Design of Experimental Set-up for Establishing Empirical Relationship for Chaff Cutter Energized by Human Powered Flywheel Motor

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Abstract A human powered chaff cutter has been developed in the absence of any data. Literature survey reveals thata system for pumping using muscular energy in the flywheel is feasible and then the energy stored in flywheel can used for different applications. Accordingly, it was decided to work on chaff cutter with this concept and to establish an empirical relationship for human powered chaff cutting process. Since this is a man-machine system, it is rather difficult and unreliable to adopt total theoretical approach for the development, thus, the experimental approach was adopted. This set up consists of three subsystems namely. (i) Human powered flywheel motor (HPFM) i.e. energy unit. (ii)Torque amplification gears and clutch unit and (iii) process unit i.e. chaff cutter. This paper reports the design of experimental setup for carrying out the experimentation to establish empirical relationship for chaff cutter energized by human powered flywheel motor.

Key words: Human powered Flywheel Motor, Chaff cutter.

Introduction

In India, animal husbandry is an integral part of the rural economy. The forage (dry or wet) production requires high labor, coupled with a lack of sufficient land for forage. Production and forage scarcity during the dry season means that available forage must be efficiently used to minimize waste.

Traditionally, the farmers chop forage into small pieces for easy consumption by the animals as shown in Figure1. This method is tedious, time consuming and quite dangerous to operator, as well as low output and lack of

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uniformity. Presently mechanized forage cutters are electric motor driven or hand driven. But today, there is huge scarcity of electricity almost everywhere in India, which results in six totwelve hours load shedding. The power cut (load shedding) badly affects daily needs that require power supply. Hand muscles are weaker than leg muscles, so it can be used in muscle-driven flywheel motor to replace the electric driven cutting unit.

Fig. 1. Traditional hand chopping

Materials and methods

Formulation of the Present Problem

In the human powered flywheel motor concept, the bicycle mechanism for converting and transmitting human energy through paddling to rotational kinetic energy of flywheel is hereby proposed. The energy stored in the flywheel can then used for actual cutting process. This human energy output is in the low range and the processes could be operated intermittently can be considered for utilization. Modak (2004) gave a landmark paper, in which he presented reports on (1) functional feasibility and (2) economic viability of human powered flywheel motor. A schematic representation of the set-up is given in figure 2. A driver sits on seat and paddles the bicycle mechanism converting oscillating motion of things into rotational motion of flywheel with increasing speed.
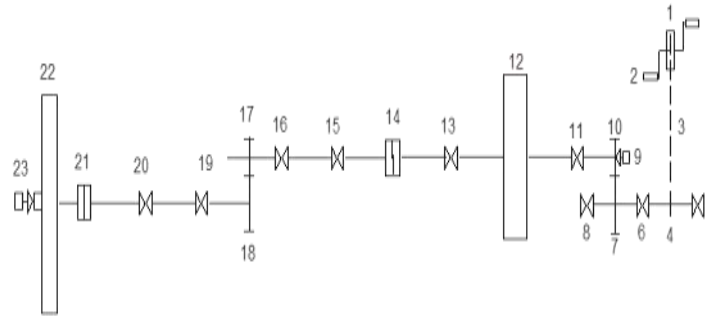
Fig. 2. Schematic of Human Powered Flywheel Motor.
1-Chain Sprocket 2-Pedal 3-Chain 4-Freewheel 5,6-Bearings for bicycle side 7-Gear-I 8-Bearing 9-Tachogenerator for flywheel shaft 10-Pinion-I 11-Bearing for flywheel shaft 12-Flywheel 13-Bearing for flywheel 14-Two jaw clutch 15,15-Bearing of intermediate shaft 17-Pinion II 18-Gear II 19,20-Bearing for process unit shaft21-Coupling 22-Chaff Cutter blade 23-Tachogenerator for chaff Cutter shaft

The load on the thighs (legs) of the rider is only the inertia load of the flywheel. The rider pumps energy in the flywheel at an energy input rate convenient to him. Thus, this man-machine system brings out ways for energy conversion of human muscular energy into rotational kinetic energy of flywheel. It was necessary to develop the energy unit of this man-machine system for chaff cutter scientifically. An approach of methodology of experimentation (Schenk, 1962) is adapted and worked out for the detailed design of experimentation for chaff cutter.

Formulation of the dimensional Equations

Dimensional Analysis of the parameters affecting the chaff cutter energized by human powered flywheel motor was identified and listed in Table 1. Dimensional analysis was used to express the required functional relationship between the different parts of the chaff cutting process. The main advantage of this analysis is the reduction of the number of variables.

Dimensional equations for response variables is :

1. Resistive Torque (Tc) :
$$(D/gI)\ T_c = f\ \{(\ dW_b t_b/D^3\ ),(D^4/gI)\ E,G,n,\ \alpha,\ ,(\sqrt{D/g})\ \omega,(\sqrt{g/d})t_c,\}\qquad(1)$$

2. Number of cuts (C_p):
$$(\sqrt{D/g})C_p = f\{(\ dW_b t_b/D^3\ ),(D^4/gI)\ E,G,n,\ \alpha,\ ,(\sqrt{D/g})\ \omega,(\sqrt{g/d})t_c,\}\qquad(2)$$

3. Process time for cutting (t_p):

$$(\sqrt{g/D})t_p = f\{(\,dW_b t_b/D^3\,),(D^4/gI)\,E,G,n,\,\alpha,\,,(\sqrt{D/g})\,\omega,(\sqrt{g/d})t_c,\} \qquad (3)$$

Where:

d = Hub Diameter of blade	W_b = Width of cutting blade
t_b = Thickness of cutting blade	D = Tip diameter of blade
g = Acceleration due to gravity	I = Moment of inertia of flywheel
E = Young's modulus of elasticity of cutting blade	▊ = Cutting blade angle
G = Gear ratio	n = Number of blades
e = Kinetic energy of flywheel	ω = Angular velocity
T_c = Instantaneous torque on cutting blade	C_p = Number of cuts during cutting
t_p = Process time for cutting	

Table 1. Dimensional Matrix

	d	W_b	t_b	D	g	I	E	α	n	ω	t_c	e	G	T_c	C_p	t_p
M	0	0	0	0	0	0	1	0	0	0	0	1	0	1	0	0
L	1	1	1	1	1	1	-2	0	0	0	0	2	0	2	0	0
T	0	0	0	0	-2	-2	-1	0	0	-1	1	-2	0	-2	0	1

The variables listed were combined to from dimensionless ratio or π terms on the basis of nature of physical quantities. These formed groups have been used for experimentation, as given in Table 2.

Table 2. Dimensionless ratio or π terms

Sr. No.	Independent dimensionless ratio or π terms	Nature of basic physical quantities
1	$\pi_1 = \dfrac{dw_b t_b}{D^3}$	Geometric Variables
2	$\pi_2 = \dfrac{D^4 E}{g\,I}$	Material of blade
3	$\pi_3 = \sqrt{\dfrac{D}{g}}\,\omega_c$	Instantaneous Terminal velocity of cutter
4	$\pi_4 = G$	Gear Ratio
5	$\pi_5 = \alpha$	Cutting blade angle
6	$\pi_6 = n$	No. of cutting blade
7	$\pi_7 = \sqrt{\dfrac{g}{D}}\,t_c$	Cutting time
8	$\pi_8 = \dfrac{D}{2g}\,\omega_f{}^2$	Terminal speed of flywheel
9	$\pi_{D1} = \dfrac{D}{gI}\,T_c$	Resistive torque

| 10 | $\pi_{D2} = C_p$ | No. of cuts by cutter |
| 11 | $\pi_{D3} = \sqrt{\dfrac{g}{D}}\, t_p$ | Process time |

Finalization of Test Envelope and Test Point

It is necessary to decide the range of parameters which would be varied during experimentation. The test envelope, test points and test sequence for every independent π terms is given in Table 2.

Table 2. The test envelope, test points and test sequence

Sr. No	Ratio	Test Envelop Range	Test Point	Test Sequence							
				1		2		3		4	
01	$\pi_1 = dWbtb/D^3$	5.33×10^{-3}	Constant								
02	$\pi_2 = (D^4/gI)E$	1.46×10^{9}	Constant								
03	$\pi_3 = G$	2,3,4		2.3.4	2.3.4	2.3.4	2.3.4	2.3.4	2.3.4	2.3.4	2.3.4
04	$\pi_4 = \alpha$	0.122	Constant								
05	$\pi_5 = n$	2,3		2	3	2	3	2	3	2	3
06	$\pi_6 = (D/2g)\omega f^2$	31.428-62.857		31.426		41.904		52.380		62.857	

Next step is to design and fabrication of an experimental set-up for the proposed experimentation.

Design of Experimental set up

Flywheel

$$\Delta E = \left(\frac{W}{g} k^2 \right) K_s \omega_m^{\,2}$$

(4)

Where,
K_s = Coefficient of speed Fluctuation, ΔE = The maximum fluctuation of energy. k = Radius of gyration
$K_s = \{0.12[D_o^2 + (D_o - 2h)^2]\}^{1/2}$

ω_m = mean velocity in radians/second, h = rim thickness, b = rim width
$\therefore$ Considering $\Delta E = 42{,}035$ J

$$\left(\Delta E = \left[\frac{1}{2} I \omega_{max}^{\,2} - \frac{1}{2} I \omega_{min}^{\,2} \right] \right)$$

(5)

$$K_s = \frac{N_{max} - N_{min}}{N_{mean}}$$

$$\Delta E = \left(\frac{W}{g} k^2\right) K_s \omega_m^2$$

$$= \left(mk^2\right) K_s \omega_m^2$$

$$= I K_s \omega_m^2$$

Where,

I = Moment Of Inertia of Flywheel.

$\Delta E = 42{,}035$ J, $K_s = 2$, $\omega_m = 41.866$ rad/sec

$\therefore \Delta E = I K_s \omega_m^2$

$\therefore 42{,}035 = I \times 2 \times (41.866)^2$

$\therefore I = 11.99$ kg-m^2

We consider, I = 12 kg-m^2

Assume D_m = flywheel mean diameter. =0.98m

$\therefore k = 0.49$m

Assuming that the rim provides 95 percent of the required moment of inertia,

$mk^2 = 0.95 \times I = 0.95 \times 12 = 11.4$ (6)

$m = 11.4 / k^2 = 11.4/0.49^2$ $\therefore m = 47.48$ kg

Other dimensions of flywheel:

b = 100 mm, h = 19.7mm, D_o = 1000 mm, No. of arms = 6.

Stresses in the flywheel:

Centrifugal stress = 13.53 MPa, Bending stress = 187.21 MPa, resultant stress = 56.79 MPa

Chain Drive

Rated Power, PR= 900 W, Design Power(P_d) : $Pr \times K_1$ (7)

Where,

K_1 =Load Factor, From design data book it should be 1.2 for moderate shock and service of 10 hours per day [Shiwalkar B. D., 2004].

$P_d = 900 \times 1.2 = 1080$ W =1.44 hp

Speed of smaller sprocket = 120 rpm.

From design data book graph, chain No. 50 for which Pitch (p) = 15.875 was selected. The chain sprocket with the 24 teeth on the smaller sprocket and 48 teeth on the larger sprocket is available in the market.

Pitch Diameter of smaller Sprocket (D_{p2})

$$D_{p2} = \frac{p}{\sin\left[\dfrac{180}{T_2}\right]} \qquad (8)$$

T_2 = No. of teeth on smaller sprocket = 24

$$D_{p2} = \frac{1.1875}{\sin\left[\dfrac{180}{24}\right]} = 106.5 \text{ mm}$$

Pitch Line Velocity(V_p) :

$V_{p2} = (\pi \times D_{p2} \times N_2)/60 = (3.14 \times 106.5 \times 10^{-3} \times 120)/60$, $V_{p2} = 0.66$ m/sec

Power capacity per strand:

$$P = p^2 \left[\frac{V}{104} - \frac{V^{1.41}}{526}\left(26 - 25\cos\frac{180}{T}\right)\right] \times 10^3 \qquad (9)$$

where, p= chain pitch=15.875mm, V =V_{p2} =0.66m/sec , T =T_2 =21

$$P = 15.875^2 \left[\frac{0.66}{104} - \frac{0.66^{1.41}}{526}\left(26 - 25\cos\frac{180}{24}\right)\right] \times 10^3$$

P = 1275.6W

No. of strands = 1080/1275.6 =0.84

No. of strands =1

Tooth Load (F_t) = P_d/V_p = 1080/0.66 = 1636.36 N

Other dimensions

Pitch diameter of larger sprocket = 232.17 mm, center distance C = 120.82 mm, recommended C_{min} = 285.42mm, length of chain in pitch Lp = 72.75, Outer Diameter of the smaller sprocket = 130.10 mm, outer diameter of the larger sprocket = 251.73mm, width of the sprocket 9 mm.

Gear Design (Stage 1)

P_d = 2000 W, Module = m, pitch diameter, D_p=20×m,

V_p = 0.8373×m,

F_t = P_d/V_p = 2000/0.8373m = 2388.63/m

Assuming 1045 steel with heat treatment, S_o= 210 MPa, (Shiwalkar B. D., 2004; Bhandari V.B.(2005).

Bending strength,

F_b = S_o.C_v.b.Y.m =210×0.30×0.3415×10m×m = 21.145m²

Basic strength, So = 210 MPa

Velocity factor, C_v=0.3 (trial value), face width of gears, b= 10m (trial value.)

Modified Lewis form factor,

$Y = 0.485 – 2.87/t_p = 0.485 - 2.87/20 = 0.3415$

$F_b = 215.145m^2$

Equating $F_b = F_t$, $215.145m^2 = 2388.63/m$, $m = 2.23$, select module, $m = 3$.

$D_p = m.t_p = 3 \times 20 = 60$ mm.

After calculation, the actual values of V_p, C_v and F_t are as follows :

$V_p = 2.5119$ m/sec, $C_v = 0.544$, $F_t = 796.21$ N

Calculated dimensions for other gears

Pinion   Gear

m=3   m=3

T=20   t=80

D_p=60mm D_p=240mm

Design of shaft
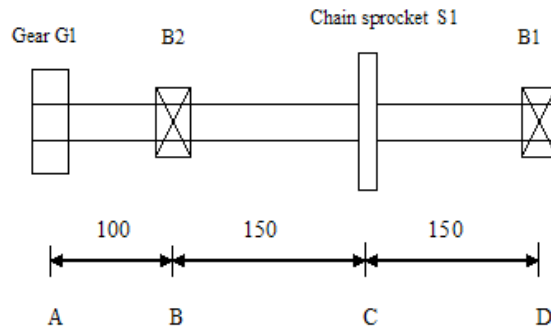
Fig. 3. Schematic arrangement of Shaft 1

The moment of inertia of flywheel, $I = 12$ kg-m^2

Here, gear G1 is on the driving shaft and the pinion G2 is on the flywheel shaft. Assume that the number of teeth on the gear and pinion are 80 and 20 respectively. Now, the value of torque to be transmitted to the pinion,

Torque $= I.\alpha$ (10)

The moment of Inertia of Flywheel, $I = 12$ kg-m^2

Angular acceleration of flywheel, $\alpha = (\omega_2 - \omega_1)/t$

ω_2 = Maximum speed attained by flywheel after peddling time t = 60 sec

N_2 = Max. Speed which can be attained by flywheel = 800 rpm

$\omega_2 = (2*3.14*800)/60 = 83.73$ rad/sec., $\omega_1 = 0$ (Initially, flywheel is at rest)

$\alpha = (83.73 - 0)/60 = 1.3955$ rad/sec^2

The moment of Inertia of Flywheel, $I = 12$ kg-m^2

The gear ratio, G=no. of teeth on gear (driving)/no. of teeth on pinion (driven) = 80/20 = 4

The load torque can be overcome (because of the M.I.) of the flywheel=I. $\alpha =$ G.I. $\alpha = 4 \times 12 \times 83.73 = 66.984$ N-m.

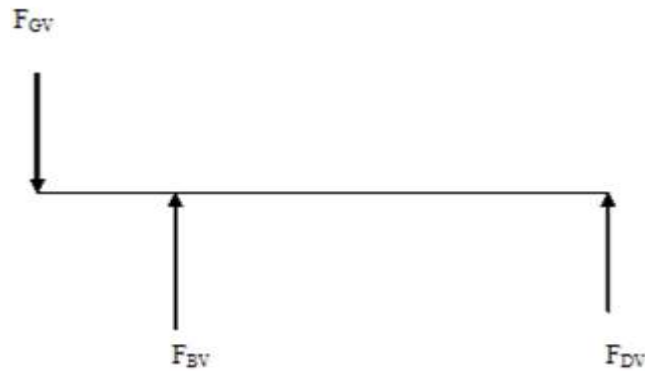The maximum torque to be transmitted from gear G_1 to the pinion $G_2 = 66.984$ N-m.

Fig. 4. Forces in vertical plane on Shaft 1

F_{GV} = tangential force acting on gear tooth, F_{GC} = radial force acting on the gear tooth

$T = F_{GV} \times r$ (r = pitch circle radius of gear) , T = Torque

$66.984 = F_{GV} \times 120$

$F_{GV} = 558.2$ N, $F_{GV} = 558.2$ N, $F_{GC} = 558.2$

$\tan 20 = 203.168$ N

Let F_t be the tension in the chain

$F_t . (\dfrac{D_{p2}}{2}) = 558.2 \times 120$, $F_t (106.51/2) = 558.2 \times 120$, $F_t = 1257.79$ N

$F_{BV} + F_{DV} = 558.2$

$-8.2 \times 100 - F_{DV} \times 300 = 0$

$F_{DV} = (-558.2 \times 100) / 300 = -186$ N

$F_{BV} = 744.2$ N

Bending moment at A $= 0$

Bending moment at B $= -558.2 \times 100 = -55820$ N-mm

Bending moment at C $= -186 \times 150 = -27900$ N-mm

Bending moment at D $= 0$

Forces in H.P.

$F_{BH} + F_{DH} = F_{GC} + F_t$

$F_{BH} + F_{DH} = 203.168 + 1257.79 = 1460.98$

$\sum M_{@B} = 0$

$203.168 \times 100 + 1257.79 \times 150 + F_{DH} \times 300 = 0$

F_{DH} = 561.17, F_{BH} = 899.788

Bending moment at A =0

Bending moment at B = 203.16 $\times$ 100 = 20316.8 N-mm

Bending moment at C = -561.17 $\times$ 150 = -84175.5. N-mm

Bending moment at D = 0.

Resultant Bending Moment at B = $\sqrt{55820^2 + 20316.8^2}$ = 59402.39 N-mm

Resultant Bending Moment at C = $\sqrt{27900^2 + 84175.5^2}$ = 88678.77 N-mm

Maximum Bending Moment = M = 88678.77 N-mm

Selecting Material of shaft as SAE1030 for which :

S_{yt} = 296MPa & S_{ut} = 527Mpa, $\tau_{max} < 0.30 S_{yt}$ or $\tau_{max} < 0.18\ S_{ut}$

Therefore the design shear stress should be

S_{ds} = 0.30S_{yt} = 0.30*296 = 88.8MPa or S_{ds} = 0.18 S_{ut}= 0.18*527 =94.86 MPa

We consider provision for Keyways, S_{ds} should be reduced by 25 percent

S_{ds} = 0.75*88.8 =66.6 Mpa , τ_{max} = 66.6 Mpa

D = 18.54mm

If we consider the factors for gradually load

K_b =1.5 & K_t = 1 (Bhandari V.B.,2005; Shiwalkar B. D.,2004),

By Calculation, D = 22.5 mm. Select D = 25 mm.

Similarly by calculations,

Diameter of shaft 2, D = 25.13 mm. Select D = 30 mm.

Diameter of shaft 3, D = 25.51 mm. Select D = 30 mm.

Bearing

For shaft 1:

The horizontal and vertical components of the reactions at two bearings B1 & B2 are already calculated while designing the shaft.

The reactions at two bearings are given by:

$$R_{B1} = \sqrt{R_{DH}^2 + R_{DV}^2} = \sqrt{561.17^2 + 186^2} = 591.19 \text{ N} \qquad (11)$$

$$R_{B2} = \sqrt{R_{BH}^2 + R_{BV}^2} = \sqrt{899.788^2 + 744.2^2} = 1167.66 \text{ N}$$

The expected bearing life (L_{10}) :

$$L_{10} = \frac{60\,n\,L_{10h}}{10^6} \qquad (12)$$

Where, n = speed of rotation of shaft= 200 rpm.

L_{10h} = Expected bearing life = 20000 hours (for the machines used for eight hours of service per day –

L_{10} = (60 $\times$ 200 $\times$ 20000)/10^6 = 240 million revolutions.

The dynamic load carrying capacity (C_1 & C_2):

Load factor = 1.5 (for Chain Drive)

$C_1 = P_1 [L_{10}]^{1/3}$(Load Factor)

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Considering no axial load, $P_1 = R_{B1} = 591.19$ N
$C_1 = 591.19 \times 240^{1/3} \times 1.5 = 5510.89$ N
$C_2 = P_2[L_{10}]^{1/3}$(Load Factor)
Considering no axial load, $P_2 = R_{B2} = 1167.66$ N
$C_2 = 1167.66 \times 240^{1/3} \times 1.5 = 10884.57$ N
For the Shaft diameter 25 mm following Bearings are available:
i) No. 61805 (C =3120 N), ii) No. 16005 (C = 7610 N), iii) No. 6005 (C =11200 N)
For required dynamic load carrying capacity, bearing no. 6005 is suitable for B1 & B2.
Other dimensions of the bearing:
Inner diameter of the bearing = 25 mm, outer diameter of the bearing = 47 mm, Axial width of the bearing = 12 mm.
Similarly by calculations,
Bearings available for shaft 2 and 3,
i) No. 61806 (C =3120 N), ii) No. 16006 (C = 11200 N), iii) No. 6006 (C =13300 N).
For required dynamic load carrying capacity, Bearing No. 16006 is suitable at B3 & B4.
Other dimensions of the bearing:
Inner diameter of the bearing = 30 mm, Outer diameter of the bearing =55 mm, Axial width of the bearing = 9 mm.
An experimental set-up for human powered chaff cutter (Fig. 5) was developed, constructed and tested.

Fig. 5. View of the experimental set-up.

Results and discussions

The assembly was checked for its sturdiness and was found to be reliable. There were no vibrations in the set-up. The driver paddles for 1 minute and it was noticed that the flywheel shaft reached a speed of 350 RPM with a gear ratio of 1:2. The paddling was stopped after one minute and the set-up was checked for its free running. The flywheel shaft came to rest after 25 minutes. It proved that the alignment of bearing and other parts of the experimental set-up was satisfactory. Thus now the experimental set-up is ready to perform the desired task.

The developed machine is similar to that reported by Dhale and Modak (2010), wherein they formulated the approximate generalized data based model for oilseed presser using human powered flywheel motor as an energy source. Other human powered machines had been developed such as the one used for lime-fly ash-sand bricks (Modak, 1882), manually driven flywheel motor that operates on wood turning process (Modak and Bapat, 1993) and manually powered apparatus for keyed bricks production (Sohoni, *et al*., 1997).

The output of this research coincides with the research of Deshpande *et al*. (2009), who had confirmed the application of human powered flywheel motor as an energy source for rural generation of electrical energy for rural applications along with computer aided analysis of battery charging process.

Acknowledgements

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