Design of Power Transmission System

Semester 5 CEP

Machine Design – II

DHA Suffa University

Department of Mechanical Engineering

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Problem Statement

Design a power transmission system consisting of shafts, gears, belts and bearings to run a blower and a compressor. The blower has a rated power of 12kW and weighs 150 kg while the compressor has a rated power of 15 kW and weighs 100 kg. The compressor is powered by gear mechanism while the blower is powered by a belt drive.

In this exercise you have to take care of the following:

- Select the starting motor
- Design the drive shaft
- Design the driven shafts
- Design the belt drive
- Design the gears
- Select the bearings (optional)

You also have to select the materials for the machine elements you are designing. A schematic drawing of your proposed design is also required.

Component Selection

Components List

- 1x Motor
- 1x Blower
- 1x Compressor
- 2x Gears
- 2x Pulleys
- 1x Belt
- 3x Shafts
- 8x Bearings
- 3x Couplings

Motor

Defined Specifications

- Single Shaft Motor
- Power: 30 KW
- Speed: 900 RPM
- Weight: 339 Kg

Regarding the requirements, selected motor from:

https://www.electrotechdrives.co.uk/product/30kw-three-phase-electric-motor-40hp-6-pole-900rpm-225-frame/

Compressor

Defined Specifications

- Model: OMEDUAL15/12
- Power: 15 kW
- Speed: 1030 RPM
- Weight: 50 Kg

Regarding the requirements, selected compressor from:

https://www.keepital.com/company/my/one-machine-engineering-sdn-bhd/product/air-compressor-10hp-bare-head

Blower

Defined Specifications

- Model: SAMOS SI 1150 E1 (A)
- Power: 16.9 HP (12.6 kW after conversion)
- Speed: 3600 RPM
- Weight: 205 Kg
- Frequency: 60 Hz

Regarding the requirements, selected blower from:

https://www.buschvacuum.com/us/en/products/samos-si-0540-1150-e1.html?product_technical_data=US_60

Shaft

Defined Specifications

Note: No specifications defined

Bearing

Defined Specifications

Note: No specifications defined

Note: Bearings not included in calculations, only used in moment diagrams for positions

Calculation of Belt and Pulley

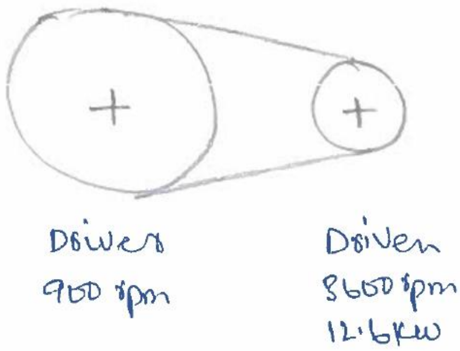
Assumptions

- Horizontal Orientation
- Weight of Pulleys = 0 Kg
- Type of Belt Drives = Heavy Drive (see 18.3)
- Type of Belt = Flat Belt (see 18.4)
- Belt Configuration = Open Belt
- Belt Material = Leather Chrome Tanned (see Table 18.1)
- Belt Density = 1000 kgm^{-3} (see Table 18.1)
- Pulley Material = Cast Iron, Steel (Greasy) (see Table 18.2)
- Friction Coefficient between pulley and Belt = 0.22 (see Table 18.2)
- Allowable Stress for Leather Belt = 2.8 MPa (see URL in References section)
- Distance between Pulleys (x) = 1 m
- Diameter of Pulley 1 = 200 mm
- Overload = 0%
- Slip = 0%
- Friction Loss at Pulley = 0%

Working

$$v = \frac{\pi D N}{60} = \frac{\pi(200\times10^{-3})(3600)}{60}$$
$$= 37.70\ m\,s^{-1}$$

($v > 10\ m\,s^{-1}$
hence, T_c included)



Diameter of pulley ②:

$$v = \frac{\pi D N}{60} = \frac{\pi D(900)}{60} = 37.70$$
$$D_2 = 0.8\,m = 800\,mm$$

∵ Since Open Belt:

$$\alpha = \sin^{-1}\left(\frac{D_1 - D_2}{2x}\right)$$
$$= \sin^{-1}\left(\frac{0.8-0.2}{2(1)}\right) = 0.30469\ rad$$

$$\theta = \pi - 2\alpha = \pi - 2(0.30469)$$
$$= 2.5322\ rad$$

$$P = (T_1 - T_2)v \quad , \quad T_1 = T_2 e^{0.22\theta}$$
$$12.6\times10^3 = (T_1 - T_2)(37.70) \quad , \quad T_1 = T_2 e^{0.22(2.5322)}$$

∵ After simultaneously solving

$$T_1 = 782.5024\ N \quad , \quad T_2 = 448.2770\ N$$

$A = bt$, $T_c = mv^2$, $\rho = \frac{m}{V} = \frac{m}{l} \times \frac{1}{A}$

$$\frac{m}{l} = A\rho$$

$$T_c = \rho A v^2$$

$$\sigma = \frac{F}{A} = \frac{T_1 + T_c}{A} = \frac{T_1 + \rho A v^2}{A}$$

$$= \frac{T_1 + \rho t d v^2}{td}$$

$$2.8 \times 10^6 = \frac{782.80 + [(1000)\, td\, (37.70)^2]}{td}$$

$$d = \frac{782.80}{t\,((2.8 \times 10^6) - (1000 \times 37.70^2))}$$

∵ See thickness and width of belt table in References section. Only 1 selection of $t = 6.5mm$, calculates value of d that lies with range.

$$d = \frac{782.80}{(6.5 \times 10^{-3})((2.8 \times 10^6) - (1000 \times 37.70^2))}$$

$$= 87.32\ mm \approx 90\ mm$$

$$T_c = \rho A v^2 = (1000)(90 \times 10^{-3})(6.5 \times 10^{-3})(37.70)^2$$

$$= 831.45\ N$$

$$T_0 = \frac{T_1 + T_2 + 2T_c}{2} = 1018.022\ N$$

$$L = 0.5\pi (D_1 + D_2) + 2x + \frac{(D_1 - D_2)^2}{4x}$$

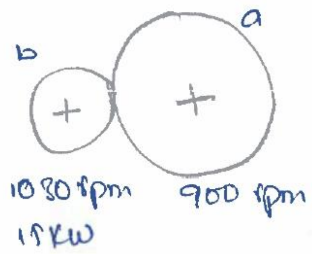
$$= 5.62\ m$$

Calculation of Gear and Pinion

Assumptions

- Horizontal Orientation
- Material of Gear and Pinion = Cast Steel, Untreated (see Table 28.4)
- Allowable Stress = 140 MPa (see Table 28.4)
- Distance between shafts = 300 mm
- Type of Load = Medium Stock (see Table 28.10)
- Type of Service = 8 - 10 hours per day (see Table 28.10)
- Service Factor (Cs) = 1.54 (see Table 28.10)
- System of Gear and Pinion teeth = 20° stub involute system (see 28.11)

Working



$$VS = \frac{Na}{Nb} = \frac{Db}{Da}$$

$$Db = \frac{Na\,Da}{Nb}$$

$$= \frac{900\,Da}{1030} = \frac{90}{103} Da$$

$$0.3 = \frac{Da}{2} + \frac{Db}{2} \quad , \quad \frac{90}{103} Da - Db = 0$$

∴ Solving both equations simultaneously.

$$Da = 0.32\,m \quad , \quad Db = 0.28\,m$$

∴ Design using smaller diameter

$$v = \frac{\pi D N}{60} = \frac{\pi(0.28)(1030)}{60} = 15.10\,m\,s^{-1}$$

∴ Refer to Cv selection image in Reference section

for $v > 12.5\,ms^{-1}$ and $v < 20\,ms^{-1}$

$$Cv = \frac{6}{6+v} = \frac{6}{6+15.10} = 0.28435$$

∴ Refer to y selection image in Reference section

since 20° stub involute system

$$y = 0.175 - \frac{0.841}{T} \qquad \because \quad m = \frac{D}{T}$$

$$= 0.175 - \frac{0.841\,m}{D} \qquad T = \frac{D}{m}$$

$$W_T = \frac{P}{v} \times C_s$$

$$= \frac{15 \times 10^3}{15.10} \times 1.54 = 1529.74\ N$$

$$W_T = \sigma \times C_v \times \pi \times m \times b \times y$$

$$1529.74 = 140 \times 10^6 \times 0.28435 \times \pi \times m \times b \times y$$

$$1.223 \times 10^{-5} = m \times b \times \left(0.175 - \frac{0.841 m}{D}\right)$$

∵ See Module and face width table in Reference section. Only, module = 2.5mm value calculates value of b that lies within range

$$1.223 \times 10^{-5} = (2.5 \times 10^{-3}) \times b \times \left(0.175 - \frac{0.841 \times 2.5 \times 10^{-3}}{0.28}\right)$$

$b = 29.21$ mm

(range verified)

$$T_b = \frac{D_b}{m} = \frac{0.28}{2.5 \times 10^{-3}} = 112 \text{ teeth}$$

Motor shaft gear:

$W_T = 1529.74\,N$, $W_N = 1627.92\,N$

$b = 29.21\,mm$, $m = 2.5\,mm$

$$T_a = \frac{Da}{m} = \frac{0.32}{2.5\times10^{-3}} = 128 \text{ teeth}$$

$$\begin{aligned} W_g \text{ (weight of gear)} &= 0.00118\, T_a\, b m^2 \\ &= 0.00118(128)(29.21)(2.5)^2 \\ &= 27.57\,N \end{aligned}$$

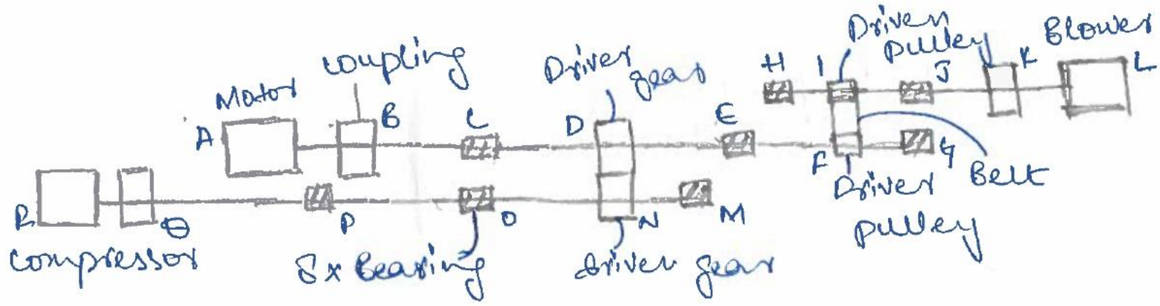
$$\begin{aligned} W_R &= (W_N^2 + W_g^2 + 2(W_N)(W_g)\cos\phi)^{1/2} \\ &= (1627.92^2 + 27.57^2 + 2(1627.92)(27.57)\cos 20)^{1/2} \\ &= 1653.85\,N \end{aligned}$$

Calculation of Shaft

Assumptions

- Motor Shaft Length = 10 m
- Blower Shaft Length = 6 m
- Compressor Shaft Length = 8 m
- Shaft Material = Steel
- Type of Shafts = Solid Shafts
- Allowable Shear Stress = 40 MPa
- Rotating Shafts with Suddenly applied load with minor shocks only = Km (1.5 to 2.0) and Kt (1.5 to 2.0) (see Table 14.2)
- Km = 1.75 (see Table 14.2)
- Kt = 1.75 (see Table 14.2)

Working



Compressor Shaft:

$W_T = 1529.74\,N$, $\phi = 20°$ $\quad \because P = \frac{2\pi N T}{60}$

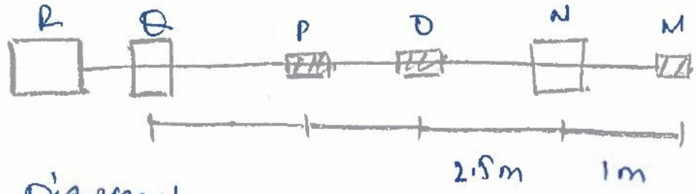
$$W_N = \frac{W_T}{\sin\phi} = \frac{1529.74}{\cos 20} = 1627.92\,N \qquad T = \frac{15\times10^3 \times 60}{2\pi \times 1030}$$

W_g (weight of pinion) $= 0.00118\,T_b\,bm^2$ $\quad = 139.07\,Nm$

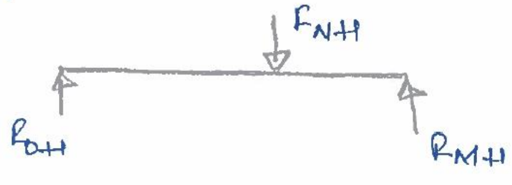
$= 0.00118(112)(29.21)(2.5)^2$

$= 24.13\,N$

$$W_R = (W_N^2 + W_g^2 + 2W_N W_g \cos\phi)^{1/2}$$
$$= (1627.92^2 + 24.13^2 + 2(1627.92)(24.13)\cos(20))^{1/2}$$
$$= 1650.61\,N$$



Horizontal Load Diagram:



$F_{NH} = W_N \sin\phi$

$= 1627.92 \sin 20$

$= 556.78\,N$

$\circlearrowleft + \sum M_{OH} = 0$:

$-F_{NH}(2.5) + R_{MH}(3.5) = 0$

$R_{MH} = 397.70\,N$

$+\uparrow \sum F_i = 0:$

$R_{MH} + R_{OH} = F_{NH}$

$397.70 + R_{OH} = 556.78 \longrightarrow R_{OH} = 159.08\,N$

BM at $N_H = R_{OH} \times 2.5 = 397.70\,N$

$= R_{MH} \times 1 = 397.70\,N$

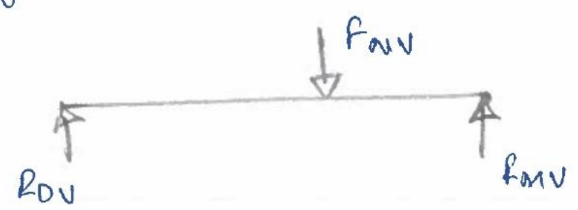
Vertical load Diagram:

$F_{NV} = W_T + W_g$

$= 1529.74$

$+ 24.13$

$= 1553.87\,N$



$\circlearrowleft + \sum M_{OV} = 0:$

$-F_{NV}(2.5) + R_{MV}(3.5) = 0 \longrightarrow R_{MV} = 1109.91\,N$

$+\uparrow \sum F = 0:$

$R_{OV} + R_{MV} = F_{NV}$

$R_{OV} = 1553.87 - 1109.91 = 443.96\,N$

BM at $N_V = R_{MV} \times 1 = 1109.91\,N$

$= R_{OV} \times 2.5 = 1109.91\,N$

$M = (M_{NV}^2 + M_{NH}^2)^{1/2}$

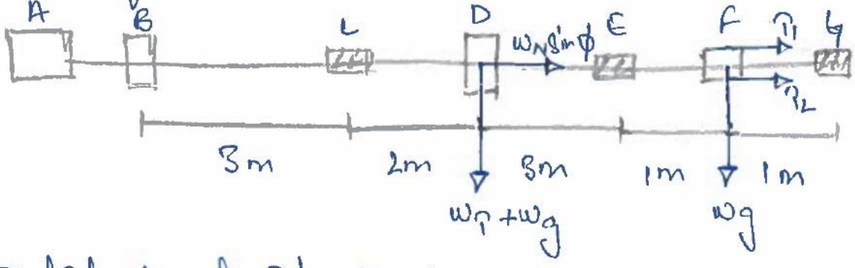
$= (1109.91^2 + 897.70^2)^{1/2} = 1179.01\,Nm$

$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2} = \frac{\pi}{16} \tau d^3$

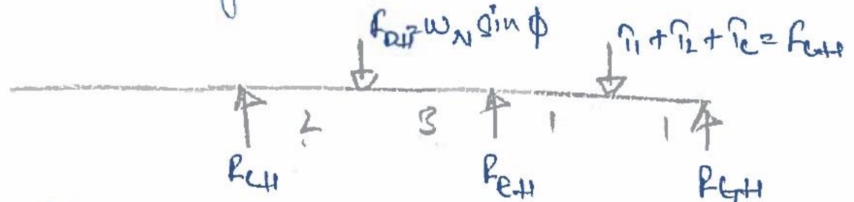
$= ((1.75 \times 1179.01)^2 + (1.75 \times 139.07)^2)^{1/2} = \frac{\pi}{16} \times 40 \times 10^6 \times d^3$

$d = 64.19\,mm \approx 70\,mm$

Motor shaft:



Horizontal load Diagram:



$\circlearrowleft + \sum M_{CH} = 0$:

$-F_{DH}(2) + R_{EH}(5) - F_{CH}(6) + R_{GH}(7) = 0$

$-2F_{DH} + 5R_{EH} - 6F_{CH} + 7R_{GH} = 0$ — (1)

$\because F_{DH} = 1627.92 \sin 20$
$= 556.78\,N$
$F_{CH} = 2062.23\,N$

$\circlearrowleft + \sum M_{EH} = 0$:

$-R_{CH}(5) + F_{DH}(3) - F_{CH}(1) + R_{GH}(1) = 0$

$-5R_{CH} + 3F_{DH} - F_{CH} + R_{GH} = 0$ — (2)

$\circlearrowleft + \sum M_{GH} = 0$:

$-R_{CH}(7) + F_{DH}(5) - R_{EH}(2) + F_{CH}(1) = 0$

$-7R_{CH} + 5F_{DH} - 2R_{EH} + F_{CH} = 0$ — (3)

$5R_{EH} + 7R_{GH} - 13486.94 = 0$ — (1)

$-5R_{CH} + R_{GH} - 391.89 = 0$ — (2)

$-7R_{CH} - 2R_{EH} + 4846.13 = 0$ — (3)

∴ After simultaneously solving.

$R_{CH} = 435.41\,N$, $R_{EH} = -899.13\,N$, $R_{GH} = 2568.90\,N$

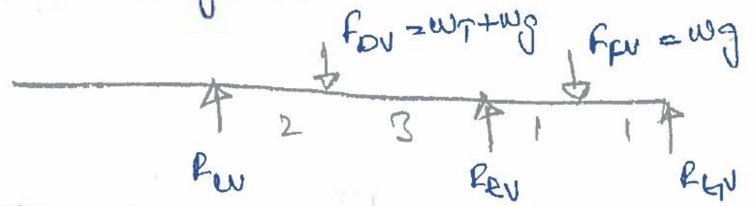
BM at DH $= R_{CH} \times 2 = 870.82\,Nm$

$= R_{EH} \times 3 = 2697.39\,Nm$ (highest)

BM at FH $= R_{EH} \times 1 = 899.13\,Nm$

$= R_{GH} \times 1 = 2568.90\,Nm$ (highest)

Vertical load Diagram:



$F_{DV} = W_T + W_g = 1529.74 + 27.57 = 1557.31\,N$

$F_{FV} = 0\,N$ (weight of pulley) $= W_g$

$\circlearrowright + \Sigma M_{CV} = 0$:

$-F_{DV}(2) + R_{EV}(5) - \overset{0}{F_{FV}}(6) + R_{GV}(7) = 0$

$-2F_{DV} + 5R_{EV} + 7R_{GV} = 0$ —— (4)

$\circlearrowright + \Sigma M_{EV} = 0$:

$-R_{CV}(5) + F_{DV}(3) - \overset{0}{F_{FV}}(1) + R_{GV}(1) = 0$

$-5R_{CV} + 3F_{DV} + R_{GV} = 0$ —— (5)

$\circlearrowright + \Sigma M_{GV} = 0$:

$-R_{CV}(7) + F_{DV}(5) - R_{EV}(2) + \overset{0}{F_{FV}}(1) = 0$

$-7R_{CV} + 5F_{DV} - 2R_{EV} = 0$ —— (6)

$5R_{EV} + 7R_{GV} - 3114.62 = 0$ —— (4)

$-5R_W + R_{GV} + 4671.93 = 0$ —— (5)

$-7R_W - 2R_{EV} + 7786.55 = 0$ —— (6)

∴ After simultaneously solving

$R_W = 934.39\,N,\ R_{EV} = 622.92\,N,\ R_{GV} = 0\,N$

BM at DV $= R_W \times 2 = 1868.78\,N$

$R_{EV} \times 3 = 1868.78\,N$ (highest)

BM at EV $= R_{EV} \times 1 = 622.92\,N$ (highest)

$R_{GV} \times 1 = 0\,N$

$M_D = (M_{DV}^2 + M_{DH}^2)^{1/2} = (1868.78^2 + 2697.89^2)^{1/2}$

$= 3281.50\,Nm$ (highest)

$M_E = (M_{EV}^2 + M_{EH}^2)^{1/2} = (622.92^2 + 2568.90^2)^{1/2}$

$= 2643.35\,Nm$

$P = \dfrac{2\pi N T}{60} \rightarrow T = \dfrac{30 \times 10^3 \times 60}{2\pi \times 900} = 318.31\,Nm$

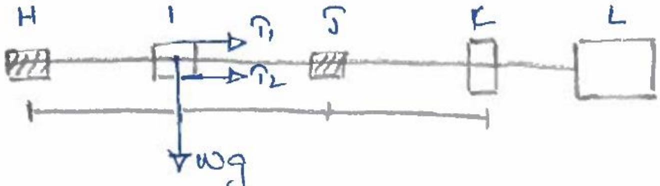
$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2} = \dfrac{\pi}{16} \tau d^3$

$= ((1.75 \times 3281.50)^2 + (1.75 \times 318.31)^2)^{1/2} = \dfrac{\pi}{16} \times 40 \times 10^6 \times d^3$

$d = 90.23\,mm$

$\approx 100\,mm$

Blower shaft:



wg = 0 = weight of pulley

$$M = (T_1 + T_2 + T_c) \times 2$$

$$= (782.50 + 448.28 + 831.45) \times 2$$

$$= 4124.46\ \text{Nm}$$

$$P = \frac{2\pi N T}{60} \longrightarrow T = \frac{12.6 \times 10^3 \times 60}{2\pi \times 3600}$$

$$= 33.42\ \text{Nm}$$

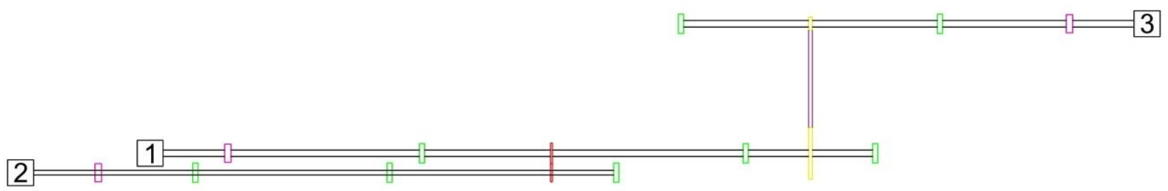
$$T_e = \sqrt{(K_m \times M)^2 + (K_t \times T)^2} = \frac{\pi}{16} \tau d^3$$

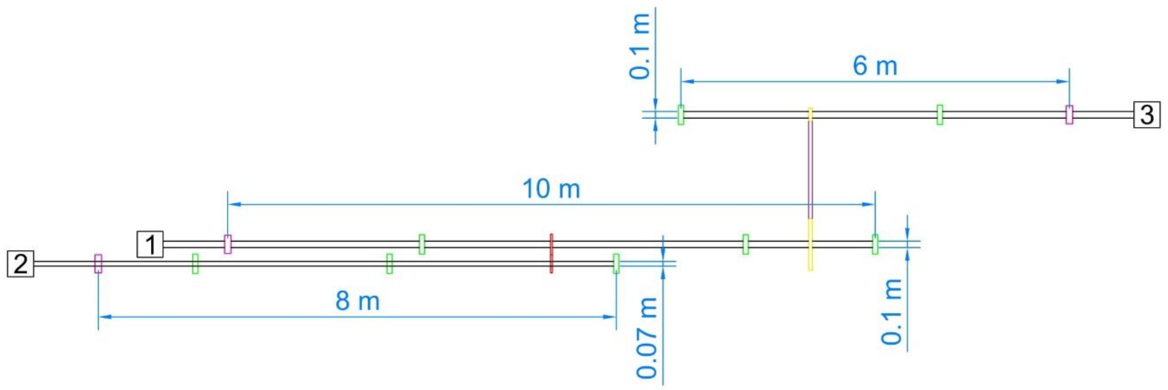
$$= \left((1.75 \times 4124.46)^2 + (1.75 \times 33.42)^2\right)^{1/2} = \frac{\pi}{16} \times 40 \times 10^6 \times d^3$$

$$d = 97.23\ \text{mm}$$

$$\approx 100\ \text{mm}$$

Final Design





Legend	
1	Motor
2	Compressor
3	Blower
Pink	Couplings
Green	Bearings
Red	Gears
Yellow	Pulleys
Purple	Belt
White	Shafts

References

https://www.engineersgallery.com/working-stresses-in-belts-density-of-belt-materials/#:~:text=The%20ultimate%20strength%20of%20leather,give%20a%20reasonable%20belt%20life.

Thickness and Width of Belt Table

Thickness of Belt (t/mm)	Width of Belt (Calculated) (w/mm)	Width Lower Limit (w/mm)	Width Upper Limit (w/mm)	Within Limits
5.0	113.51	35.00	63.00	No
6.5	87.32	50.00	140.00	Yes
8.0	70.94	90.00	224.00	No
10.0	56.76	125.00	40.00	No
12.0	47.30	250.00	600.00	No

Module and Face Width Table

Module (m/mm)	Module (m/m)	Face Width (b/m)	Face Width (b/mm)	b Lower Limit/m	b Lower Limit/mm	b Upper Limit/m	b Upper Limit/mm	Within Limits
1.00	0.001	0.0711	71.10	0.01	9.00	0.02	15.00	No
1.25	0.001	0.0571	57.13	0.01	11.25	0.02	18.75	No
1.50	0.002	0.0478	47.82	0.01	13.50	0.02	22.50	No
2.00	0.002	0.0362	36.18	0.02	18.00	0.03	30.00	No
2.50	0.003	0.0292	29.21	0.02	22.50	0.04	37.50	Yes
3.00	0.003	0.0246	24.56	0.03	27.00	0.05	45.00	No
4.00	0.004	0.0188	18.76	0.04	36.00	0.06	60.00	No
5.00	0.005	0.0153	15.29	0.05	45.00	0.08	75.00	No
6.00	0.006	0.0130	12.98	0.05	54.00	0.09	90.00	No
8.00	0.008	0.0101	10.12	0.07	72.00	0.12	120.00	No
10.00	0.010	0.0084	8.43	0.09	90.00	0.15	150.00	No
12.00	0.012	0.0073	7.33	0.11	108.00	0.18	180.00	No
16.00	0.016	0.0060	6.02	0.14	144.00	0.24	240.00	No
20.00	0.020	0.0053	5.32	0.18	180.00	0.30	300.00	No
25.00	0.025	0.0049	4.89	0.23	225.00	0.38	375.00	No
32.00	0.032	0.0048	4.84	0.29	288.00	0.48	480.00	No
40.00	0.040	0.0056	5.56	0.36	360.00	0.60	600.00	No
50.00	0.050	0.0098	9.78	0.45	450.00	0.75	750.00	No

Table 14.2. Recommended values for K_m and K_t

Nature of load	K_m	K_t
1. Stationary shafts		
(*a*) Gradually applied load	1.0	1.0
(*b*) Suddenly applied load	1.5 to 2.0	1.5 to 2.0
2. Rotating shafts		
(*a*) Gradually applied or steady load	1.5	1.0
(*b*) Suddenly applied load with minor shocks only	1.5 to 2.0	1.5 to 2.0
(*c*) Suddenly applied load with heavy shocks	2.0 to 3.0	1.5 to 3.0

14.5 Standard Sizes of Transmission Shafts

The standard sizes of transmission shafts are :

25 mm to 60 mm with 5 mm steps; 60 mm to 110 mm with 10 mm steps ; 110 mm to 140 mm with 15 mm steps ; and 140 mm to 500 mm with 20 mm steps.

The standard length of the shafts are 5 m, 6 m and 7 m.

Table 18.1. Density of belt materials.

Material of belt	*Mass density in kg / m³*
Leather	1000
Convass	1220
Rubber	1140
Balata	1110
Single woven belt	1170
Double woven belt	1250

Table 18.2. Coefficient of friction between belt and pulley.

Belt material	*Pulley material*						
	Cast iron, steel			*Wood*	*Compressed paper*	*Leather face*	*Rubber face*
	Dry	*Wet*	*Greasy*				
1. Leather oak tanned	0.25	0.2	0.15	0.3	0.33	0.38	0.40
2. Leather chrome tanned	0.35	0.32	0.22	0.4	0.45	0.48	0.50
3. Convass-stitched	0.20	0.15	0.12	0.23	0.25	0.27	0.30
4. Cotton woven	0.22	0.15	0.12	0.25	0.28	0.27	0.30
5. Rubber	0.30	0.18	—	0.32	0.35	0.40	0.42
6. Balata	0.32	0.20	—	0.35	0.38	0.40	0.42

18.3 Types of Belt Drives

The belt drives are usually classified into the following three groups:

1. *Light drives*. These are used to transmit small powers at belt speeds upto about 10 m/s as in agricultural machines and small machine tools.

2. *Medium drives*. These are used to transmit medium powers at belt speeds over 10 m/s but up to 22 m/s, as in machine tools.

3. *Heavy drives*. These are used to transmit large powers at belt speeds above 22 m/s as in compressors and generators.

18.4 Types of Belts

Though there are many types of belts used these days, yet the following are important from the subject point of view:

1. *Flat belt*. The flat belt as shown in Fig. 18.1 (*a*), is mostly used in the factories and workshops, where a moderate amount of power is to be transmitted, from one pulley to another when the two pulleys are not more than 8 metres apart.

18.10 Standard Belt Thicknesses and Widths

The standard flat belt thicknesses are 5, 6.5, 8, 10 and 12 mm. The preferred values of thicknesses are as follows:

(*a*) 5 mm for nominal belt widths of 35 to 63 mm,

(*b*) 6.5 mm for nominal belt widths of 50 to 140 mm,

(*c*) 8 mm for nominal belt widths of 90 to 224 mm,

(*d*) 10 mm for nominal belt widths of 125 to 400 mm, and

(*e*) 12 mm for nominal belt widths of 250 to 600 mm.

The standard values of nominal belt widths are in R10 series, starting from 25 mm upto 63 mm and in R 20 series starting from 71 mm up to 600 mm. Thus, the standard widths will be 25, 32, 40, 50, 63, 71, 80, 90, 100, 112, 125, 140, 160, 180, 200, 224, 250, 280, 315, 355, 400, 450, 500, 560 and 600 mm.

Table 28.2. Minimum number of teeth on the pinion in order to avoid interference.

S. No.	*Systems of gear teeth*	*Minimum number of teeth on the pinion*
1.	$14\frac{1}{2}°$ Composite	12
2.	$14\frac{1}{2}°$ Full depth involute	32
3.	20° Full depth involute	18
4.	20° Stub involute	14

Table 28.4. Values of allowable static stress.

Material	*Allowable static stress (σ_o) MPa or N/mm²*
Cast iron, ordinary	56
Cast iron, medium grade	70
Cast iron, highest grade	105
Cast steel, untreated	140
Cast steel, heat treated	196
Forged carbon steel-case hardened	126
Forged carbon steel-untreated	140 to 210
Forged carbon steel-heat treated	210 to 245
Alloy steel-case hardened	350
Alloy steel-heat treated	455 to 472
Phosphor bronze	84
Non-metallic materials	
Rawhide, fabroil	42
Bakellite, Micarta, Celoron	56

Table 28.10. Values of service factor.

Type of load	*Type of service*		
	Intermittent or 3 hours per day	*8-10 hours per day*	*Continuous 24 hours per day*
Steady	0.8	1.00	1.25
Light shock	1.00	1.25	1.54
Medium shock	1.25	1.54	1.80
Heavy shock	1.54	1.80	2.00

28.11 Systems of Gear Teeth

The following four systems of gear teeth are commonly used in practice.

1. $14\frac{1}{2}°$ Composite system, **2.** $14\frac{1}{2}°$ Full depth involute system, **3.** 20° Full depth involute system, and **4.** 20° Stub involute system.

The $14\frac{1}{2}°$ ***composite system*** is used for general purpose gears. It is stronger but has no inter-changeability. The tooth profile of this system has cycloidal curves at the top and bottom and involute curve at the middle portion. The teeth are produced by formed milling cutters or hobs. The tooth profile of the $14\frac{1}{2}°$ ***full depth involute system*** was developed for use with gear hobs for spur and helical gears.

The tooth profile of the **20°** ***full depth involute system*** may be cut by hobs. The increase of the pressure angle from $14\frac{1}{2}°$ to 20° results in a stronger tooth, because the tooth acting as a beam is wider at the base. The **20°** ***stub involute system*** has a strong tooth to take heavy loads.

The values of the velocity factor (C_v) are given as follows :

$$C_v = \frac{3}{3+v}, \text{ for ordinary cut gears operating at velocities upto 12.5 m / s.}$$

$$= \frac{4.5}{4.5+v}, \text{ for carefully cut gears operating at velocities upto 12.5 m/s.}$$

$$= \frac{6}{6+v}, \text{ for very accurately cut and ground metallic gears operating at velocities upto 20 m / s.}$$

$$= \frac{0.75}{0.75+\sqrt{v}}, \text{ for precision gears cut with high accuracy and operating at velocities upto 20 m / s.}$$

$$= \left(\frac{0.75}{1+v}\right) + 0.25, \text{ for non-metallic gears.}$$

Module, $m = D / T$

Note : The recommended series of modules in Indian Standard are 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40 and 50.

The modules 1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5,5.5, 7, 9, 11, 14, 18, 22, 28, 36 and 45 are of second choice.

$$y = 0.124 - \frac{0.684}{T}, \text{ for } 14\tfrac{1}{2}° \text{ composite and full depth involute system.}$$

$$= 0.154 - \frac{0.912}{T}, \text{ for 20° full depth involute system.}$$

$$= 0.175 - \frac{0.841}{T}, \text{ for 20° stub system.}$$