DESIGN AND DEVELOPMENT OF SUGAR CANE SPROUT CUTTER MACHINE BY HUMAN POWERED FLYWHEEL MOTOR CONCEPT

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ABSTRACT: ***In India conventional system of sugarcane cultivation, about six to nine tones seed cane is used as planting material. This big mass of planting material poses a great problem in transport, handling and the state of being stored of seed cane and undergoes rapid deterioration. It is well understood that hand operated sugar cane sprout cutter is a very time consuming. A common to solve above problem is made by motor operated cutter machine to cut sprout in minimum time. But again it's a bulky, complicated to required electricity, to cut the whole sugar cane can't remaining into single piece. Hence it is important to study a very new true mechanism known as Design and Development of Sugar Cane Sprout Cutter Machine by Human Powered Flywheel Motor Concept in which there will be very less friction and wear during its operation. We using our machine unspecified part's are two chain drive, flywheel, gear drive and cam-follower arrangement. By increasing input speed of first and second shaft due to the use of chain drive. Again speed of is increasing by use of gear drive. By using cam-follower sugarcane cutter are easily reciprocates and cut the sugarcane sprouts. In hand operating cutter machine time is more require to cut the sprout and in motor operating cutter machine to need of the electricity. But, in our pedal operating cutter machine above both the problem has reduce or eliminate.***

Keywords: Design, Human Power, fabrication, Flywheel energy, Crank Rocker Mechanism

1. INTRODUCTION:

1.1 History:

Agriculture is one of the most significant sectors of the Indian Economy. Horticulture is the only stingy of living for on the point of two thirds of the workers in India. There are number of crops grown by farmers. These include different food crops, in demand crops, oil seeds etc. sugarcane is one of the important in demand crop grown in India. Sugarcane is the main source of sugar in Asia and Europe. Sugarcane is growing primarily in the equatorial and sub-tropical zones of the southern hemisphere. Sugarcane is the raw material of the production of white sugar, jiggery and khandsari. It is also used for chewing and extraction of juice for purpose. The sugarcane cultivation and sugar industry in India plays a vital role towards socio-economic development in the rural areas by mobilizing rural resources and generating higher income and employment opportunities.

At first a simple hand operated cutter was used to make work easier for the farmer. The literature says that the sugarcane was widely used since ancient times and spread all over the world. In the case of small farms, cutting is done by beating the sprout by hand or foot and requires a large amount of hard physical labour. Hand operated sugarcane cutter machine has compact dev ice to cut sugarcane sprout. This machine has to be easily moved from one place to another place. For these machine no need of any power to drive the machine. On this machine one person can easily cut the sugarcane sprouts. The cost of this machine is more affordable in rural area.

Figure 1.1 Piece of sugarcane sprout

A motor operated sugarcane sprout cutting machine has to take less time to cut the sprouts. For this machine no need of skill person to operate the machine. For this machine require large amount of electricity to run machine. But in rural area most of the time electricity problem will create. But efficiency of these machine is very high.

2. METHODOLOGY:

1. First of all, we see that the how perform old devices of sugar cane sprout cutter. Then we realise to require

effort was for cutting sugar cane sprout i.e. indicating the need or purpose for which the machine is to be made.
2. After need of design, we give the suitable title for this model or machine.
3. Selecting the possible mechanism or group of mechanism which will give the desired motion or output
4. After synthesis we were estimate the required power or calculating rated power for run the model.
5. We find out the forces acting on each member of the machine or model and motion transmitted by each part's.
6. Select the best suitable material for each part of the machine.
7. Find the shape and size of each part's of the machine by considering forces acting on the member and permissible stresses for the used material.
8. Draw the detailed delineation of assembly of the component with complete specification for the suggested manufacturing process.
9. As per the drawn delineation drawing, the component is manufactured and massed in the garage and workshop.
10. After manufacturing the machine we were plot the result and check the reliability of our machine.

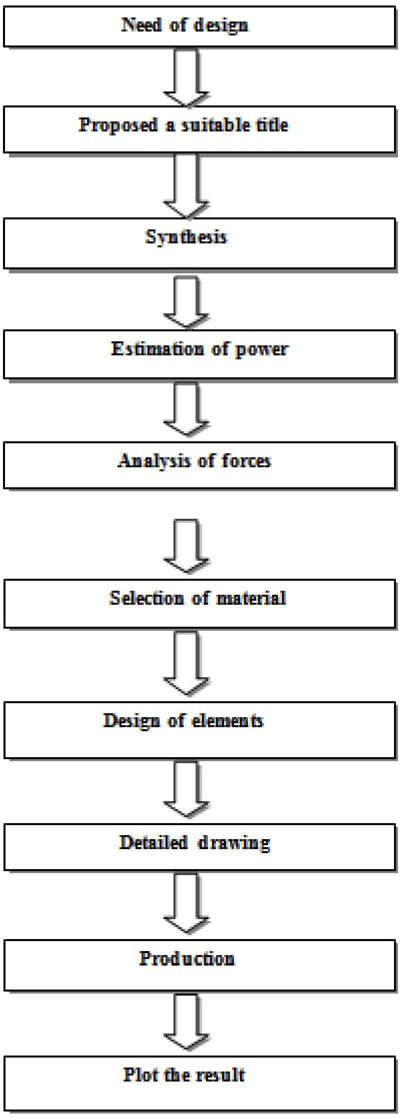

Figure 2 Flowchart of project methodology

3. DESIGN AND CALCULATION:

3.1 Estimation Of Demand Power:

It is planned that before estimating design power there should be some consideration keeping objectives in mind to achieve it. The parameters considered in the design and development of sugar cane sprout cutter machine by human powered flywheel motor concept is as follows.

1. Overall height of the machine to expedite ease of operation by a rural farmer.
2. Comprehensive width and breadth of the machine for purposes of storage space in the rural Farmer's granaries
3. Weight of the equipment for motility
4. Cost effectiveness.

For that we have,

Power Requirement Of Machine:

The power to drive the process unit (P)
Now to determine this,

Power to drive the process unit (P) = F*V

But, F= M*g

Where,

M = Mass of the cutter=6 kg

H = Height of the cutter=0.45m

V = Velocity of cutter $\sqrt{2*g*h} = 1.77 m/s$

So,

$$F = 6 * 9.81 = 58.86 N$$

And, $P = 58.86 * 1.77 = 104.18$ watt = 0.1396hp

Now, taking factor of safety in consideration

We assume factor of safety as 2

So the Total power = 2×0.1396= 0.2793 hp
Thus from above calculations, it is somehow confirmed that the demand power for sugarcane is equal to 0.2793hp. However, at present the torque due to sprockets and frictional torque due to bearings are not considered so far. Hence, it is assumed that to overthrown these torques and overload conditions it may further demand 0.15 hp. Therefore the total power which is to be supplied to the machine will be approximately 0.422 hp.

3.2 Design Of Flywheel

In the upcoming design following terminologies are used
M = mass of flywheel
P = density of material
D_o= outside diameter of flywheel
Rim Diameter = 350mm
Rim width = 50mm

Rim thickness = 50mm

3.2.1Calculate the mass of the flywheel

Material of the flywheel = cast iron (ρ = 7200 kg/m^3)

m = ρ*w*h*π*D=7200*0.050*0.050*π*0.350 =19.7 (Referring data book)

$m \cong 21$ kg

Moment of inertia of the flywheel,

$$I = mk^2$$

Since, k = D/2= 0.350/2= 0.175m……………… (Referring data book)

So, $I = 21 \times 0.175^2 = 0.643125 \text{ kgm}^2$

3.2.2 Calculate the K.E.

$$K.E. = \frac{1}{2} I\omega^2$$

Here, since N= 275, so

$$\omega = \frac{2\pi N}{60} = 28.7979 \text{ rad/sec}$$

So, $K.E. = \frac{1}{2} \times 0.6431 \times 28.79^2 = 266.6675$

Now after calculating the K. E. one can calculate required power from human power. Suppose the time required per second

$$\text{Power} = \frac{K.E.}{t} = \frac{266.67}{1} = 266.67 \text{ watt} = \frac{266.67}{746} = 0.35746 \text{ hp}$$

Let, the rim cross section be A m^2.

$$m = \rho \times \omega \times \pi \times D_m$$

$$A = \frac{m}{\rho \times \pi \times D_m} = \frac{21}{7200 \times \pi \times 0.350} = 2.6525 \times 10^{-3} m^2$$

We Know, $b = 2h$ and $A = 2b \times h$ So,

$$2.6525 \times 10^{-3} = 2h^2$$

So, $h = 0.03641 \text{ mm} = 364 \text{ mm}$

And

$b = 2h = 0.0728365 \text{ m} = 72 \text{ mm}$

Now find out stress in the flywheel,

For that assume $V_s < 1600 \text{ m/min}$

$V_s = \frac{\pi D_o N}{60} = \frac{\pi \times 0.350 \times 275}{60} = 5.039 \text{ m/sec}$……………… (Referring data book)

3.2.3 Calculate Stresses

Centrifugal stress, $\sigma_1 = \rho \times V_s^2 = 7200 \times 5.039^2 = 0.18282 \times 10^6 \text{N/m}^2 = 0.18282 \text{N/mm}^2$ ……………… (Referring data book)

3.2.4 Flywheel construction detail,

a) Hub diameter, $D_h = 2 \times d_s = 2 \times 25 = 50 \text{ mm}$

b) Hub length, $L_h = 2 \times d_s = 2 \times 25 = 50\text{mm}$ …… (Referring data book)

3.3 Design Of Shaft 1

Speed of small sprocket N_1 = 275 rpm

Pitch circle dia. of small sprocket = 65 mm

PCD of gear =261 mm

Wt. of small sprocket = 0.6 kg = 5.886 N

Wt of gear= 0.7 kg = 6.86 N

Wt of flywheel = 21 kg =206.01 N

Φ =20

Power received = 0.42hp = 0.42x746 = 313.32 Watt

F_t = Tangential force

F_r = Radial force

3.3.1 Design Torque

$T_d = \frac{60 \times p \times k}{2 \times \pi \times N} = \frac{60 \times 313.32 \times 2 \times 1000}{2 \times 3.14 \times 275} = 21.75 \times 10\text{^}3 \text{ Nmm}$… ..(Referring data book)

3.3.2 Calculation of force due to chain tension

$F_{t1} = \left(\frac{\text{Torque}}{\text{Pitch radius of sprocket}}\right) = \frac{21.75 \times 10\text{^}3}{32.5}$ =669.23 N

Resolving force F_{t1V} and F_{t1H}

$F_{t1V} = F_{t1} \cos(80) = 669.23 \cos(80) = 116.21 \text{ N}$

$F_{t1H} = F_{t1} \sin(80) = 669.23 \sin(80) = 659.06 \text{ N}$

$F_{r1} = F_{t1} \tan\varphi = 669.23 \tan(20) = 243.57 \text{ N}$

Resolving force F_{r1V} and F_{r1}H

$F_{r1V} = F_{r1} \cos(80) = 243.57 \cos(80) = 42.29 \text{ N}$

$F_{r1H} = F_{r1} \sin(80) = 243.57 \sin(80) = 239.86 \text{ N}$

Forces due to another sprocket gear

$F_{t2} = \frac{\text{Torque}}{PCD/2} = \frac{21.75 \times 10^3}{38.57} = 563.86 \text{ N}$

$F_{r2} = F_{t2} \tan\varphi = 563.86 \tan(20) = 205.22 \text{ N}$

Resolving force F_{t2V} and F_{t2H}

$F_{t2V} = F_{t2} \cos(80) = 563.86 \cos(80) = 97.91 \text{ N}$

$F_{t2H} = F_{t2} \sin(80) = 563.86 \sin(80) = 555.29 \text{ N}$

Resolving force F_{r2V} and F_{r2}H

$F_{r2V} = F_{r2} \cos(80) = 205.22 \cos(80) = 35.64 \text{ N}$

$F_{r2H} = F_{r2} \sin(80) = 205.22 \sin(80) = 202.10 \text{ N}$

3.3.3 Virtical Load Diagram

Taking Moment at "A" get,

∑MA=0

$R_{BV} \times 0.6 = 206.01 \times 0.3 + 133.55 \times 0.45 + 158.5 \times 0.15$

$R_{BV} = 242.79 \text{ N}$

∑Fy=0

$R_{AV} + 242.79 - 206.01 - 158.5 - 133.55$

$R_{AV} = 255.27 \text{ N}$

BM at C= M_{CV}=255.27×0.15=38.29 Nm

BM at D = M_{DV} = 255.27×0.3-158.5×0.15 = 52.8 Nm

BM at E = M_{DV} = 36.41 Nm

3.3.4 Horizontal Load Diagram

Taking moment about "A", we get

$R_{BH} \times 0.6 = 898.92 \times 0.15 + 757.39 \times 0.45$

$R_{BH} = 792.77 \text{ N}$

∑Fy=0

$R_{AH} + 792.77 - 898.92 - 757.39$

$R_{AH} = 863.54 \text{ N}$

BM at C = M_{CH}=863.54×0.15= 129.53 Nm

BM at E = M_{EH} = 863.54×0.45-898.92×0.3 = 118.91 Nm

Resultant BM at C = $\sqrt{38.29^2 + 129.53^2}$ = 135 Nm

Resultant BM at E = $\sqrt{36.41^2 + 118.91^2}$ = 124.34 Nm

Equivalent twisting moment (T_e),

$$T_e = \sqrt{(K_m \times M)^2 + (K \times T)^2}$$

$$= \sqrt{(1 \times 135000)^2 + (1 \times 21750)^2}$$… (Referring data book)

Since,

$$T_e = \frac{\pi}{16} \times \tau \times d^3$$

$$\sqrt{(1 \times 135000)^2 + (1 \times 21750)^2} = \frac{\pi}{16} \times 42 \times d^3$$

So, we get $d = 23.22 \cong 25\text{mm}$

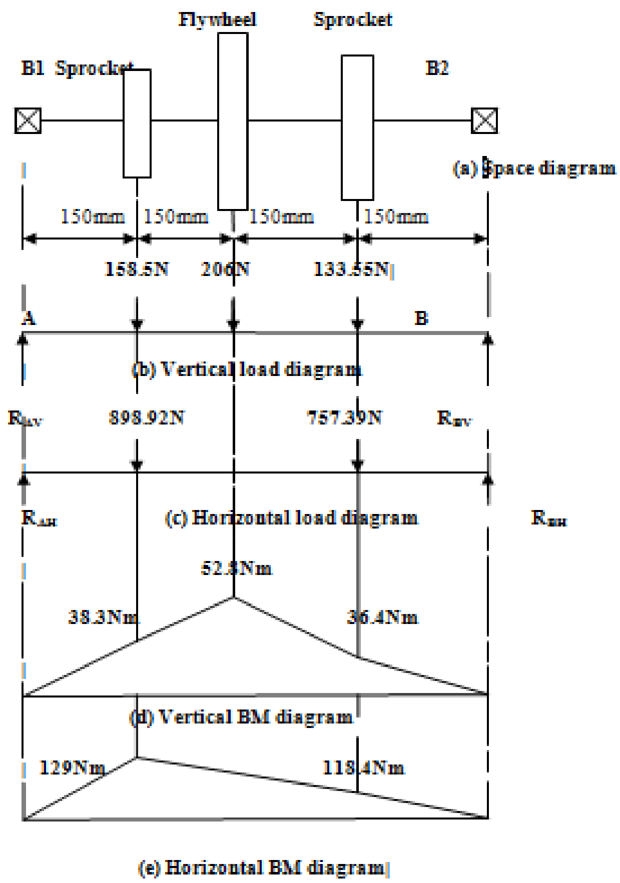

Figure 3 BMD diagram of shaft 1

3.4 Design Of Gear:

Materials:-

Gear – Plain Carbon Steel
Pinion – Alloy Steel
(Sut)pinion = 800 N/mm^2
(Sut)gear = 700 N/mm^2
P = 0.313 Kw
ng = 756 rpm
ka = 1.5
Nf = 1.35
b = 12times module
km = 1.4
Φ = 20
(BHN)p = 400
(BHN)g = 350
Assume gear ratio (G) = 2.5
G = np/ng
2.5= np/756
np = 1890 rpm
If Zp = 20
G = zg/zp
2.5 = zg/20
zg = 50

3.4.1 Beam Strength:

σbp = sup/3
800/3
σbp = 266.67 N/mm^2
σbg = sug/3
=700/3
σbp = 233.33 N/mm^2
Lewis Form Factor:
ϒp = 0.484-2.87/zp
= 0.484-2.87/20
ϒp = 0.3405
ϒg = 0.484-2.87/zg
=0.484-2.87/50
ϒg = 0.4266
σbp*ϒp = 266.67*0.3405 = 90.80 N/mm^2
σbg*ϒg = 233.33*0.4266 = 99.54 N/mm^2
As σbp*ϒp < σbg*ϒg , the piniom is greater than gear in bending hence if it is necessary to design pinion for bending
There fore
Fbp = σbp*b*m*ϒp
Fbp = 266.67*12m*m*0.3405 N
Fbp = 1089.61 m^2 N

3.4.2 Wear Strength:

dp = m*zp
dp = m*20
dp = 20m
Q = 2*zp/zp+zg
2*50/20+50
Q = 1.429
K = 0.16[BHN/100]^2
= 0.16[350/100]^2
(taking lowest value of surface hardness)
K = 1.96 N/mm^2
Fw = dp*b*Q*k
=20m*12m*1.429*1.96
Fw = 672.20 m^2 N
As Fw < Fb , gear pair is greater in pitting hence if should be designed for safely against pitting failure

3.4.3 Effective Load:

V= π*dp*bp/60*1000
= π*20m*1890/60*1000
= 1.98m , m/s
Ft = P/V
=313/1.98m
Ft = 158.08/m , N
Kv = 3/3+V
= 3/3+1.98

Feff = Ka*Km*Ft/Kv
Feff = 1.5*1.4*313/{3/(3+V)}*1.98*m
= 110.65(3+1.98*m)/m , N

3.4.4 Estimation of Module:
In order to avoid the pitting failure
Fw = Nf*Feff
672.20*m^2 = 1.35*110.65(3+1.98*m)/ m , N
672.20*m^3 – 295.78 m – 448.13 = 0
m = 1.039 mm

Dimensions of Gear Pair:
m = 1.25 mm zp = 20
zg = 50 b = 12*m = 12*1.25 = 15 m
dp = m*zp dg = m*zg
dp = 1.25*20 = 25 mm dg = 1.25*50 = 63mm
ha = 1*m = 1.25*m hf = 1.25*m = 1.25*1.25 = 1.56 m
a = (dp+dg/2)
= 25+63/2
a = 44 mm

Precise Estimation Of Dynamic Buckingham's Equation
We take grade 7
e = 11.0+0.9 (m+0.25√d)
for pinion ep = 11.0 + 0.9 (m + 0.25 √dp)
= 11.0 + 0.9 (1.25 + 0.25√25)
= 13.25μm
for gear ep = 11.0+0.9(m+0.25 √dg)
= 11.0+0.9(1.25+0.25√63)
= 13.91 μm
e = ep + eg
e = 13.25+13.91
e = 27.16*10^-3 mm
The Buckingham's Equation For The Dynamic Load In The Tangential Direction Is
Fd = 21*V(b c + Ftmax)/21*V+√b c + Ftmax
= 158.08/m
= 126.46 N
V = 1.98*1.25 = 2.475 m/s
Ftmax = Ka*Km*Ft
` = 1.5*1.4*126.46
= 265.54 N
C = K*e [Ep.Eg/Ep+Eg]
= 0.111*27.1*10^-3 * [207*10^3/207*10^3 + 207*10^3]
= 312.02 N/m
Fd = 21*2.475(15*32.02 + 265.56)/21*2.475 + √15*312.02+265.56
= 257061.07/122.30
Fd = 2101.85 N

Available Factor Of Safety
Feff = Ka*Km*Ft + fd
= 1.4*1.5*126.46 + 2101.85
Feff = 2367.42 N
Fw = 672.20 m^2
= 672.20(1.25)^2
Fw = 1050.31 N
Hence available FOS is
Nf = Fw/Feff
= 1050.31/2367.42
= 0.44
= 0.44 < 1.35
As the available FOS is very lower than required FOS
Hence The Design is unsafe
<u>Rough Estimation of Next standard module</u>
Nf.Eeff = 1.25*2367.42
= 2959.28 N
For
m = 2 mm
Fw = 672.20(2)^2
= 2688.8 N
2688.8 N < Nf. Feff
Hence
m = 2 mm is unsafe
For
m = 2.5
Fw = 672.20(2.5)^2
= 4201.2 N
4201.25N > Nf.Feff
Hence Design Is To Modified For
m = 2.5 mm

Modified dimension of Gear Pair
m = 2.5 mm
dp = m.zp = 2.5*20 = 50 mm
dg = m.zg = 2.5*50 = 125 mm
ha =1m = 2.5 mm
hf = 1.25*2.5 = 3.125 mm
a = dp+dg/2
a = (50+125/2) = 87.5 mm

3.4.5 Force Analysis of spur gear
1. Tangential component or (Tangential Force)
Ft = P/V
= 313/1.98
= 158 N

2. Radial Force
Fr = Ft.tanΦ
Φ = pressure angle
Fr = 158 tan 20
Fr = 57.54 N
Fr = 58 N

3.5 Design Of Shaft 2
Given:
Ft = 158 N
Fr = 58 N
Torque To Be Transmitted
T = Ft * Dg/2
= 158*15/2
T = 9.875*10^3 N.mm

3.5.1 Vertical Force Analysis:
Td = 60*313.13*2/2*π*756
Td = 7.91*10^3 N.mm
Ft = 7.91910^3*2/45 = 352 N
This Force is resolved in to two force
Ftv = 35cos80 = 61.24 N
Fth = 35sin80 = 346.65 N
The Radial Force
Fr = Ft tan Φ
= 352 tan 20
Fr = 128 N
This Force Is Reso lved In To Tow Force

Frv = 128 cos 80 = 22.22 N
Frh = 128 sin 80 = 126 N
∑MA = 0
58*0.2+83.43*0.45-Rbv*0.6 = 0
Rbv = 81.92 N
∑Fy = 0
Rav+81.92-58-83.43 = 0
Rav = 60 N
Mc = 60*0.2 = 12 N.m
Md = 60*0.45-58*0.25
Md = 12.5 N

3.5.2 Horizontal Force Analysis:

∑MA = 0
158*0.2+472*0.45-Rbh*0.6 = 0
Rbh = 406.67 N
∑Fy = 0
Rah+406.67-158-472 = 0
Rah = 224 N
Mc = 224*0.2 = 44.8 N.m
Md = 224*0.45-158*0.25 = 61.3 N.m
Bmax = 61.3*10^3 N.mm
Te = √(km * M)^2 + (kt*Te)^2
Te = √(1*61.310^3)^2+(1*9.875*10^3)^2
Te = 62.09*10^3 N.mm
Te = π/16*Ʈ*d^3
d^3 = 62.09*10^3*16/π*55.57
d = 17.85
d = 18 mm

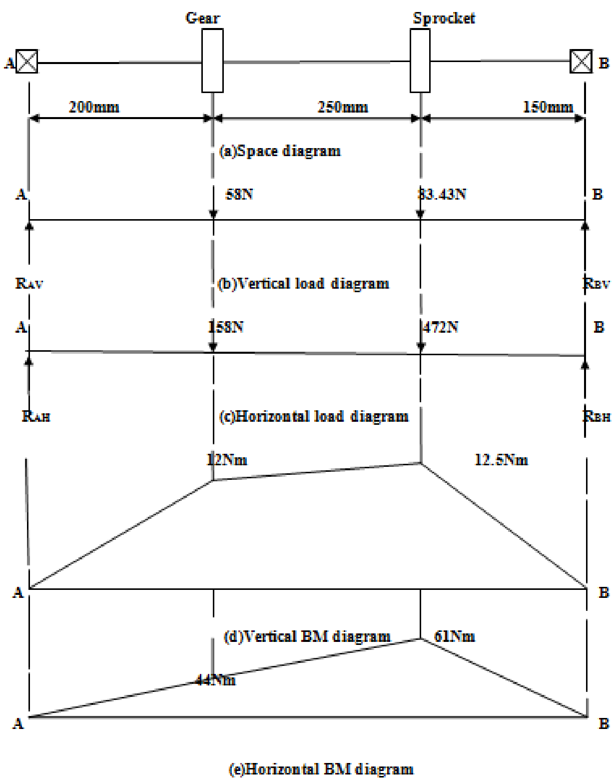

Figure 4 BMD diagram of shaft 2

3.6 Design Of Shaft 3:

Given:
Ft = 158 N
Fr = 58 N
Weight Of Cutter is 58.86 N
Torque to be transmit
T = Ft*Dp/2
= 158*50/2
= 3.950*10^3 N.mm

3.6.1 Vertical Force Analysis:

∑MA = 0
-58.86*0.15+58*0.2-Rbv*0.6+58.86*0.75 = 0
Rbv = 78.19 N
∑FY = 0
-58.86+Rav-58+78.19-58.86 = 0
Rav = 91.53 N
Ma = -58.86*0.15 = -8.829 N.m
Mc = -58.86*0.35+97.53*0.2 = -1.095 N.m
Mb = -58.86*0.75+97.53*0.6-58*0.4
Mb = -8.827 N.m

3.6.2 Horizontal Force Analysis:

∑MA = 0
158*0.2-Rbh*0.6 = 0
Rbh = 52.66 N
∑FY = 0
Rah+52.66-158 = 0
Rah = 105.34 N
Mc = 164.19*0.2 = 21.068 N
Bmax = 21.068*10^3 N.m
Te = √(km*Mmax)^2 + (kt*T)^2
Te = √(1*21.068*10^)^2+(3.95*10^3)^2
Te = 21.068*10^3 N.mm
Te = π/16 *Ʈ*d^3
d^3 = 21.068*10^3*16/π*55.57
d = 12.52 mm d =15 mm

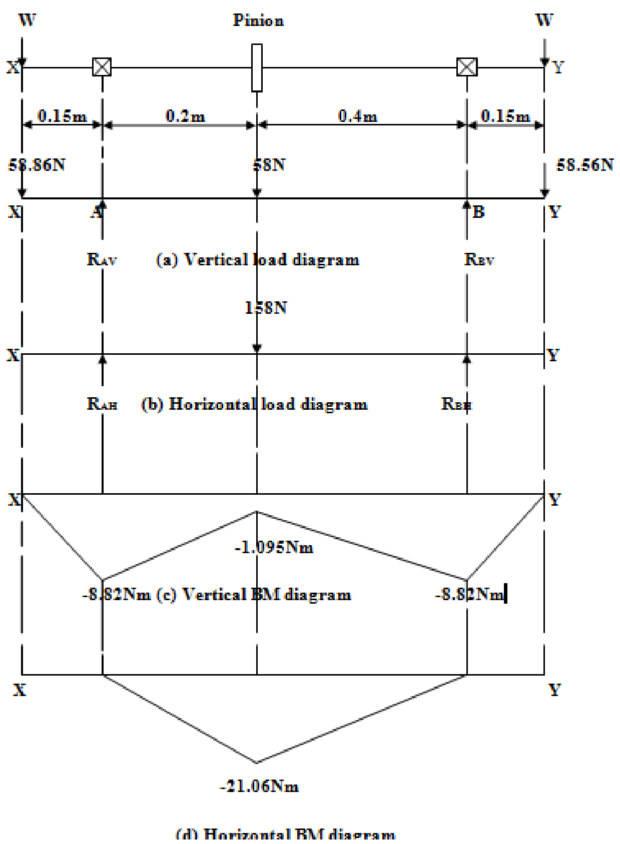

Figure 5 BMD diagram of shaft 3

3.7 Design Of Antifriction Bearing:

There are two antifriction bearing used in the experimental set up

Maximum Reaction developed at Shft 1^{st} is

Far = √(Fav)^2+(Fah)^2

Far = √(255.27)^2 + (863.34)^2

Radial Load

Far = 900 N

Axial Load = 0

Equivalent Load Coming On Bearing (Fe)

Fe = [XFr + YFa]*KsKoKpKv

Fr = 900N

Fa = 0

e = Fa/Fr = 0

Selecting Self – aligning ball Bearing

X = 1 , Y = 2.3

Kp = 1

Kr = 1

Ks = 2

Fe = (1*900+0)*1*1*1*2

Fe = 1800 N

Life of Bearin (L)

L10 = (C/Fe)^n*krel

[C = Dynamic load capacity,krel = reliability factor, n = outer race, d = inner race, b = bearing width]

L = 50 (depenstration model)

n = 3 (ball bearing)

krel = reliability (90%)

C = (L10)^1/n *Fe

C = (500)^1/3 * 1800

C = 14286.60 N

C = 14.28 KN

Selecting Series 6205[C = 14800]

d = 25 mm

D = 52 mm

B = 15 mm

4. DESIGNED MODEL OF SUGARCANE SPROUT CUTTING MACHINE BY HUMAN POWER:

As per the designed and analytical calculations made in the previous chapter following CAD model is developed by using different commands of Catiya software. This CAD software provides the tools needed to perform modeling of different parts of the proposed machine efficiently and free from tedious and time consuming task. The front view a of the model is shown below

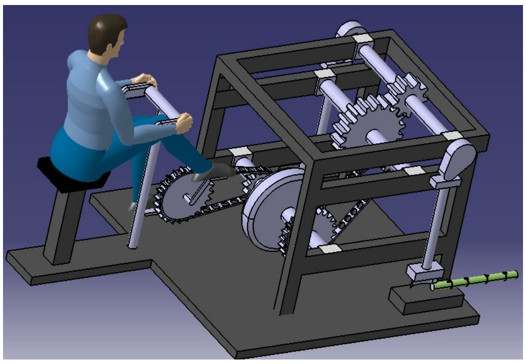

Figure 6 Isometric view of the model

5. FABRICATION AND WORKING

5.1 Fabrication

The manufacturing process used in the fabrication of Human Powered sugarcane sprout cutting machine is such that the total cost of fabrication is low and also one that can make use of the available materials.

5.2 Components:

5.2.1 Shaft

Shaft is a rotating member, usually of circular cross section. It is used to transmit power as well as motion. It provides the axis of rotation or oscillation for elements like gears , pulleys, flywheel, sprockets, cranks, etc. The power transmitted to the shaft by some tangential force and the resultant torque set up within the shaft transmits the power to machines connected with the shaft.

Figure 8 shaft

Shafts are generally divided into two types-
1) Transmission shafts
2) Machine shafts

5.2.2 Chain Drive

The chain drive assumes a special position in the large group of drive mediums for the transmission of torque and power. The great advantage of this type of positive controlled connection is the constant relative speed between driving and driven shafts and the complete elimination of slip, compared with non-positive drives(e.g. belt drives) where slip very often can only be prevented by high belt tension. Steel roller chains are used as drive, transport or conveyor chains for static or alternating loads with slow or high chain speeds transmitting power between two parallel shafts.

Figure 9 Chain drive

5.2.3 Flywheel

A flywheel is an inertial energy-storage device. It absorbs mechanical energy and assist as a reservoir, storing energy all along the period when the supply of energy is more than the requirement and releases it during the period when the requirement of energy is more than the supply.

The main function of a flywheel is to smoothen out variations in the speed of a shaft caused by torque fluctuations. If the antecedent of the driving torque or load torque is fluctuating in nature. Many machines have load template that cause the torque time function to vary bygone the cycle. Internal combustion engines with one or two cylinders are a exemplary example. Piston compressors, punch presses, rock crushers etc. are the another systems that have fly wheel. The amount of energy required for the desired degree of smoothening must be found and the (mass) moment of inertia needed to absorb so that energy determined. Then flywheel geometry must be defined so that caters the required moment of inertia in a reasonably sized package and is safe against failure at the designed speeds of operation. A flywheel used in machines serves as a reservoir, which stores energy during the span when the supply of energy is more than the requirement, and releases it all along the span when the requirement of energy is more than the supply.

Figure 10 Flywheel

5.2.4 Gear Drive

Spur gear is a cylindrical shaped gear in which the teeth are parallel to the axis. It is a easy to manufacture and it is mostly used in transmitting power from one shaft to another shaft up to certain distance & it is also used to vary the speed & torque e.g. watches & gearbox etc. The cost of replacement of spur gear is very high and also the system down time is one of the effect in which these gears are part of the system. Failure of gear causes breakdown of system which runs with help of gear example automobile vehicle. When gear is subjected to load high stresses developed at the root of the teeth, due to the high Stresses, possibility of fatigue failure at the root of teeth of spur gear increases. There is a higher chance of fatigue failure at these locations. So avoid fatigue failure of the gear, the stresses should be minimum to maximum stress. Number of studies has been done by various authors to analyze the gear for stresses. Gears have been analyzed for different points of contact on the tooth profile and the corresponding points of contact on the pinion. In this study the variation stress in root fillet region is found, which is a used for the study of variation of various parameters of stress reducing feature. The effect and use of stress relief feature in geometry of gear is studied as reported by research.

Figure 11 Gear drive

5.2.5 Cam and follower:

If the contact end of the follower is perfectly flat then it is called as flat faced follower. If this flat faced follower is circular then it is called as mushroom follower. In these followers the side thrust between the guide and follower is much reduced, but the side thrust due to friction will exist. These followers are used to operate the valves of automobile engine. Cam-follower systems are frequently used in all kinds of machines. The valves in your automobile engine are opened by cams. Machines used in the manufacture of many consumer goods are full of cams. Compared to linkages, cams are easier to design to give a specific output function, but they are much more difficult and expensive to make than a linkage. Cams are a form of degenerate four bar linkage in which the coupler link has been replaced by a half joint.

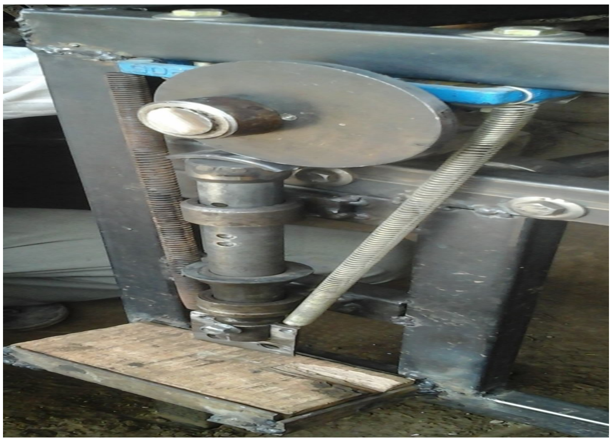

Figure 12 Cam and follower

5.3 Working:

A chain drive CH will be used to speed the shaft s1.a smaller sprocket sp2 will be placed on the shaft. On the other hand bigger sprocket sp2 will be placed on the shaft with the pedal arrangement. The speed of the shaft will amplify the speed of the flywheel because they are placed on the same shaft. On this flywheel shaft there will be another bigger sprocket connected to the other shaft s2 with the chain drive ch2.the speed of shaft will be increased with the help of chain drive. On the shaft s2 there is gear. Pinion attached with other shaft s3.When the driver drives the pedal, the human energy will be converted into rotational kinetic energy stored in flywheel and then it transferred with the help of another chain drive to spur gear to the cutting unit.

6. RESULT AND DISCUSSION

6.1 DISCUSSION ABOUT TESTING OF SUGARCANE SPROUT CUTTING MACHINE BY HUAMAN PEDAL POWER

It is the common phrase that any new concept which is being evolved it needs to be verified to check its performance parameters, so that one could compare the newly evolved concept with the existing technology. For the testing purpose the different types of sugarcane sprout cutter have been used. Before cutting, the moisture content is removed and kept up to 15 to 18%.

Afterwards the following procedure is adopted

1. Firstly the known quantities of sugarcane have been taken to process unit to cut the sprout.
2. Next to this the pedaling is started this pedaling is carried out for one minute.
3. As soon as the shaft is rotates it helps to transmit force to the process unit and helps in cutting the sugarcane sprout. It means available energy stored in the flywheel becomes helpful for this purpose.
4. When the cutting activity takes place, simultaneously the measurement of the speed of the flywheel and the torque required and time is note down with the help of measuring speed logger instruments.
5. Step 2 to 5 is known as one cycle.
6. Once again, rider does pedaling energizes the flywheel note the measurements of speed of the flywheel shaft and process unit shaft and time for cutting. This process unit lasting until the amount of sprouts are fully cut without breaking of sprout.
7. Steps 1 to 6 are also carried out for the known amount of number of sprouts like 15

During testing it is found that the maximum speed gained by the flywheel shaft is 400 rpm.

6.2 MEASUREMENT OF EFFICIENCY

The efficiency of sugarcane sprout cutting machine can be defined as the ratio of the number of sprouts cut to the total number of sprouts are on sugarcane. The total number of sprouts is obtained by adding the number of sprouts cut to the number of uncut (damage) sprouts remaining on the sugarcane after cutting.

Cutting efficiency (%) = $\frac{x}{x+y} * 100$

Where

X = number of sprouts cut

Y = number of uncut (damage) sprouts remaining on the sugarcane after cutting.

6.3 PERFORMANCE EVALUATION

Five sugarcanes were selected

6.3.1 PARAMETERS

i). Cutting efficiency

$$=\frac{\text{number of sprouts cut (kg}*3600/\text{hr}}{\text{number of sprouts cut+number of uncut (damage) sprouts}}*100\%$$

6.3.2 CUTTING EFFICIENCY:

Cutting efficiency

$$=\frac{\text{number of sprouts cut (kg}*3600/\text{hr}}{\text{number of sprouts cut+number of uncut (damage)sprouts}}*100\%$$

Trial No.1

Output = $\frac{14}{14+2}$ * 100 = 87.5%

Trial No. 2

Output = $\frac{13}{13+2}$ * 100 = 86.66%

Trial No. 3

Output = $\frac{15}{15+1}$ * 100 =93.75%

Trial No. 4

Output = $\frac{14}{14+3}$ * 100 = 82.35%

Trial No. 5

Output =$\frac{14}{14+4}$ * 100 =77.78%

Table 1: Cutting efficiency

Number of sprouts cut	Cutting time (sec)	Cutting efficiency %
14	58	87.5
13	55	86.66
15	61	93.75
14	60	82.35
14	63	77.78

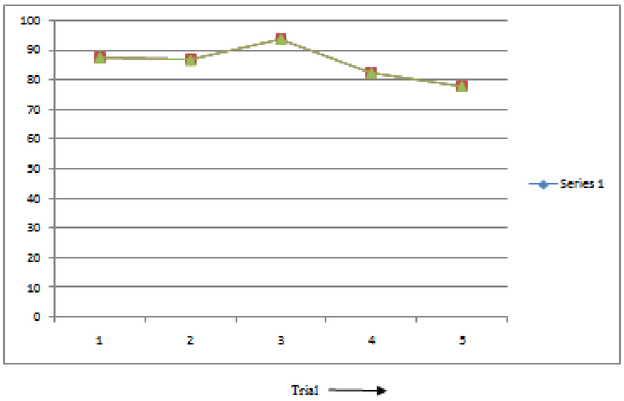

Figure 13 Cutting time vs cutting efficiency

The variety of sprout cutter available in the market is broadly classified as hand operated and other with power operated process. The existing sprout cutter uses motors having capacity varying from 2hp to 12 hp and more.

The per hour consumption is quite high. At the same time it needs proper maintenance time to time. An average electricity consumption of existing cutters is around 2 -3 units per hour. This adds to around 16 units per day. At the same time the initial cost of the machine is low as compared to together machine

The present sprout cutter which is electrically powered totally rely on the availability of electricity. In the absence of electricity due to loading shading or any other reason the present sprout cutter cannot work. Further at some remote areas the electricity is not yet made available.

In country with the existing machine sprout cutter the proposed human powered does not rely on the electricity hence it can be easily run uninterrupted even in rural areas where there is interrupted supply of electricity also due to absence of electric motor its maintenance is quite easy and less expensive

The initial cost the machine is very less as Rs 10000 to 16000 as comparison with other sugarcane sprout cutter machines.

The time taken for cutting of sugarcane sprout is almost same as the existing machines. Hence it is said that the proposed machine is viable best suited for remote areas and energy efficient.

7. CONCLUSIONS

A new concept in terms of human powered of flywheel is applied for sugarcane sprout cutting process which finds suitable and viable and reduces human effort compared to hand operated machine as hand operated machine requires two operators whereas new machine requires only one. This machine provides comfort for seating arrangement for different position depending upon ergonomics. The demand power of the estimated machine estimated as 0.5hp.After the rider pedals, the speed of flywheel shaft increases due to gear ratio given due to get the necessary required speed. The time required for cutting depends upon the size of the sugarcane which is to be taken. It is seen that for higher size the time requirement will be aggrandized. For the instant work machine production rate is estimated as 85.6 per hour. It is also seen that if moisture contained in the sugarcane is more in that condition there will be more breaking of sprout and time required to cut the sprout will be more.

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