DESIGN AND DEVELOPMENT OF GROUNDNUT OIL EXTRACTING MACHINE BY HUMAN PEDAL FLYWHEEL MOTOR CONCEPT

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Abstract - *A critical appraisal of technologies for oil extraction from oil-bearing agricultural products is presented. Different types of oil- bearing agricultural products are discussed. The products include; groundnut, coconut, sheanut , castor, sunflower, sesame, oil-palm, etc . In India, most of land use for agricultural purpose which produces semi-finished product or goods. Groundnut also one of the agricultural semi-finished goods. Groundnut is grown on small scale farmers in developing countries like India. The average kernel price is approximately twice the price of pod. A research-work for design, fabricate, and performance evaluation of a groundnut oil extracting consisting of feed hopper with a flow rate control device and power system. So working on the above points, we design and fabricate a new medium production capacity machine and today we proudly present this machine called groundnut extracting machine*

Key Words: **Groundnut, Human power machine , Bicycle technology , Screw presser , Flywheel**

1.INTRODUCTION

Oil extraction is the process of recovering oil from oil-bearing agricultural products through manual, mechanical, or chemical extraction. The agricultural products are classified into oil-seeds (cotton, castor, sunflower, etc), nuts (coconut, groundnut, sheanut, etc) and mesocarps or fruits (oil palm). Bicycle technology is one of oldest technology in world, it can be used to transmit power. Current scenario of power supply in rural areas is worse load shedding of 10 to 12hrs daily is experienced so electrical machines are rarely used. groundnuts are important seed used for oil extraction, our state is larger producer of groundnuts but oil extraction in industry is less beneficial for farmer and farmer is unable to extract oil due to limited resources so our aim is to develop an machine which runs on human power and helps easier oil extraction .there are Various Method of Extraction of Oil such as Distillation, Expression, Effleurage, Maceration, Extraction of Oil By Oil Seed Presser. The Human Powered oil seed presser can be used to extract oil from seeds for a small scale quantity. For domestic purposes human mechanical power can be used to extract oil from various seeds at a lower scale.

1.1 Concept

Introducing low cost automation was to overcome problems with the current manual traditional method. The concept of the work is,

(1) Observe the manual methods to identify the important process variables.

(2) Develop a prototype automation system which could control over all of the process.

(3) Produce a specification for a low cost automated system.

(4) Refined design of the machine & fabricate the machine, as this plays a major role in rural area.

The above considering point we design the semi-automated machine which replace manual process.

1.2 *Objective*

The main aim of this project is to overcome the traditional method.

(1) To reduce wastage due to crack or crushed groundnut.

(2) To increase the efficiency.

(3) To reduce the hard work and To reduced time to shell the groundnut.

(4) To develop a low cost machine which can be used by farmer to convert their semi-finished (shell groundnut) into finished product (groundnut).

(5) It satisfies the need of village people to earn more money.

2. WORKING PRINCIPLE

Groundnut Oil Extracting is operated on the shearing action blowering action and separating action. Firstly the inputs i.e. the groundnut are fed to the machine through the hopper. Then groundnuts come in contact with the two members, one is semicircular net and another is roll shaft. Semicircular net is a stationary member while the roll shaft is rotating member. When the groundnut comes in contact with these two members then the shearing action takes place here. Due to shearing action (crushing) the groundnuts gets shelled and divided into two parts. i.e. in the peanut and outer shell of the groundnuts. There clearance is provided between the net and roll shaft. The clearance provided is depends upon the size of the groundnuts which is to be decocted. After shelleing the groundnut the peanut and shells of the groundnut gets dropped from the semicircular net, in downward direction then a centrifugal force is applied by a fan on the peanut and shell of the groundnut. Due to more weight, the peanuts gets moved downward and collected in the separator. But due to lighter weight the shell of the groundnuts are thrown outside the machine and which are collected from the backside of the machine. From the shelling chamber the unshelled groundnuts also gets dropped in the tray (7% to 10%). This groundnut gets dropped from the clearance made among the grill. The three kinds of the nets can be used with different size of capsule slots, size vise small, medium and large for various size of groundnuts. In this way the "GROUNDNUT OIL EXTRACTING NACHINE" works.

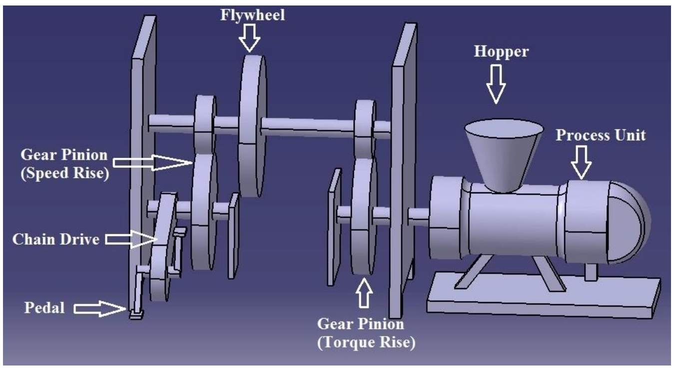

Fig. Modeling of Oil Extracting Machine

2.1 The Screw Press Operation

Continuous pressing by means of expellers (also known as screw press) is a widely applied process for the extraction of oil from oil seeds and nuts. It replaces the historical method for the batch wise extraction of oil by mechanical or hydraulic pressing. The expeller consists of a screw (or worm), rotating inside a cylindrical cage (barrel). The material to be pressed is fed between the screw and barrel and propelled by the rotating screw in a direction parallel to the axis. The configuration of the screw and its shaft is such that the material is progressively compressed as it moves on, towards the discharge end of the cylinder. The compression effect can be achieved, for example by decreasing the clearance between the screw shaft and the cage (progressive or step-wise increase of the shaft diameter) or by reducing the length of the screw flight in the direction of the axial movement. The gradually increasing pressure releases the oil which flows out of the press through the slots provided on the periphery of the barrel, while the press cake continues to move in the direction of the shaft, towards a discharge gate installed at the other extremity of the machine

2.2 Modeling and Fabrication

After that design complete semi-automize machine, then regarding development done on shelling machine. Parameters will be selected according to objectives. 3D diagrams & photos of each components and assembled machine and line diagrams with labeling. Main objectives of this project were to develop the first prototype of an easy to use, low priced and efficient ground nut decorticator and test its performance. Taking leads from previous researchers following design constraints were established. Design should be easy to maintain and should not require highly skilled labor, which is difficult to befound in rural areas.Design should be based on easily available material in rural areas.Manufacturing process should be simple and based on locally available machines in rural areas. The various instruments used for fabrication of machine. Following are the main components of machine:-

(1) Hopper
(2) Flywheel
(3) Shaft
(4) Bearing
(5) Chain Drive
(6) Gear Pinion
(7)Process Unit

2.3 Advantage

1. Easy in operation
2. Simple in construction
3. Light weight
4. Easy to maintain
5. Portable

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 of oil expeller by human power flywheel motor are as follows.

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

The total power requirement of the machine is the sum of the power to drive the process unit (P_D) and power to extract the oil (P_E) [11], so

$$\boldsymbol{P_T = P_D + P_E}$$

Now to determine this,

Power to drive the process unit(P_D) = $T_D \times \omega_D$

But, $T_D = W_D \times R_D$

Where,

W_D = Weight of the pressing Screw=2.12×9.81=20.7972N

R_D = radius of the pressing screw= 0.022m

ω_D = Angular velocity of pressing screw $= \frac{2\pi \times 190.8}{60} = 19.9805\ rad/sec$

So,

$$T_D = 20.7972 \times 0.022 = 0.4575 \cong 0.46Nm$$

And,

$$P_D = 0.46 \times 19.9805 = 9.1416\ watt$$

Now, power to extract oil

i) Average surface area of a groundnut seed = $0.000121m^2$
ii) Force required to crush a groundnut seed = 46.2N

So, stress (τ) in the seed can be given

$$\tau = \frac{Force}{Area} = \frac{46.2}{0.000121} = 381818.1818\ N/m^2$$

Now, power to extract oil is defined by,

$$P_E = T_s \times \omega_n$$

$$T_S = \frac{\pi \times {D_O}^3 \times \tau}{16}$$

Where,

P_E=power required to extract oil, Watt

T_S=Torque of screw in relation with shear stress of groundnut

D_O= Major diameter of screw (0.044m)

$$T_S = \frac{\pi \times 0.044^3 \times 381818.1818}{16} = 6.3829Nm$$

And,

$$P_E = T_S \times 19.98 = 6.3829 \times 19.98 = 127.53Watt$$

Now total power required,

P_T = 9.1416+127.53= 136.6716 Watt= 0.183205 hp

Now, taking factor of safety in consideration, when the seeds fed to the screw are more and the angular velocity variation of screw,

We assume factor of safety as 2

So the Total power = 2×0.183205= 0.3664 hp

Thus from above calculations, it is somehow confirmed that the demand power for oil expeller is equal to 0.3664hp. 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 overcome these torques and overload conditions it may further demand 0.1335 hp. Therefore the total power which is to be supplied to the machine will be approximately 0.5 hp.

3.2 Chain Drive 1 Design

In present case,

Rated power =0.5hp
Speed of pinion = N_p= 275 rpm
Teeth on pinion = 16
Centre distance = 580mm
From Table XIV-1

1) **Design power, P_d**

$P_d = P_r * k_1$ ……………………… (*Referring data book*)

Since, K_1=1.4, for heavy shocks, so…………….. (*Referring data book*)

= 0.5*1.4

= 0.7hp = 522.2 watt = 0.522KW

For smaller sprockets running at 275rpm and P_d=0.5222KW select a single strand,
Chain No. = 40
Pitch in mm = 12.7

2) **To find pitch circle diameter**

PCD of smaller sprocket

$D_p = \left(\frac{P}{\sin(180/T_p)}\right) = \frac{12.7}{\sin(180/16)}$ = 65mm…………………………. (*Referring data book*)

Now find the teeth of bigger sprocket,

V.R = 2.75

$$V.R = \frac{N.P}{N.G}$$

$$2.75 = \frac{275}{N.G}$$

N.G. = 100

P.C.D of bigger sprocket

$$D_g = \left(\frac{P}{\sin(180/T_g)}\right) = \frac{12.7}{\sin(180/44)}$$

= 178.02

$$T_g = T_p * \frac{N_p}{N_g}$$

$$= 16 * \frac{275}{100}$$

$$T_g = 44$$

3) **now find pith line velocity,**

$$V_p = \frac{\pi * D_p * N_p}{60*100} = \frac{\pi*65*275}{60*1000}$$
$$= 0.9359 = 0.9359 * 60 = 56.155\ rad/sec$$

4) **Now to find out power capacity/strand**

$$P = p^2 * \left[\left(\frac{V_p}{104}\right) - \left(\frac{V_p^{1.41}}{526}\right) * \left(26 - 25\cos\left(\frac{180}{16}\right) * K_c\right)\right]$$

……………… (*Referring data book*)

$$P = 12.7^2 * \left[\left(\frac{0.935}{104}\right) - \left(\frac{0.935^{1.41}}{526}\right) * \left(26 - 25\cos\left(\frac{180}{16}\right) * 10^3\right)\right]$$

$$P = 0.6536 * 10^3 Watt$$

$$P = 0.6536\ KW$$

No. of strands on sprocket= $\frac{P_d}{Power\ capacity/Strand}$

$$= \frac{0.5222}{0.6536} = 0.85 \cong 1$$

5) **Length of the chain in pitches**

$L_p = \left[\left(\frac{T_g+T_p}{2}\right) + \left(\frac{2C}{p}\right) + \left(\frac{p*(T_g-T_p)^2}{40C}\right)\right]$……… (*Referring data book*)

For assuming the C.D. we must assume, Θ_{min} to be 120° or $\frac{2}{3}\pi\ rad$

So,

Recommended C_{min} = Dia of larger sprocket + 1/2 (Dia of smaller sprocket)

$$= 178 + \frac{1}{2} * 65$$

$$= 210.5\ mm$$

But we know that C.D. is 580 mm.

$$L_p = \left[\left(\frac{T_g + T_p}{2}\right) + \left(\frac{2C}{p}\right) + \left(\frac{p * (T_g - T_p)^2}{40C}\right)\right] = 121.35 \cong 122$$

6) **Recommended wear load, F_w**

$F_w = 0.35 * p^2 = 0.35 * 12.7^2 = 56.45$........ (*Referring data book*)

7) **Maximum permissible bore**

$d < \left(\frac{T_p - 5}{4}\right) * p$ .... (*Referring data book*)

$$d < \left(\frac{16-5}{4}\right) * 12.7$$

$$d < 34.92$$

Hence design is safe.

8) **Other dimensions of the sprocket**

1) **Outer diameter of the small sprocket**

$D_{op} = P * \left[0.6 + cot\left(\frac{180}{T_p}\right)\right]$ .... (*Referring data book*)

$$D_{op} = 12.7 * \left[0.6 + cot\left(\frac{180}{16}\right)\right] = 71.46mm$$

2) **Outer diameter of the bigger sprocket**

$D_{og} = P * \left[0.6 + cot\left(\frac{180}{T_g}\right)\right]$... (*Referring data book*)

$$D_{op} = 12.7 * \left[0.6 + cot\left(\frac{180}{44}\right)\right] = 185.18mm$$

9) **Roller Chain dimensions,**

Roller diameter = $d_r = \frac{5}{8} * p = 7.93$

Chain Width = w = $\frac{5}{8} * p = 7.93$

Pin dia = $d_p = \frac{5}{16} * p = 3.96$

Thickness of link plate = $\frac{1}{8} * p = 1.58$

Max height of pin link plate = H_p = 0.82*p
= 10.414

Max height of roller link plate = H_r = 0.95*p
= 12.065

10) **Standard roller chain sprocket dimension**

Width of the sprocket for a single strand chain = t_o
= 0.58p – 0.13 =7.216mm

3.3 Chain Drive 2 Design

In present case,

Rated power =0.5hp

Speed of pinion = N_p= 190.8 rpm

Teeth on pinion = 11

Centre distance = 298mm

1) **Design power, P_d**

$P_d = P_r * k_1$................. (*Referring data book*)

Since, K_1=1.4, for heavy shocks, so … (*Referring data book*)

= 0.5*1.4 = 0.7hp = 522.2 watt = 0.522KW

For smaller sprockets running at 275rpm and P_d=0.5222KW select a single strand,

Chain No. = 40

Pitch in mm = 12.7

2) To find pitch circle diameter

PCD of smaller sprocket

$D_p = \left(\frac{P}{\sin(180/T_p)}\right) = \frac{12.7}{\sin(180/11)}$= 45.078mm.................

(*Referring data book*)

Now find the teeth of bigger sprocket,

V.R = 1.727

$V.R = \frac{N.P}{N.G}$

$1.727 = \frac{190.8}{N.G}$

N.G. = 110.4806

P.C.D of bigger sprocket

$D_g = \left(\frac{P}{\sin(180/T_g)}\right) = \frac{12.7}{\sin(180/19)}$ = 77.1592mm

.... (*Referring data book*)

$$T_g = T_p * \frac{N_p}{N_g} = 11 * \frac{190.8}{110.4806} = 18.997 \cong 19$$

3) now find pith line velocity,

$V_p = \frac{\pi * D_p * N_p}{60*100}$ .... (*Referring data book*)

$= \frac{\pi * 45.078 * 190.8}{60*1000}$

$= 0.45039 = 0.45039 * 60 = 27.0204\, rad/sec$

3.4 Design Of Flywheel

In the forthcoming 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

1) Calculate the mass of the flywheel

Material of the flywheel = cast iron (ρ = 7200 kg/m³)

m = ρ*w*h*π*D=7200*0.050*0.050*π*0.350 =19.79............ (*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, $I = 21 \times 0.175^2 = 0.643125\ kgm^2$

2) Calculate the K.E.

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

Here, since N= 275, so

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

So,

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

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

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

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, $h = 0.03641\ m = 364\ mm$

And

$$b = 2h = 0.0728365\ m = 72\ mm$$

Now find out stress in the flywheel,

For that assume $V_s < 1600\ m/min$

$$V_s = \frac{\pi D_0 N}{60} = \frac{\pi \times 0.350 \times 275}{60} = 5.039\ m/sec$$ ....

(*Referring data book*)

4) **Calculate Stresses**

Centrifugal stress, $\sigma_1 = \rho \times V_s^2 = 7200 \times 5.039^2 = 0.18282 \times 10^6 N/m^2 = 0.18282 N/mm^2$

5) **Flywheel construction detail,**

a) Hub diameter,

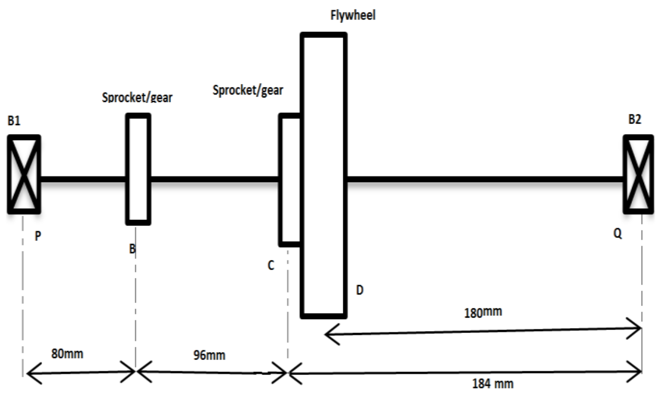
$$D_h = 2 \times d_s = 2 \times 25 = 50\ mm$$

b) Hub length

$$L_h = 2 \times d_s = 2 \times 25 = 50mm$$

3.5 Desigh 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= 0.445 kg = 4.365 N
Wt of flywheel = 21 kg =206.01 N
Φ =20
Power received = 0.5hp = 0.5x746 = 343 Watt
F_t = Tangential force
F_r = Radial force



1) Design Torque

$$T_d = \frac{60 \times p \times k}{2 \times \pi \times N} = \frac{60 \times 343 \times 2}{3 \times 3.14 \times 275} = 23.821\ Nmm$$ ....

(*Referring data book*)

2) Calculation of force due to chain tension

$$F_{t1} = \left(\frac{Torque}{Pitch\ radius\ of\ sprocket}\right) = \frac{23.821}{32.5} = 732.9\ N$$

Resolving force F_{t1V} and F_{t1H}

$$F_{t1V} = F_{t1}\cos(80) = 732.9\cos(80) = 127.2667N$$

$$F_{t1H} = F_{t1}\sin(80) = 732.9\sin(80) = 721.76N$$

$$F_{r1} = F_{t1}\tan\varphi = 732.9\tan(20) = 266.753N$$

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

$$F_{r1V} = F_{r1}\cos(80) = 266.753\cos(80) = 46.321N$$

$$F_{r1H} = F_{r1}\sin(80) = 266.753\sin(80) = 262.7004\ N$$

Forces due to another sprocket gear

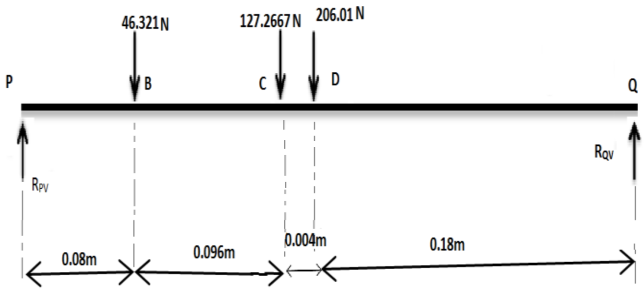
$$F_{t2} = \frac{Torque}{PCD/2} = \frac{23.821\times10^3}{24.5} = 972.2857\ N$$

$$F_{r2} = F_{t2}\tan\varphi = 972.2857\tan(20) = 353.883\ N$$

Calculate the moment due to flywheel

$$M = \frac{WL}{8} = \frac{196.2\times220}{8} = 5395.5\ Nmm$$

Virtical Load Diagram



Vertical load diagram of shaft 1

Taking Moment at P new get,

$$R_{QV}\times0.36 = 206.01\times0.18 + 127.2667\times0.176 + 46.321\times0.08$$

$$R_{QV}\times0.36 = 63.18641$$

$$R_{QV} = 175.5183\ N$$

Now taking moment at Q we get,

$$R_{PV}\times0.36 = 46.321\times0.28 + 127.2667\times0.184 + 206.01\times0.18$$

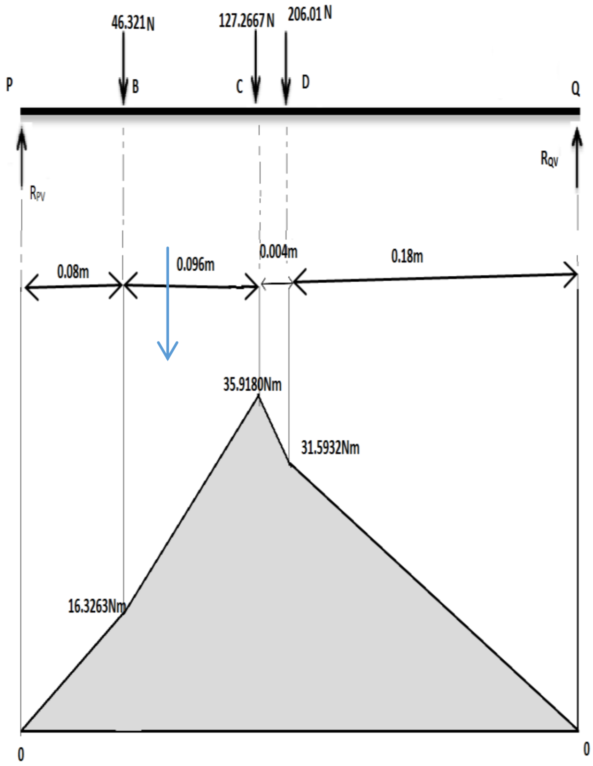
$$R_{PV}\times0.36 = 73.4687$$

$$R_{PV} = 204.07986\ N$$

BM at B= M_{BV}=204.0798×0.08=16.3263Nm

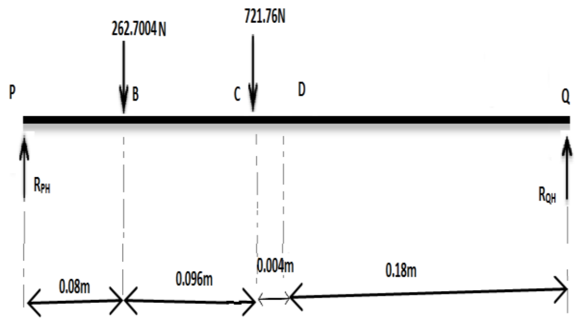
BM at C = M_{CV} = 204.0798×0.176 = 35.9180Nm

BM at D = M_{DV} = 175.5183×0.18=31.5932Nm



B.M.D.1 of shaft 1

Horizontal Load Diagram



horizontal load diagram of Shaft 1

$R_{PH}+R_{QH}=262.7004+721.76=984.4604N$

Taking moment about P, we get

$R_{QH} \times 0.36 = 262.7004 \times 0.08 + 721.76 \times 0.176$

$R_{QH} = 411.2382N$

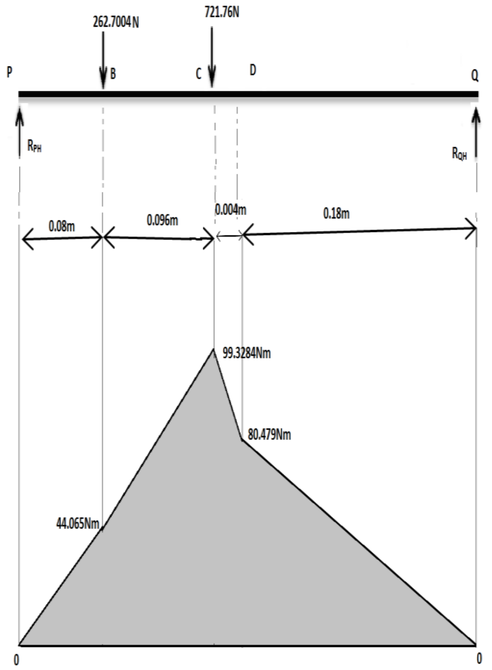
Now, taking moment about Q,

$R_{PH} \times 0.36 = 262.7004 \times 0.28 + 721.76 \times 0.18$

B.M.D.2 of shaft 1

$R_{PH} = 565.2025N$

We know BM at p and Q, $M_{PH}=M_{QH}=0$



BM at B =

$M_{BH}=565.2025\times0.08=45.2162Nm$

BM at C = M_{CH} = 565.2025×0.176 = 92.61164Nm

BM at D = M_{DH} = 411.2382×0.18 = 74.0228Nm

Resultant BM at B $=\sqrt{16.32^2 + 45.21^2} = 48.065Nm$

Resultant BM at C $=\sqrt{35.91^2 + 92.61^2} = 99.3284Nm$

Resultant BM at D $=\sqrt{31.59^2 + 74.02^2} = 80.479Nm$

Equivalent twisting moment (T_e),

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

$= \sqrt{(1 \times 99.3284)^2 + (1 \times 23821)^2}$... (*Referring data book*)

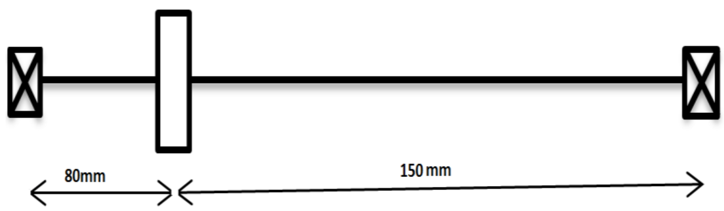
Since,

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

So, we get $d = 23.76 \cong 25mm$

3.6 Design Of Shaft 2

$$\sqrt{(1 \times 99.3284)^2 + (1 \times 23821)^2} = \frac{\pi}{16} \times 42 \times d^3$$



shaft 2

Torque Transmitted by the shaft,

$T_d = \frac{60 \times P \times k}{2\pi \times N}$................. (*Referring data book*)

$= \frac{60\times343\times2}{2\times3.14\times190.8} = 34.33\ Nm = 34.33 \times 10^3\ Nmm$

Tangential force on gear,

$$F_t = \frac{2T}{d} = \frac{2 \times 34.33 \times 10^3}{150} = 457.73\ N$$

Normal load acting,

$$W = \frac{F}{\cos\alpha} = \frac{457.733}{\cos 20} = 487.109\ N$$

$$M = \frac{W \times a \times b}{150} = \frac{487.109 \times 80 \times 70}{150} = 18185.40\ Nmm$$

$$T_e = \sqrt{(M)^2 + (T)^2} = \sqrt{18185.4^2 + 34330^2} = 38849.16\ Nmm$$

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

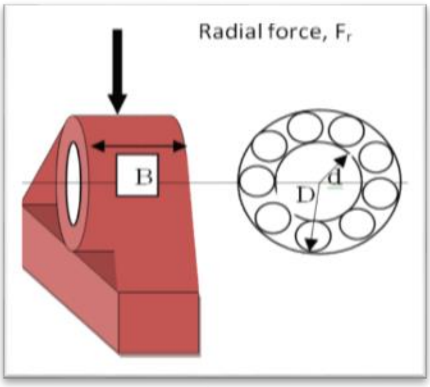
$$38849.16 = \frac{\pi}{16} \times 72 \times d^3$$

So, $d = 14.006 \cong 15mm$

3.7 Design Of Antifriction Bearing

There are two antifriction bearings B1 and B2 used in the experimental setup. The maximum reaction developed at bearing B2 i.e. R_{PH} = 565.2025 N

is considered for designing the bearing.



Schematic diagram of antifriction bearing

1. Equivalent load coming on bearing, Fe, N

Fe = (XF_r+ YF_a) $K_sK_oK_pK_r$ ....(Fr = radial load, NFa = axial load, N)

F_r= 565.2025 N

F_a= 0 N

e = F_a/ F_r

e = 0

selecting self-aligning ball bearing

X = 1, Y = 2.3 ....(X, Y = constants)

Kp = 1 (no preloaded bearing)

Kr = 1(outer race fixed inner race rotating).

Ks = 2 (moderate shock load)

Fe = (XF_r+ YF_a) $K_sK_oK_pK_r$ ....(Kp= preloading factor,Kr= rotational factor,Ks= service factor.)

= (1x 565.2025 + 0) x 1 x 1 x 1 x 2

= 1130.405 N

2. Life of bearing, L (million revolution)

L = $(C/Fe)^n$ Kret.

(C = dynamic load capacity;Kret = reliability factor;D = outer race diameter;d = inner race diameter;B = bearing width.)

L = 500 (demonstration model)

n = 3 for ball bearing

Kret = 1 (reliability = 90%)

C =$(500)^{(1/3)}$ x Fe

C = 4486.015215 N

Selecting series 02xx (C = 11000)

Dimension d = 25 mm,

D = 52 mm,

B = 15 mm

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