AGE 401: Design of Agric. and Food Processing Machines I (3 Units)

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Synopsis:

- Design and analysis of individual machine components – shafts, gears, chains, linkages, bearings, keys, keyways, belts, clutches, etc. Component assemblies and machine systems.
- Design project.

The Power Transmission Shaft

The shaft is design based on strength and rigidity criteria.

A. Strength Criterion

The required diameter for a solid shaft having combined bending and torsional loads is obtained from ASME code equation (Hall, et al. 1980) as

$$D^3 = \underline{16}\ \sqrt{(K_bM_b)^2 + (K_tM_t)^2}\ / \pi S_s$$

Where, at the section under consideration :

S_s = Allowable combined shear stress for bending and torsion

= 40MPa for steel shaft with keyway.

K_b= Combined shock and fatigue factor applied to bending moment

= 1.5 to 2.0 for minor shock.

K_t = Combined shock and fatigue factor applied to torsional moment

= 1.0 to 1.5 for minor shock.

M_b = Bending moment (Nm)

M_t = Torsional moment (Nm) = 55.59Nm (section 4.6.1.5)

D = Diameter of solid shaft (m).

The bending load is due to the weight of the pulley, the summation of tensions on the belts acting vertically downward, and the weight of the threaded shaft.

The shaft is supported at point A and C by two bearings. The reactions R_A and R_c at the two supports are determined as follows :

$R_A + R_c = W_s + (T_1 + T_2) + W_p$ ....

where : W_s = weight of threaded shaft =50N

$T_1 + T_2$ = sum of tensions on vertical belts = 1030N

W_p = weight of pulley = 50N

$R_A + R_c = 50 + 2144 + 50$

$R_A + R_c = 2244$

Taking moment about A,

$R_c (0.485) = 50(0.3025) + 2194(0.605)$

$R_c = 1343N$

$R_A = 2244 – 1343 = 902N$

The shear force and bending moment diagrams

The maximum bending moment occurs at B and it is 273Nm.

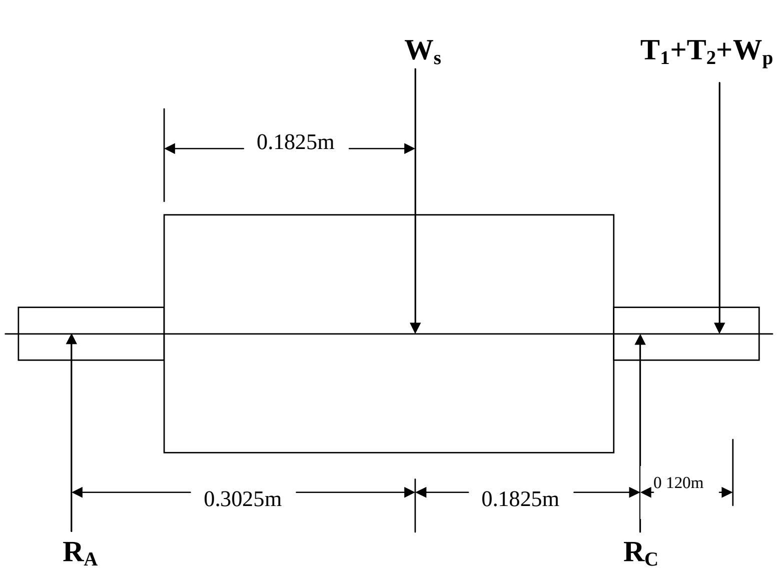


Figure 1 : Bending Loads on the Wormshaft

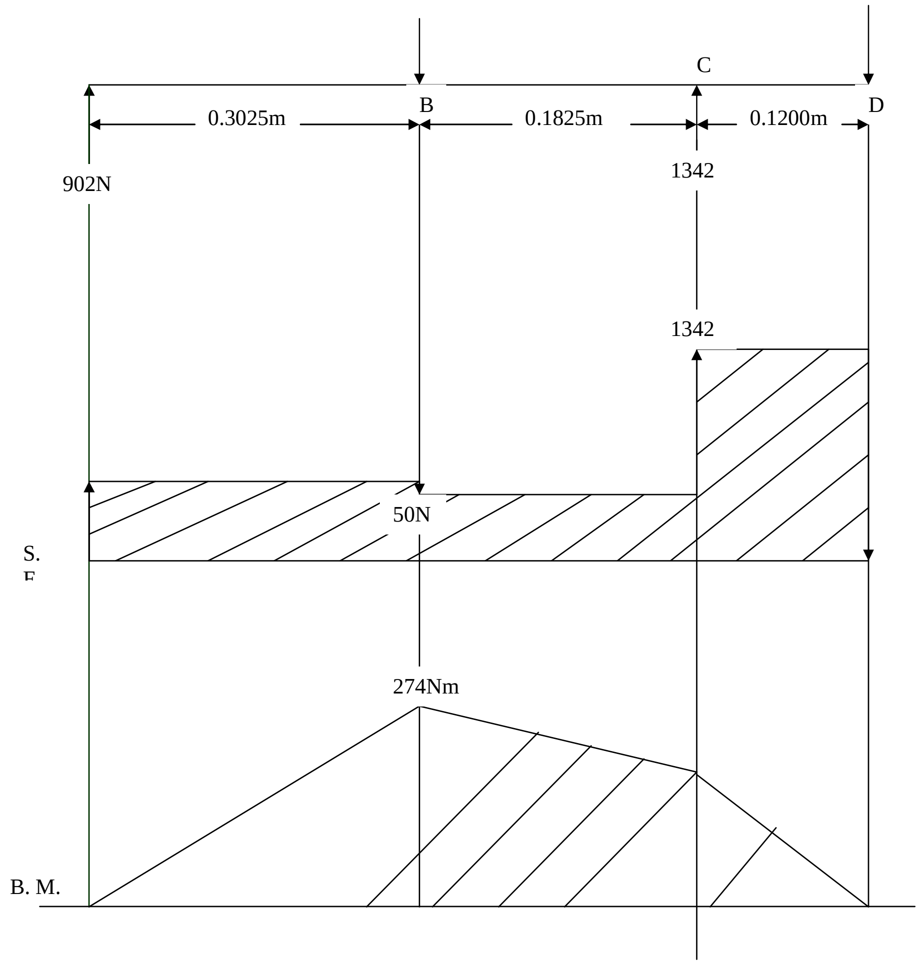


Figure 4.19: Shear Force and Bending Moment Diagrams

From equation, $D^3 = \dfrac{16 \sqrt{(1.0 \times 273)^2 + (1 \times 55.6)^2}}{3.142 \times 40 \times 10^6}$

$= 2.259 \times 10^{-5}$

$D = (2.259 \times 10^{-5})^{1/3}$

$= 28.77mm$

The calculated diameter is less than the least chosen diameter (30mm). Therefore, strength criterion is satisfied.

B. Rigidity Criterion

The design of shaft for torsional rigidity is based on the permissible angle of twist. This is 3deg/m for steel shaft (Hall et. al., op. cit.). For a tapered shaft,

$$\phi = \frac{2TL}{3\pi G} \{(1/D_i^3) - (1/D_o^3)\}$$

where : ϕ = angle of twist (deg)

T = Torsional moment = 55.6Nm

L = Length of tapered section = 0.325m

G = Modulus of rigidity = $80GN/m^2$ for steel shaft

D_i= Inlet diameter of tapered section = 0.0475m

D_0 = Outlet diameter of tapered section = 0.0600m

$$\phi = \frac{2 \times 55.6 \times 0.325 \{(1/0.0475^3) - (1/0.0600^3)\}}{3 \times 3.142 \times 80 \times 10^6}$$

$$= 4.14 \times 10^{-6} deg$$

This is less than the permissible angle of twist (3deg/m). Hence, torsional deflection is satisfied.

EXAMPLE

The solid shaft of a stone crusher is transmitting 1MW at 240 r.p.m. Determine the diameter of the shaft if the maximum torque transmitted exceeds the mean torque by 20%. Take the maximum allowable shear stress as 60MPa.

Given: $P = 1 \text{ MW} = 1 \times 10^6 \text{ W}$;
$N = 240$ r.p.m.;
$T_{max} = 1.2\, T_{mean;}$
$\tau = 60 \text{ MPa} = 60 \text{ N/mm}^2$

Let d = Diameter of the shaft

Mean torque transmitted by the shaft,

$$T_{mean} = \frac{P \times 60}{2\pi N}$$

$$= \frac{1 \times 10^6 \times 60}{2\pi \times 240}$$

$= 39784$N-m

$= 39784 \times 10^3$N-mm

Therefore, maximum torque transmitted,

$$T_{max} = 1.2\, T_{mean}$$

$= 1.2 \times 39784 \times 10^3$

$= 47741 \times 10^3$N-mm

Also, maximum torque transmitted,

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

$$= \frac{\pi}{16} \times 60 \times d^3$$

$= 11.78d^3$

Therefore, 47741×10^3N-mm $= 11.78d^3$

$$d^3 = \frac{47741 \times 10^3 \text{N-mm}}{11.78}$$

$= 4053 \times 10^3$

$d = 159.4$ say 160mm

Belt Design

For a chosen 1hp,180rpm electric gear motor, the belt type is a - B section with dimension 17 x $11mm^2$. The diameter, d =75mm is used at the gear motor shaft. The expeller pulley's diameter,

$$D = \frac{N_m d}{N_e}$$

where,

N_m = Speed of the electric motor = 180rpm

d = Diameter of Driving Pulley = 75mm

N_e = Wormshaft Speed = 45rpm

$$D = \frac{180 \times 75}{45}$$

$$= 300mm$$

The minimum centre distance,

$$C_d = \frac{d + D}{2} + d$$

$$= \frac{75 + 300}{2} + 75$$

$$= 263mm.$$

To take care of the bigger pulley, a – 500mm centre distance is chosen.

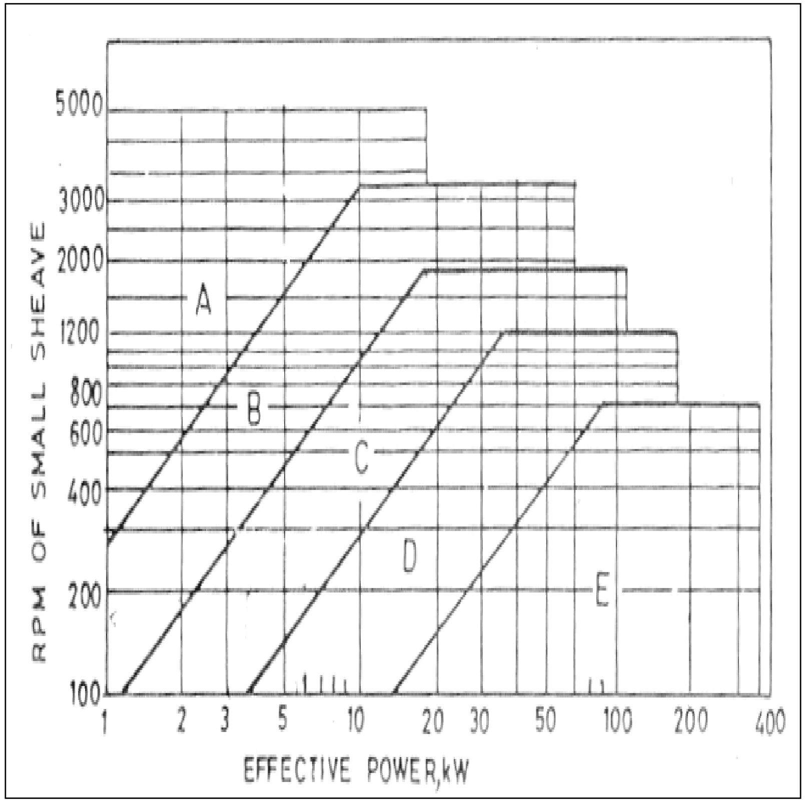
The pitch length of the belt,

$$L = 2c_d + 1.57\,\frac{(d + D)}{2} + \frac{(D - d)^2}{4C_d}$$

$$L = 2 \times 500 + 1.57\,\frac{(75 + 300)}{2} + \frac{(300 - 75)^2}{4 \times 500}$$

$$= 873mm$$

From table 4.1, the nearest standard pitch length is 932.2mm for which the nominal length is 838mm. A – 2 B33 - synchronous (toothed) belt arrangement which combines the characteristics of belts and chains will be used. This will guide against slippage, hence maintaining a constant speed ratio between the driving and the driven shafts.



Source: Mubeen, 1998

Figure 4.16: Effective Power of Belts as a Function of RPM of Small Sheaves

Table 4.1: Standard V – Belts Pitch Lengths

Nominal Length mm (inches)	Standard Pitch Length, mm (inches)		
	A – Section	B – Section	C – Section
660 (26)	696 (27.4)	-------	--------
787 (31)	823 (32.4)	-------	--------
838 (33)	874 (34.4)	-------	--------
889 (35)	925 (36.4)	932.2 (36.7)	--------
965 (38)	1001 (39.4)	1008.4 (39.7)	--------
1067 (42)	1102 (43.4)	1110 (43.7)	--------
1168 (46)	1204 (47.4)	1212 (47.7)	--------
1219 (48)	1252 (49.4)	--------	--------
1295 (51)	1331 (52.4)	1339 (52.7)	1351 (53.2)
1295 (53)	1382 (54.4)	1389 (54.7)	--------
1397 (55)	1433 (56.4)	1440 (56.7)	--------
1524 (60)	1561 (61.4)	1567 (61.7)	1580 (62.2)
1575 (62)	1610 (63.4)	1618 (63.7)	--------
1625 (64)	1661 (65.4)	1669 (65.7)	--------
1727 (68)	1762 (69.4)	1770 (69.7)	1783 (70.2)
1905 (75)	1941 (76.4)	1948 (76.7)	1961 (77.2)
1981 (78)	2017 (79.4)	2024 (79.7)	--------
2032 (80)	2067 (81.4)	--------	--------
2057 (81)	--------	2101 (82.7)	2113 (83.2)
2108 (83)	2144 (84.4)	2151 (84.7)	--------
2159 (85)	2195 (86.4)	2202 (86.7)	2215 (87.2)
2286 (90)	2322 (91.4)	2329 (91.7)	2342 (92.2)
2438 (96)	2474 (97.4)	--------	2499 (98.2)
2464 (97)	2499 (98.4)	2507 (98.7)	--------
2667 (105)	2702 (106.4)	2710 (106.7)	2723 (107.2)
2845 (112)	2880 (113.4)	2888 (113.7)	2901 (114.2)
3048 (120)	3084 (121.4)	3091 (121.7)	3104 (122.2)

Source : Mubeen, 1998.

Determination of Tensions in the Belt

Figure 4.17 shows the belt geometry and according to Hall et.al. 1980, the angle of wrap

$$\alpha = 180 \pm 2\sin^{-1}\{(R - r)/C\}$$

where :

R = radius of the larger pulley =150mm

r = radius of the smaller pulley = 37.5mm

C = centre distance = 500mm

$\therefore$ $\alpha_1 = 180 + 2\sin^{-1}\{(150 – 37.5)/500\}$= 206deg. = 3.6rad

and $\alpha_2 = 180 - 2\sin^{-1}\{(150 – 37.5)/500\}$= 154deg. = 2.7rad.

To obtain T_1 and T_2, the following equations are solved simultaneously :

$$(T_1 - T_2) V = P_\tau$$

and
$$\frac{T_1 - mv^2}{T_2 - mv^2} = e^{\mu\alpha/\sin(\theta/2)}$$

where :

T_1 = tension in the tight side
T_2 = tension in the slack side
m = bte
b = belt width = 17mm

t = belt thickness = 11mm

e = belt density 970kg/m^3 for leather belt

$\therefore$ m = 17 x 11 x 10^{-3} x 970 = 0.18kg/m

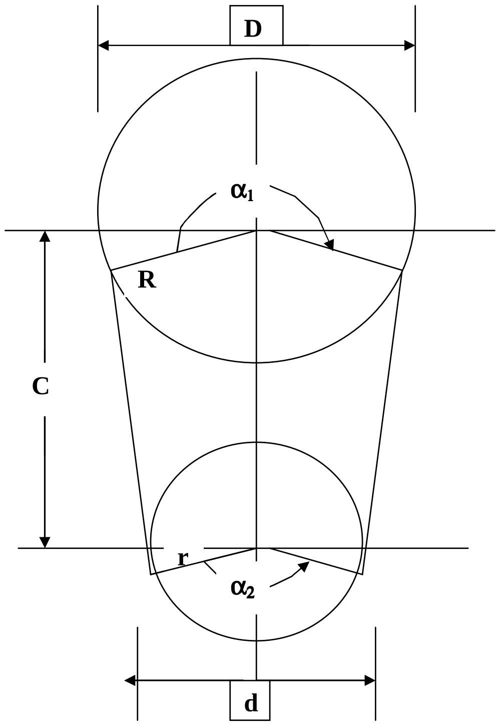


Figure 3. Geometry of Belt Drive

μ = coefficient of friction between belt

= (0.15 for leather belt on steel)

v = belt velocity = r ω = $2\pi r N_m$/ 60 m/s

$$= \frac{2 \times 3.142 \times 37.5 \times 10^{-3} \times 180}{60}$$

= 0.71m/s

θ = 40deg. (most common angle of groove)

For small pulley, $e^{\mu_1 \alpha_1 / \sin(\theta/2)} = e^{0.15 \times 2.7 / \sin 20} = 3.27$

and For big pulley, $e^{\mu_2 \alpha_2 / \sin(\theta/2)} = e^{0.15 \times 3.6 / \sin 20} = 4.80$

The pulley with smaller value governs the design. In this case, the smaller pulley governs the design.

$$\frac{T_1 - mv^2}{T_2 - mv^2} = 3.27$$

$$\frac{T_1 - 0.18 \times 0.716^2}{T_2 - 0.18 \times 0.716^2} = 3.27$$

$T_1 - 0.093 = 3.27\,T_2 - 0.302$

$3.27\,T_2 - T_1 = 0.302 - 0.093$

$3.27\,T_2 - T_1 = 0.209$

But Power (Kw) = $(T_2 - T_1)V$

P = 1Hp = 0.746KW

V = 0.71m/s

$\therefore T_2 - T_1 = 1.042$

$T_1 = 1.042 + T_2$

$3.27 T_2 - (1.402 + T_2) = 0.209$

$3.27 T_2 - T_2) = 0.209 + 1.402$

$T_2 = 0.55KN$

and $T_1 = 1.042 + 0.55 = 1.593KN$

Example

The belt drive of a maize sheller consists of two V – belts in parallel, on grooved pulleys of the same size. The angle of groove is 30^0. The cross – sectional area of each belt is 750 mm^2 and $\mu = 0.12$. The density of the belt material is 1.2 Mg/m^3 and the maximum safe stress in the material is 7 MPa. Calculate the power that can be transmitted between pulleys of 300 mm diameter rotating at 1500 r.p.m.

Given: $n = 2$;

$2\beta = 30^0$ or $\beta = 15^0$

$a = 750\ mm^2 = 750 \times 10^{-6}\ m^2$

$\mu = 0.12$;

$\rho = 1.2\ Mg/m^3 = 1200 kg/\ m^3$

$\sigma = 7\ MPa = 7 \times 10^6\ N/m^{2;}$

$d = 300\ mm = 0.3\ m$

$N = 1500$ r.p.m.

The mass of belt per metre length,

M = area X length X density

$= 750 \times 10^{-6} \times 1 \times 1200$

$= 0.9$ kg/m

Speed of the belt,

$$v = \frac{\pi dN}{60}$$

$$= \frac{\pi \times 0.3 \times 1500}{60} = 23.56 m/s$$

Therefore, Centrifugal Tension,

$T_c = mv^2$

$= 0.9 (23.36)^2 = 500N$

and maximum tension, $T = \sigma \times a$

$= 7 \times 10^6 \times 750 \times 10^{-6} = 5250N$

Tension in the tight side of the belt,

$T_1 = T - T_c$

$= 5250 – 500 = 4750N$

Let T_2 = Tension in the slack side of the belt

Since the pulley are of the same size, angle of lap, $\theta = 180^0 = \pi$ rad

But $2.3 \log T_{1/} T_2 = \mu.\theta \operatorname{cosec} \beta$

$= 0.12 \times \pi \times \operatorname{cosec} 15^0 = 0.377 \times 3.8637 = 1.457$

$\log T_{1/} T_2 = 1.457/2.3$

$= 0.6335$

$T_{1/} T_2 = 4.3$

and $T_2 = T_1/4.3 = 1105N$

Power transmitted,

$P = (T_{1/} - T_2)\ v \times n$

$= (4750 – 1105)\ 23.56 \times 2 = 171750W$

$= 171.75$ kW

DESIGN OF KEYS

A - 45mm diameter rice thresher shaft is made of steel with a yield strength of 400 MPa. A parallel key of size 14 mm wide and 9 mm thick made of steel with a yield strength of 340 MPa is to be used. Find the required length of key, if the shaft is loaded to transmit the maximum permissible torque. Use maximum shear stress theory and assume a factor of safety of 2.

Given: $d = 45$ mm;

σ_{yt} for shaft $= 400\ MPa = 400\ N/mm^{2;}$

$w = 14$ mm;

$t = 9$ mm;

σ_{yt} for key $= 340\ MPa = 340\ N/mm^{2;}$

F.S. = 2

Let l = length of key

According to maximum shear stress theory, the maximum shear stress for the shaft,

$$\tau_{max} = \frac{\sigma_{yt}}{2 \times F.S}$$

$$= \frac{400}{2 \times 2}$$

$= 100\ N/mm^2$

and maximum shear stress for key,

$$\tau_{key} = \frac{\sigma_{yt}}{2 \times F.S}$$

$$= \frac{340}{2 \times 2}$$

$= 85\ N/mm^2$

The maximum torque transmitted by the shaft and key

$$T = \frac{\pi}{16} \times \tau_{max} \times d^3$$

$= \pi \times 100 \times (45)^3$

16

$= 1.8 \times 10^6$ N-mm

Considering failure of key due to shearing, the maximum torque transmitted

$T = l \times w \times \tau_{key} \times d/2$

$= l \times 14 \times 85 \times 45/2$

$= 26775l$

1.8×10^6 N-mm $= 26775l$

Therefore, $l = 67.2$ mm

Considering failure of key due to crushing, the maximum torque transmitted by the shaft and key

$T = l \times t/2 \times \sigma_{ck} \times d/2$

$= l \times 9/2 \times 340/2 \times 45/2$

$= 17213l$

(taking $\sigma_{ck} = \sigma_{yt}/2$)

1.8×10^6 N-mm $= 7213l$

Therefore, $l = 104.6$ mm

Taking the larger of the two values, we have

$l = 104.6$ mm say 105 mm

DESIGN OF BEARINGS

The shaft of a yam extruder rotating at constant speed is subjected to variable load. The bearings supporting the shaft are subjected to stationary equivalent radial load of 3 kN for 10 per cent of time, 2 kN for 20 per cent of time, 1 kN for 30 per cent of time and no load remaining is 20×10^6 revolutions at 95 per cent reliability, calculate dynamic load rating of the ball bearing. For ball bearing, b and k are taking as 1.17 and 3 respectively.

Given: $W_1 = 3$ kN;

$n_1 = 0.1n$;

$W_2 = 2$ kN;

$N_2 = 0.2n$;

$W_3 = 1$ kN;

$N_3 = 0.3n$;

$W_4 = 0$;

$N_4 = (1 - 0.1 – 0.2 – 0.3)\ n = 0.4n$

$L_{95} = 20 \times 10^6$ rev

$b = 1.17$

$k = 3$

Let L_{90} = Life of the bearing corresponding to reliability of 90 per cent.

L_{95} = Life of the bearing corresponding to reliability of 95 per cent.

$= 20 \times 10^6$ revolutions (given)

But $$\frac{L_{95}}{L_{90}} = \frac{(\log_e (1/R_{95}))^{1/b}}{(\log_e (1/R_{90}))^{1/b}}$$

$$= \frac{(\log_e (1/0.95))^{1/1.17}}{(\log_e (1/0.95))^{1/1.17}}$$

$= (0.0513/0.1054)^{0.8547}$

$= 0.54$

$L_{90} = L_{95} / 0.54 = 20 \times 10^6 / 0.54 = 37 \times 10^6$ rev

Equivalent radial load

$$W = \frac{(n_1(W_1)^3 + n_2(W_2)^3 + n_3(W_3)^3 + n_4(W_4)^3)^{1/3}}{(n_1 + n_2 + n_3 + n_4)^{1/3}}$$

$$= \frac{(0.1n \times 3^3 + 0.2n \times 2^3 + 0.3n \times 1^3 + 0.4n \times 0^3)^{1/3}}{(0.1n + 0.2n + 0.3n + 0.4n)^{1/3}}$$

$= (2.7 + 1.6 + 0.3 + 0)^{1/3}$

$= 1.663$ kN

Dynamic load rating,

$C = W (L_{90} /10^6)^{1/k}$

$= 1.663 (37 \times 10^6 /10^6)^{1/3}$

$= 5.54$ kN

DESIGN OF GEARS

A bronze spur pinion rotating at 600 r.p.m. drives a cast iron spur gear of a Coconut oil extractor at a transmission ratio of 4:1. The allowable stastic stresses for the bronze pinion and cast iron gear are 84 MPa and 105 MPa respectively. The pinion has 16 standard 20^0 full depth involute teeth of module 8 mm. The face width of both gears is 90 mm. Find the power that can be transmitted from the standpoint of strength.

Given: $N_p = 600$ r.p.m.;

V.R. $= T_G/T_P = 4$

$\sigma_{0p} = 84$ MPa $= 84$ N/mm^2

$\sigma_{0G} = 105$ MPa $= 105$ N/mm^2

$T_p = 16$

$M = 8$ mm

$b = 90$ mm

Pitch circle diameter of the pinion,

$D_p = m.\ T_p$

$= 8 \times 16 = 128$ mm $= 0.128$ m

Therefore, pitch line velocity,

$$v = \frac{\pi\ D_p\ .\ N_p}{60}$$

$$= \frac{\pi \times 0.128 \times 600}{60} = 4.02 \text{ m/s}$$

Since the pitch line velocity (v) is less than 12.5m/s. therefore velocity factor,

$$C_v = \frac{3}{3 + v} = \frac{3}{3 + 4.02} = 0.427$$

For 200 full depth involute teeth, tooth form factor for the pinion,

$$y_p = 0.154 - \frac{0.912}{T_p}$$

$$= 0.154 - \frac{0.912}{16}$$

$$= 0.097$$

and tooth form for gear

$$y_G = 0.154 - \frac{0.912}{T_G}$$

$$= 0.154 - \frac{0.912}{4 \times 16} \qquad \text{(since } T_G/T_p = 4 \text{ given)}$$

$$= 0.14$$

Therefore,

$$\sigma_{0p} \times y_p = 84 \times 0.097 = 8.148$$

$$\sigma_{0G} \times y_G = 105 \times 0.14 = 14.7$$

Since ($\sigma_{0p} \times y_p$) is less than ($\sigma_{0G} \times y_G$) , therefore the pinion is weaker.

Now using the lewis equation for the pinion, we have tangential load on the tooth (or beam strength of the tooth),

$$W_T = \sigma_{wp}.\, b.\pi.m.y_p = (\sigma_{0p} \times C_v)\, b.\pi.m.y_p$$

$$= 84 \times 0.427 \times 90 \times \pi \times 8 \times 0.097 = 7870\text{N}$$

Therefore, Power that can be transmitted. $P = W_T \times v$

$$= 7870 \times 4.02 = 31640 \text{ W} = 31.64 \text{ kW}$$

CHAIN AND SPROCKET DESIGN

A chain drive is to actuate a compressor from 15 kW electric motor running at 1000 r.p.m., the compressor speed being 350 r.p.m. The compressor operates 16 hours per day.

(i) Calculate the velocity ratio.

(ii) Given that the number of teeth on the smaller sprocket is 25, determine the number of teeth on the larger sprocket.

(iii) Using standard values for service inputs, calculate the design power.

Given: Rated power = 15 kW

N_1 = 1000 r.p.m.;

N_2 = 350 r.p.m.;

T_1 = the number of teeth on the smaller sprocket = 25

Velocity ratio of chain drive

V.R. = N_1/N_2

= 1000/350 = 2.86 say 3

Number of teeth on the larger sprocket or gear

$T_2 = T_1 \times N_1/ N_2$

= 25 X 1000/350 = 71.5

Design power

P = rated power X service factor (K_S)

But $K_S = K_1 \times K_2 \times K_3$

where K_1 = load factor for variable load with heavy shock = 1.5

K_2 = lubrication factor for drop lubrication = 1.0

K_3 = rating factor for 16 hours per day = 1.25

Therefore, K_S = 1.5 X 1 X 1.25 = 1.875

and design power = 15 X 1.875 = 28.125 kW

4.7 Cost Estimation of the Developed Beniseed Oil Plant

The developed plant is made of two major equipment viz: the oil expeller and the oil filter press. The cost of materials for the constuction of these equipment are as shown in tables 4.3 and 4.4.

Table 4.3: Bill of Materials for the Construction of the Designed Oil Expeller

Qty.	Material	Specifications	Rate (#)	Amount (#)
	MECHANICAL COMPONENTS			
6	Angle Iron	One Length, 50mm x 50mm^2	800	4800
1	Galvanized Metal Sheet	240cm x 120cmx 2mm	5200	5200
1	Mild Steel Solid Shaft	100cm long, θ65mm	5000	5000
6	Mild Steel Bar	10mm x 10mm x 1m	400	2400
1	Hollow Pipe	θ80mm x 25mm thick x 50cm long,	1000	1000
1	Driven Pulley	θ75mm Double Groove	500	500
1	Driving Pulley	θ300mm Double Groove	2200	2200
2	Pillow Bearings	θ30mm Inner Bore	3200	6400
2	Leather Belts	B35; V – Type	800	1600
1	Mild Steel Plate	120cm x 60cmx 5mm	5000	5000
1Pkt.	Mild Steel Electrode	Gauge 10	1200	1200
1Pkt.	Mild Steel Electrode	Gauge 12	900	900

Qty	Material	Specifications	Rate (#)	Amount (#)
24	Bolts & Nuts	M10 Hex. (50mm)	25	600
4	Cutting Stones	θ300mm Size	180	180
2	Grinding Stones	θ300mm Size	150	150
2	Hack Saw Blades	300mm Long	120	240
4	Drill Bits	3, 5, 7 & 10mm	110	440
		Sub Total		**#38,440**
	ELECTRICAL COMPONENTS			
1	Electric Gear Motor	3 – Phase, 2Hp @ 180rpm	30000	30000
1	Motor Starter	2 Buttons (ON & OFF)	5000	5000
1	Switch Gear Box	30Amp. (MEM)	5000	5000
15Pcs.	PVC Cables	3 - Core X 6mm X 1m	60	900
		Sub Total		**#40,900**
	Machining of Wormshaft and Barrel			15000
	Fabrication (Bending, Rolling, Shearing)			5000
		TOTAL		**#99,340 ≅ #100,000**

Sample Question

INSTRUCTION: ANSWER **QUESTION ONE AND ANY OTHER THREE TIME**: $2^{1}/_{2}$HOURS

(a) The shaft of a Coconut oil extractor supported at the ends by ball bearings carries a straight tooth spur gear at its mid span and is to transmit 7.5 kW at 300 r.p.m. The pitch circle diameter of the gear is 150 mm. The distances between the centre line of bearings and gear 100mm each. If the shaft is made of steel the allowable shear stress is 45MPa. Determine the diameter of the extracting shaft. Show in a sketch how the gear will be mounted on the shaft; also indicate the ends where the bearings will be mounted? The pressure angle of the gear may be taken as 20^0

A cashew juice hydraulic press exerts a total load of 3.5 MN. This load is carried by two steel rods, supporting the upper head of the press. If the safe stress is 85 MPa and E = 210 kN/mm^2, determine

i. diameter of the rods,
ii. extension in each rod in a length of 2.5m.

The hollow shaft a boom sprayer is required to transmit 600 kW at 110 rpm, the maximum torque being 20% greater than the mean. The shear stress is not to exceed 63 MPa and twist in a length of 3 metres not to exceeed 1.4 degrees. Find the external diameter of the shaft, if the internal diameter to external diameter ratio is 3 : 8. Take modulus of rigidity as 84 GPa.

Design the rectangular key for an Ofada rice thresher shaft of 50mm diameter. The shearing and crushing stresses for the key are 42 MPa and 70 MPa respectively.

The belt drive of a maize sheller consists of two V – belts in parallel, on grooved pulleys of the same size. The angle of groove is 30^0 . The cross – sectional area of each belt is 750 mm^2 and μ = 0.12. The density of the belt material is 1.2 Mg/m^3 and the maximum safe stress in the material is 7 MPa. Calculate the power that can be transmitted between pulleys of 300 mm diameter rotating at 1500 r.p.m. Find also the shaft speed in r.p.m. at which the power transmitted would be a maximum.

The vertical screw of a grain combine harvester with single start square threads of 50 mm mean diameter and 12.5 mm pitch is raised against a load of 10 kN by means of a hand wheel, the boss of which is threaded to act as a nut. The axial load is taken up by a thrust collar which supports the wheel boss and has a mean diameter of 60 mm. The coefficient of friction is 0.15 for the screw and 0.18 for the collar. If the tangential force applied by each hand to the wheel is 100N, find suitable diameter of the hand wheel.