For the following exercise write the MatLab code to execute the design process. Create a separate folder for design of shafts. Ensure to name the files both main and function files as instructed.

Submit the Assignment to the designated mail id: cvcassign5@gmail.com on or before 14.09.2020

Subject for the mail: "CDME: Shaft: Your Name and USN"

Other important instructions for coding:

1. **Start the program with**

   % Program for design of shaft, Problem No. 01, Write the problem statement

   % Name and USN

   % Date

   clc

   clear all

2. **There will be *maximum of 10 exercises* under each topic will be posted with *complete solutions for minimum 05 exercises*, for your ready reference. However, *you will be completing the coding for all exercises posted.***
3. **Main program to be named after the exercise number like: Ex_1_Shaft**
4. **All relevant Equations to be created as function file with file name after the Equation number like: Eqn_3_1**
5. **All relevant Tables to be created as script file with file name after the Table number like: Table_3_5a**
6. **Each and every one's exercises will be run at my end for evaluations and marks will be awarded accordingly. Neat and clean coding will get more marks**

1. A shaft is required to transmit 1MW power at 220 rpm. The twist of shaft must not exceed more than 1^0 on a length of 15diameter. Determine the diameter of the shaft and shear stress induced. Take G = 80 kN/mm^2.

Solution:

Data:

$P = 1MW = 1 \times 10^3 kW$

$n = 220\, rpm$

$\theta = 1^0$

$L = 15d$

$G = 80 kN/mm^2 = 80 \times 10^3 N/mm^2$

Using Eqn. 3.3(a),

$$T = \frac{9.55 \times 10^6 (P)}{n} = \frac{9.55 \times 10^6 (1 \times 10^3)}{220}$$

$T = 43.41 \times 10^6 N - mm$

[Express it in terms of 10^3 and 10^6]

Using Eqn. 3.2,

$$\theta = \frac{584TL}{Gd^4} \Rightarrow 1 = \frac{584 \times 43.41 \times 10^6 \times 15d}{80 \times 10^3 \times d^4}$$

$\therefore d = 168.14\, mm$

Adopting standard diameter for the shaft, using *Table 3.5(a),*

$d = 180\, mm$

Using Eqn. 3.1,

$$\tau = \frac{16T}{\pi d^3} = \frac{16 \times 43.41 \times 10^6}{\pi \times 180^3}$$

$\therefore \tau = 37.9\, N/mm^2$

2. Find the diameter of a solid circular shaft to transmit a power of 60 kW at a rated speed of 1,200 rpm selecting *C50* material and using an appropriate value for factor of safety. Replace the solid shaft by a hollow circular shaft assuming the value of 0.6 for the ratio of diameters selecting the same material and the same factor of safety. Also find the percentage of reduction in weight assuming the same length for solid and hollow shafts, and the ratio of torsional stiffness of the hollow shaft to that of the solid shaft.

   Solution:

   From DDHB, Table I.8, for C50 material, we have,

   $\sigma_u = 647\ N/mm^2 (Consider\ Lower\ value)$
   $\sigma_e = 373\ N/mm^2$
   According to ASME, Assuming No key-way as it is not mentioned,
   $\tau_{max} = 0.3 \times\ \sigma_e = 0.3 \times\ 373 = 111.90\ N/mm^2\ and$
   $\tau_{max} = 0.18 \times\ \sigma_u = 0.18 \times\ 647 = 232.92\ N/mm^2\ and$

3. A line shaft is transmitting 650 kW of power at 500 rpm and is made of *C35* material. The shaft carries a central load of 1,000 N and is simply supported between the bearings 3 m apart. Determine the diameter of the shaft, considering *ASME* code and steady load for rotating shaft.

Solution:

$P = 650\ kW$

$n = 500\ rpm$

$F = 1{,}000\ N$

$L = 3m = 3 \times 10^3 mm$

From Table, I.8, Page No. 463, for C35 material, we have

$\sigma_u = 510\ N/mm^2\ (Lower\ Value)\ and\ \sigma_e = 304\ N/mm^2$

Assume No key-way, if key-way is not mentioned

According to ASME code,

$\sigma_{max} = 0.6 \times \sigma_e = 0.6 \times 304 = 182.4\ N/mm^2\ and$

$\sigma_{max} = 0.36 \times \sigma_u = 0.36 \times 510 = 183.6\ N/mm^2$

Consider the lower value,

$i.e., \quad \sigma_{max} = 182.4\ N/mm^2$

$\tau_{max} = 0.3 \times \sigma_e = 0.3 \times 304 = 91.2\ N/mm^2\ and$

$\tau_{max} = 0.18 \times \sigma_u = 0.18 \times 510 = 91.8\ N/mm^2$

Consider the lower value,

$i.e., \quad \tau_{max} = 91.2\ N/mm^2$

Using Eqn. 3.3(a),

$$T = \frac{9.55 \times 10^6 (P)}{n} = \frac{9.55 \times 10^6 (650)}{500}$$

$T = 12.42 \times 10^6 N - mm$

From Table, 1.4,

$$M_{max} = \frac{FL}{4} = \frac{1000 \times 3 \times 10^3}{4}$$

$M_{max} = 750 \times 10^3 N - mm$

For steady load for rotating shaft,

$C_m = 1.5\ and\ C_t = 1$

According to Maximum Normal Stress Theory:

Using Eqn. 3.6(a),

$$d = \left[\frac{16}{\pi \sigma_{max}}\left(C_m M + \sqrt{(C_m M)^2 + (C_t T)^2}\right)\right]^{1/3}$$

$d = 72.41\ mm$

According to Maximum Shear Stress Theory:

Using Eqn. 3.6(b),

$$d = \left[\frac{16}{\pi \tau_{max}}\left(\sqrt{(C_m M)^2 + (C_t T)^2}\right)\right]^{1/3}$$

$d = 88.64\ mm$

Considering the higher value and adopting the standard size for the shaft, from Table 3.5

$d = 90\ mm$

4. A shaft is mounted between bearings located 9.5 m apart and transmits 10, 000 kW at 90 rpm. The shaft weighs 66,000 N has outside diameter 450 mm and inner diameter 300 mm. Determine the stresses induced in the shaft and the angular deflection between the bearings. Do not neglect the weight the shaft. Take G = 80 kN/mm^2.

Solution:

Data:

L = 9.5 m = 9,500 mm P = 10, 000 kN

n = 90 rpm W = 66, 000 N

d_0 = 450 mm d_i = 300 mm

$$k = \frac{d_i}{d_0} = \frac{300}{450} = \frac{2}{3}$$

$$T = \frac{9.55 \times 10^6 \times P(kw)}{n} = 10.61 \times 10^8\ N\,mm \qquad 3.3(a)$$

Consider the weight of the shaft, it is like UDL

For simply supported beam with UDL, we have

$$M_{max} = \frac{W \times l}{8}$$

$$M_{max} = \frac{66000 * 9500}{8} = 78.375 \times 10^6\ N - mm$$

Stresses Induced:

Maximum Normal Stress induced,

$$d_0 = \left(\frac{16\left(M + \sqrt{M^2 + T^2}\right)}{\pi \sigma_{max}}\left(\frac{1}{1 - k^4}\right)\right)^{\frac{1}{3}} \qquad 3.5(a)$$

$\sigma_{max} = 82.44\ N/mm^2$

Maximum Shear Stress induced,

$$d_0 = \left(\frac{16\left(\sqrt{M^2 + T^2}\right)}{\pi \tau_{max}}\left(\frac{1}{1 - k^4}\right)\right)^{\frac{1}{3}} \qquad 3.5(b)$$

$$\tau_{max} = \left(\frac{16\left(\sqrt{M^2 + T^2}\right)}{\pi {d_0}^3}\left(\frac{1}{1 - k^4}\right)\right) = \left(\frac{16\left(\sqrt{(78.375 \times 10^6)^2 + (10.61 \times 10^8)^2}\right)}{\pi \times 450^3}\left(\frac{1}{1 - \left(\frac{2}{3}\right)^4}\right)\right)$$

$\therefore \tau_{max} = 74.10\ N/mm^2$

5. A 1.2 m hollow shaft is subjected to a bending moment of 900 N-m and a twisting moment of 600 N-m. The shaft is also subjected to an end thrust 1.2 kN. Considering the ratio of inner diameter to outer diameter of the hollow shaft as 0.7 and cold rolled steel *C30*, determine the size of the shaft for heavy shock conditions. Assume FOS = 3 and use maximum shear stress theory only.

Data:

L = 1200 mm; $M = 900\ Nm = 900 \times 10^3\ N\ mm\ ; T = 600\ Nm;\ F = 1.2\ KN = 1200N$

$k = 0.7\ and\ n = 3;\ material\ is\ C30\ cold\ rolled\ from\ T1.8$

$\sigma_y = 294N/mm^2$

Outer diameter of the shaft is subjected to bending torsion and axial load we use maximum stress theory, According to maximum stress theory

$$d_0 = \left[\frac{16}{\pi * \tau_{max}}\left\{\sqrt{\left(C_m * M + \frac{\alpha * F * d_0(1+k^2)}{8}\right)^2 + (C_t * T)^2}\right\} * \left(\frac{1}{1-k^4}\right)\right]^{\frac{1}{3}} \quad \ldots..3.8(b)$$

$from\ table\ 3.1\ for\ heavy\ shock\ laod$

$C_m = 2\ to\ 3\ consider\ higher\ value\quad C_m = 3$

$C_t = 1.5\ to\ 3\ consider\ higher\ value\quad C_t = 3$

$$\tau_y = \frac{\sigma_y}{2} = \frac{294}{2} = 147\ N/mm^2$$

$$there\ fore\ \tau_{all} = \frac{\tau_y}{n} = \frac{147}{3} = 49\ N/mm^2$$

Using relation in Page No. 52

$$\alpha = \frac{1}{1 - 0.0044\left(\frac{L}{K}\right)}$$

Where K is radius of gyration

$$K = \sqrt{\frac{I}{A}} = \frac{\sqrt{{d_0}^2 + {d_i}^2}}{4} \qquad (T1.3\ P14)$$

$$or\ K = \frac{1}{4} \times d_0\sqrt{1+k^2} = \frac{1}{4} \times d_0\sqrt{1+0.7^2}$$

$$K = 0.3052d_0$$

$$\alpha = \frac{1}{1 - 0.0044\left(1.2 \times \frac{10^3}{0.3052d_0}\right)} = \frac{1}{1 - \frac{17.3}{d_0}};\ \therefore \alpha = \frac{d_0}{d_0 - 17.3}$$

$$d_0 = \left[\frac{16}{\pi \times 49}\left\{\sqrt{\left(3 \times 900 \times 10^3 + \left(\frac{d_0}{d_0 - 17.3}\right)\frac{1200 \times d_0(1+0.7^2)}{8}\right)^2 + (3 \times 600 \times 10^3)^2}\right\} \times \left(\frac{1}{1-0.7^4}\right)\right]^{\frac{1}{3}}$$

$by\ trial\ and\ error\ method$

$d_0 = 71mm$

$d_i = 0.7 \times 71 = 56mm$

6. A line shaft is to transmit 30 kW at 160 rpm. It is driven by a motor placed directly below it by means of a belt running on a 1 m diameter pulley keyed to the end of shaft. The tension on the tight side of the belt is 2.5 times that on the slack side and the centre of the pulley overhangs 150mm beyond the centre line of the end bearing. Determine the diameter of the shaft, if the allowable shear stress is 60 N/mm^2 and the pulley weighs 1600 N. Assume minor shock conditions.

Solutions:

$P = 30kw\ and\ \ n = 160rpm$

$d = 1000mm\ and\ r = 500mm$

$T_1 = 2.5T_2 \tau_{all} = 60\frac{N}{mm^2}, \qquad W = 1600N$

$$T = \frac{9.55 \times 10^6 \times P}{n}$$

$$T = \frac{9.55 \times 10^6 \times 30}{160} = 17.91 \times 10^5\ N\ mm$$

For belt drive

$T = (T_1 - T_2) \times r$

$17.91 \times 10^5 = (2.5T_2 - T_2) \times 500$

$T_2 = 2388\ N$

$T_1 = 5970\ N$

Total load acting at the shaft = total belt tension + pulley weight

$F = T_1 + T_2 + W_p$

$F = 2388 + 5970 + 1600$

$F = 9958N$

$M_A = 0$

$M_B = -F \times L = -9958 \times 150$

$M_B = -14.94 \times 10^5\ N - mm$

$C_m = 2\ and\ C_t = 1.5$

$$d = \left[\frac{16}{\pi \times \tau_{all}} \times \sqrt{(C_m \times M)^2 + (C_t \times T)^2}\right]^{\frac{1}{3}}$$

$$d = \left[\frac{16}{\pi \times 60} \times \sqrt{(2 \times 14.94 \times 10^5)^2 + (1.5 \times 17.91 \times 10^5)^2}\right]^{\frac{1}{3}}$$

$d = 69.87mm$

$d = 71\ mm$

7. The layout of a transmission shaft carrying two pulleys, B and C, and supported on bearings A and D is shown in *Figure 2*. Power is supplied to the shaft by means of a vertical belt on the pulley B, which is then transmitted to the pulley C carrying a horizontal belt. The maximum tension in the belt on the pulley B is 2.5 kN. The angle of wrap for both the pulleys is 180^0 and the coefficient of friction is 0.24. The shaft is made of plain carbon steel with yield stress of 400 N/mm^2 and FOS = 3. Determine the diameter of the shaft on the strength basis.

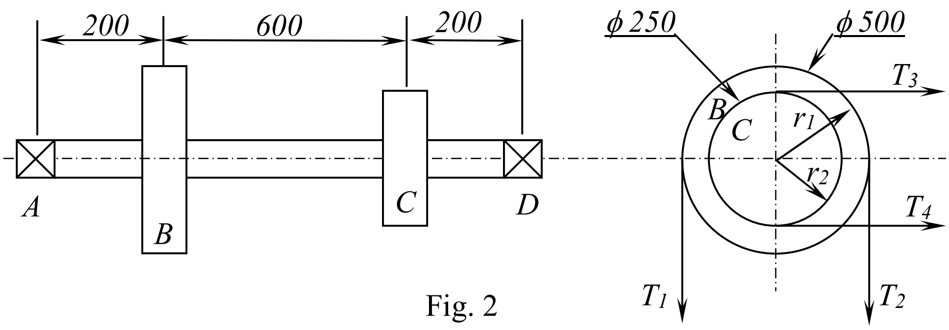


Fig. 2

$$Maximum\ Tension, T_1 = 2.5 \times 10^3 N \qquad \sigma_e = 400 N/mm^2$$
$$Angle\ of\ wrap, \theta = 180^0\ or\ \pi\ rad \qquad \mu = 0.24$$
$$L = 1{,}400\ mm$$
$$FOS = 3$$
$$d_1 = 500\ mm \qquad \therefore r_1 = 250\ mm$$
$$d_2 = 250\ mm \qquad \therefore r_2 = 125\ mm$$

Using Eqn. 14.3(a),

$$\frac{T_1}{T_2} = e^{\mu\theta}$$

$$\therefore\ T_2 = \frac{T_1}{e^{\mu\theta}} = \frac{2.5 \times 10^3}{e^{0.24\times\pi}} = 1.17 \times 10^3 N$$

Torque supplied to the shaft,

$$T = (T_1 - T_2)r_1 = (2.5 - 1.17) \times 10^3 \times 250 = 3.325 \times 10^5 N - mm$$

But, $T = (T_3 - T_4)r_2$

$$\therefore (T_3 - T_4) = \frac{T}{r_2} = \frac{3.325 \times 10^5}{125} = 2.66 \times 10^3 N \quad (1)$$

$$and\ \frac{T_3}{T_4} = e^{\mu\theta}\ \ OR\ \ T_3 = e^{\mu\theta}.T_4 = e^{0.24\times\pi}.T_4$$

$$\therefore T_3 = 2.125T_4$$

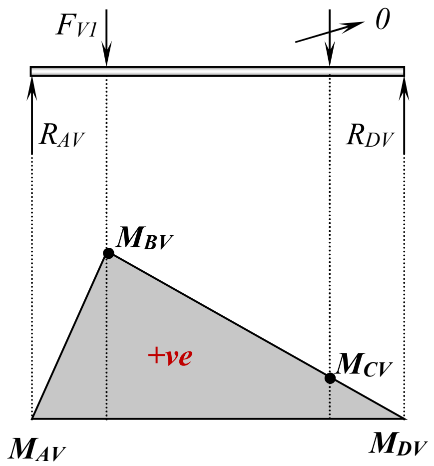
Putting this value in Eqn.(1), we get

$$(T_3 - T_4) = 2.66 \times 10^3 \Rightarrow (2.125T_4 - T_4) = 2.66 \times 10^3 \Rightarrow 1.125T_4 = 2.66 \times 10^3$$

$$\therefore T_4 = \frac{2.66 \times 10^3}{1.125} = 2.36 \times 10^3 N\ and$$

$$\therefore T_3 = 2.125 \times T_4 = 2.125 \times 2.36 \times 10^3 = 5.02 \times 10^3 N$$

Bending Moment due to Vertical load on the shaft:



[Note: M_{AV} and M_{DV} are equal to zero at supporting points]

Vertical load on the shaft at pulley *B* is due to T_1 and T_2, [note that weight of the pulley *B* is not given and is ignored]

$$F_{V1} = T_1 + T_2 + W_1 = (2.5 + 1.17) \times 10^3$$
$$\therefore F_{V1} = 3.67 \times 10^3 N$$

Taking moment about point *D*, we get

$$R_{AV} \times 1000 - F_{V1} \times (600 + 200) = 0$$
$$\therefore R_{AV} = \frac{3.67 \times 10^3 \times (600 + 200)}{1000} = 2.936 \times 10^3 N$$

But,

$$R_{AV} + R_{DV} = F_{V1} \quad OR \quad R_{DV} = (3.67 - 2.936) \times 10^3$$
$$\therefore R_{DV} = 0.734 \times 10^3 N$$

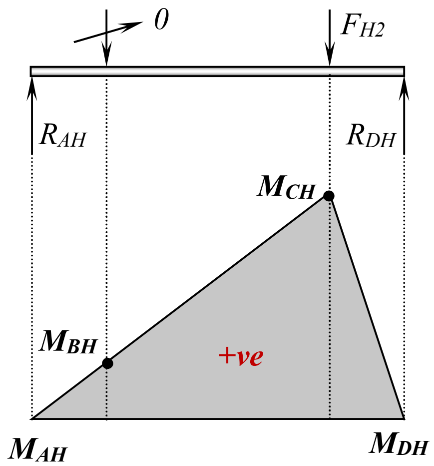
Bending moment at point *B* due to vertical load,

$$M_{BV} = R_{AV} \times 200 = 2.936 \times 10^3 \times 200$$
$$\therefore M_{BV} = 5.872 \times 10^5 N - mm$$

Bending moment at point *C* due to vertical load,

$$M_{CV} = R_{DV} \times 200$$
$$\therefore M_{CV} = 1.468 \times 10^5 N - mm$$

Bending Moment due to Horizontal load on the shaft:



[Note: M_{AH} and M_{DH} are equal to zero at supporting points]

Horizontal load on the shaft at pulley *C* is due to T_3 and T_4, [note that weight of the pulley *C* is not given and is ignored]

$$F_{H2} = T_3 + T_4 + W_2 = (5.02 + 2.36) \times 10^3$$
$$\therefore F_{H2} = 7.38 \times 10^3 N$$

Taking moment about point *D*, we get

$$R_{AH} \times 1000 - F_{H2} \times 200 = 0$$
$$\therefore R_{AH} = \frac{7.38 \times 10^3 \times 200}{1000} = 1.476 \times 10^3 N$$

But,

$$R_{AH} + R_{DH} = F_{H2} \quad OR \quad R_{DH} = (7.38 - 1.476) \times 10^3$$
$$\therefore R_{DH} = 5.904 \times 10^3 N$$

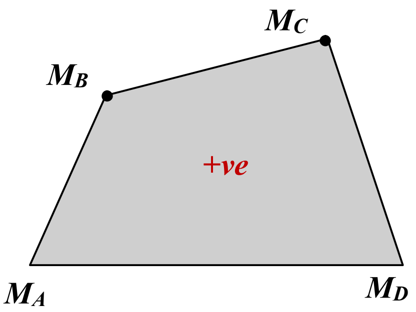
Bending moment at point *B* due to horizontal load,

$$M_{BH} = R_{AH} \times 200 = 1.476 \times 10^3 \times 200$$
$$\therefore M_{BH} = 2.952 \times 10^5 N - mm$$

Bending moment at point *C* due to horizontal load,

$$M_{CH} = R_{DH} \times 200 = 5.904 \times 10^3 \times 200$$
$$\therefore M_{CH} = 11.808 \times 10^5 N - mm$$

Resultant Bending Moment due to both Vertical and Horizontal load on the shaft:

Resultant Bending Moment Diagram  	Resultant Bending moment at point *B*, $M_B = \sqrt{M_{BV}^2 + M_{BH}^2} = \sqrt{(5.872^2 + 2.952^2) \times (10^5)^2}$ $\therefore M_B = 6.57 \times 10^5 N - mm$ and Resultant Bending moment at point *B*, $M_C = \sqrt{M_{CV}^2 + M_{CH}^2} = \sqrt{(1.468^2 + 11.808^2) \times (10^5)^2}$ $\therefore M_C = 11.89 \times 10^5 N - mm$ Consider the maximum Bending Moment, i.e., Bending moment at point *C*, $\therefore M = 11.89 \times 10^5 N - mm$

Now, Using Eqn. 3.5(b),

Diameter of the shaft, according to Maximum Shear Stress Theory:

$$d = \left[\frac{16}{\pi \tau_{max}}\left(\sqrt{M^2 + T^2}\right)\right]^{1/3}$$

$$where, \tau_{max} = \frac{\sigma_e}{2FOS} = \frac{400}{2 \times 3} = 66.67\ N/mm^2$$

$$\therefore d = \left[\frac{16}{\pi \times 66.67}\left(\sqrt{(11.89^2 + 3.325^2) \times (10^5)^2}\right)\right]^{1/3}$$

$$\therefore d = 45.52\ mm$$

Adopting the Standard Size for the Shaft, from Table 3.5,

$d = 50\ mm$

8. A steel shaft 900 mm long between bearings receives power of 18 kW at 900 rpm through a 20^0 involute spur gear of 200 diameter located 250 mm to the right of left hand bearing by a gear directly below the shaft. The power is transmitted by a 400 mm diameter pulley to another pulley which is behind the shaft and above it. The belt drive axis is inclined at 60^0 to the horizontal. The pulley is located at a distance of 250 mm to the left of the right hand bearing. The angle of lap of belt is 180^0 and the coefficient of friction is 0.3. The shaft rotates in the clockwise direction as seen from the left side bearing. Design the shat diameter if the allowable shear stress for the shaft material is 60 MPa.

Solution :

Torque tansmitted $M_t = 9550 \times 1000 \times \frac{N}{n} = 9550 \times 1000 \times \frac{18}{900} = 191000 \text{ Nmm}$

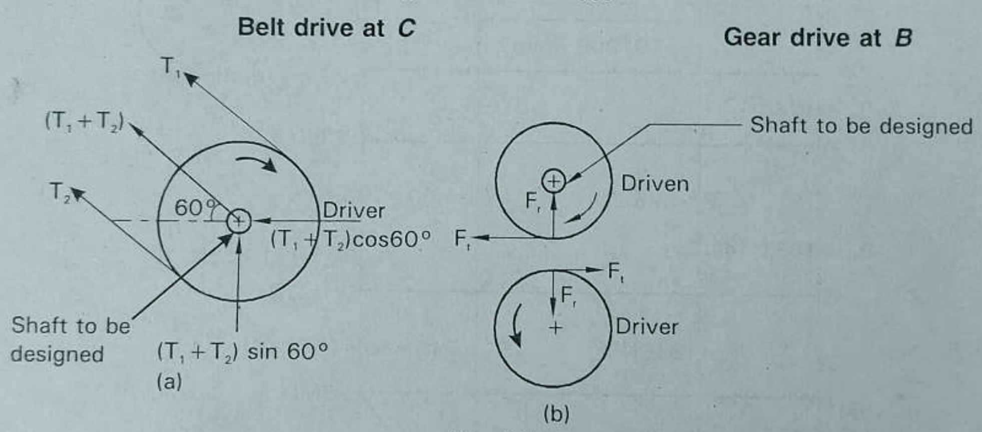


Fig. 5.30

Resultant bending moment at $B = \sqrt{(256\,292.5)^2 + (420360)^2} = 465240.954 \text{ Nmm}$

Resultant bending moment at $C = \sqrt{(388282.5)^2 + (328940)^2} = 508885.865 \text{ Nmm}$

$\therefore$ Maximum bending moment $M_b = 508885.865 \text{ Nmm}$

Assume suddenly applied load with minor shocks

From Table 14.2 (DDHB), $K_b = 1.5 - 2.0$; $K_t = 1.0 - 1.5$

Take $K_b = 1.75$ and $K_t = 1.25$

According to maximum shear stress theory, diameter of shaft subjected to combined bending and torsion.

$$D = \left[\frac{16}{\pi \tau_{ed}}\left\{(K_b M_b)^2 + (K_t M_t)^2\right\}^{1/2}\right]^{1/3} = \left[\frac{16}{\pi \times 60}\left\{(1.75 \times 508885.865)^2 + (1.25 \times 191000)^2\right\}^{1/2}\right]^{1/3}$$

---- 14.12 (DDHB)

$= 42.774 \text{ mm}$

Adopt standard diameter of shaft $D = 45$ mm [Table 14.6 (New DDHB)]
For detailed procedure, refer previous Examples.