"Design Methodology of Garlic Peeling Machine"

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Abstract

This paper presents the design calculation of garlic peeling machine. Assumptions and references are taken for designing the garlic peeling machine. The design calculation of garlic peeling machine is done. For designing the garlic peeling machine, it is calculate required design parameters such as speed of crank, strokes of garlic pot, design of worm and worm wheel and design of connecting rod.

Keywords: Design of V-Belt, Design of Pulley, Design of Connecting Rod, Design of Worm and Worm Gear

I. INTRODUCTION

Capacity of machine or mass of peeling garlic, M_{PG} = 2 Kg
Mass of garlic pot with cage, M_{GP} = 3 Kg
Stroke length, h = 152 mm = 0.152 m
Taking obliquity ratio, n = 6 = Lc/r
Where, Lc is length of connecting rod
r is length of crank
We know that stroke is double of crank length i.e. Stroke, h = 2r
Therefore crank length, r = h/2 = 152/2 = 76 mm = 0.076 m
Then, length of connecting rod, Lc
$Lc = 6r = 6\times76 = 456$ mm = 0.456 m
Total weight, W_T = Mass of peeling garlic + Mass of garlic pot
$W_T = (2 + 3) \times 9.8 = 49$ N
Force required to break the garlic bulb varies between
F_{Break} = 4.88 N to 6.1 N
Force required to peel the garlic clove varies between
$F_{Peeling}$ = 1.83 N to 3.05 N
Force required to crush the garlic clove varies between
F_{Crush} = 7.32 N to 8.54 N

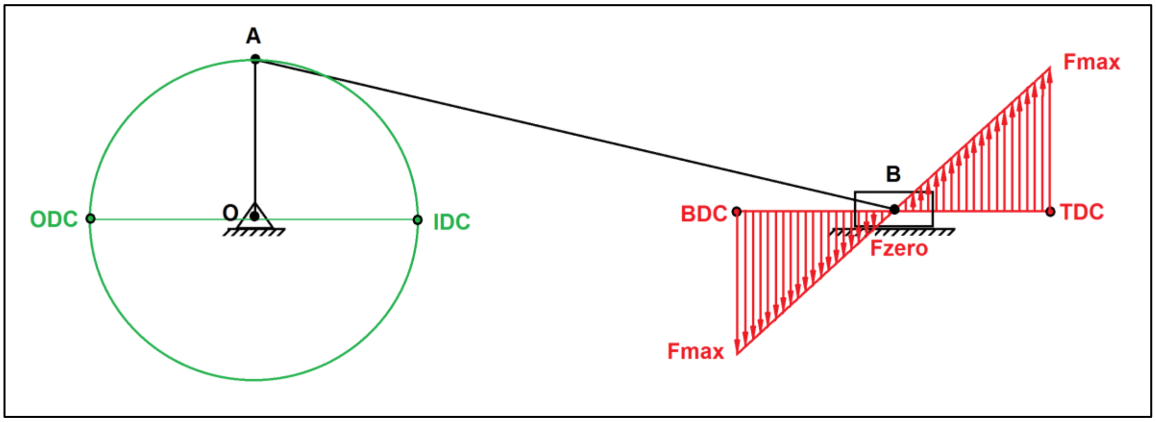

Fig. 1: Slider Crank Mechanism (Force)

In the machine, first impact force will be breaking the garlic bulb and then it will peel the garlic clove.
Stress developed due to impact load, σ_{Impact}

$\sigma_{Impact} = [F/A]\,[1 \pm (1 + (2hAE/FL))^{0.5}]$
Where, F, is a gradual force, N
A, is area of garlic clove, m^2
h, is height of impact, m
E, is young's modulus of garlic, N/m^2
L, is length of garlic clove, m
Therefore,
F = 6.1 N, h = 0.15 m,
L = 3 cm = 0.03 m, d = 15 mm = 0.015 m,
$A = 1.767 \times 10^{-4}\ m^2$, $E = 1.62 \times 10^5\ N/m^2$
$\sigma_{Impact} = 204469.95\ N/m^2$
Therefore, F_{Impact}
$F_{Impact} = \sigma_{Impact} \times A$
$F_{Impact} = 36.13$ N
Total force, F_T
F_T = Total weight + Impact force
$F_T = 49 + 36.13$
$F_T = 85.13$ N
Kinetic energy, KE
KE = Total force x Stroke
KE = 12.94 Nm
Also Kinetic energy $KE = 0.5 \times M_T \times V_{GP}^2$
Where, M_T = Total mass in Kg
$M_T = M_{PG} + M_{GP}$
$M_T = 5$ Kg
And
V_{GP} = Velocity of garlic pot in m/s
$\therefore 12.94 = 0.5 \times 5 \times V_{GP}^2$
$\therefore V_{GP} = 2.275$ m/s
i.e. Linear velocity of garlic pot, $V_{GP} = 2.275$ m/s
Now considering slider crank mechanism at 90^o

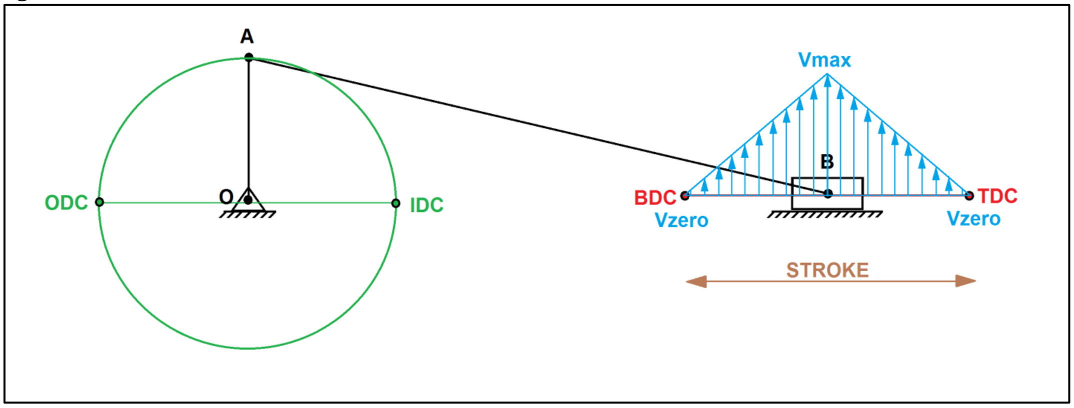

Fig. 2: Slider Crank Mechanism (Velocity)

From figure, when crank at 90^o and 270^o, the point B having maximum linear velocity V_{Max} . When point A at IDC or ODC then point B will be at TDC or BDC respectively and also having zero linear velocity i.e. V_{zero} . Also at this position, the crank velocity and slider velocity will be same i.e. point A and point B having same velocity at 90^o. So we can take,

Velocity of slider = Velocity of crank
$V_{GP} = V_{crank} = 2.275$ m/s
Now using relation $V_{crank} = r \times \omega_{crank}$
Where, V_{crank} is linear velocity of crank, m/s
r is length of crank, m
ω_{crank} is angular velocity of crank, rad/s
$\therefore \omega_{crank} = 29.93$ rad/s
Also, $\omega_{crank} = 2\pi N/60$
$\therefore N = N_{crank} = 285.8$ RPM

II. SPECIFICATION OF ELECTRIC MOTOR

The specifications of electric motor available in market are as follows.
AC electric motor
Power – 1 HP
Rotation – 1440 RPM

III. DESIGN OF V-BELT AND PULLEY

A. Design of V-Belt:

From Table – XV – 9 (DDB)
Design power, $P_d = P_R \times k_l$
Where, P_R is Rated power = 0.746 KW
k_l is load factor
from table – XV – 2 (DDB)
load factor, $k_l = 1.15$
$\therefore$ Design power, $P_d = 0.858$ KW
Selection of belt on the basis of design power i.e. $P_d = 0.858$ KW from design data book.
Form table – XV – 8 (DDB)
Belt designation is – A
Normal width, w = 13 mm
Normal thickness, t = 8 mm
Recommended minimum pulley diameter, D = 75 mm
Taking velocity ratio $V_R = 2.7$
Now by using the velocity ratio
$V_R = N_2/N_1 = D_1/D_2$
$D_1 = V_R \times D_2 = 2.7 \times 75$
$D_1 = 202.5 \text{ mm} \approx 203 \text{ mm}$
Peripheral velocity, V_P
$V_P = \pi D_1 N_1/60$
$V_P = 15.30$ m/s
Centre distance between two pulleys, C
From Table – XV – 10 (DDB)
$C = D_1+D_2$ where, D_1 – Bigger pulley diameter
$C = 203+75$ D_2 – Smaller pulley diameter
$C = 278 \text{ mm} = 0.278 \text{ m}$

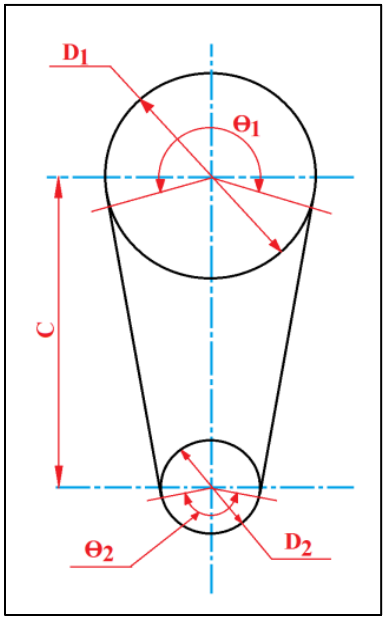

Fig. 3: Details of V Belt and Pulley

Length of V – Belt, L
$L = (\pi/2) \times (D_1+D_2) + 2C + \{[(D_1-D_2)^2]/4C\}$
L = 1050.44 mm = 1.05 m
From table – XV – 1 (DDB)
$\theta = \pi \pm (D_1-D_2)/C$
Where, θ – angle of lap or contact
θ_1 – angle of lap or contact on larger pulley
θ_2 – angle of lap or contact on smaller pulley
$\theta_1 = \pi + (D_1-D_2)/C$
$\theta_1 = 3.6$ rad
$\theta_2 = \pi - (D_1-D_2)/C$
$\theta_2 = 2.68$ rad
From table – XV – 11 (DDB)
For $D_2 = 75$ mm
Angle $\alpha = 34^o$
Power rating per belt, Watt
From table – XV – 9 (DDB)
$(Power/Belt) = (F_W - F_C)\{[e^{(\mu\theta/\sin(\alpha/2))} - 1]/[e^{(\mu\theta/\sin(\alpha/2))}]\}V_P$
Where,
F_W = Working load, N
$F_W = w^2$
$F_W = 169$ N
F_C = Centrifugal Tension, N
$F_C = k_c (V_P/5)^2$
k_c = Centrifugal Tension factor
From Table – XV – 8 (DDB)
$k_c = 2.52$
$F_C = 23.6$ N
θ = Arc of contact on smaller pulley, rad
θ = 2.68 rad
μ = Co-efficient of friction
From Table – XV – 10 (DDB)
μ = 0.3
α = Cone angle = 34^o
∴ (Power/Belt) = 2082.4 W = 2.08 KW
Number of strand, n
Number of strand = Design power, P_d /Power per belt
Number of strand = 0.858 /2.08
Number of strand = 0.4125
Taking number of strand is – 1
Bending load, F_b , N
From Table – XV – 9 (DDB)
$F_b = k_b / D$ Where, k_b – Bending stress factor and
D – Pulley diameter in mm
From table – XV – 8 (DDB)
$K_b = 17.6 \times 10^3$
For larger pulley, D_1
Bending load, $F_b = 86.7$ N
For Smaller pulley, D_2
Bending load, $F_b = 234.67$ N
Taking maximum bending force, F_{bmax}
∴ $F_{bmax} = 234.67$ N
Belt tension ratio
For V – Belt, belt tension ratio
$(T_1/T_2) = e^{(\mu\theta/\sin(\alpha/2))}$
Where, μ = 0.3
θ = 2.68
$\alpha = 34^o$
∴ $T_1/T_2 = 15.64$
Also, From Table – XV – 1 (DDB)

Belt Tension, T_1, T_2, N
$(T_1\text{-}T_2) = P_d/V_P$ where, P_d = Design Power = 858 W
$\therefore (T_1\text{-}T_2) = 56.08$ N V_P = Peripheral Velocity = 15.30 m/s But, $T_1/T_2 = 15.64$
$\therefore T_1 = 15.64\ T_2$
$(15.64\ T_2\text{-}T_2) = 56.08$ N
$\therefore T_2 = 3.83$ N and
$\therefore T_1 = 59.91$ N
From Table – XV – 1 (DDB)
Initial Tension, T_i , N
$2(Ti)^{0.5} = (T1)^{0.5} + (T2)^{0.5}$
$\therefore T_i = 23.51$ N
From Table – XV – 9 (DDB)
Maximum total force, $F_1 = T_i + F_C + F_{bmax}$
$\therefore F_1 = 281.78$ N

B. Smaller Pulley:

From table – XV – 7 (DDB)
Material of pulley – cast iron
Type of construction
Diameter below 150 mm – Web construction
Diameter above 150 mm – Arm construction

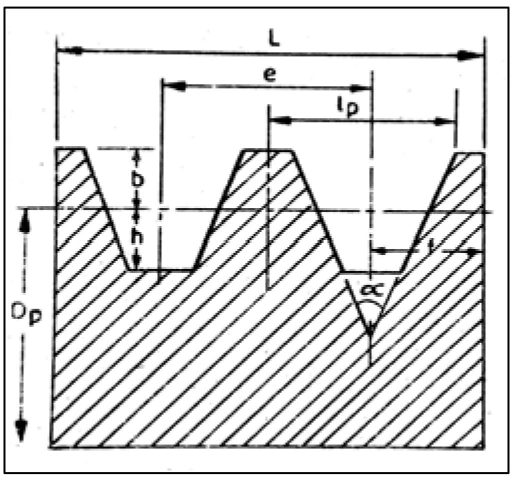

Fig. 4: Details of Pulley

So, smaller pulley having 75 mm diameter, therefore construction will be web construction.
From Table – XV – 11 (DDB)
Groove section – A

l_p – 11 mm
b – 3.3 mm
h – 8.7 mm
e – 15±0.3 mm
f – 9 To 12 ≈ 10.5 mm
D_P – 75 mm
α – 34^o

Width of pulley, w
$w = (n\text{-}1)\ e + 2f$ where, n – number of belt = 1
$\therefore w = 21$ mm

C. Shaft Design for Smaller Pulley:

From Table – XI – 1 (DDB)
Design torque, T_d
$T_d = (60\times P\times K_1)/2\pi N$ Nm
Where, P = Rated power = 746 W

K_l = Load factor
From table – XI – 5 (DDB)
$K_l = 1.75$
N = Rotation = 3888 RPM
$\therefore T_d = 3.21$ Nm
Taking solid shaft
From table – XI – 1 (DDB)
Maximum stress, τ_{Max}
$\tau_{Max} = (16/\pi D^3)[(K_b \times M)^2 + (K_t \times T_d)^2]^{0.5}$
Where, $\tau_{Max} < 0.30\ S_{yt}$ or
$\tau_{Max} < 0.18\ S_{ut}$
From table – II – 7
For shaft selecting material SAE 1030
$S_{ut} = 527$ MPa
$S_{yt} = 296$ MPa
From table – I – 20 (A) (DDB)
Taking factor of safety = 2
$\therefore$ FOS = S_{ut} / Working Stress or
FOS = S_{yt} / Working Stress
$\therefore \tau_{Max} < 0.30$ Working stress or
$\tau_{Max} < 0.30\ S_{yt} / 2 = 44.4\ N/mm^2$ or
$\tau_{Max} < 0.18\ S_{ut} / 2 = 47.43\ N/mm^2$
Taking minimum stress i.e.
$\tau_{Max} = 44.4\ N/mm^2$
Weight of smaller pulley
Mass of pulley = Density × Volume
$M = \rho AL$
For cast iron, $\rho = 7800\ Kg/m^3$
A = Area of pulley = $(\pi/4) \times D_S^2$ D_S = 75 mm = 0.075 m and L = w = 21 mm = 0.021 m
$\therefore A = 4.42 \times 10^{-3}\ m^2$
$\therefore M = 0.7236$ Kg
$\therefore$ Weight of smaller pulley, W_{Small}
$W_{Small} = M \times g$
$W_{Small} = 7.1$ N

Now considering the smaller pulley is mounted on the drive shaft at 75 mm away from fix end and shaft is cantilever type. On the pulley the self acted in downward direction and tension forces acted in horizontal direction.

D. Vertical Plane:

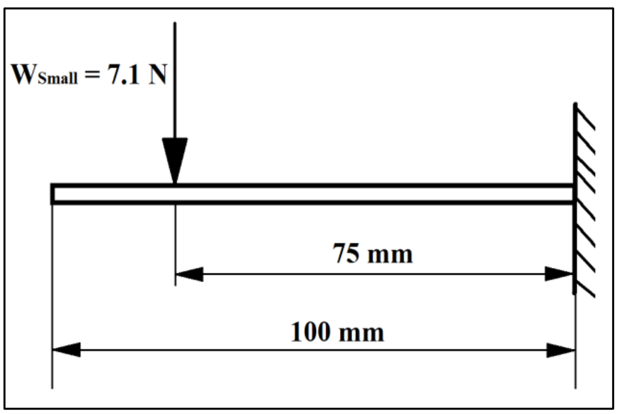

Fig. 5: Force Diagram in Vertical Plane

From Table – I – 2 (DDB)
$M_V = W \times b$ Where, W = 7.1 N and b = 75 mm
$\therefore M_V = 532.5$ Nmm = 0.5325 Nm

E. Horizontal Plane:

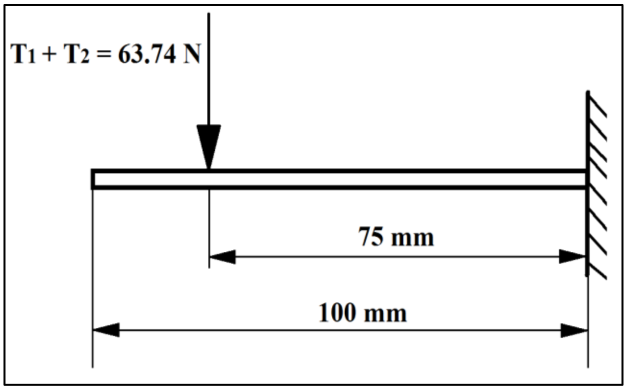

Fig. 6: Force Diagram in Horizontal Plane

From Table – I – 2 (DDB)
$M_H = W \times b$ Where, W = 63.74 N and b = 75 mm
$\therefore M_H = 4780.5$ Nmm = 4.78 Nm
So, Resultant Bending Moment, M_R
$M_R = (M_V^2 + M_H^2)^{0.5}$
$M_R = 4.81$ Nm
Then equivalent torque, Te
$T_e = [(k_b \times M)^2 + (k_t \times T_d)^2]^{0.5}$
From Table – XI – 3 (DDB)
For rotating shaft, suddenly applied load (Heavy shock)
K_b – 2.0 to 3.0 and K_t – 1.5 to 3.0
$\therefore T_e = 17.35$ Nm
$T_e = \tau_{Max} \times (\pi/16) \times D^3$
D = 0.01258 m = 12.58 mm
Also,
Equivalent moment, M_e
$M_e = \{(k_b \times M) + [(k_b \times M)^2 + (k_t \times T)^2]^{0.5}\}/2$
$M_e = 15.89$ Nm
$M_e = \sigma_t \times (\pi/32) \times D^3$ where, $\sigma_t = S_{ut}$ / FOS = 263.5×10^6 N/m^2
D = 0.0085 m = 8.5 mm
So, taking maximum diameter of driven shaft i.e. D = 12.58 mm
From table – XI – 4 (DDB)
Selecting Shaft diameter, D = 14 mm
$\therefore$ Shaft diameter of smaller pulley, D_S = 14 mm
From table – XV – 7 (DDB)
Hub proportions
Diameter of hub, $D_h = 1.5D_s + 25$
Diameter of hub, D_h = 46 mm
Length of hub, $L_h = 1.5D_s$
Length of hub, L_h = 21 mm

F. Larger Pulley:

By using the smaller pulley diameter and velocity ratio, we already have larger pulley diameter.
D_{Larger} = 203 mm
From table – XV – 7 (DDB)
Material of pulley – Cast Iron
Type of Construction – Diameter above 150 mm – Arm construction
Number of arms – 4
Number of set – 1
Hub proportions
Diameter of hub, $D_h = 1.5D_s + 25$
Diameter of hub, D_h = 55 mm

Length of hub, $L_h = 1.5D_s$
Length of hub, L_h = 30 mm

IV. DESIGN OF WORM AND WORM GEAR

In reduction gear box, there is worm and worm wheel is used.
Available rotation at the worm, N_W = 3888 RPM
Required speed of crank, N_{crank} = 285.8 RPM i.e.
Rotation of worm gear, N_G = 285.8 RPM
So, reduction ratio, $VR = N_W/N_G = 13.6$
$\therefore$ Approximate efficiency of worm gear drive, η
$\eta = 1-0.005\times VR$
$\eta = 93.2$ %
Available rated power, P_R = 0.746 KW
Calculated design power, P_d = 0.858 KW
From table – XVI – 16 (DME)
Selecting number of teeth on worm, $t_w = 2$ for VR between 12 – 36
Number of teeth on gear, $t_g = VR \times t_w = 27.2 \approx 29$
Lead angle $\lambda = 6^o$ per worm tooth $= 6\times2 = 12^o$
For compact design $\lambda = \tan^{-1} (Ng/Nw)^{1/3}$
$\lambda = 22.73^o$
Pressure angle, ϕ_n
For lead angle $\lambda = 20^o$ to 25^o, pressure angle $\phi_n = 22.5^o$
Let the module of gear be 'm' mm.
$\therefore$ $D_g = t_g \times m = 29m$ and
$V_P = \pi\times(D_g/1000)\times(N_g/60)$
$V_P = 0.43m$, m/sec
From table – XVI – 15 (DME)
Tooth load, $F_t = P_d/V_P = 1995.35/m$, N
Beam strength by Lewis equation, $F_B = S_O \times C_V \times b \times Y \times m$
Where S_O – Basic strength
From table – XVI – 10 (DME)
Material of construction
Worm – SAE 3120 steel
Gear – Ph bronze SAE 65, $S_O = 84$
$C_V = 0.75$ (trial value)
b – Face width $= 2.38\pi m = 7.47m$
$Y = 0.314+0.0151(\phi_n - 14.5)$
$Y = 0.435$
So, $F_B = 376.27\ m^2$
Now equating F_B to F_t
$376.27\ m^2 = 1995.35/m$
$\therefore$ m = 1.744
From table – XVI – 7 (DME)
Selecting a recommended module, m = 4 mm
Therefore,
$D_g = t_g \times m = 116$ mm
$V_P = 0.43m = 1.72$ m/sec
$b = 2.38\ \pi\ m + 6.25 = 36.16$ mm
$F_t = 1995.35/m = 1995.35/4 = 498.84$ N
$F_B = S_O \times C_V \times b \times Y \times m$
$C_V = 6 / (6+V_P) = 0.777$
$F_B = 4106.56$ N
$\therefore F_B > F_t$ So design is O.K.
To check for wear
$F_d = F_t / C_V = 642.01$ N
From table – XVI – 15 (DME)
Limiting wear strength, $F_W = D_g \times b \times K_2$
From table – XVI – 17 (DME)

$K_2 = 0.70$ MPa
$F_W = 2936.19$ N
$\therefore F_W > F_d$ So design is O.K.
From Table – XVI – 19 (DME)

A. Worm:

Normal pressure angle – $\phi_n = 14.5^o$
Pitch Dia. of worm bored for shaft – $D_W = (2.4\ \pi m) + 27.5 =$
$D_W = 57.66$ mm
Face Length – $L_W = (4.5+0.02t)\ \pi m = 57.05$ mm
Depth of tooth – $h = 0.686\ \pi m = 8.62$ mm

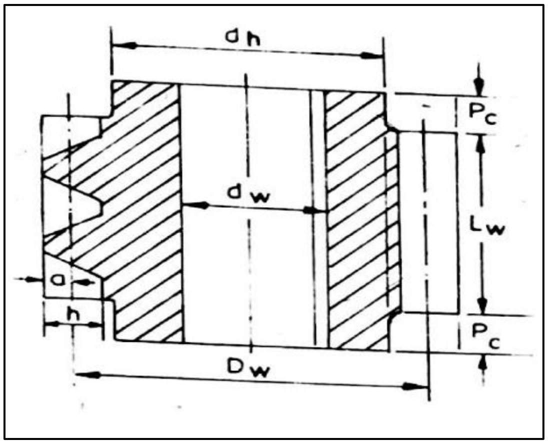

Fig. 7: Details of Worm

Addendum – $a = 0.318\ \pi m = 3.996$ mm
Hub diameter – $d_h = (1.66\ \pi m) + 25 = 45.86$ mm
Minimum bore of shaft – $d_w = \pi m + 16 = 28.56$ mm

B. Gear:

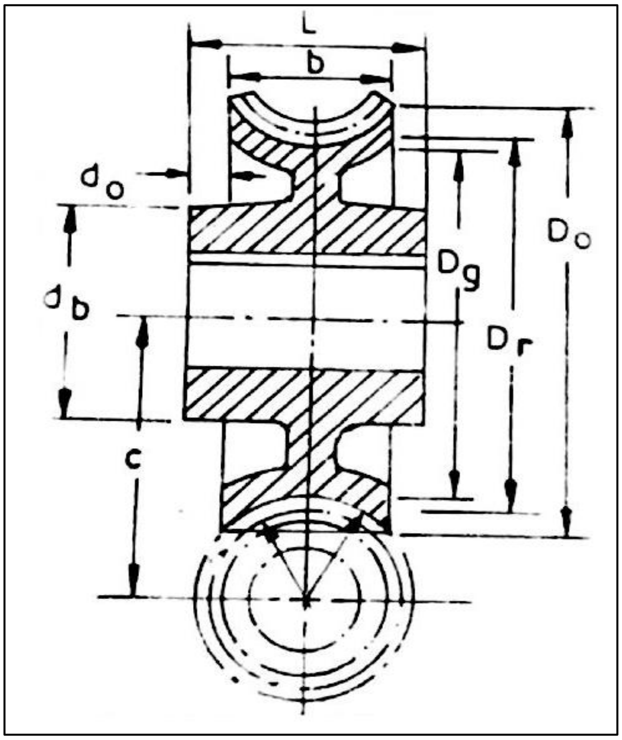

Fig. 8: Details of Worm Gear

Normal pressure angle – $\phi_n = 14.5^o$
Outside diameter – $D_o = D_g + 1.0315\ \pi m = 128.96$ mm
Throat diameter – $D_r = D_g + 1.0636\ \pi m = 129.36$ mm
Face width – $b = 2.38\ \pi\ m + 6.25 = 36.16$ mm
Radius of gear face – $r = 0.882\ \pi m + 13.75 = 24.83$ mm
Radius of gear rim – $r_b = 2.2\ \pi m + 13.75 = 41.39$ mm
Radius of edge – $r_r = 0.25\ \pi m = 3.14$ mm
Hub diameter – $d_b = 1.875\ \pi m = 23.56$ mm

V. DESIGN OF CONNECTING ROD

From table – II – 6 (DDB)
Material for connecting rod is SAE 1040
From table – II – 7 (DDB)
For SAE 1040 – $S_{ut} = 632$ MPa
$S_{yt} = 350$ MPa
Total force generated at the end of connecting rod, $F_T = 85.13$ N
Length of connecting rod, $L_C = 0.456$ m = 456 mm
Length of crank, r = 0.076 m = 76 mm
Obliquity ratio, $L_C/r = n = 6$
Stroke, h = 0.152 m = 152 mm
Design of connecting rod will be done under the critical load, F_{cr}
$\therefore$ Critical Load, $F_{cr} = (S_y \times A_c) / (1+a\times\zeta^2)$
Where, F_{cr} – Critical Load, N
S_y – Yield strength of material, N/mm^2
A_c – Cross sectional area of connecting rod, mm^2
a – Factor depending on material and end fixity. For steel and iron part it is assumed as
a = 1/7500
ζ – Slenderness ratio = L_c/K
where, Lc – Length of connecting rod, mm
K – Radius of gyration, mm

For safety operation, the critical load should be at least 5 to 6 times of the actual load on the connecting rod. Taking cross section of connecting rod is rectangular type.

For rectangular section, from table – I – 1
$I = bt^3/12$
$Z = bt^2/6$
$K = 0.289\ t$
A_C = btand Taking b = 2t
$\therefore I = (2t\times t^3)/ 12 = t^4/6$
$\therefore K = 0.289\ t$
Taking factor of safety is 12
So, critical force = FOS × Total Force
$F_{cr} = FOS \times F_T$
$F_{cr} = 1021.56$ N
Slenderness ratio, $\zeta = L_C/K$
$\zeta = 456/0.289t$
$\zeta = 1577.85/t$
Substituting all values in equation of critical force i.e.
$F_{cr} = (S_{yt}\times A_c) / (1+a\times\zeta^2)$
$1021.56 = (350\times 2t^2) / (1+ (1/7500)\times(1577.85/t)^2)$
$700t^4 - 1021.56t^2 - 339106.84 = 0$
Putting $t^2 = T$
$\therefore 700T^2 - 1021.56T - 339106.84 = 0$
$\therefore T = 22.75 = t^2$
$\therefore t = 4.77\ mm \approx 5$ mm and
$\therefore B = 10$ mm

VI. CONCLUSION

The designs of various parts and parameters are taken into consideration and above values obtained successfully. These values are implemented of fabrication of same machine which is working successfully.

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