Problem 3: Solution

1 mark

→ Material: AISI 1050 HR Steel, $S_{ut} = 620\ MPa$
$S_{yt} = 340\ MPa$ (table A-20)

→ $K_{ts} = 1.6$, $L = 610\ mm$, $F_{min} = 0.25\ kN$
$F_{max} = 0.50\ kN$, $q_s = 0.88$ at $r = 3.2\ mm$

$T_{max} = 0.50 \times 610 = 305.0\ N.m$

2 mark

$T_{min} = 0.25 \times 610 = 152.5\ N.m$

$K_{fs} = 1 + q(K_{ts} - 1) = 1 + 0.88(1.6 - 1) = 1.53$

→ $$\tau_{max} = \frac{16 K_{fs} T_{max}}{\pi d^3} = \frac{16 \times (1.53) \times 152.5}{\pi (22.23 \times 10^{-3})^3} = 106.91\ MPa$$

2 mark

$$\tau_{min} = \frac{16 K_{fs} T_{min}}{\pi d^3} = \frac{16 \times (1.53) \times 152.5}{\pi (22.23 \times 10^{-3})^3} = 53.46\ MPa$$

→ $$\tau_m = \frac{\tau_{max} + \tau_{min}}{2} = 80.19\ MPa$$

2 mark

$$\tau_a = \frac{\tau_{max} - \tau_{min}}{2} = 26.73\ MPa$$

→ checking for yielding with MSS (chp 5):

$$n_y = \frac{S_{yt}/2}{\tau_{max}} = \frac{340/2}{80.19} = 2.12$$

1 mark

→ Marin's factors:

$S_e' = 0.5(620) = 310\ MPa$

$K_a = 57.7(620)^{-0.718} = 0.570$

$d_e = 0.37(22.23) = 8.23\ mm$

2 mark

$K_b = 1.24(8.23)^{-0.107} = 0.99$

$K_c = 0.59$

$S_{es} = 0.57 \times 0.99 \times 0.59 \times 310 = 103.21\ MPa$

1 mark

→ From Eq. 6-54: $S_{us} = 0.67 S_{ut} = 0.67(620)$ 1 mark

$S_{us} = 415.4$ MPa

→ From table 6-7: 3 mark

$$n_f = \frac{1}{2}\left(\frac{S_{us}}{\tau_m}\right)^2 \cdot \frac{\tau_a}{S_{es}}\left[-1 + \sqrt{1 + \left(\frac{2\tau_m S_{se}}{S_{su} \cdot \tau_a}\right)}\right]$$

$$= \frac{1}{2}\left(\frac{415.4}{80.19}\right)^2 \cdot \frac{26.73}{103.21}\left[-1 + \sqrt{1 + \left(\frac{2 \times 80.19 \times 103.21}{415.4 \times 26.73}\right)}\right]$$

$\therefore n_f = 2.01$

Problem 4: Solution

Problem - 4

→ Material: $S_{ut} = 848$ MPa (123 kpsi), $S_{yt} = 648$ MPa (94 kpsi) → table A-21 AISI 1030 Q&T Steel

$S_e' = 0.5 S_{ut} = 424$ MPa 1 mark

$K_a = 4.51(848)^{-0.265} = 0.755$ (table 6-2)

$K_e = 0.897$ (table 6-5) 1 mark

$K_b = 1.51(60)^{-0.157} = 0.794$

K_c is considered for each load separately

$K_d = K_f = 1$

⇒ $S_e = 0.755 \times 0.794 \times 0.897 \times 424 = 228.0$ MPa 1 mark

→ $r/d = 6/60 = 0.1$, $\frac{D}{d} = \frac{80}{60} = 1.333$.

Axial loading

$K_c = 0.85$

$K_t = 1.7$ from table A-15-7 1 mark

$\sqrt{a} = 0.246 - 3.08(10^{-3})(123) + 1.51(10^{-5})(123)^2 - 2.67(10^{-8})(123)^3$

└ from Eq. (6-35 a)

$\sqrt{a} = 0.04592$ 1 mark

$$q = \frac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = 0.98$$

$K_f = 1 + q(K_t - 1) = 1.687$ 1 mark

$$K_{eff} = \frac{1}{K_c \cdot K_f} = 0.697$$

axial load is Constant ⇒ it will only create mean stress

$$\sigma_m = \frac{F}{A} = \frac{15000}{\frac{\pi(60)^2}{4}} = 5.31 \text{ MPa}$$ 1 mark

Bending Moment

$K_c = 1$

$K_t = 1.51$ from Fig. A-15-9

$\sqrt{a} = 0.04592$ From Eq 6-35a

$q = 0.98$ 1 mark

$K_f = 1 + q(K_t - 1) = 1.5$ 1 mark

$$K_{eff} = \frac{1}{K_c K_f} = 0.67$$

→ Bending moment is fluctuating ⇒ σ_m & σ_a

$\sigma = \frac{Mc}{I}$, $I = \frac{\pi (60)^4}{64} = 636.173 \times 10^3 \ mm^4$

1 mark

$c = \frac{60}{2} = 30 \ mm$

$M_{min} = 20 \times 10^3 \ N.mm$

$M_{max} = 200 \times 10^3 \ N.mm$

$M_m = \frac{M_{max} + M_{min}}{2} = 110 \times 10^3 \ N.mm$

2 mark

$M_a = \frac{M_{max} - M_{min}}{2} = 90 \times 10^3 \ N.mm$

$\sigma_m = \frac{M_m \, c}{I} = 5.187 \ MPa$

2 mark

$\sigma_a = \frac{M_a \cdot C}{I} = 4.244 \ MPa$

$\sigma'_{ae} = \frac{\sigma_a}{K_c \cdot K_{eff}} = 6.335 \ MPa$

Torsion

$K_c = 1.0$

$K_t = 1.42$ from Fig. A-15-8

$\sqrt{a} = 0.190 - 2.51(10^{-3})(123) + 1.35(10^{-5})(123)^2 - 2.67(10^{-8})(123)^3$

$\therefore \sqrt{a} = 0.0358$

$q = \dfrac{1}{1 + \frac{\sqrt{a}}{\sqrt{r}}} = 0.986$ 1 mark

$K_{fs} = 1 + q(K_t - 1) = 1.414$ 1 mark

$K_{eff} = \dfrac{1}{K_c K_f} = 0.707$

→ Torsion is fluctuating ⇒ τ_m & τ_a

$J = \dfrac{\pi (60)^4}{32} = 1.27235 \times 10^6 \text{ mm}^4$, $C = \dfrac{60}{2} = 30 \text{ mm}$

$T_{min} = -50 \times 10^4$ N.mm

$T_{max} = 150 \times 10^4$ N.mm

$T_m = \dfrac{T_{max} + T_{min}}{2} = 50 \times 10^4$ N.mm

$T_a = \dfrac{T_{max} - T_{min}}{2} = 100 \times 10^4$ N.mm 2 mark

$\tau_m = \dfrac{T_m \cdot C}{J} = 11.8$ MPa

$\tau_a = \dfrac{T_a \cdot C}{J} = 23.6$ MPa 2 mark

$\tau'_{ae} = \dfrac{\tau_a}{K_c K_{eff}} = 33.35$ MPa

→ Total results:

$\sigma_m = 5.31 + 5.187 = 10.5$ MPa (bending + axial force)

$\sigma_{ae} = 6.34$ MPa 1 mark

$\tau_m = 11.8$ MPa , $\tau_{ae} = 33.35$ MPa

→ Using von-Mises 2D Stress State:

$$\sigma_m' = \sqrt{\sigma_m^2 + 3\tau_m^2} = \sqrt{10.5^2 + 3(11.8)^2} = 22.98 \text{ MPa}$$ 1 mark

$$\sigma_a' = \sqrt{\sigma_{ae}^2 + 3\tau_{ae}^2} = \sqrt{6.34^2 + 3(33.35)^2} = 58.11 \text{ MPa}$$ 1 mark

→ To check for first circle yielding:

$$\sigma_a' + \sigma_m' = \frac{S_{yt}}{n} \Rightarrow n_y = \frac{648}{(22.98 + 58.11)}$$ 1 mark

$$n_y = 8$$

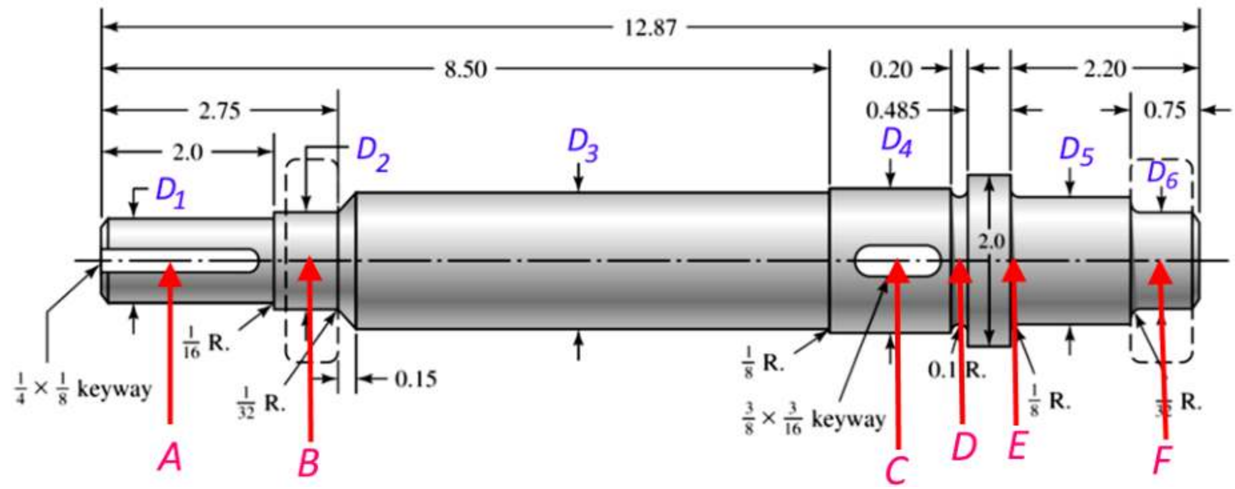
→ Using modified Goodman theory:

$$\frac{\sigma_a'}{S_e} + \frac{\sigma_m'}{S_{ut}} = \frac{1}{n} \rightarrow n = 3.55$$ 1 mark

Since $n_y > n$, it means the part will fail by Fatigue (not yielding)

∴ Factor of Safety, $n = 3.55$

Problem 5: Solution



A: location of fan

B: left bearing

C: gear

D: groove

E: shoulder

F: right bearing

Approach to solve the problem as follows:

(a) For determining suitable shaft dimensions:

1. *Identify the critical location*
2. *Choose material*
3. *Find S_e and S_e'*
4. *Estimate stress concentration factors*
5. *Estimate shaft diameters at the given location*
6. *Verify the estimated diameters*

(b) For check conformity of deflection to design standard:

1. *By neglecting smaller details, utilize a design software to calculate approximate deflection and slope developed at various points on the shaft (MD solids is always useful here)*
2. *Determine if the slopes and deflections conforms to the catalogue standard given in Table 7-2.*

Step-1: Indentifying critical location

Reaction forces 1 mark

-Since there is only one gear at C,

$F_g = \sqrt{230^2 + 633^2} = 673.5\ lbf$ – resultant force at C

-Due to shaft rotation, the bending stress is completely reversed case

-Assuming the fan load to be much smaller than the bending stress caused by gears, $F_{fan} \approx 0$

-Applying equations of equilibrium:

$R_1 + R_2 = 637.5\ lbf$ and $F_g\ x\ 6.98 - R_2\ x\ 10.12 = 0$

$\rightarrow R_1 = 209.9\ lbf$ and $R_2 = 464.5\ lbf$

Bending moment 1 mark

OB: $M_{OB} = 0$

BC: $M_{BC} = R_1 x$ → M = 1465 lbf.in at C

CF: $M_{CF} = R_1 x - F_g(x - 6.98)$ → M = 1115 lbf.in at D, 845 lbf.in at E and 0 at F

Torque 1 mark

-Torque is constant between points A and C, because a steady torque was applied.

-Torque is given as: $T = F_t\ x\ \frac{d}{2} = 633\ x\ \left(\frac{8}{2}\right) = 2532\ lbf.in$

Critical locations 2 mark

-From the BM and Torque diagrams below, potential critical locations can be spotted as follows:

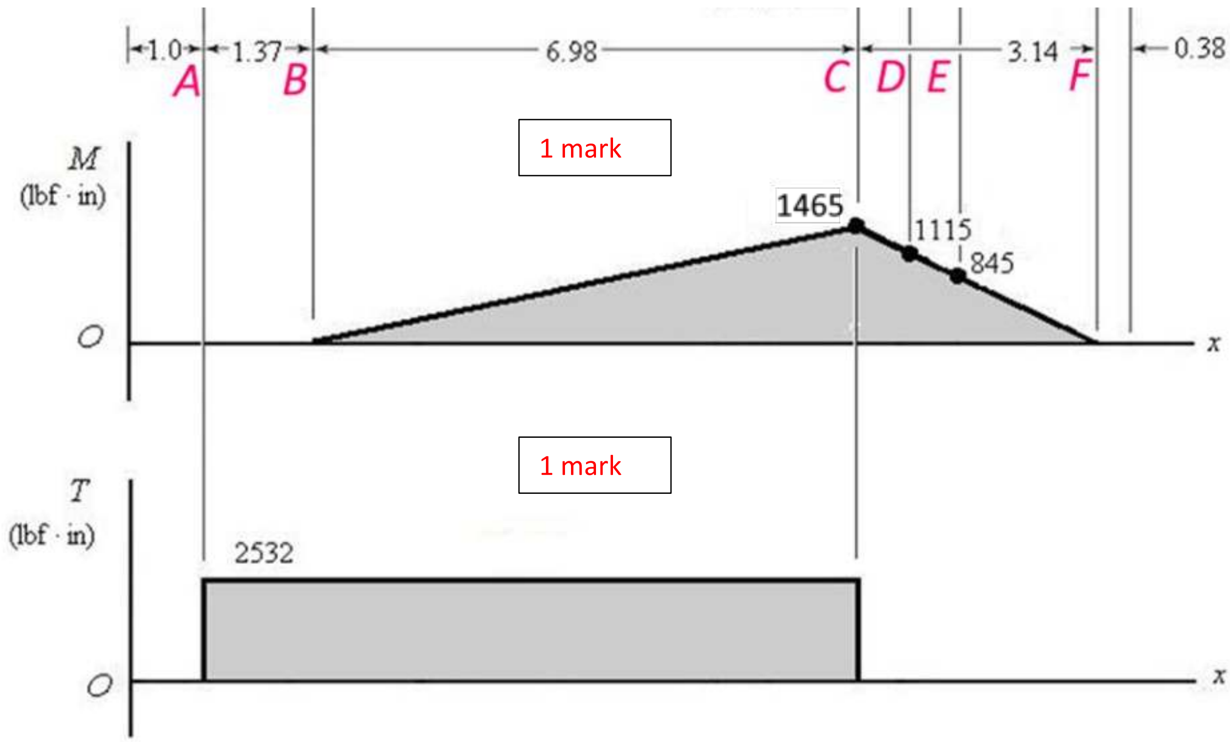
1. Keyway at C, where bending moment is maximum, torque is high and stress concentration exists.

2. Keyway at A, where torque is high, diameter is expected to be smaller and stress concentration exists.

3. Groove at D, where diameter is expected to be smaller, bending moment is still high and stress concentration exists.

4. Shoulder at E, where diameter is expected to be smaller, bending moment is still high and stress concentration exists.

Since the BM is higher at C, we will start analysis from that location.



Step-2: Selection of material 1 mark

-Let us select an inexpensive material, i.e. 1020 CD steel with

Table A-20: S_{ut} = 68 kpsi, S_y = 57 kpsi

<u>Keyway at C</u>

Step-3: Find S_e and $S_e{}'$ 2 mark

Eq. (6-8): $S_e' = 0.5(68) = 34.0 \text{ kpsi}$

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

$k_b = 0.9$, initial geuss

$k_c = k_d = k_e = k_f = 1$

$S_e' = k_a k_b k_c k_d k_e k_f S_e' = 27.02\ kpsi$

Step-4: Estimate stress concentration factors at C 1 mark

-Assuming $\frac{r}{d} = 0.02$, $k_t = 2.14$, $k_{ts} = 3.0$ → for end-milled key seat, Table 7-1

-From Fig. 6-20: $q = 0.66$ and $q_s = 0.72$ → $k_f = 1.8$ and $k_{fs} = 2.4$

Step-5: Estimate shaft diameter at C 2 mark

-We use DE-Goodman criterion which is conservative and good for initial design with $n = 1.5$

-$M_a = 1465.1\ lbf.in$, $M_m = T_a = 0$, $T_m = 2532\ lbf.in$ → at C

Using:

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} \left[4(K_f M_a)^2 + 3(K_{fs} T_a)^2 \right]^{1/2} + \frac{1}{S_{ut}} \left[4(K_f M_m)^2 + 3(K_{fs} T_m)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$d = 1.389''$ → Hence, let us select $D_4 = 1.4$" based on recommened values from table A-14 and A-17.

Let us check our initial estimate: $k_b = \left(\frac{1.4}{0.3}\right)^{-0.107} = 0.848 \approx 0.9$. Hence, initial value of k_b used is acceptable.

Nevertheless, utilizing $k_b = 0.848$ yields $S_e' = 25.45\ kpsi$. And for keyways made from end-milled cutters, $\frac{r}{d} \sim 0.02$, $k_t = 2.14$ and $k_{ts} = 3.0$. Hence, recalculating the d using the new values:

$$d = \left(\frac{16n}{\pi} \left\{ \frac{1}{S_e} \left[4(K_f M_a)^2 + 3(K_{fs} T_a)^2 \right]^{1/2} + \frac{1}{S_{ut}} \left[4(K_f M_m)^2 + 3(K_{fs} T_m)^2 \right]^{1/2} \right\} \right)^{1/3}$$

$d = 1.415''$ → Hence, let us selecting D_4 *to be* 1.4" is acceptable based on the recommened values from table A-14 and A-17.

-Check for yielding: 1 mark

$\sigma_a' = \frac{32M_a}{\pi d^3} = 5.438\ kpsi$ and $\sigma_m' = 0$

$n_y = \frac{S_y}{\sigma_a' + \sigma_m'} = \frac{57}{5.438} \sim 10.5$, hence fatigue failure will occur first.

<u>Groove at D</u>

First, we need to assume dimensions and then check the factor of safety.

-will assume Fig. A-15-14 is applicable

-will assume the 2'' diameter specified to the right of the groove is relatively narrow and will not allow the stress to flow (see Fig. 7-9 for stress flow concept) 1 mark

$\frac{r}{d} = \frac{0.1}{1.2} = 0.0833$ and → $d = 1.4 - 2(0.1) = 1.2$ and $r = 0.1$ (given)

$\frac{D}{d} = \frac{1.4}{1.2} = 1.167$

Step-3: Find S_e and $S_e{'}$ 1 mark

Eq. (6-8): $S_e' = 0.5(68) = 34.0 \text{ kpsi}$

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

$$k_b = \left(\frac{1.2}{0.3}\right)^{-0.107} = 0.862$$

$$k_c = k_d = k_e = k_f = 1$$

$$S_e' = k_a k_b k_c k_d k_e k_f S_e' = 25.88\ kpsi$$

Step-4: Estimate stress concentration factors at D 1 mark

-$k_t = 1.9$ for $\frac{r}{d} = 0.0833$ and $\frac{D}{d} = \frac{1.4}{1.2} = 1.167$

-From Fig. 6-20: $q = 0.8$ → $k_f = 1.72$

Step-5: Estimate factor of safety at groove 2 mark

-We use DE-Goodman criterion which is conservative and good for initial design with $d = 1.2$

-$M_a = 1115\ lbf.in,\ M_m = T_a = 0,\ T_m = 2532\ lbf.in$ → at D

Using:

$$\frac{1}{n} = \frac{16}{\pi d^3}\left\{\frac{1}{S_e}\left[4(K_f M_a)^2 + 3(K_{fs} T_a)^2\right]^{1/2} + \frac{1}{S_{ut}}\left[4(K_f M_m)^2 + 3(K_{fs} T_m)^2\right]^{1/2}\right\}$$

$n = 2.16$ → Acceptable

Hence, $D_4 = 1.4$" is good value for diameter at C.

-Check for yielding: 1 mark

$\sigma_a' = \frac{32M_a}{\pi d^3} = 6.572\ kpsi$ and $\sigma_m' = 0$

$n_y = \frac{S_y}{\sigma_a' + \sigma_m'} = \frac{57}{6.572}$ ~8.67, hence fatigue failure will occur first.

Shoulder at E

First, we need to assume dimensions and then check for the factor of safety.

- $r = \frac{1}{8}''$ (given) and $D = 2.0''$ (specified on diagram and added to balance fan weight on far left) 1 mark

-adopting typical $\frac{D}{d}$ ratio for a shoulder to be 1.2 (recommended by bearing catalogue—slides)

- $\frac{D}{d} = 1.2$ → $d = 1.67 \approx 1.7''$

- $\frac{r}{d} = \frac{0.125}{1.7} = 0.074$

Step-3: Find S_e and S_e' 1 mark

Eq. (6-8): $S_e' = 0.5(68) = 34.0 \text{ kpsi}$

Eq. (6-19): $k_a = 2.70(68)^{-0.265} = 0.883$

$k_b = \left(\frac{1.7}{0.3}\right)^{-0.107} = 0.831$

$k_c = k_d = k_e = k_f = 1$

$S_e' = k_a k_b k_c k_d k_e k_f S_e' = 23.54\ kpsi$

Step-4: Estimate stress concentration factors at D 1 mark

- $k_t = 1.95$ for $\frac{r}{d} = 0.074$ and $\frac{D}{d} = \frac{1.7}{1.2} = 1.2$

-From Fig. 6-20: $q = 0.75$ → $k_f = 1.71$

Step-5: Estimate factor of safety at shoulder 2 mark

-We use DE-Goodman criterion which is conservative and good for initial design with $d = 1.7$

- $M_a = 845\ lbf.in$, $M_m = T_a = 0$, $T_m = 0$ → at D

Using:

$$\frac{1}{n} = \frac{16}{\pi d^3}\left\{\frac{1}{S_e}\left[4(K_f M_a)^2 + 3(K_{fs} T_a)^2\right]^{1/2} + \frac{1}{S_{ut}}\left[4(K_f M_m)^2 + 3(K_{fs} T_m)^2\right]^{1/2}\right\}$$

$n = 2.76$ → Acceptable

Hence, $D_5 = 1.7$" is good value for diameter at E.

-Check for yielding: 1 mark

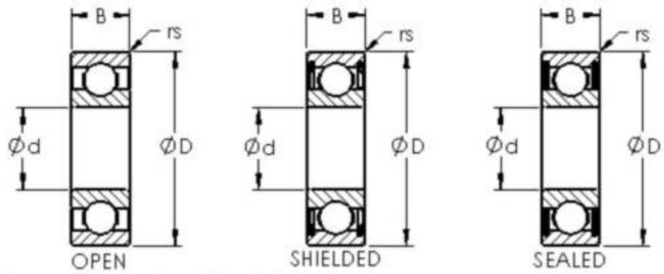
$\sigma_a' = \frac{32M_a}{\pi d^3} = 1.751\ kpsi$ and $\sigma_m' = 0$

$n_y = \frac{S_y}{\sigma_a' + \sigma_m'} = \frac{57}{6.572}$ ~32.54, hence fatigue failure will occur first.

Diamater of other sections 2 mark

The locations of the bearings and other locations of shaft are subjected to less bending and/or torgue, we expect the factor of safety to be higher than the required value at these locations. Hence, the diameters at the other locations can be obtained from standard catalogue for bearings and shafts as follows:

-$D_2 = D_6 = 0.875''$ standard bore size from bearing catalogue



Open, Shielded and Sealed - Inch

1-30 of 75 Products Sort by: Default Compare

Product	Bearing Type	Bore Dia (d) (in)	Outer Dia (D) (in)	Width (B) (in)	Radius (min) (rs) (in)	Dynamic Load Rating (Cr) (lbs)	Static Load Rating (Cor) (lbs)
CI38	Open	0.7500	2.0000	0.5625		2,270	1,290
CI40	Open	0.8750	2.0000	0.5625		2,265	1,315
CI41	Open	1.0000	2.0000	0.5625		2,260	1,345

https://www.astbearings.com/catalog?cid=inch_series_1600

-$D_3 = 1.2''$ from table A-14 and A-17

Keyway at A 1 mark

Since there is only steady torsion here, the diameter at pulley end need only be estimated based on a static failure criteria. Since the tension of pulley belt are neglected, we'll use the MSS theory (from Chp5) to find the diameter by considering only torque.

-Assuming the fan is held in place by Marine Propeller Shaft Hub, https://www.techboat.com/Bibliotheque/SAE_%20dimension.pdf

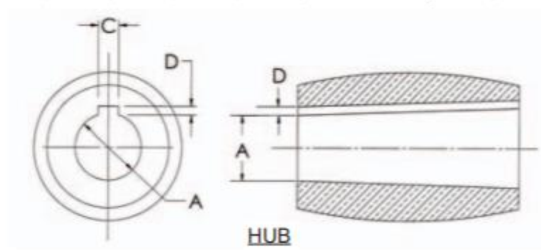
MARINE PROPELLERS HUB BORE DIMENSIONS

Taper: Per Foot = ¾"

Per Inch - 1/16"   Angle with centerline = 1° 47' 24"

Std. Taper	Dia. Small End "A"		Keyway Width "C"			Keyway Side Depth "D"		
	Min.	Max.	Nom.	Min.	Max.	Nom.	Min.	Max.
¾	0.608	0.610	3/16	0.1865	0.1875	3/32	0.098	0.100
⅞	0.710	0.712	¼	0.249	0.250	⅛	0.129	0.131
1	0.812	0.814	¼	0.249	0.250	⅛	0.129	0.131
1⅛	0.913	0.915	¼	0.249	0.250	⅛	0.129	0.131
1¼	1.015	1.017	5/16	0.3115	0.3125	5/32	0.162	0.165
1⅜	1.116	1.118	5/16	0.3115	0.3125	5/32	0.161	0.164
1½	1.218	1.220	⅜	0.374	0.375	3/16	0.195	0.198
1¾	1.421	1.423	7/16	0.4365	0.4375	7/32	0.226	0.229
2	1.624	1.626	½	0.499	0.500	¼	0.259	0.262
2¼	1.827	1.829	9/16	0.561	0.5625	9/32	0.291	0.294
2½	2.030	2.032	⅝	0.6235	0.625	5/16	0.322	0.325
2¾	2.233	2.235	⅝	0.6235	0.625	5/16	0.322	0.325
3	2.437	2.439	¾	0.7485	0.750	5/16	0.323	0.326
3¼	2.640	2.642	¾	0.7485	0.750	5/16	0.323	0.326
3½	2.843	2.845	⅞	0.8735	0.875	5/16	0.324	0.327
3¾	3.046	3.048	⅞	0.8735	0.875	5/16	0.324	0.327
4	3.249	3.251	1	0.9985	1.000	5/16	0.326	0.329
4½	3.796	3.798	1⅛	1.123	1.125	⅜	0.388	0.391
5	4.218	4.220	1¼	1.248	1.250	7/16	0.450	0.453
5½	4.640	4.642	1¼	1.248	1.250	7/16	0.450	0.453
*6	4.749	4.751	1⅜	1.373	1.375	½	0.517	0.520
*6½	5.145	5.147	1⅜	1.373	1.375	½	0.516	0.519
*7	5.541	5.543	1½	1.498	1.500	9/16	0.579	0.582
*7½	5.937	5.939	1½	1.498	1.500	9/16	0.579	0.582
*8	6.332	6.334	1¾	1.748	1.750	9/16	0.582	0.585

Overseas specifications on request.



PROPELLER BORING

To insure retention of inherent factory accuracy, order your propeller factory-bored whenever possible. When bored in the field, propellers should be bored to the pilot hole, NOT to the hub or blade edges.

* 6" through 8" shaft has 1 inch per foot taper, 1/12" per inch taper. Angle with centerline is 2° 23' 9".

-Fan hub bore size of 3/4 or 0.75'' has been selected for point O

-Hence, the factor of safety is estimated at:

$$\tau = \frac{Tr}{J} = \frac{2532(0.75/2)}{\pi(0.75^4)/32} = 30.6 \text{ kpsi}$$

Eq. (5-3): $$n_y = \frac{S_y/2}{\tau} = \frac{57/2}{30.6} = 0.93$$ → Not acceptable

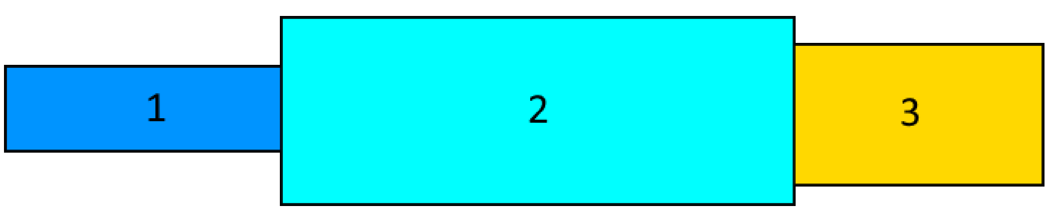
-Now, selecting hub bore size of 7/8 or 0.875'', the factor of safety becomes:

$$\tau = \frac{Tr}{J} = \frac{2532(0.875/2)}{\pi(0.875^4)/32} = 19.2 \text{ kpsi}$$

Eq. (5-3): $$n_y = \frac{S_y/2}{\tau} = \frac{57/2}{19.2} = 1.48 - 1.5$$ → Acceptable, since fan weight is very insignificant.

Hence, D_1 = 0.875''

(b) The deflection will not be much affected by the details of fillet radii, grooves, and keyways, so these can be ignored. Also, the slight diameter changes, as well as the narrow 2.0 in diameter section, can be neglected. Hence, we will model the shaft with the following three sections: 1 mark



Section	Diameter (in)	Length (in)
1	0.875	2.90
2	1.20	7.77
3	1.70	2.20

2 mark

The deflection problem can be solved easily with singularity functions or castigliano's theorem. For examples, see Ex. 4-7, p. 173, or the solution to Prob. 7-24. Alternatively, structural analysis software (such as MDsolids) or finite element software (such as ANSYS, COMSOL, Abaqus) may be used.

Using MDSolids, *initial estimate for the deflection and slope at various locations becomes*:

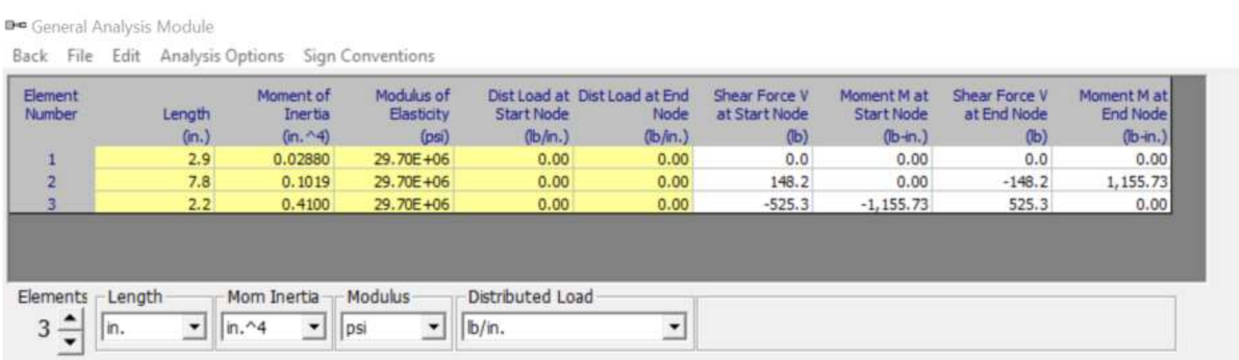
Location	Slope (rad)	Deflection (in)
Left bearing *A*	0.000730	0.000000
Right bearing *F*	0.000864	0.000000
Fan *O*	0.000730	0.002118
Gear *C*	0.000759	0.001823

2 mark

Comparing these values to the recommended limits in Table 7-2, we find that they are all within the recommended range.

1 mark

<u>**Sample Deflection Calculations (Not required in grading)**</u>

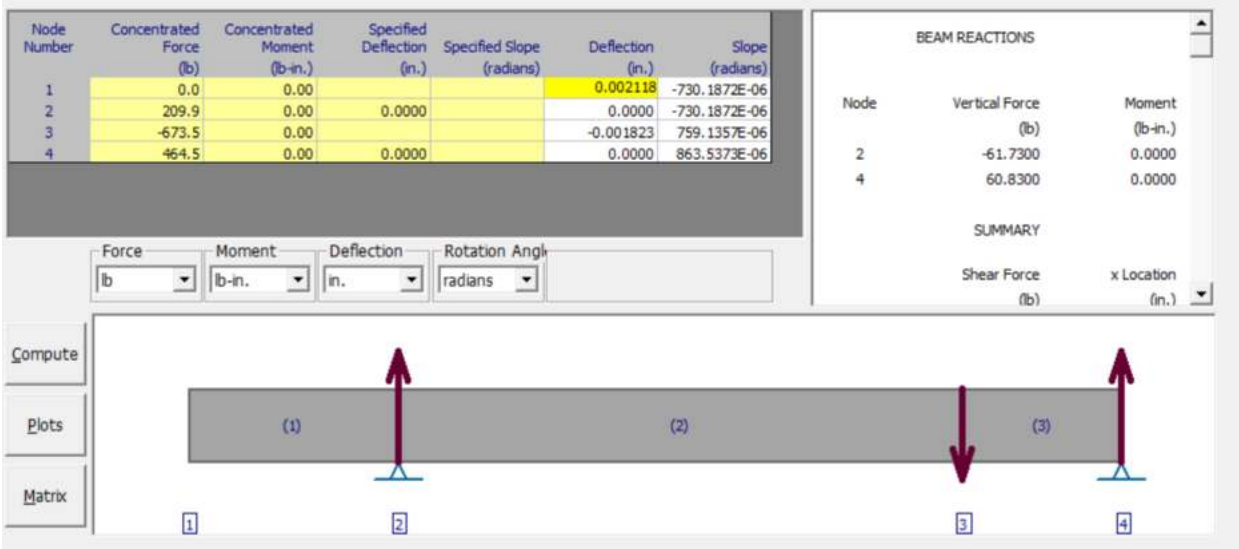


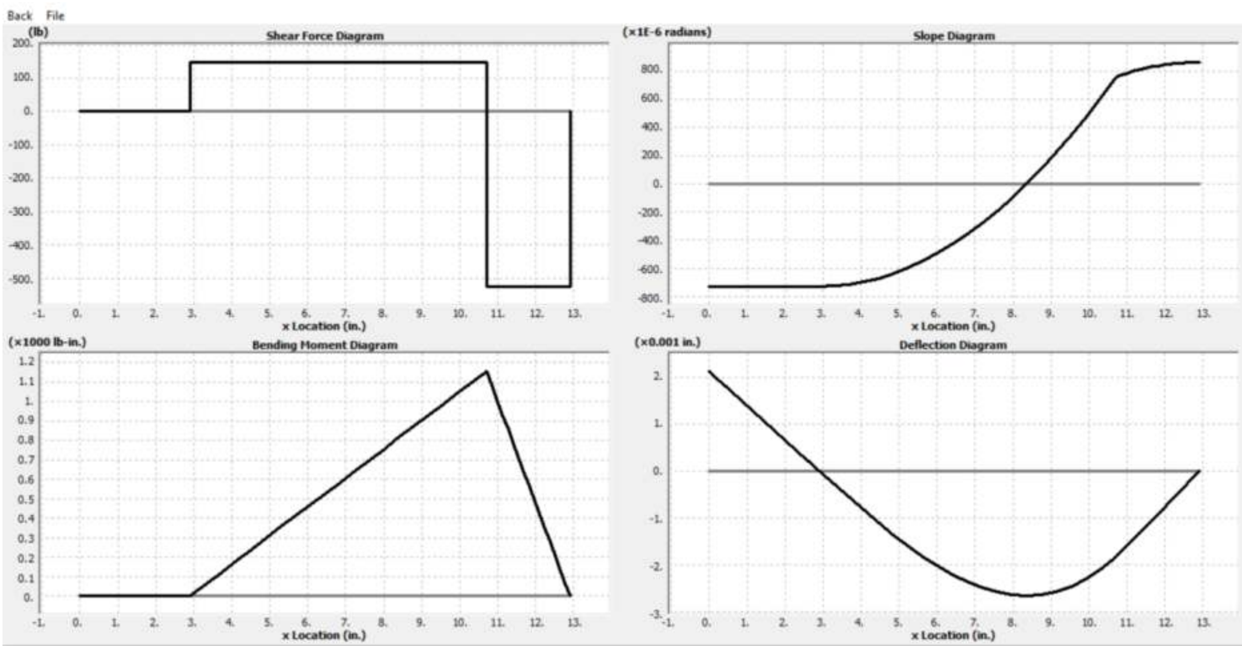
General Analysis Module

Back File Edit Analysis Options Sign Conventions

Element Number	Length (in.)	Moment of Inertia (in.^4)	Modulus of Elasticity (psi)	Dist Load at Start Node (lb/in.)	Dist Load at End Node (lb/in.)	Shear Force V at Start Node (lb)	Moment M at Start Node (lb-in.)	Shear Force V at End Node (lb)	Moment M at End Node (lb-in.)
1	2.9	0.02880	29.70E+06	0.00	0.00	0.0	0.00	0.0	0.00
2	7.8	0.1019	29.70E+06	0.00	0.00	148.2	0.00	-148.2	1,155.73
3	2.2	0.4100	29.70E+06	0.00	0.00	-525.3	-1,155.73	525.3	0.00

Elements 3 Length in. Mom Inertia in.^4 Modulus psi Distributed Load lb/in.


Node Number
Concentrated Force (lb)
Concentrated Moment (lb-in.)
Specified Deflection (in.)
Specified Slope (radians)
Deflection (in.)
Slope (radians)
1 0.0 0.00 0.002118 -730.1872E-06
2 209.9 0.00 0.0000 0.0000 -730.1872E-06
3 -673.5 0.00 -0.001823 759.1357E-06
4 464.5 0.00 0.0000 0.0000 863.5373E-06
BEAM REACTIONS
Node Vertical Force (lb) Moment (lb-in.)
2 -61.7300 0.0000
4 60.8300 0.0000
SUMMARY
Shear Force (lb) x Location (in.)
Force lb
Moment lb-in.
Deflection in.
Rotation Angl radians
Compute
Plots
Matrix
(1)
(2)
(3)
1
2
3
4


Back File
Shear Force Diagram
(lb)
Slope Diagram
(x1E-6 radians)
Bending Moment Diagram
(x1000 lb-in.)
Deflection Diagram
(x0.001 in.)
x Location (in.)