legs on which it rests. Another frame is created on a pair of legs for mounting of shaft 2 (which is meant to drive the sieve). Main frame is split into two parts for mounting of blower and crusher. The support frame also provides three members for supporting the sieve so that it moves in horizontal constrained plane only.

E. Power train

The main shaft is given power from external source. The shaft gives power to crusher, Blower and sieve. A direct drive through belt is provided for main shaft. The sieve is given power through quarter turn belt drive. The main shaft rotated at 750rpm whereas the rpm given to sieve is reduced to half through pulleys. It is 375rpm.

| Sr. No. | Model Specification | Power (HP) | Price (Rs.) | Capacity(Kg/hr) |
|---------|--|------------|-------------|-----------------|
| 1 | Double Wheel Triple Fan 1  | 25 | 1,44,000 | 550-600 |
| 2 | Double Wheel Triple Fan 2  | 20 | 1,39,500 | 400-450 |
| 3 | Double Wheel Triple Fan 3  | 15 | 1,36,500 | 300-350 |

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| | | | | |
|---|---|---|--------|---------|
| 4 | Single Wheel Double Fan  | 5 | 58,000 | 150-200 |
| 5 | Single Wheel Double Fan (Our Machine)  | 2 | 18,000 | 50-100 |

COMPARISON WITH AVAILABLE MACHINE

VI. DESIGN CALCULATIONS

A. Power Calculation

The power required for harvester is,
 Power can be given as,

$$P = 2\pi NT/60$$

Where,

$$N = \text{Speed of rotating shaft} = 720\text{rpm}$$

$$T = \text{Torque acting on shaft}$$

Torque can be calculated as follows,

Consider 100 N crushing force acted upon Kanis by thresher ring.

$$F = 100\text{ N}$$

$$\text{Torque, } T = F \times R$$

Where, $R = \text{radius of thresher ring} = 0.150\text{ m}$

$$T = 100 \times 0.150$$

$$T = 15\text{ Nm}$$

Therefore, required power is given as,

$$P = (2 \times \pi \times 720 \times 15)/60$$

$$= 1130.9733\text{ W}$$

$$P = 1.13\text{KW}$$

The power to be required is 1.13KW i.e. 1.51 HP

Motor selected for application is of 2 HP.

B. Selection of Belt

1) For drive pulley

$$\text{Motor Speed} = 1440\text{ rpm}$$

$$\text{Power to be transmitted} = 2\text{HP} = 1.492\text{ KW}$$

$$\text{Shaft Speed} = 720\text{ rpm}$$

$$\text{Centre Distance (C)} = 510\text{ mm}$$

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Selection of V belt from manufacture catalogue –

Steps: –

1) Correction Factor (F_a) –

It depends on type of driving unit, type of driven machine and the operational hours per day.

For normal torque, squirrel cage, motor and for heavy duty machine for 8 hours of operation per day,

Correction factor (F_a) = 1.2 ... [Table 13.15, Page No. 525, V. B. Bhandari]

2) Calculation of design power –

$$\begin{aligned} \text{Design power} &= F_a(\text{Transmitted power}) \\ &= 1.2 \times 1.492 \end{aligned}$$

$$\text{Design power} = 1.7404 \text{ KW}$$

3) Type of cross – section of belt –

By plotting design power as X co – ordinate and input speed as Y co – ordinate, cross section of belt is decided by this point

Cross – section of Belt is B ... [Table 13.24, Page No. 523, V. B. Bhandari]

4) Calculation of pitch diameters of pulleys

The pitch diameter of bigger pulley is given as,

$$D = d (\text{Speed of smaller pulley}) / (\text{Speed of bigger pulley})$$

$$D = d (\text{input speed}) / (\text{output speed})$$

$$D = d (N_1 / N_2)$$

Let, $d = 50 \text{ mm}$

$$D = 50 \times (1440 / 720)$$

$$D = 100 \text{ mm}$$

Hence, $D = 100 \text{ mm}$ and $d = 50 \text{ mm}$... [Table 13.13, Page No. 524, V. B. Bhandari]

5) Calculation of pitch length

$$L = 2C + \pi(D + d) / 2 + (D - d)^2 / 4C$$

$$L = 2 \times 480 + \pi(100 + 50) / 2 + (100 - 50)^2 / 4 \times 480$$

$$L = 1196.844 \text{ mm}$$

6) Corrected pitch length

$$L = 1210 \text{ mm} \quad \dots [\text{Table 13.14, Page No. 524, V. B. Bhandari}]$$

7) Corrected centre distance

$$L = 2xc + \pi(D + d) / 2 + (D - d)^2 / 4C$$

$$1210 = 2C + \pi(100 + 50) / 2 + 50^2 / 4C$$

$$C^2 - 974.386 C + 625 = 0$$

$$\text{Hence, } C = 510 \text{ mm}$$

8) Correction factor (F_c) for belt pitch length

On basis of cross section type and pitch length,

Correction factor (F_c) = 0.87 ... [Table 13.21, Page No. 534, V. B. Bhandari]

9) Arc of contact for smaller pulley

$$\alpha_s = 180 - 2 \sin^{-1} (D - d) / 2xC$$

$$\alpha_s = 180 - 2 \sin^{-1} (100 - 50) / 2 \times 510$$

$$\alpha_s = 174.380$$

Correction factor (F_d) for arc of contact

$$F_d = 0.99 \quad \dots [\text{Table 13.22, Page No. 534, V. B. Bhandari}]$$

10) Power ratings for V belt (P_r)

It depends upon speed of faster shaft, pitch diameter of smaller pulley and the speed ratio.

$$P_r = 1.12 + 0.19 \quad \dots [\text{Table 13.17, Page No. 528, V. B. Bhandari}]$$

$$P_r = 1.31$$

Selection of Belt for Quarter turn drive –

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Input Speed = 720 rpm

Power to be transmitted = 2HP = 1.492 KW

Output Speed = 360 rpm

Centre Distance (C) = 350 mm

Selection of V belt from manufacture catalogue

Steps: –

1) Correction Factor (Fa) –

It depends on type of driving unit, type of driven machine and the operational hours per day.

For normal torque, squirrel cage, motor and for heavy duty machine for 8 hours of operation per day,

Correction factor (Fa) = 1.2 ... [Table 13.15, Page No. 525, V. B. Bhandari]

2) Calculation of design power

Design power = Fa (Transmitted power)
= 1.2 x 1.492

Design power = 1.7404 KW

3) Type of cross section of belt

By plotting design power as X co – ordinate and input speed as Y co
– ordinate, cross section of belt is decided by this point

Cross section of Belt is B ... [Table 13.24, Page No. 523, V. B. Bhandari]

4) Calculation of pitch diameters of pulleys

The pitch diameter of bigger pulley is given as,

$D = d (\text{Speed of smaller pulley}) / (\text{Speed of bigger pulley})$

$D = d (\text{input speed}) / (\text{output speed})$

$D = d (N1/N2)$

Let, $d = 50 \text{ mm}$

$D = 50 \times (1440/720)$

$D = 100 \text{ mm}$

Hence, $D = 100 \text{ mm}$ and $d = 50 \text{ mm}$... [Table 13.13, Page No. 524, V. B. Bhandari]

5) Calculation of pitch length

$L = 2C + \pi(D + d)/2 + (D - d)^2/4C$

$L = 2 \times 480 + \pi(100 + 50)/2 + (100 - 50)^2/4 \times 480$

$L = 1196.844 \text{ mm}$

6) Corrected pitch length

$L = 1210 \text{ mm}$... [Table 13.14, Page No. 524, V. B. Bhandari]

7) Corrected centre distance

$L = 2xc + \pi(D + d)/2 + (D - d)^2/4C$

$1210 = 2C + \pi(100 + 50)/2 + 502/4xC$

$C^2 - 974.386 C + 625 = 0$

Hence, $C = 510 \text{ mm}$

8) Correction factor (Fc) for belt pitch length

On basis of cross section type and pitch length,

Correction factor (Fc) = 0.87 ... [Table 13.21, Page No. 534, V. B. Bhandari]

9) Arc of contact for smaller pulley

$\alpha_s = 180 - 2 \sin^{-1} (D - d) / 2xC$

$\alpha_s = 180 - 2 \sin^{-1} (100 - 50) / 2 \times 510$

$\alpha_s = 174.380$

Correction factor (Fd) for arc of contact

$Fd = 0.99$... [Table 13.22, Page No. 534, V. B. Bhandari]

10) Power ratings for V belt (Pr) –

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It depends upon speed of faster shaft, pitch diameter of smaller pulley and the speed ratio.

$$Pr = 1.12 + 0.19 \quad \dots [\text{Table 13.17, Page No. 528, V. B. Bhandari}]$$

$$Pr = 1.31$$

Selection of Belt for Quarter turn drive –

$$\text{Input Speed} = 720 \text{ rpm}$$

$$\text{Power to be transmitted} = 2\text{HP} = 1.492 \text{ KW}$$

$$\text{Output Speed} = 360 \text{ rpm}$$

$$\text{Centre Distance (C)} = 350 \text{ mm}$$

Selection of V belt from manufacture catalogue

Steps: –

1) *Correction Factor (Fa) –*

It depends on type of driving unit, type of driven machine and the operational hours per day.

For normal torque, squirrel cage, motor and for heavy duty machine for 8 hours of operation per day,

$$\text{Correction factor (Fa)} = 1.2 \quad \dots [\text{Table 13.15, Page No. 525, V. B. Bhandari}]$$

2) *Calculation of design power*

$$\begin{aligned} \text{Design power} &= Fa (\text{Transmitted power}) \\ &= 1.2 \times 1.492 \end{aligned}$$

$$\text{Design power} = 1.7404 \text{ KW}$$

3) *Type of cross section of belt*

By plotting design power as X co – ordinate and input speed as Y co – ordinate, cross section of belt is decided by this point

$$\text{Cross section of Belt is B} \quad \dots [\text{Table 13.24, Page No. 523, V. B. Bhandari}]$$

4) *Calculation of pitch diameters of pulleys*

The pitch diameter of bigger pulley is given as,

$$D = d (\text{Speed of smaller pulley}) / (\text{Speed of bigger pulley})$$

$$D = d (\text{input speed}) / (\text{output speed})$$

$$D = d (N1/N2)$$

$$\text{Let } d = 80\text{mm}$$

$$D = 80 \times (1600/800)$$

$$D = 160 \text{ mm}$$

Hence, D = 160 mm and d = 80 mm ... [Table 13.13, Page No. 524, V. B. Bhandari]

5) *Calculation of pitch length –*

$$L = 2C + \pi (D + d) / 2 + (D - d)^2 / 4C$$

$$L = 2 \times 350 + \pi (160 + 80) / 2 + (160 - 80)^2 / 4 \times 350$$

$$L = 1081.56 \text{ mm}$$

6) *Corrected pitch length –*

$$L = 1100 \text{ mm} \quad \dots [\text{Table 13.14, Page No. 524, V. B. Bhandari}]$$

7) *Corrected centre distance –*

$$L = 2xc + \pi (D + d) / 2 + (D - d)^2 / 4C$$

$$1100 = 2C + \pi (160 + 80) / 2 + 80^2 / 4 \times C$$

$$C^2 - 361.505 C + 800 = 0$$

$$\text{Hence, } C = 360 \text{ mm}$$

8) *Correction factor (Fc) for belt pitch length –*

On basis of cross section type and pitch length,

$$\text{Correction factor (Fc)} = 0.85 \quad \dots [\text{Table 13.21, Page No. 534, V. B. Bhandari}]$$

9) *Arc of contact for smaller pulley –*

$$\alpha_s = 180 - 2 \sin^{-1} (D - d) / 2 \times C$$

$$\alpha_s = 180 - 2 \sin^{-1} (160 - 80) / 2 \times 350$$

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$$\alpha_s = 169.220$$

Correction factor (F_d) for arc of contact –

$$F_d = 0.97 \quad \dots [\text{Table 13.22, Page No. 534, V. B. Bhandari}]$$

10) Power ratings for V belt (P_r) –

It depends upon speed of faster shaft, pitch diameter of smaller pulley and the speed ratio.

$$P_r = 0.66 + 0.10 \quad \dots [\text{Table 13.17, Page No. 528, V. B. Bhandari}]$$

$$P_r = 0.76$$

C. Design of Sieve Shaft

1) Belt tension for design of shaft –

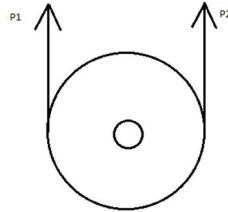


Figure 1. Reaction on shaft

Load on shaft due to shaking of sieve is considered 50N in horizontal direction.

Pitch diameter of pulley = 160mm

So, $R = 80\text{mm}$

Angle of wrap for pulley = $\theta = 180^\circ$

Coefficient of Friction = $\mu = 0.24$

Calculation of torque on shaft –

$$P = 2\pi NT/60 = 1.497\text{KW} = 1.497 \times 10^3\text{W}$$

$$\text{So, } N = 360 \text{ rpm}$$

$$T = 39.70 \times 10^3\text{N.mm}$$

Calculation of load on shaft due to pulley –

By using relation,

$$T_1/T_2 = e^{\mu\theta} = e^{0.24 \times \pi} = 2.125$$

$$\text{So, } T_1 = 2.125T_2 \quad \dots \dots \dots (1)$$

The torque supplied to shaft is also given as,

$$(T_1 - T_2) \times R = Mt$$

$$(T_1 - T_2) \times 80 = 39.70 \times 10^3$$

$$\text{So, } (T_1 - T_2) = 496.25 \quad \dots \dots \dots (2)$$

From equation (1) and (2), solving,

$$T_2 = 441.11 \text{ N}$$

$$T_1 = 937.36 \text{ N}$$

Load due to Pulley on shaft = $T_1 + T_2 + \text{Weight of pulley}$

$$= 937.36 + 441.11 + 20$$

$$= 1398.47\text{N in vertical upward direction}$$

(2) The loads acting on shaft are in both directions i.e. in calculation of bending moments –

a) In vertical direction –

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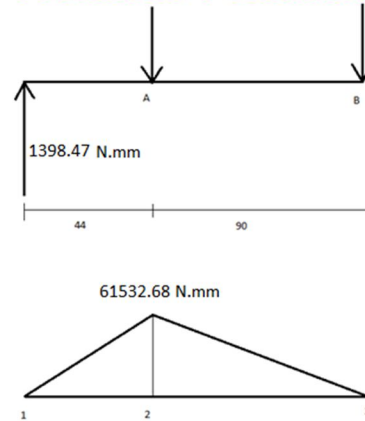


Figure 2. Bending moments on shaft

Calculation of reactions,

$$RA_v + RB_v = 1398.47N \quad \dots\dots\dots (1)$$

Taking moment about point B,

$$1398.47 \times 44 = -90 \times RB_v$$

$$\text{So, } RB_v = -683.69N$$

From equation (1),

$$RA_v = 2082.16N$$

B. M. Calculation -

$$\text{B. M. at 1} = 0$$

$$\begin{aligned} \text{B. M. at 2} &= 1398.47 \times 44 \\ &= 61532.68N. mm \end{aligned}$$

$$\text{B. M. at 3} = 0$$

Maximum bending moment in vertical direction,

$$M_b \text{ at A} = 61532.68N. mm$$

b) In horizontal direction -

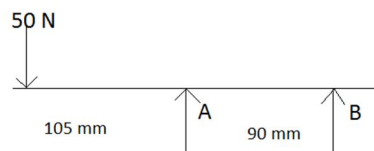


Figure 3. Bending moment on shaft

Calculations of reactions -

$$RA_h + RB_h = 50 N$$

Taking moment @ A,

$$-50 \times 105 = 90 \times RB_h$$

$$RB_h = -58.33 N$$

$$RA_h = 108.33 N$$

Bending Moment calculation-

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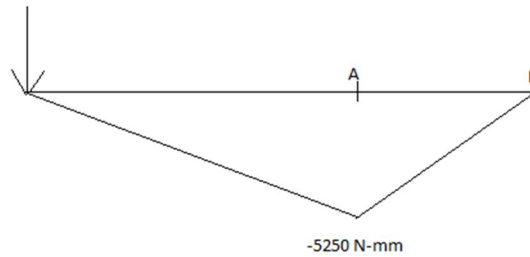


Figure 4. Bending moment on shaft

$$B.M. @ 1 = 0$$

$$B.M. @ 2 = -5250 \text{ N} - \text{mm}$$

$$B.M. @ 3 = 0$$

Maximum bending moment in horizontal direction is at A, which is given as

$$Mb_H \text{ at A} = -5250 \text{ N} - \text{mm}$$

By using relation $Mb = \sqrt{Mb_v^2 + Mb_h^2}$

Total bending moment is calculated,

$$Mb = \sqrt{61532.68^2 + (-5250)^2}$$

$$Mb = 61756.24 \text{ N} - \text{mm}$$

Torsional Moment (Mt) –

It is torque acting on the shaft which is calculated as,

$$P = (2\pi x N x T) / 60$$

$$P = 1.497 \text{ KW}$$

$$N = 360 \text{ rpm}$$

Therefore $T = 39.70 \times 10^3 \text{ N} - \text{mm}$

$$Mt = 39.70 \times 10^3 \text{ N} - \text{mm}$$

3) Design of shaft –

The design of shaft is based upon equivalent torsional moment method

The value of shock and fatigue factors are given on the basis of application of load,

For gradually applied load,

$$Kb = \text{shock factor} = 1.5$$

$$Kt = \text{fatigue factor} = 1 \quad \dots \dots \dots [\text{Table 9.2, Page No. 334, V. B. Bhandari}]$$

Material selection –

The material selected for shaft is Mild steel with grade 30C8

Properties: –

$$\text{Ultimate tensile strength} = Sut = 500 \text{ N/mm}^2$$

$$\text{Yield strength} = Syt = 400 \text{ N/mm}^2 \quad \dots [\text{Table 2.2, Page No. 31, V. B. Bhandari}]$$

The permissible shear stress can be given as

$$\tau_{max} = (0.5Sut) / F.S.$$

$$\text{Factor of safety} = 2$$

$$\tau_{max} = (0.5 \times 500) / 2$$

$$\tau_{max} = 125 \text{ N/mm}^2$$

The shaft diameter is calculated by using equivalent torsional moment method,

The permissible shear stress can be given as,

$$\tau_{max} = 16x \sqrt{(Kb \times Mb)^2 + (Kt \times Mt)^2} / \pi x d^3$$

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Putting all known values in above equation

$$125 = (16x\sqrt{(1.5 \times 61756.24)^2 + (39.70 \times 10 \times 10 \times 10)^2}) / \pi \times d^3$$

$$d = 15.56 \text{ mm} \approx 20 \text{ mm}$$

$$d = 20 \text{ mm} \quad \text{--- diameter of shaft}$$

D. Bearing selection for Sieve shaft –

The support reaction are given as follow

At point A (bearing 1) –

$$RAV = 2082.16 \text{ N}$$

$$RAH = 108.33 \text{ N}$$

At point B, (bearing 2) –

$$RBV = -686.69 \text{ N}$$

$$RBH = -58.33 \text{ N}$$

$$RA = \sqrt{RAV^2 + RAH^2}$$

$$= \sqrt{2082.16^2 + 108.33^2}$$

$$RA = 2084.97 \text{ N}$$

$$RB = \sqrt{RBV^2 + RBH^2}$$

$$= \sqrt{(-686.69)^2 + (-58.33)^2}$$

$$RB = 689.16 \text{ N}$$

Radial forces on bearings are,

$$Fr1 = RA = 2084.97 \text{ N}$$

$$Fr2 = RB = 689.16 \text{ N}$$

There is no any axial thrust is acting on bearings,

$$Fa1 = Fa2 = 0$$

Dynamic load calculations

$$P1 = Fr1 = 2084.97 \text{ N}$$

$$P2 = Fr2 = 689.16 \text{ N}$$

Calculation of dynamic load capacity: –

The life of bearings selected is on basis of machine used for eight hours

$$L_{10h} = 12000 \text{ hr} \quad \dots [\text{Table 15.2, Page No. 573, V. B. Bhandari}]$$

Speed of shaft = 360 rpm

Life of bearing in million f revolution is given as,

$$L_{10} = (60 \times n \times L_{10h}) / 10^6$$

$$= (60 \times 360 \times 12000) / 10^6$$

$$L_{10} = 259.2 \text{ million revolution .}$$

Load factor = 2 ... [Table 15.3, Page No. 573, V. B. Bhandari]

Dynamic load capacity for bearing 1 –

$$C1 = P1 \times (L_{10})^{1/3} \times (\text{load factor})$$

$$= 2084.97 \times 259.21^{1/3} \times 2$$

$$C1 = 26587.33 \text{ N}$$

Dynamic load capacity for bearing 2 –

$$C2 = P2 \times (L_{10})^{1/3} \times (\text{load factor})$$

$$= 689.16 \times 259.21^{1/3} \times 2$$

$$C2 = 8788.10 \text{ N}$$

Table 1. Bearing Specification -

| Parameter | Bearing 1 (16007) | Bearing 2 (6404) | Bearing 3 (6407) |
|-----------|-------------------|------------------|------------------|
|-----------|-------------------|------------------|------------------|

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|--------------------------------|--------|--------|--------|
| Inner diameter | 35mm | 40mm | 35mm |
| Outer diameter | 62mm | 68mm | 100mm |
| Width | 9mm | 9mm | 25mm |
| Dynamic load carrying capacity | 12400N | 13300N | 55300N |
| Static load carrying capacity | 6950N | 7800N | 31000N |

[Table 15.5, Page No. 575, V. B. Bhandari]

E. Design Of Flywheel –

Type – Solid disk flywheel

The material selected for flywheel is carbon steel which has properties as follows,

Density = $\rho = 7800 \text{ Kg/m}^3$... [Table 21.1, Page No. 751, V. B. Bhandari]

The coefficient of fluctuation of energy for flywheel is taken as,

$$C_e = 0.066 \quad \dots \text{[Table 21.3, Page No. 753, V. B. Bhandari]}$$

The coefficient of fluctuation of energy is also given as,

$$C_e = (\text{Maximum fluctuation energy}) / (\text{work done per cycle})$$

$$C_e = (U_0) / (\text{work done per cycle})$$

Work done per cycle is calculated as below

$$W. D / \text{cycle} = 4\pi x T_m$$

Where, $T_m = \text{torque on shaft}$

$$T_m = 19.85 \text{ N} - \text{m}$$

$$W. D / \text{cycle} = 4\pi x 19.85$$

$$W. D / \text{cycle} = 249.44 \text{ N} - \text{m}$$

Therefore,

$$0.066 = (U_0) / 249.44$$

$$U_0 = 16.463 \text{ N} - \text{m}$$

$$\text{Maximum fluctuation energy} = U_0 = 16.463 \text{ N} - \text{m}$$

The moment of inertia of flywheel disk is given by equation

$$I = (U_0) / (\omega^2 x C_s)$$

Where, $\omega = \text{angular speed of shaft}$

$$\omega = (2\pi x N) / 60$$

$$= (2 x 3.14 x 720) / 60$$

$$= 75.39 \text{ rad/sec}$$

$C_s = \text{coefficient of fluctuation of speed}$

$$C_s = 0.025 \quad \dots \text{[Table 21.2, Page No. 752, V. B. Bhandari]}$$

$$I = (16.463) / (75.39^2 x 0.025)$$

$$I = 0.1158 \text{ Kg. m}^2$$

The moment of inertia is also given as ,

$$I = (\pi x \rho x t x R^4) / 2$$

Where, $\rho = \text{Density of flywheel material}$

$$= 7800 \text{ Kg/m}^3$$

$$t = \text{thickness of disk} = 0.02 \text{ m}$$

$$R = \text{radius of disk (m)}$$

$$0.1158 = (\pi x 7800 x 0.02 x R^4) / 2$$

$$R = 0.150 \text{ m} = 150 \text{ mm}$$

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The mass of flywheel is calculated by relation,

$$I = (m \times R^2) / 2$$

$$0.1158 = m \times 0.152 / 2$$

$$m = 14.07 \text{ Kg} \approx 15 \text{ Kg}$$

Flywheel parameter,

$$\text{Outer diameter} = 300 \text{ mm}$$

$$\text{Inner diameter} = 40 \text{ mm}$$

$$\text{Thickness} = 20 \text{ mm}$$

Design Of Keys –

The keys to be designed are of square and flat keys.

The material selected for keys is Mild steel with grade 30C8. Which has yield strength is,

$$S_{yt} = 380 \text{ N/mm}^2 \quad \dots [\text{Table 2.2, Page No. 31, V. B. Bhandari}]$$

Steps: –

Calculation of permissible compressive and shear stresses

$$S_{yc} = S_{yt} = 380 \text{ N/mm}^2$$

$$B_c = (S_{yc}) / F.S.$$

$$= 380 / 3$$

$$= 126.67 \text{ N/mm}^2$$

According to maximum shear stress theory of failure

$$S_{sy} = 0.5 \times S_{yt} = 0.5 \times 380$$

$$= 190 \text{ N/mm}^2$$

$$\tau = (S_{sy}) / (F.S.)$$

$$= 190 / 3$$

$$\tau = 63.33 \text{ N/mm}^2$$

Key 1:

Torque transmitted by shaft

$$T = 19.85 \times 10^3 \text{ N} - \text{mm}$$

Key dimension ,

$$b = h = d / 4 = 35 / 4$$

$$b = h = d = 8.75 \approx 8 \text{ mm}$$

b = width of key

h = height of key

Length of key = 80 mm

Check for shear,

$$\tau = (2 \times M_t) / (d \times b \times l)$$

$$= (2 \times 19.85 \times 10^3) / (35 \times 8 \times 80)$$

$$\tau = 1.79 \text{ N/mm}^2 < 63.33 \text{ N/mm}^2$$

Design is safe Check for crushing

$$B_c = (4 \times M_t) / (d \times h \times l)$$

$$= (4 \times 19.85 \times 10^3) / (35 \times 8 \times 80)$$

$$B_c = 3.54 \text{ N/mm}^2 < 126.67 \text{ N/mm}^2$$

Design is safe

Key 2:

Torque transmitted by shaft

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$$T = 39.70 \times 10^3 \text{ N} - \text{mm}$$

Key dimension ,

$$b = h = d/4 = 25/4 \\ = 6.25 \approx 6 \text{ mm}$$

b = width of key

h = height of key

Length of key = 60mm

Check for shear –

$$\tau = (2xMt)/(dxbxl) \\ = (2x39.70x10^3) / (25x6x60)$$

$$\tau = 8.81 \text{ N/mm}^2 < 63.33 \text{ N/mm}^2$$

Design is safe

Check for crushing –

$$\sigma_c = (4xMt) / (dxbxl) \\ = (4x39.70x10^3) / (25x6x60)$$

$$\sigma_c = 17.64 \text{ N/mm}^2 < 126.67 \text{ N/mm}^2$$

Design is safe

F. Design of Main Shaft –

The shaft is subjected to both vertical and horizontal loads

Calculation of bending moment –

1) In vertical direction-

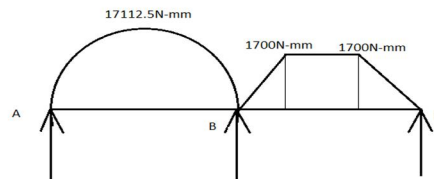
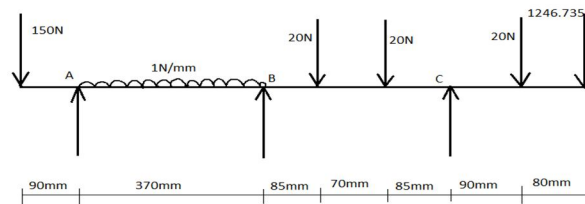


Figure 5. Moment on main shaft

Span moment for AB,

$$\text{Max B. M.} = (Wxl^2)/8 \\ = (1x370^2)/8 \\ = 17112.5 \text{ N} - \text{mm}$$

For BC,

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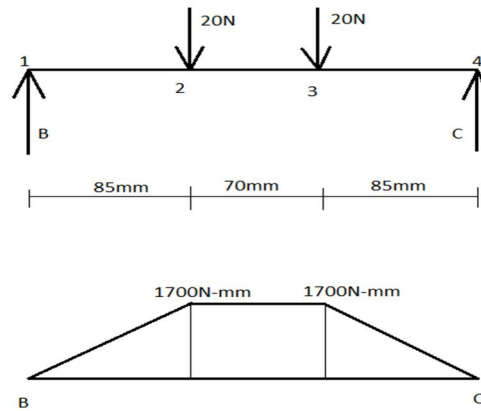


Figure 6. Reaction on main shaft

Reaction calculations –

$$R_B + R_C = 40 \text{ N}$$

Taking moment about point C

$$R_B \times 240 - 20 \times 1.55 - 20 \times 85 = 0$$

$$240R_B - 3100 - 1700 = 0$$

$$R_B = 20 \text{ N}$$

$$R_A = 20 \text{ N}$$

Bending moment calculation –

$$B. M. \text{ at point 1} = 0$$

$$B. M. \text{ at point 2} = 1700 \text{ N} - \text{mm}$$

$$B. M. \text{ at point 3} = 1700 \text{ N} - \text{mm}$$

$$B. M. \text{ at point 4} = 0$$

Support moment calculation –

Apply three moment theorem (TMT) on span AB + BC ... (SOM – Ramamrutam)

$$M_A \times l_1 + 2 \times M_B (l_1 + l_2) + M_C \times l_2 = \frac{(6 \times a_1 \times x_1^3)}{l_1} + \frac{(6 \times a_2 \times x_2^3)}{l_2}$$

Where,

$$+ M_A = 150 \times 90 = 13500 \text{ N} - \text{mm}$$

$$l_1 = 370 \text{ mm}$$

$$M_B = ?$$

$$l_2 = 240 \text{ mm}$$

$$M_C = 213744.95 \text{ mm}$$

$$a_1 = \frac{2 \times 370 \times 17112.5}{3}$$

$$= 4.22 \times 10^6 \text{ mm}^2$$

$$x_1 = 185 \text{ mm}$$

$$a_2 = \left(\frac{1 \times 85 \times 1700}{2} \right) + (1700 \times 70) + \left(\frac{1 \times 1700 \times 85}{2} \right)$$

$$a_2 = 263500 \text{ mm}^2$$

$$x_2 = \left(\frac{2 \times 85}{3} \right) + 35 + \left(\frac{1 \times 85}{3} \right)$$

$$= 116.67 \text{ mm}$$

Putting all values in above equation, we get,

$$(13500 \times 370) + (2 \times M_B \times 610) + (213744.95 \times 240)$$

$$= \frac{(6 \times 4.22 \times 10^6 \times 185)}{370} + \frac{(6 \times 263500 \times 116.67)}{240}$$

$$2.29 \times 10^6 + 1220 M_B + 51.3 \times 10^6 = 12.66 \times 10^6 + 768563.63$$

$$= 13.43 \times 10^6 \text{ N} - \text{mm}$$

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$MB = -32018.033 \text{ N-mm}$

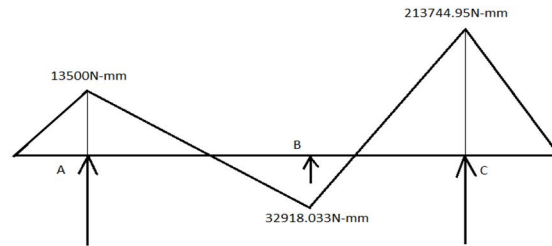


Figure 7. Moments at supports

Reaction calculation –

$$RA + RB + RC = 1826.735 \text{ N}$$

Taking moment about point B for span DAB,

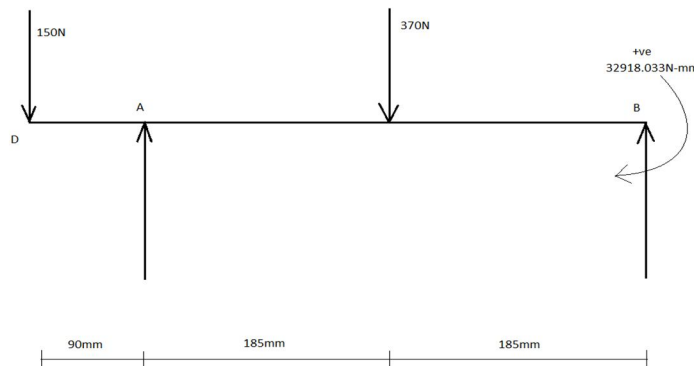


Figure 8. Reaction at supports

$$-150 \times 460 + RA \times 370 - 370 \times 185 + 32918.083 = 0$$

$$-69000 + 370RA - 68450 + 32918.033 = 0$$

$$370RA = +104531.967$$

$$RA = 282.52 \text{ N}$$

Taking moment about point B for span BCEF

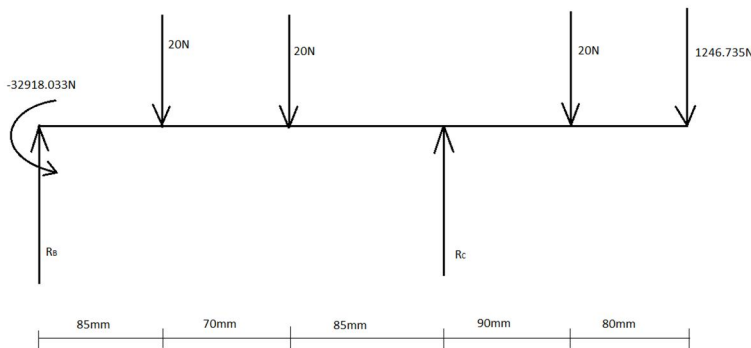


Figure 9. Moments on main shaft

$$1246.735 \times 410 + 20 \times 330 - RC \times 240 + 20 \times 155 + 20 \times 85 - 32918.033 = 0$$

$$511159.3 + 6600 - 240RC + 3100 + 1700 - 32918.033 = 0$$

$$-240RC + 489641.267 = 0$$

$$RC = 2040.172 \text{ N}$$

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$$\begin{aligned}
 RA + RB + RC &= 1826.735 \text{ N} \\
 282.52 + RB + 2040.172 &= 1826.735 \\
 RB & \\
 &= -495.96 \text{ N}
 \end{aligned}$$

2) In horizontal Reaction –

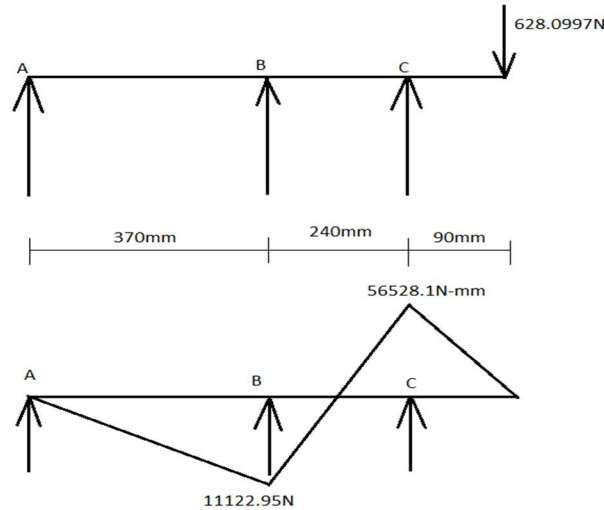


Figure 10. Horizontal reaction

Span moment –

For both spans AB and BC Span moments are zero

Support moment calculation –

$$\begin{aligned}
 MA &= 0 \\
 MB &=? \\
 MC &= 628.09 \times 90 \\
 &= 56528.1 \text{ N-mm}
 \end{aligned}$$

Apply Three Moment Theorem (TMT) on Span AB and BC

$$MAx l_1 + 2xMB(l_1 + l_2) + MC x l_2 = (6x a_1 x x_1^3) / l_1 + (6x a_2 x x_2^3) / l_2$$

$$\begin{aligned}
 MA &= 0 \\
 2MB(370 + 240) + 56528.1 x 240 &= 0 \\
 1220 MB &= -13.57 x 10^6 \\
 MB &= -11122.95 \text{ N-mm}
 \end{aligned}$$

Reaction calculation –

$$RA + RB + RC = 628.09$$

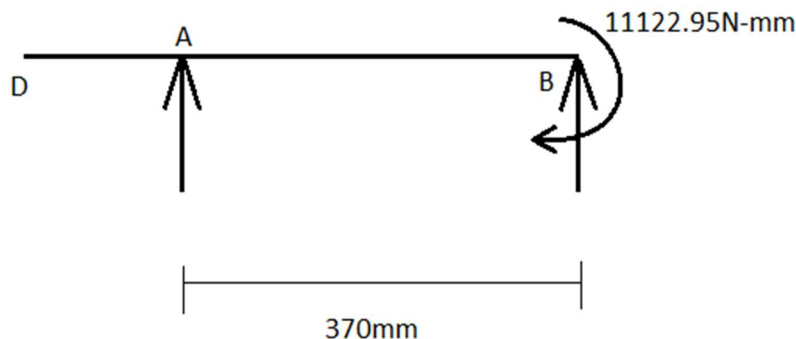


Figure 11. Span moment

Taking moment about point B, for span AB

$$\begin{aligned}
 RA x 370 + 11122.95 &= 0 \\
 \text{So, } RA &= 30.062 \text{ N}
 \end{aligned}$$

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Taking moment about point B, for span BCE

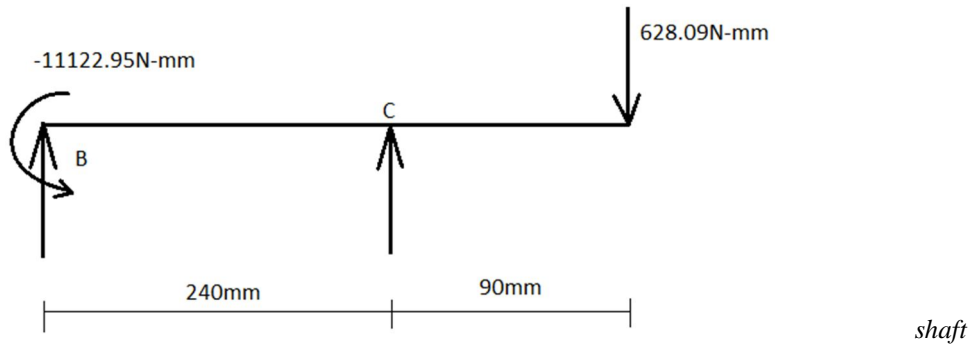


Figure 12. Moment on main

$$\begin{aligned}
 -11122.95 - RC \times 240 + 628.09 \times 330 &= 0 \\
 -240 RC &= 817.28 \text{ N} \\
 RA + RB + RC &= 628.09 \text{ N} \\
 RB + 30.062 + 817.28 &= 628.09
 \end{aligned}$$

So, $RB = -219.252 \text{ N}$

Maximum bending moment acting on shaft by considering both horizontal and vertical loads is at point C,

$$\begin{aligned}
 MB &= \sqrt{Mb^2 + Mb^2} \\
 MB &= \sqrt{213744.95^2 + 56528.1^2} \\
 &= 221093.486 \text{ N-mm}
 \end{aligned}$$

F. Design of Shaft-

The design of shaft is based upon equivalent torsional moment method.

The values of shock and fatigue factors are given on the basis of application of load.

For gradually applied load,

$$Kb = \text{shock factor} = 1.5$$

$$Kt = \text{fatigue factor} = 1 \quad \dots [\text{Table 9.2, Page No. 334, V. B. Bhandari}]$$

Material selection –

The material selected for shaft is Mild steel with grade 30C8

Properties: –

$$\text{Ultimate tensile strength} = Sut = 500 \text{ N/mm}^2$$

$$\text{Yield strength} = Syt = 400 \text{ N/mm}^2$$

... .. [Table 2.2, Page No. 31, V. B. Bhandari]

The permissible shear stress can be given as,

$$\tau_{max} = (0.5Sut) / F.S.$$

$$\text{Factor of safety} = 2$$

$$\tau_{max} = (0.5 \times 500) / 2$$

$$\tau_{max} = 125 \text{ N/mm}^2$$

The shaft diameter is calculated by using equivalent torsional moment method,

The permissible shear stress can be given as,

$$\tau_{max} = 16 \times \sqrt{(Kb * Mb)^2 + (Kt * Mt)^2} / \pi \times d^3$$

Putting all known values in above equation,

$$125 = (16 \times \sqrt{(1.5 * 221093.48)^2 + (19.85 * 10 * 10 * 10)^2}) / \pi \times d^3$$

$$d = 23.84 \text{ mm} \approx 35 \text{ mm}$$

$$d = 35 \text{ mm} \quad \text{--- diameter of shaft}$$

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Bearing selection for Main shaft –

The support reaction are given as follow,

$$RA = \sqrt{Rav^2 + Rah^2}$$

$$RA = \sqrt{282.52^2 + 30.062^2}$$

$$RA = 284.115 \text{ N}$$

$$RB = \sqrt{Rbv^2 + Rbh^2}$$

$$= \sqrt{(-495.96)^2 + (-219.252)^2}$$

$$RB = 542.261 \text{ N}$$

$$RC = \sqrt{Rcv^2 + Rch^2}$$

$$= \sqrt{2040.172^2 + (817.28)^2}$$

$$RC = 2197.78 \text{ N}$$

Radial forces on bearings are,

$$Fr1 = RA = 284.115 \text{ N}$$

$$Fr2 = RB = 542.261 \text{ N}$$

$$Fr3 = RC = 2197.78 \text{ N}$$

There is no any axial thrust is acting on bearings,

$$Fa1 = Fa2 = Fa3 = 0$$

Dynamic load calculations

$$P1 = Fr1 = 284.115 \text{ N}$$

$$P2 = Fr2 = 542.261 \text{ N}$$

$$P3 = Fr3 = 2197.78 \text{ N}$$

Calculation of dynamic load capacity: –

The life of bearings selected is on basis of machine used for eight hours

$$L10h = 12000 \text{ hr} \quad \dots [\text{Table 15.2, Page No. 573, V. B. Bhandari}]$$

$$\text{Speed of shaft} = 720 \text{ rpm}$$

Life of bearing in million f revolution is given as,

$$L10 = (60 \times n \times L10h) / 10^6$$

$$= (60 \times 720 \times 12000) / 10^6$$

$$L10 = 518.4 \text{ million revolution.}$$

Load factor = 2 – – – – – selected for V – belt drive

... [Table 15.3, Page No. 573, V. B. Bhandari]

Dynamic load capacity for bearing 1 –

$$C1 = P1 \times (L10)^{1/3} \times (\text{load factor})$$

$$= 284.115 \times 518.4^{1/3} \times 2$$

$$C1 = 4564.70 \text{ N}$$

Dynamic load capacity for bearing 2 –

$$C2 = P2 \times (L10)^{1/3} \times (\text{load factor})$$

$$= 542.261 \times 518.4^{1/3} \times 2$$

$$C2 = 8712.16 \text{ N}$$

Dynamic load capacity for bearing 3 –

$$C3 = P3 \times (L10)^{1/3} \times (\text{load factor})$$

$$= 2197.78 \times 518.4^{1/3} \times 2$$

$$C3 = 35302.35 \text{ N}$$

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Table 2. Bearing Specifications

| Parameter | Bearing 1 (16007) | Bearing 2 (6404) | Bearing 3 (6407) |
|--------------------------------|-------------------|------------------|------------------|
| Inner diameter | 35mm | 40mm | 35mm |
| Outer diameter | 62mm | 68mm | 100mm |
| Width | 9mm | 9mm | 25mm |
| Dynamic load carrying capacity | 12400N | 13300N | 55300N |
| Static load carrying capacity | 6950N | 7800N | 31000N |

VII. DESIGNED HARVESTER

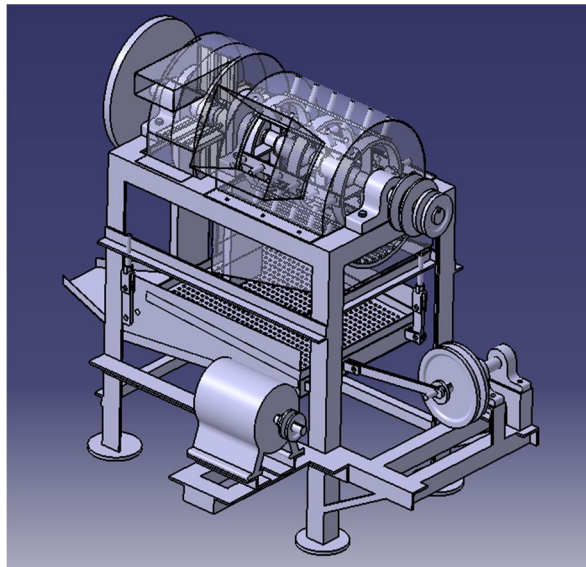


Figure 13. CATIA Model



Figure 14. Actual Model

VIII. CONCLUSION

From all the experiments and trials, it is observed that the Harvester designed by us sufficed the actual needs satisfactorily with the following conclusions drawn out:

- A. The purpose of reducing down the cost and capacity is achieved as far as the existing machines are concerned.
- B. As per the Indian scenario, the per capita land owned by farmers is low making it obvious that they can't afford the existing machines. We achieved to maintain the cost around Rs.15000-18000.
- C. The feed capacity is maintained around 50-100 Kg/hr.
- D. Low maintenance and easy to operate due to simpler design.
- E. All the parts are easily available in the market.
- F. The harvester, being portable, can be carried to the farm with fewer efforts.

IX. ACKNOWLEDGEMENT

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