'DESIGN & FABRICATION OF PERFORMANCE EVALUATION GARDEN TILLER'

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ABSTRACT: *In order to meet the food requirements of the growing population and rapid industrialization, modernization of agriculture is inescapable. Mechanization enables the conservation of inputs through precision in metering ensuring better distribution, reducing quantity needed for better response and prevention of losses or wastage of inputs applied. Mechanization reduces unit cost of production through higher productivity and input conservation.*

The problem arises from the fact that workers available to work in farmlands are insufficient. And as a ▪result most of the fields are left uncultivated. The middle class fanners cannot bear the high cost of power tillers available in the market. Most middle class fanners have small land holdings and to buy a tiller for serving their need is uneconomical. Cost of power tillers available in the market are in terms of lakhs. Also that the operation of the available power tillers is very complex, they cannot be operated without the help of skilled men. Thus it adds to the cost for the need of hiring a skilled labor.

Owing to the problems mentioned above our motive was to develop a tiller based on diesel. Diesel engines have more dead weight compared to petrol engines of same power. Thus the garden tiller basically is a tiller running on diesel which could be used effectively in cultivation of tapioca, ginger, pulses etc. And that the cost of the tiller should be affordable to the middle class fanners. Thus our objective was to design and fabricate a tiller based on diesel that is easy to operate and should come with an affordable price so that middle class fanners would not find any problem buying them.

INTRODUCTION

Status of agricultural mechanization in India

Most of the developing countries of Asia have the problem of high population and low level of land productivity as compared to the developed nations. One of the main reasons for low productivity is insufficient power availability on the farms and low level of farm mechanization. This is especially true for India. It is now realized the world over that in order to meet the food requirements of the growing population and rapid industrialization, modernization of agriculture is inescapable. It is said that on many farms, production suffers because of improper seedbed preparation and delayed sowing, harvesting and threshing. Mechanization enables the conservation of inputs through precision in metering ensuring better distribution, reducing quantity needed for better response and prevention of losses or wastage of inputs applied. Mechanization reduces unit cost of production through higher productivity and input conservation. Agricultural implement and machinery program of the Government has been one of selective mechanization with a view to optimize the use of human, animal and other sources of power. In order to meet the requirements, steps were taken to increase availability of implements ,irrigation pumps ,tractors ,power tillers, combine harvesters and other power operated machines and also to increase the production and availability of improved animal drawn implements. Special emphasis was laid on the later as more than 70% of the farmers fall in small and, marginal category. Liberal credit has helped in acquiring new machines. For example, Faridkot district in Punjab recorded 137 tractors per thousand hectares in 1986-87 where as many of the districts in the country may not have a single tractor even today. The availability of farm power through mechanical means was estimated as 2.71 hp per hectare in Punjab in 1986-87 where as many states may not have one tenth of it.

It is generally said that mechanization of small farms is difficult .But Japan having average land holding even smaller than ours, with proper mechanization has led agriculture to great heights. In order to minimize the drudgery of small fanners ,to increase efficiency and save fanner's time for taking up additional /supplementary generating activities , the use of modern time saving machines/implements of appropriate size needed to be suitably promoted.

DESIGN OF GARDEN TILLER

Selection of engine-

As per standards the torque required for the output shaft was taken as 75Nm at a speed of about 300 rpm. We studied the details of the engine available. We selected the diesel engine that could provide a torque of around lOONm. The engine selected was the GARUDA engine which had a rated power of 6.4hp @ 3600 rpm. We found that as the speed of the engine was increased above 2500 rpm large amount of emissions was obtained .Hence we decided to restrict the speed of the engine to be less than 2000 rpm and set the maximum acceleration as 2000rpm. This was achieved by controlling the fuel fo be supplied by acceleration. We know that as the speed of the engine is decreased the brake power output also decreases. We assume brake power output is proportional to the speed of the engine. Since we have set the maximum engine speed as 2000 rpm, the brake power output will be 2000*6.4/3600 = 3.56hp.But in actual practice the decrease in power will be a parabolic variation. Io order to take account of this factor we assume a service factor 1.2. Hence the output power of the engine will be = 3.56*1.2= 4.3hp.Thus we have selected a four stroke single cylinder diesel engine of GARUDA as the power source.

4 Stroke compression ignition engines-

In 4-stroke cycle engine the cycle of operation of engine, the cycle of operation of engine is completed in 4-strokes of piston or two revolutions of the crankshaft. In CI engine high compression ratio is used. During suction stroke air is alone inducted. Due to high compression ratio, the temperature at the end of the compression stroke is sufficient to ignite the fuel which is injected in to the combustion chamber. In the CI engine a high pressure fuel pump and an injector is provided to inject fuel in to combustion chamber. The ideal sequence of operation for the 4-stroke CI engine is as follows;

Suction stroke: Suction stroke 0-1 starts when the piston is at top dead centre and about to move downwards. The inlet valve is open at the time and the exhaust valve is closed. Due to the suction created by the motion of the piston towards bottom centre, the air is drawn in to the cylinder. At the end of suction stroke the inlet valve closes.

Compression stroke; The fresh air taken in to the cylinder during suction stroke is compressed by the return stroke of the piston 1-2.During this strokes both inlet and exhaust valves remain closed.

Expansion stroke OR Power stroke; Fuel is injected in the beginning of expansion stroke 2-3 The rate of injection is such that the combustion maintains the pressure constant. After the injection of fuel is over (i.e. after the fuel is cut-off) the products of combustion expand. Both valves remain closed during expansion stroke.

Exhaust stroke; at the end of the expansion stroke the exhaust valve opens, the inlet valve remaining closed, and the piston is moving from bottom dead centre sweeps out the burnt gases from the cylinder, stroke 4-0.

BELT DRIVES-

The belts are used to transmit power from one shaft to another by means of pulleys which rotate at the same speed or different speeds. The amount of power transmitted depends on the following factors:

1. The velocity of the belt.
2. The tension under which the belt is placed on the pulleys.
3. The arc of contact between the belt is used. It may be noted that

a. The shafts should be properly in line to ensure uniform tension across the belt section.

b. The pulleys should not be so far apart as to cause the belt to weigh heavily on the shafts, thus increasing the friction load on the bearings.

c. The pulleys should not **be** too close together, in order that the arc of contact on the smaller pulley may be as large as possible.

d. A long belt tends to swing from side to side, causing the belt to run out of the pulleys, which in turn develops crooked spots in the belt.

e. The tight side of the belt should be at the bottom, so that whatever sag is present on the loose side will increase the arc of contact at the pulleys.

f. In order to obtain good results with flat belts, the maximum distance between the shafts should not exceed 10 meters and the minimum should not be less than 3.5 times the diameter of the larger pulley.

2.2.1 Selection of a Belt Drive

Following are the various important factors upon which the selection of a belt drive depends:

1. Speed of the driving and driven shaft.
2. Speed reduction ratio.
3. Power to be transmitted.
4. Centre distance between the shafts.
5. Positive drive requirements.
6. Shafts layout.
7. Space available.
8. Service conditions.

Types of belt drives:

The belt drives are usually classified into following three groups:

1. Light drives. These are used to transmit small powers at belt speeds up to about lOm/s as in agricultural machines and small machine tools.
2. Medium drives. These are used to transmit medium powers at belt speeds over
3. 1 0m/s but up to 22m/s, as in machine tools

Heavy drives. These are used to transmit large powers at belt speeds above 22m/s as in compressors and generators.

Types of belts:

Though there are many types of belts used these days, yet the following are important:

1. Flat belts. It is mostly used in factories and workshops, where a moderate amount of power is to be transmitted, from one pulley to another when the two pulleys are not more than 8 meters apart.
2. V-belts. It is mostly used in factories and workshops, where a great amount of power is transmitted, from one pulley to another, when the two pulleys are very near to each other.
3. Circular belt or rope. It is mostly used in factories and workshops, where a great amount of power is transmitted, from one pulley to another, when the two pulleys are more than 8 meters apart.
4. If a huge amount of power is transmitted, then a single belt may not be sufficient, in such case, wide pulleys (for V-belts and circular belts) with a number of grooves are used. Then a belt in each groove is provided to transmit the required amount of power form one pulley to another.

Materials used for belt-

1. Leather belts. The most important material for flat belt is leather. The best leather belts are made from 1.2 meters to 1.5 meters long strips cut from either side of the backbone of the top grade steer hides. The hair side of the leather is smoother and harder than the flesh side, but the flesh side is stronger. The fibre on the hair side are perpendicular to the surface and give more intimate contact between belt and pulley and places the greatest tensile strength of the belt section on the outside, where the tension is maximum as the belt passes over the pulley.

2. Cotton or fabric belts. Most of the fabric belts are made by folding convass or cotton duct to three or more layers (depending upon the thickness desired) and stitching together. These belts are woven also into a strip of the desired width and thickness. They are impregnated with some filler like linseed oil in order to make the belt water proof and to prevent injury to the fibres. The cotton belts are cheaper and suitable in warm climates, in damp atmospheres and in exposed positions. Since the cotton belts require little attention, therefore these belts are mostly used in farm machinery, belt conveyors etc...

Rubber belt. Rubber belts are made of layers of fabric impregnated with rubber composition and have a thin layer of rubber on the faces. These belts are very flexible but are quickly destroyed if allowed to come into contact with heat, oil or grease. One of the principal advantages of these belts is that they may be easily made endless. These belts are found suitable for saw mills, paper mills where they are exposed to moisture.

Balata belts. These belts are similar to rubber belts except that balata gum is used in place of rubber. These belts are acid proof and water proof and it is not affected by animal oils or alkalis. The balata belts should not be at temperatures above 40° Celsius because at this temperature the balata begins to soften and becomes sticky. The strength of balata belts is 25% higher than rubber belts.

Advantages and disadvantages of V-belt drive over flat belt drive Advantages

1. The V-belt drive gives compactness due to small distance between centres of pulleys.
2. The drive is positive, because the slip between the belt and pulley groove is negligible.
3. Since the V-belts are made endless and there is no joint trouble, therefore the drive is smooth.
4. It provides longer life, 3-5 years.
5. It can be easily installed or removed.
6. The operation of belt and pulley is quiet.
7. The belts have the ability to cushion the shock when machines are started.
8. The high velocity ratio (maximum 10) may be obtained.
9. The wedging action of the belt in the groove gives high value of limiting ratio of tensions. Therefore the power transmitte4d by V- belts is more than flat belts for the same coefficient of friction, arc of contact and allowable tension in the belt.
10. The V-belt may be operated in either direction, with tight side of belt at the top or bottom. The centre line may be horizontal, vertical or inclined.

Disadvantages

1. The V-belt drive cannot be used with large centre distances, because of larger weight per unit length.
2. The V-belts are not as durable as flat belts.
3. The construction of pulleys for V -belts is more complicated than pulleys of flat belts.
4. Since the V-belts are subjected to certain amount of creep, therefore these are not suitable for constant speed applications such as synchronous machines and timing devices.
5. The belt life is greatly influenced with temperature changes, improper tension and mismatching of belt length.
6. The centrifugal tension prevents the use of V-belts at speeds below 5m/s and above 50m/s.

V-belts-

The V-belts are made of fabric and cords moulded in rubber and covered with fabric and rubber a.) These belts are moulded to a trapezoidal shape and are made endless. These are particularly suitable for short drives. The included angle for the V-belts is usually from 30° to 40°. The power is transmitted by the wedging action between the belt and the V- groove in pulley and sheave. A clearance must be provided at the bottom of the groove, b.) In order to prevent touching of the bottom as it becomes narrower from wear. The V-belt drive maybe inclined at any angle with tight side either at top or bottom. In order to increase the power output, several V-belts maybe operated side by side. It may be noted that in multiple V-belt drive all the belts should stretch at the same rate so that the load is equally divided between them. When one of the set of belts breaks, the entire set should be replaced at the same time. If only one belt is replaced the new unworn and unscratched belt will be more tightly stretched and will move with different velocity.

Types of V-belts and pulleys-

According to Indian standards (IS: 2494 - 1974), the V -belts are made in five types ie. A, B, C, D and E. The pulleys for V-belts maybe made of cast iron or pressed steel in order to reduce weight.

Table 2.1 Dimensions of standard V-belts according to IS 2494-1974

Type of belt	Power ranges in kW	Min. pitch dia. of pulley (D) mm	Top width (b) mm	Thickness (t) mm	Weight/m length in N
A	0.7-3.5	75	13	8	1.06
B	2-15	125	17	11	1.89
C	7.5-75	200	22	14	3.43
D	20-150	355	32	19	5.96
E	30-350	500	38	23	-

DESIGN OF THE BELT DRIVE-

Nl speed of engine shaft

N2 speed of intermediate shaft

D diameter of the pulley

9 lap angle

a area of the belt in mm^2

m mass of the belt in kg/m

v speed of the belt in m/s

Tc centrifugal tension in N

Velocity ratio for the drive-

= N1/N2 = D2/D1

= 2000 / 500

= 4

Assuming the diameter of the pulleys to be of 25cm and 6.25cm according to the standard dimensions.

The distance between the two pulleys x = 40cm

For an open belt drive,

Sin a = (0.25-0.0625)0.4

= **0.04685**

a = **27.953**

Angle of lap on the smaller pulley=

0 = 180-2a

= 124.093 degrees

= **2.165 radians.**

Area of the belt =130 mm^2

Mass of the belt = 0.052kg/m

Angular velocity of the shaft=

co = (2**TI***2000)/60

= **209.439 rad/sec.**

Speed of the belt=

=r*co

= 0.03125*209.439

= 6.544m/sec.

Centrifugal tension=

$T_c = m*v^2$

$= 0.052*6.54^2$

= 2.226N

Maximum tension in the belt=

T =2.5*130

= 325N

Tension in the tight side of the belt=

$T, = T - T_c$

= 325-2.226

= 322.774

We know that,

2.3 log($T!/T_2$) = uB cosec ᵽ

log($T,/T_2$) = (0.25*2.165*cosec 17)/2.3

$T,/T_2$ **= 6,380**

Tension in the slack side of the belt T_2 = 322.774/6.380

= 50.591N

Power transmitted = (Tl -T2)* v

= (322.774- 50.591)* 6.544

= 1.781

No. of belts = 321/ 1.781

=1.802 = 2

CHAIN DRIVES-

A chain drive consists of an endless chain wrapped around two sprockets. The chain consists of a number of links connected by pin joints, while the sprockets are toothed wheels with a special profile for teeth. The chain drive is intermediate between the belt and gear drives. It has some features of gear drive and some features of belt drive. The advantages and disadvantages of the chain drive compared to the belt and gear drives are as follows:

1) Chain drives can be used for long as well as short distances. They are particularly suitable for medium centre distance, where gear drives will require additional idler gears.

2) A number of shafts can be driven in the same or opposite direction by means of the chain drive from a single driving sprocket.

3) Chain drives are used for shafts which are parallel, whereas some gears like bevel and worm gears can be used for non parallel shafts.

4) Chain drives are more compact than belt and gear drives.

5) A chain drive does not slip and to that extend, it is a positive drive compared to belt drive. However it is unsuitable where precise motion is required due to polygonal effect and wear in the joints. They also require adjustment for slack, such as a tensioning device.

6) Compared to belt drives chain drives require precise alignment of shaft. However, the center distance is not as critical as in case of gear drives.

7) The efficiency of chain drives is high at times as high as 98%.

8) Compared to belt drives chain drives requires proper maintenance, particularly lubrication and slack adjustment. However, chains can be easily replaced.

Roller chains-

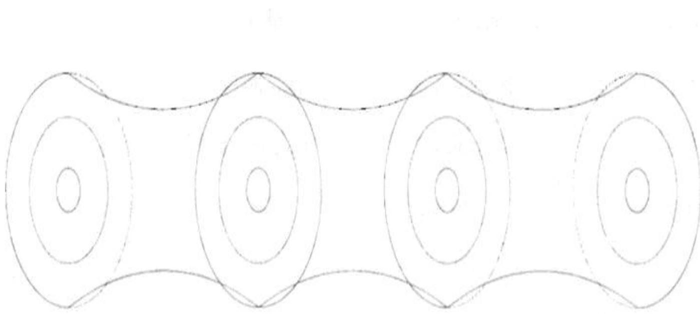


Figure 2.2 construction of roller chain

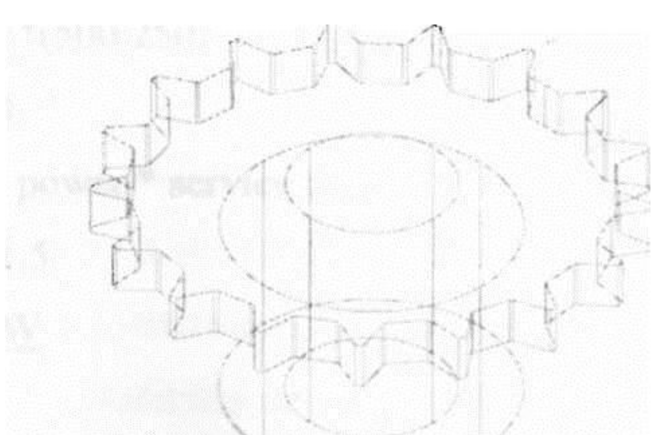
There are five parts for a roller chain namely pin, bushing, roller and inner and outer link plates. The pin is press fitted to two outer link plates, while bushing is press fitted to the two inner link plates. The bush and the pin form a swivel joint and the outer link is free to swivel with respect to inner link. The rollers are freely fitted on bushes and, during engagement, turn with the teeth of the sprocket wheels. This results in rolling friction instead of sliding friction between the roller and the sprocket teeth and reduces wear. The pins bushes and rollers are made of alloy steels.

The pitch of the chain is defined as the linear distance between axes of adjacent rollers. Roller chains are standardized and are manufactured on the basis of pitch. These chains are available in single -row or multi- row constructions such as simple duplex or triplex strands.

Sprocket wheels-

A wheel that drives or is driven by a chain is usually

referred to as a sprocket. Small sprockets up to 100mm in diameter are usually made of a disc or a solid disc with a hub on one side. They are machined from low carbon steel bars. Large sprockets with diameter more than 100mm diameter are either welded to steel hubs or bolted to cast iron hubs. In general for most applications the sprockets are made of low carbon or medium carbon steels and, on rare occasions, stainless steel is also used for making sprockets. When the chain velocity is less than 180m/min, the teeth of the sprocket wheels are heat-treated to obtain a hardness of 180 B.H.N. For high speed applications the hardness recommended is 300 to 500 B.H.N. The teeth are hardened either by carburizing in case of low carbon steels or by quenching and tempering in case of high carbon steels. The sprocket that we used is shown below.



DESIGN OF CHAIN DRIIVE FOR THE BLADE SHAFT-

P power transmitted in watts

Z| no. of teeth on the smaller sprocket or pinion

Z_2 no. of teeth in the larger sprocket or gear

p pitch of the chain in mm

WB breaking load of the chain in N

PCD pitch circle diameter in mm

v pitch line velocity in m/s

Power transmitted = 3210watts Velocity Ratio N1/N2 = 500/250 **= 2**

For roller chain, the number of teeth on the smaller sprocket or pinion (Zi) for a velocity ratio of 2 is 27.

Therefore number of teeth on the larger sprocket or gear, Z_2 =Zi *(N1/N2) = 17*(500/250) **= 54**

Design Power = rated power * service factor

=3.2 *1.5 = 4.8 KW

For the range 3-15 kW

Chain no. 10 B should be used.

For chain 10B

Pitch = 15.875 mm

Breaking load = 22.2Kn

PCD for pinion di = p*cosec (180/Zi)

= 15.875*cosec (180/27)

= 136.74mm

PCD for sprocket = p*cosec(1**8O/Z2)**

= 15.875* cosec(180/54) **= 273.02mm**

Pitch line velocity on smaller sprocket,vl **=**

(7r*d!.Ni)/60 **=(7t***0.136*500)/60 **=3.56m**/sec

Load on the chain, W

= 3.210/3.56 **=901.68N**

SHAFT-

The term transmission shaft is usually referred to a rotating machine element, circular in cross section, which supports transmission elements like gears, pulleys and sprockets and transmits power. Such shafts are subjected to bending tensile, bending or torsional shear stresses, or to a combination of these. The design of transmission consists of determining the correct shaft diameter from strength and rigidity considerations. The materials usually used are mild steel and alloy steels, such as nickel, nickel-chromium and molybdenum steels.

The shaft used here is subjected to a combination of both bending and torsional moments. The shaft material used is standard steel which is tensile in nature and hence the principle shear stress theory of failure is used to determine the shaft diameter.For a shaft subjected to a bending moment Mb, the bending stress at any fiber is given by

$o_b = (M_b * y)$ *II*

where,

a_b — Bending stress at a distance y from the neutral axis in N/mm^2.

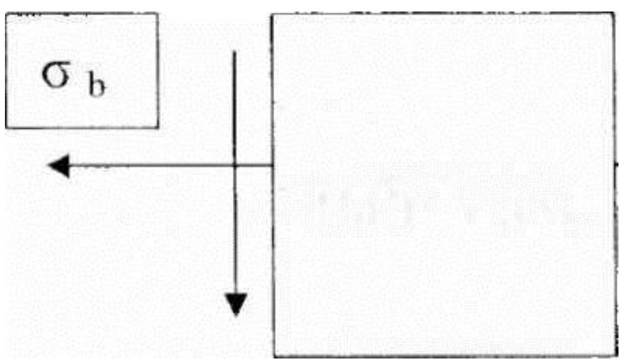
Mb — Applied bending moment in Nmm.

I — moment of inertia of the cross section about the neutral axis in mm^4

For circular cross section=

the moment of inertia, I = fld^4/64.

The maximum shear stress on the shaft can be determined by constructing the Mohr's circle-



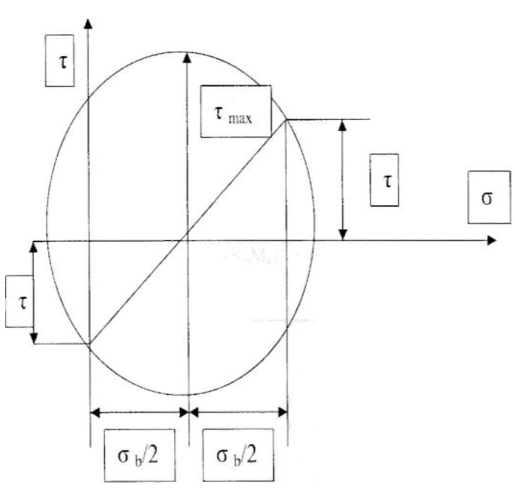


Figure 2.5 Mohr's circle

From the Mohr's circle diagram the maximum shear stress

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Tmax = √ {(0$_b$/2)^2+X^2}.

Substituting the value of maximum shear and bending stress from above equations number 1 and 2, we get,

Tmax = 16/(IId^3)* √ {(M$_b$)^2+(M,)^2}

From the above equation the diameter of the solid shaft subjected to both bending and torsion,

d = [16/ (n*x$_{imx}$) {(M$_b$)^2 +(M$_t$)^2}$^{(a5)}$]$^{(1/3)}$ -3

Ti tension on the tight side of the pulley in N

T$_2$ tension on the slack side of the pulley in N

|i coefficient of friction

9 lap angle

N rpm

P power transmitted in watts

T torque to be transmitted in Nm

F$_T$ chain load

F$_{TH}$ horizontal component of the chain load

F$_{TV}$ vertical component of the chain load

M design bending moment in Nm

T$_e$ Torque effective

f$_s$ shear stress in N/mm^2

We know that,

T,/T$_2$ = $e^{(\mu*\theta)}$

$e^{0.25*\pi}$

= 2.193 **T$_2$**

Power transmitted

P = (2**TI*N*T**) / 60

= (3210 * 60)/ (T 2?r* 2000)

= 15.326Nm

(Ti-T$_2$) * 0.125 = 15.326

From 1

(2.193 T$_2$-T$_2$)*0.125 = 15.326

1.193 T$_2$*0.125 =15.326 T$_2$

= **102.772 N** Ti = **122.606N**

Chain load =

F$_T$ =3210/3.56

= 901.68N

F$_T$ cos 60= 901.68*cos 60

=450.93

F$_T$ sin 60= **781.03N**

Bending moments-

Horizontal

M$_A$ = 22.537Nm

M$_c$ = 33.824Nm

Vertical

M$_A$ = 6Nm

M$_c$ = 75.10Nm

Resultant bending moments

Res **M$_A$ = (22.**53**^{2}+6^2)$^{1/2}$**

= 23.315Nm

Res $\mathbf{M}_c = (33.824^2+75.103^2)^{1/2}$

= 82.368Nm

Torque effective, $T_e = (\mathbf{M}^2+T2)^{1/2}$

=83.788Nm

K_t*T_e **=(7t/16)**$*f_s*d^3$

$1.5*83.788 = (\mathbf{7i}/16)*44*10^6*d^3$ d

=**24.41mm**

Nearest standard size available = **25mm**

Design of the blade shaft-

R_a reaction at the left side bearing, in N

R_b reaction at the right side bearing, in N

P power transmitted in Watts

N speed of the shaft in revolutions per second

T torque to be transmitted in Nm

d diameter of the shaft, in mm

$\mathbf{F_T}$ chain load in N

$\mathbf{F_{TV}}$ vertical component of chain load

FJH horizontal component of chain load

T_e torque effective in Nm

f_s shear stress in N/mm^2

M bending moment in Nm

d diameter of shaft in mm

Chain load acting on the shaft $\mathbf{F_T}$ = 902 N

FTV = $F_T*\sin 30$

=902*sin30==**45 IN**

FTH =$\mathbf{F_T}$ ***cos**30

=902***cos**30= **781.15**

Power Torque

= (2TT*N*T)/60

=(2 7t*250*T)/60

=122.23Nm

Bending moments-

Vertical

$\mathbf{M_A}$ **-106.2**

$\mathbf{M_B}$ **=106.2**

Horizontal

$\mathbf{M_A}$ **=156.3**

MR =156.3

Resultant bending moment,$M=(M_A^2+M_B^2)^{1/2}$

$=(106.2^2+156.3^2)^{1/2}$ =**188.96Nm**

Torque effective, $T_e =(M^2+T^2)^{1/2-}$

$=(188.96^2+123^2)^{1/2}$ =**225.465Nm**

K_t*T_e **= (Tt/16)**$*f_s*d^3$

$1.875*225.465 =(\boldsymbol{n}/16)*42*10^6*d^3$

d=37.14mm

Nearest standard size available = **40mm**

BEARINGS-

A bearing is a machine part whose function is to support a moving element and to guide or confine its motion, while preventing the motion in the direction of applied load. They take up the radial and axial loads imposed on the shaft or axle they carry, and transmit these to the casing or machine frame.

Classification-

Bearing can be classified in the following many ways:

1) Depending up on the direction of load to be supported:

a) Radial bearing

b) Thrust bearing.

2) Depending up on the nature of contact between the working surfaces:

a) Sliding contact bearing

b) Rolling contact bearing.

3) Depending up on the type of loading:

a) Bearing with steady load

b) Bearing with variable or fluctuating load.

Bearing selection based on mechanical requirements-

Table 2.2 Bearing selection based on mechanical requirements

SLNO	CHARACTERISTICS	SLIDING	ROLLING
1	LOAD: One way Both ways Unbalance Shock Starting	GOOD GOOD GOOD FAIR POOR	EXCELLENT EXCELLENT EXCELLENT EXCELLENT EXCELLENT
2	SPEED	Usual value is up to 75 m/s	Usual value is up to 50 m/s
3	STARTING FRICTION	POOR	GOOD
4	TYPE OF FAILURE	Often permits limited emergency operation after failure	Limited operation may continue after fatigue failure but not after lubricant failure
5	DAMPING OF VIBRATIONS	GOOD	POOR
6	NOISE	Quiet	May be noisy depending up on the quality of bearing and resonance of mounting
7	TYPE OF LUBRICANT	Oil or other fluids, grease ,dry lubricants air or gas	Oil or grease
8	QUANTITY OF LUBRICANT	Large ,except in low speed boundary lubrication	Very small except where large amount of heat must be removed

Sliding bearings or plain bearings

In sliding bearings the primary motion between the bearing and the moving element is sliding one. The sliding bearings can be classified in to two groups depending up on the nature of motion of the moving element.

They are 1) linear bearings: - In these bearings sliding action is guided in a straight line.

2) Sliding bearing with motion of rotation: - In these bearing, the sliding motion between the bearing and the moving element, is he motion of rotation.

Sliding bearings are also classified according to the type of friction present between the bearing and the moving element:

Dry friction bearing: No lubricant is supplied in between the rubbing surface and there is metal contact. The coefficient of friction may range fromO.l to 0.25.

1) Boundary friction bearings: - In these bearings, the lubricant is supplied to the rubbing surfaces in scarce quantities. The lubricant is neither constant nor abundant .Such bearings are suitable for low load and low speed conditions since at increased loads, the thin film of lubricant will break and the bearing surfaces approach each other, resulting in metal to metal contact and wear the surfaces.

2) Semi fluid friction: - This is the transition between the boundary and fluid type of friction. Lubricant is supplied constantly but not abundantly and so the lubricant film is very thin and when it breaks, there is metal to metal contact. The coefficient of friction range from 0.005 to 0.10.

4) Bearing with fluid friction: - In these bearings there is always a thick film of lubricant between the working surfaces and there will never be **a** metal to metal contact.

Rolling contact bearings

In these bearings the contact between the bearing elements is rolling instead of sliding as in plain bearings. Since rolling friction is very less as compared to the sliding friction, such bearings are also known as "antifriction bearings".

Types of rolling contact bearings

Following are the two types of rolling contact bearings

1) Ball bearings
2) Roller bearings

The ball and roller bearings consists of an inner race which is mounted on the shaft or journal and an outer race which is carried by the housing or casing. In between the inner and outer race, there are balls or rollers. A number of balls or rollers are used and these are held at proper distances by retainers so that they do not touch each other. The retainers are thin strips and are usually in two parts which are assembled after the balls have been properly spaced. The ball bearings are used for light loads and the roller bearings are used for heavier loads.

The roller bearings depending up on the load to be carried are classified as

1) Radial bearings
2) Thrust bearings.

Types of radial ball bearings

1) Single row deep groove bearings: A single row deep groove bearings are used due to their high load carrying capacity and suitability for high running speeds. The load carrying capacity of a ball bearing is related to the size and number of the balls. This bearing is usually made with deep groove. This is most widely used type of ball bearing.

2) Filling notch bearing: These bearings have notches in the inner and outer races which permits more balls to be inserted than deep groove ball bearing.

3) Angular contact bearing: These bearings have one side of the outer race cut away to permit the insertion of more balls than in deep groove bearing but without having a notch cut in to both races.

4) Double row bearing: These bearings may be made with radial or angular contact between the balls and races. The double row bearing is narrower than two single row bearings.

5) Self aligning bearings: These bearings permit shaft deflection with in 2-3 degrees. Following are the two types of self aligning bearings

DESIGN OF BEARINGS-

L bearing life in millions of revolutions

Co static load capacity in **N**

C dynamic load capacity in **N**

N rotation of the shaft in rpm

L_H bearing life in hours

X radial thrust factor

V race rotation factor

The required life of bearing in millions of revolution is given by **L= ((60*n*L„) /10^a6)**

Where,

n= 500 rpm

From table 24.40, typical values of bearing life for various applications. **L_H=600hrs,** for light vehicle Therefore **L= ((60*500*600)/10^a6)**

=18 millions of revolutions

The radial load acting on the bearing,(Rb**),Fr =865N.**

Since the shaft is horizontal the axial load acting on the bearing is assumed to be zero i.e., **Fa=0 N.**

Index **F_L** of dynamic stressing varies from 1.4-1.9.

Here we take **F_L =1.4**

Therefore radial load, **F_r=865*1.4**

=1211N

From table 24.60, deep groove ball bearing -diameter series-2, **(SKF)**

Assume the bearing number is 6205.

Then, d**=25mm.**

D=52mm. B=15mm. C_0**=6965 N.**

C=10690N

e=Fa/Co=**=0/6965**

Therefore **(F,,.Fa**/C_0)= **13.8*0/10690=0**

Since there is no axial load is acting on the bearings, we take **X=l**

The equivalent dynamic load is given by

P=X*V*Fr +Y*Fa

Where

X=l

V=l, for all types of bearing if the inner race is rotating

Fa=0

Substituting the values in the above equation, we get

=1211 N

The required life of bearing in millions of revolution is given by **L= ((60*n*L,,) /10**^{a6}**)**

n=250 rpm.

From table 24.40, typical values of bearing life for various applications.

Ln=600hrs, for light vehicle

Therefore, life of bearings in millions of revolutions, **L=((60*250*600)/10**^{A6}**) =9** millions of revolutions

The radial load acting on the bearing, (Rb**),Fr =3500N.**

Since the shaft is horizontal the axial load acting on the bearing is assumed to be i.e., **Fa=0 N.**

Index **FL** of dynamic stressing varies from 1.4-1.**9.** Here we take F_L **=1.4**

Therefore radial load, F_R**= 3500*1.4=4900 N**

From table 24.60, deep groove ball bearing -diameter series-2, **(SKF)**

Assume the bearing number is 6208, since we start from light series.

Then, **d=40mm.**

D=80mm. B=18mm.

C_0**=15495 N**

c=22165N

e= Fa/Co

=0/1549=0

Factor F_0 ,for deep groove ball bearing is 13.8

Therefore **(**F_0**.Fa**/C_0)= **13.8*0/15495=0**

Since , there is no axial load is acting on the bearings ,we take **X=l and Y=0**

The equivalent dynamic load is given by

P=X*V*Fr +Y*Fa

Substituting the values in the above equation, we

get =1*4900

Therefore equivalent dynamic load, **P** =4900 **N**

Dynamic load carrying capacity C=22165 **N**

p=3, for ball bearings

P=4900N **L**=(22165/4900)^{a3} = 93 **millions** of revolutions

Since life of selected bearing is more than required life, the bearing suitable for the purpose.

So we select the bearing 6208

SPECIFICATION OF BLADES FOR ROTAVATOR OF POWER TILLER-

Blade is an important soil engaging component of the rotavator. It wears out earlier than other components causing its replacement very often. For the purpose of deciding whether a particular requirement of this standard is compiled with , the final value ,observed or calculated, expressing the result of a test or analysis ,shall be rounded off in accordance with IS : 2-1960. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

Scope-

This standard specifies material, hardness, dimensions and other requirements for blades used in rotavators operated by power tillers.

Types-

a) Type A- Straight blade.

b) Type B- Hatchet blade.

The blade that we are using is the hatchet blade.

Materials-

The chemical composition of the steels to be used for the manufacture of blades shall be as follows.

a) Carbon steel:

Carbon 0.70 to 0.85 %

Silicon 0.10 to 0.40%

Manganese 0.50 to 1.0 %

Sulphur 0.05 % ,Max

Phosphorous 0.05 % **,Max**

Silicon 1.50 to 2.00%

Manganese 0.50 to 1.00%

Sulphur 0.05 %, Max

Phosphorous 0.05 %, Max

Some of the typical steels that may be used are T 70 Mn 65, T 75, T 80 Mn65 and 55 Si 2 Mn 90.

Hardness-

The blades shall be heat treated, quenched and tempered. The hardness in edge portion shall be 56 ± 3 HRC and shank portion shall be 37 to 45 HRC.

Dimensions and tolerances-

The essential dimensions of Type B blades are given below.

SI no	Description	Dimension	Tolerance
1	A	25.0,26.0	-0.3 -0.8
/ 2	*B*	*10.0*	*±0.5*
3	*C*	*25.0*	*±0.5*
4	D	10.5	+0.3 0.0
5	E	210OR225	±0.5
6	E	240,245	±0.5

All dimensions are in millimeters.

A mechanism for tightening the belts over the pulley is made using a **2B**-type pulley. A handle is made out of a GI pipe and is fixed on to the angle iron frame. The accelerator switch is fixed on to the handle for easy manipulation of the engine power as required. The whole assembly was checked for any loose or faulty weldings. The assembly was painted and put to test.

Conclusion-

The model of the garden tiller was constructed successfully and was found to work as per the requirements. The garden tiller that we developed can be used for many activities including the cultivation of tapioca, pulses ginger, turmeric etc. This can be achieved by using special attachments some of which are available with KAMCO. The cost of the garden tiller that we developed is around Rs.25000, whereas the cost of the power tiller is in terms of lakhs. Thus the equipment that we developed will be accessible to middle class farmers who are in deep crisis due to the unavailability of sufficient labor for working in farmland. The big scale farmers could only bear the costly equipments used in farmlands that have very specialized purpose. Thus this multipurpose equipment would be a boon to the small scale farmers.

Future scope-

Providing of power to the ***wheels*** can greatly improve the mobility of the device. This will ease the effort of the worker. Additional accessories can be incorporated which helps us to use the device for various applications. In order to improve safety we can use flap to prevent mud from striking the worker.

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