طراحی اجزاء ۱

فصل هفتم: طراحی شفت

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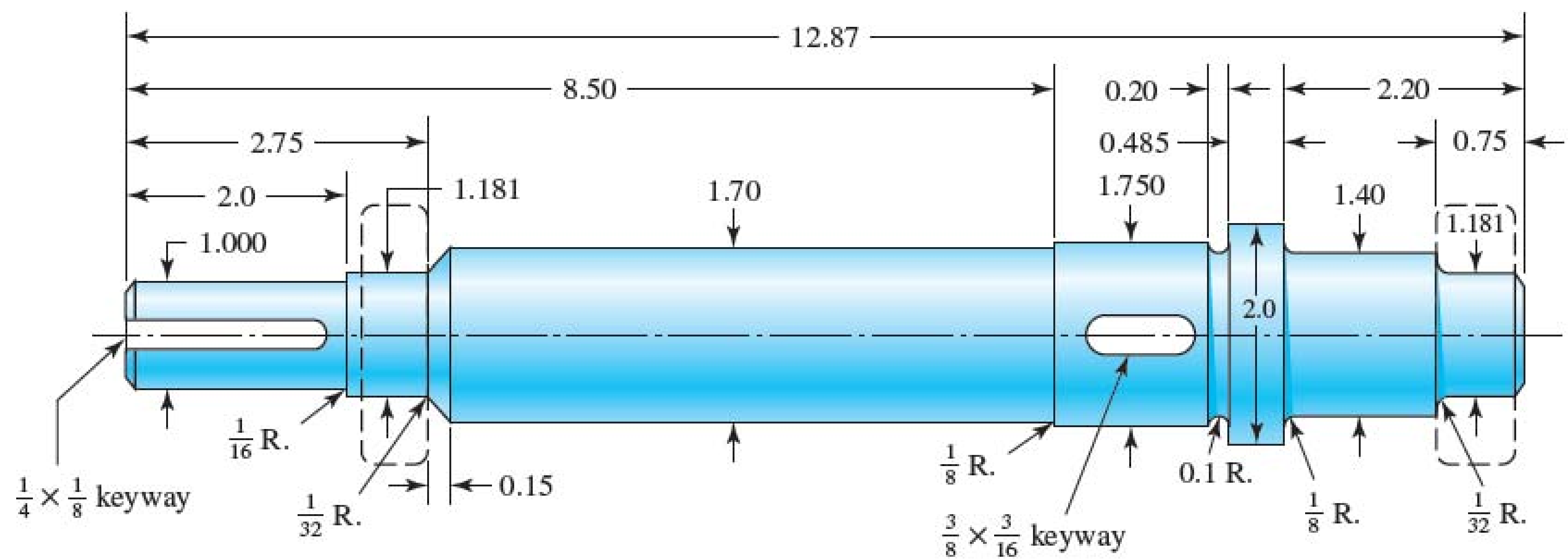
Design of Mechanical Elements

جلسه پانزدهم

شفت یا میل محور یکی از المانهای اصلی ماشین آلات صنعتی می باشد. شفت اغلب توسط دو بیرینگ مهار می شود و قطعاتی مثل چرخدنده، پولی و چرخ زنجیر روی آن سوار می شود



Material Shaft

Necessary strength to resist loading stresses affects the choice of materials and their treatments. Many shafts are made from low-carbon, cold-drawn or hot-rolled steel, such as AISI 1020–1050 steels.

AISI 1045 ≡ CK45

becomes an issue. The cost of the material and its processing must be weighed against the need for smaller shaft diameters. When warranted, typical alloy steels for heat treatment include AISI 1340–50, 3140–50, 4140, 4340, 5140, and 8650.

فولادهایی که مناسب انجام عملیات حرارتی هستند.

Shafts usually don't need to be surface hardened unless they serve as the actual journal of a bearing surface. Typical material choices for surface hardening include carburizing grades of AISI 1020, 4320, 4820, and 8620.

مناسب سخت کاری سطح

فولادهای متداول موجود در بازار

✓ CK45 or AISI 1045

$S_{ut} \simeq 600\ MPa$ ، $S_y \simeq 450\ MPa$

$S_{ut} = 570$ تا 700

آهن معمولی

st37

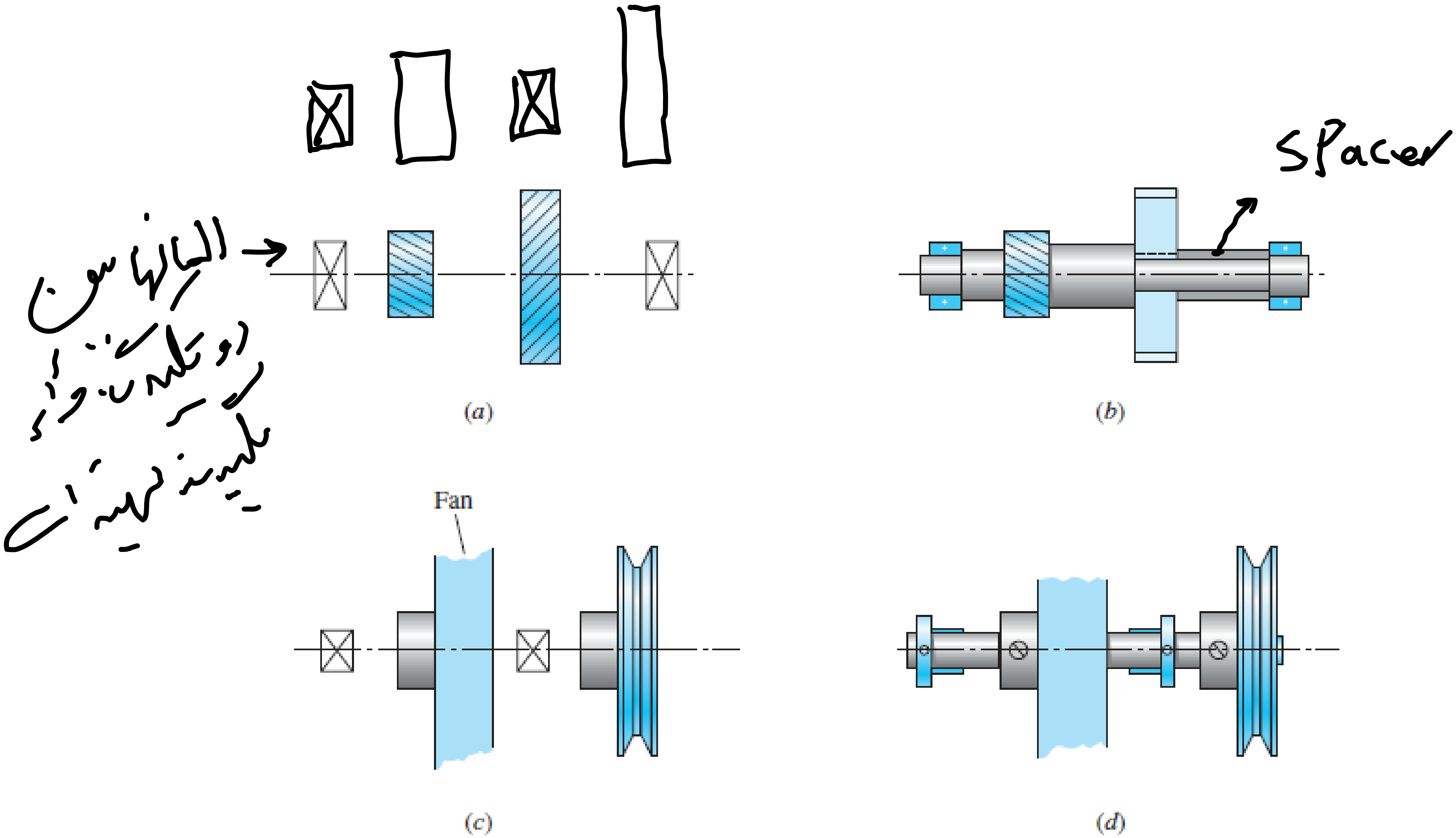
$S_{ut} = 370\ MPa$

Mo40 AISI 4140

$S_{ut} = 700\ MPa$ ، $S_y = 420\ MPa$

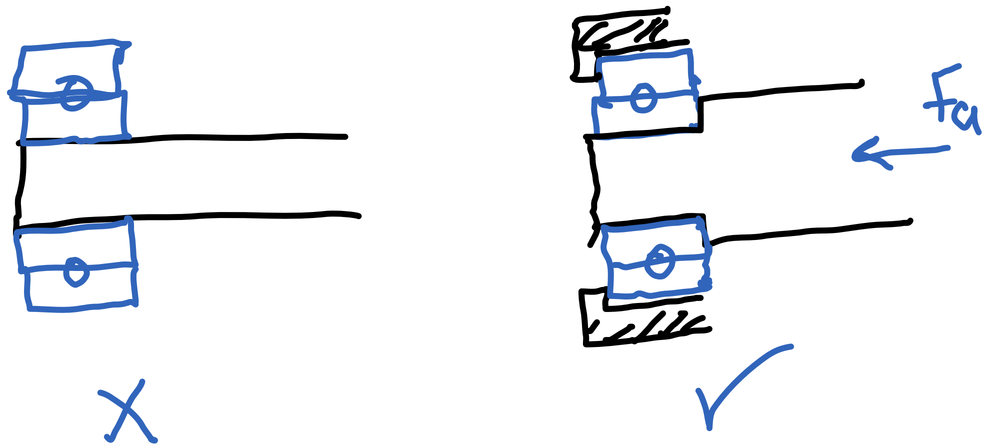
$S_{ut} = 700$ تا 850 $\xrightarrow[\text{حرارتی}]{\text{عملیات}}$ $S_{ut} = 800$ تا $1000\ MPa$

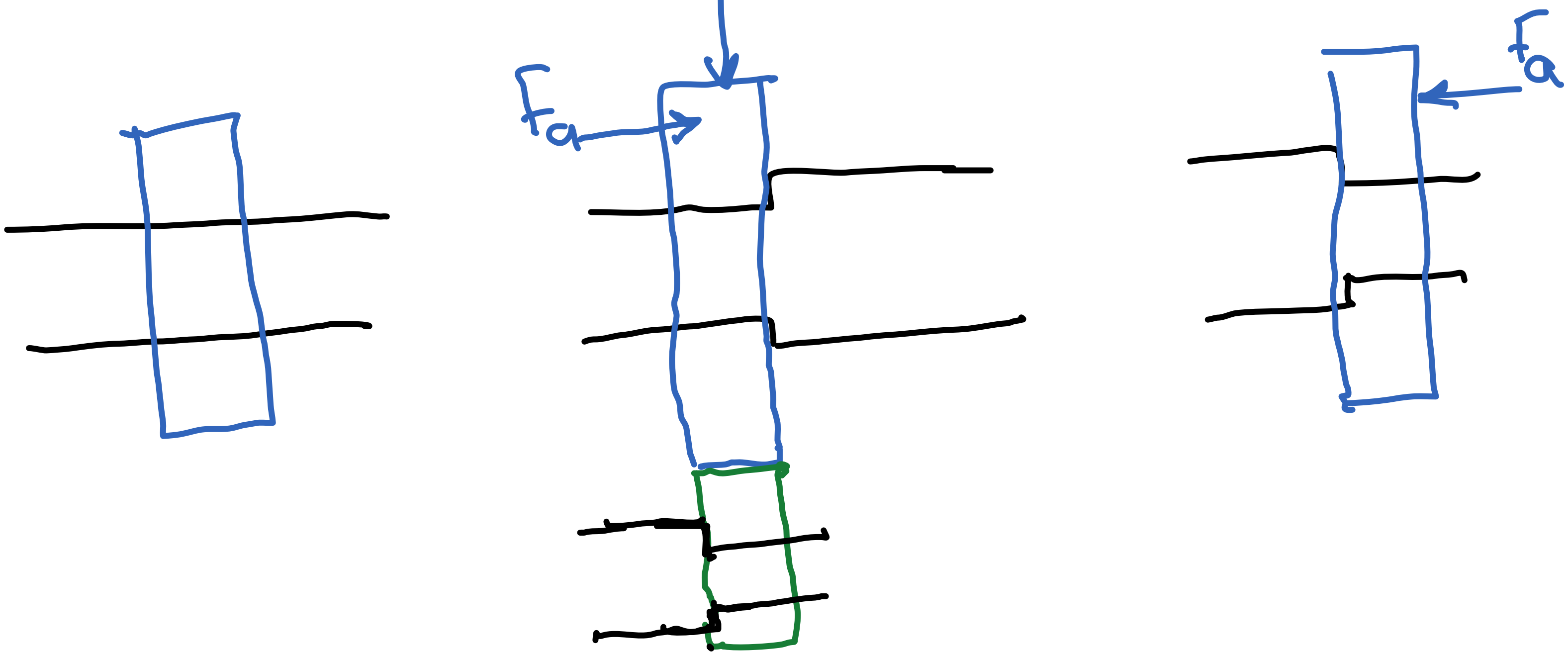
Shaft layout طرح بندی شفت

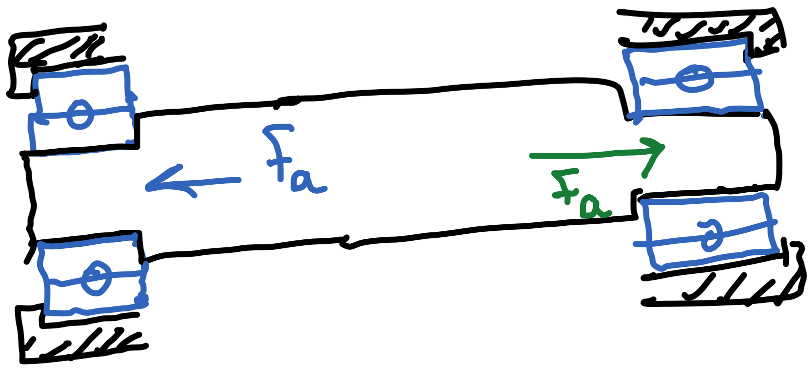


(a) (b) (c) (d)

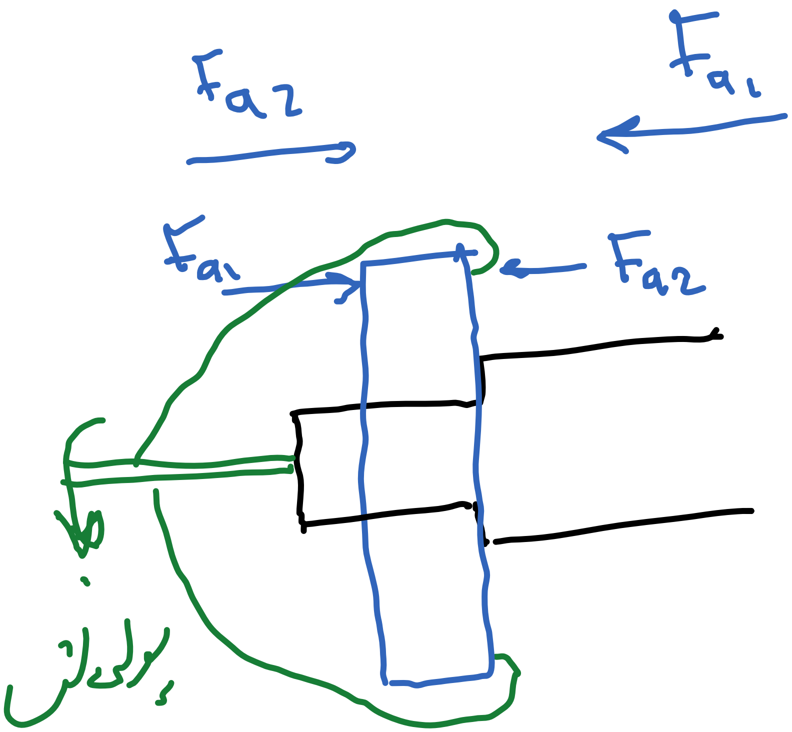
تحمل بار محوری







اگر بار محوری دو طرفه باشد



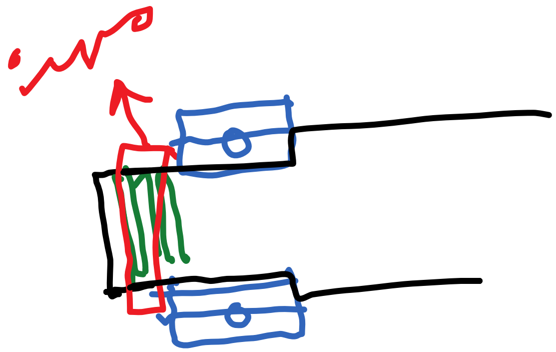
جا زدن به صورت فشاری وقتی

shrink fit

اما دمونتاژ آن سخت است

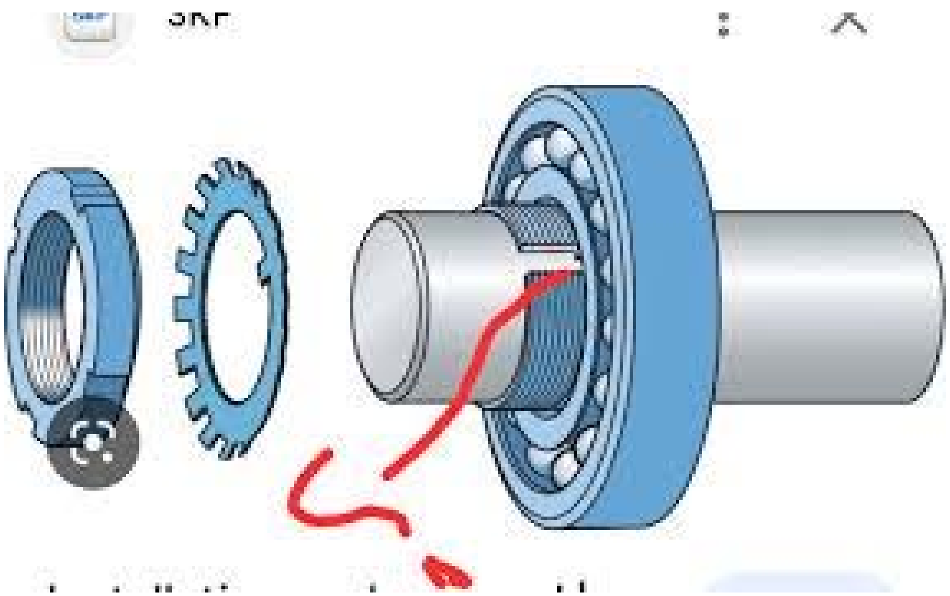
جا زدن با گیره یا با پرس

در آوردن ، یا پولی کش



استفاده از مهره در انتهای شفت

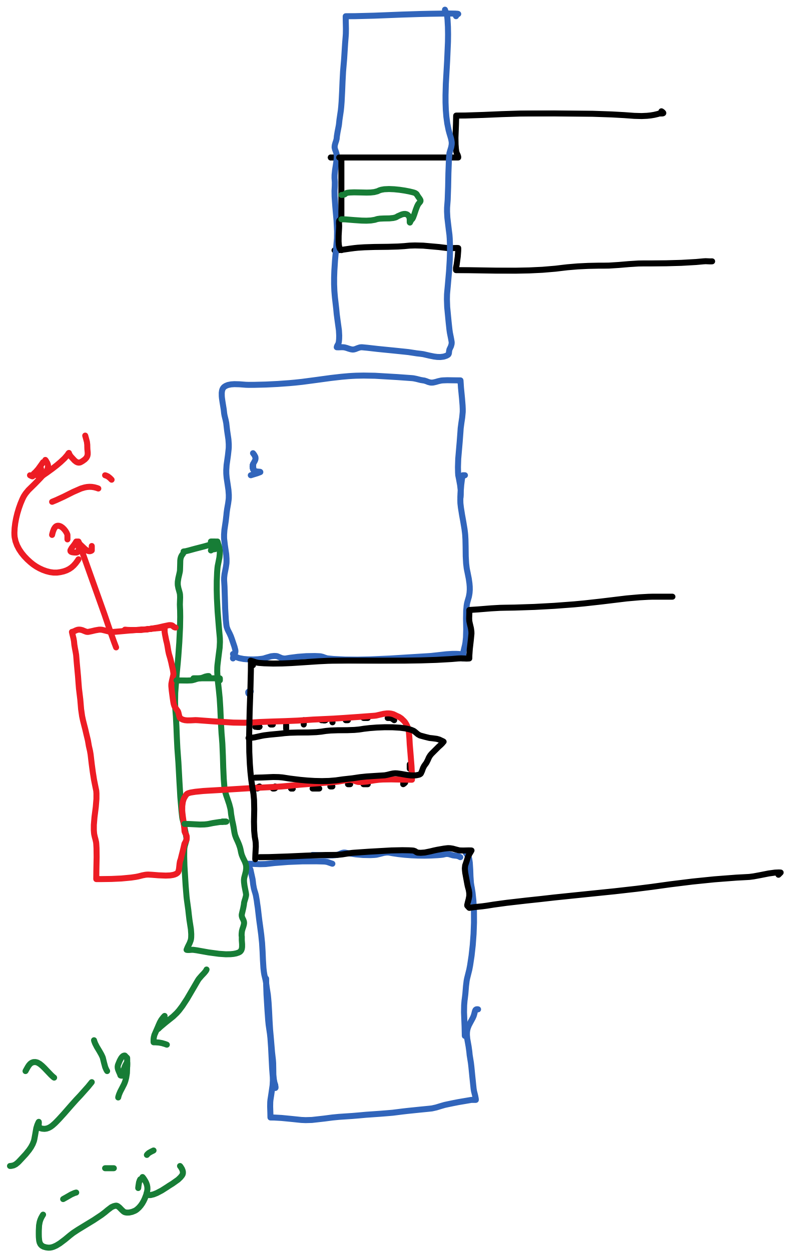
locknut

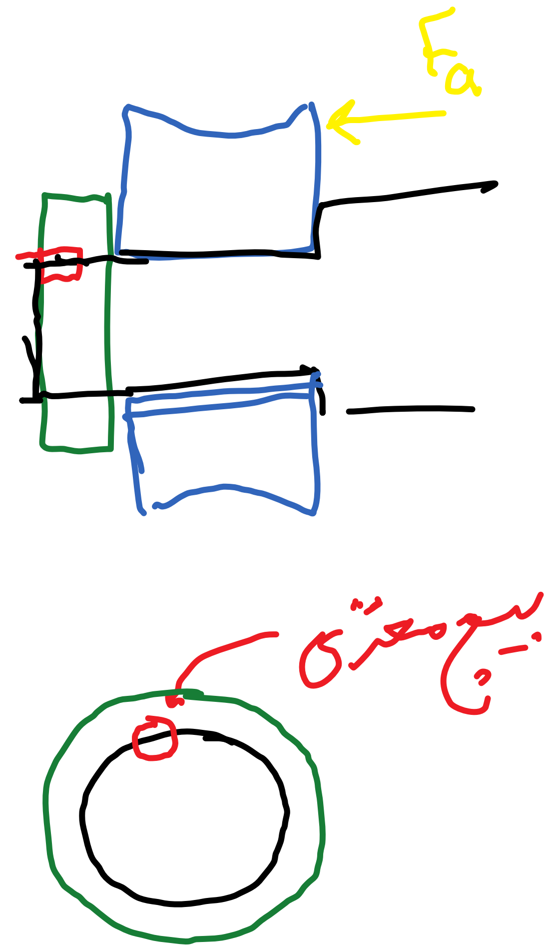


مهره چاکنت

SKf → bearing

واشر چاکنت

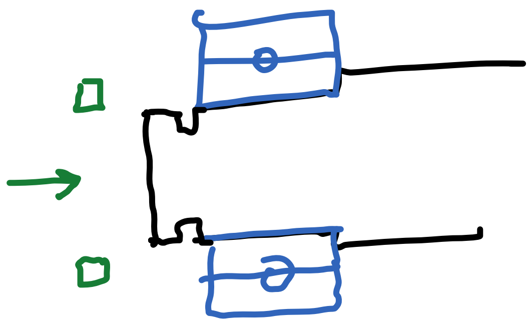

پیچ
واشر تخت


Fa
پیچ مغزی

برای حالتی که بار محوری کمی داشته باشیم

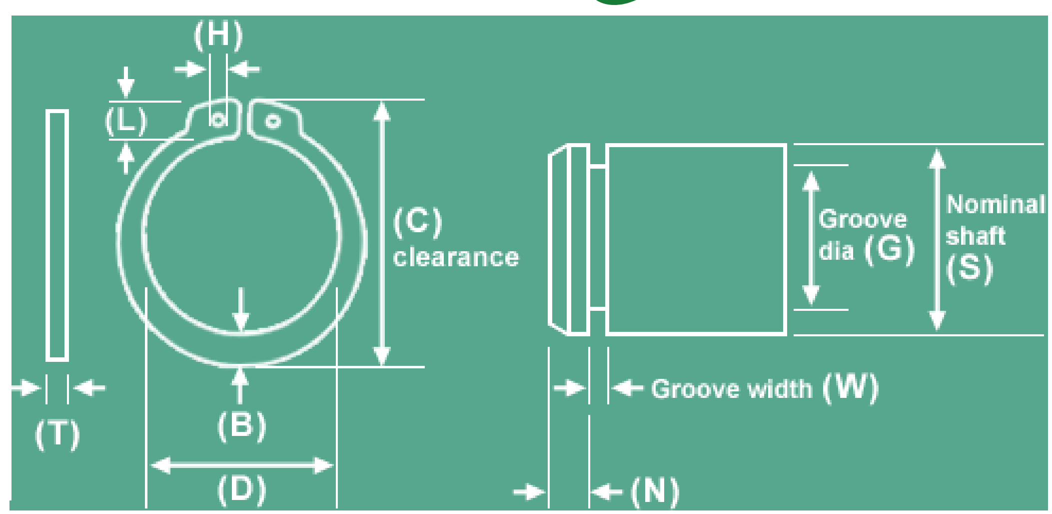
خار فنری چشمی

Retaining ring

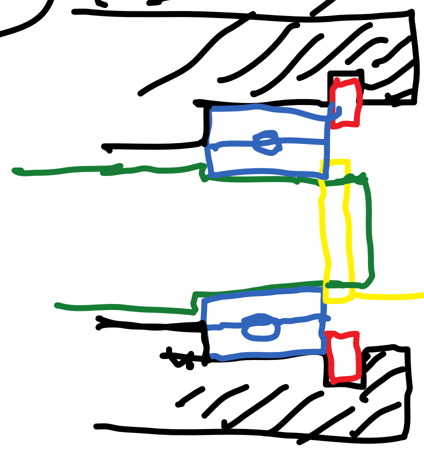


a external خارجی

روی شفت



هوزینگ گیربکس



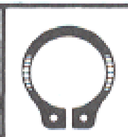
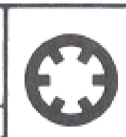
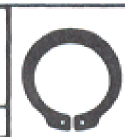
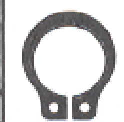
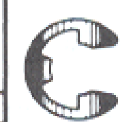
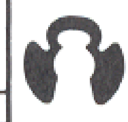
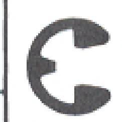
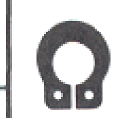
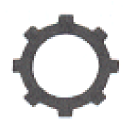
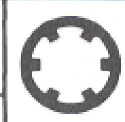
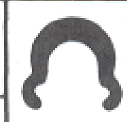
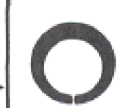
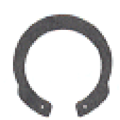
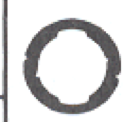
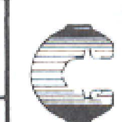
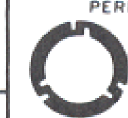
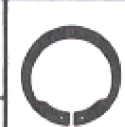
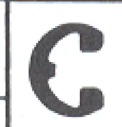
خار فنری داخلی

خار فنری خارجی

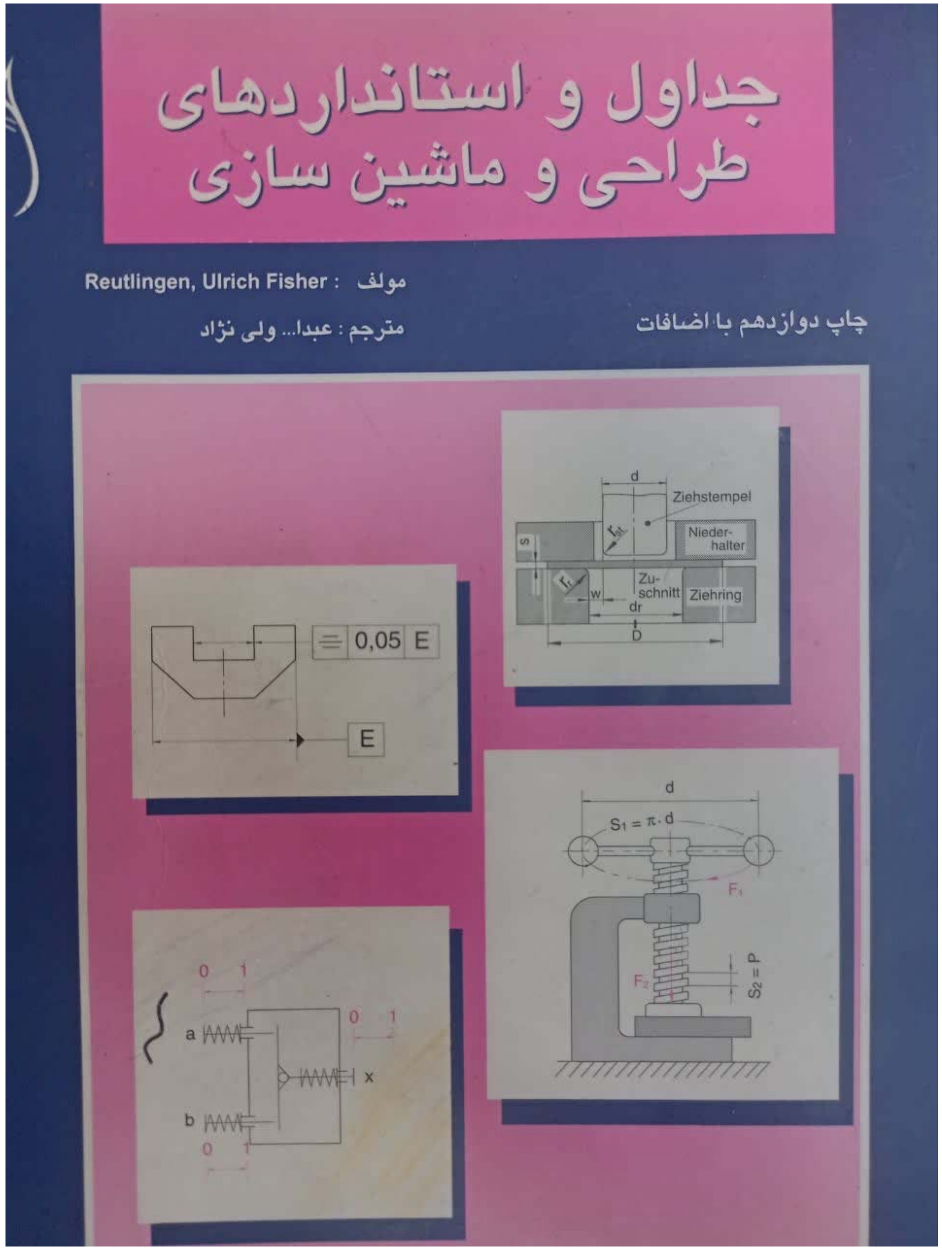
داخلی

خارجی


INTERNAL
EXTERNAL
C-STYLE SIDE MOUNT
E-STYLE SIDE MOUNT
KLIPRING SIDE MOUNT
CRIMP
PUSH ON
SPIRAL

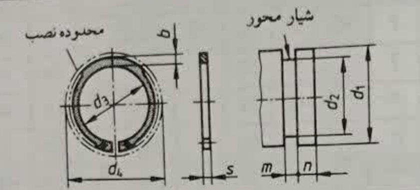
					Size Range	
	INTERNAL	BASIC	N5000	For housings and bores	.250—10.0 in	6.4—254.0 mm
	EXTERNAL	BOWED	5101*	For shafts and pins	.188—1.750 in.	4.8—44.4 mm.
	EXTERNAL	REINFORCED	5115	For shafts and pins	.094—1.0 in	•
	EXTERNAL	HEAVY-DUTY	5160	For shafts and pins	.394—2.0 in.	10.0—50.8 mm.
	INTERNAL	BOWED	N5001*	For housings and bores	.250—1.750 in.	6.4—44.4 mm.
	EXTERNAL	BEVELED	5102	For shafts and pins	1.0—10.0 in	25.4—254.0 mm
	EXTERNAL	BOWED E-RING	5131	For shafts and pins	.110—1.375 in	2.8—34.9 mm
	EXTERNAL	KLIPRING®	5304 T-5304	For shafts and pins	.156—1.000 in.	4.0—25.4 mm.
	INTERNAL	BEVELED	°N5002/*N5003	For housings and bores	° 1.0—10.0 in. * 1.56—2.81 in.	° 25.4—254.0 mm * 39.7—71.4 mm
	EXTERNAL	CRESCENT®	5103	For shafts and pins	.125—2.0 in.	3.2—50.8 mm.
	EXTERNAL	E-RING	5133	For shafts and pins	.040—1.375 in.	1.0—34.9 mm.
	EXTERNAL	GRIPRING®	5555	For shafts and pins	.079—.750 in.	2.0—19.0 mm.
	INTERNAL	CIRCULAR	5005	For housings and bores	.312—2.0 in	•
	EXTERNAL	CIRCULAR	5105	For shafts and pins	.094—1.0 in.	•
	EXTERNAL	RADIAL GRIPRING®	5135	for shafts and pins	.094—.375 in.	2.4—9.5 mm.
	EXTERNAL	HIGH-STRENGTH	5560*	For shafts and pins	.101—.328 in.	•
	INTERNAL	INVERTED	5008	For housings and bores	.750—4.0 in.	19.0—101.6 mm.
	EXTERNAL	INTERLOCKING	5107*	For shafts and pins	.469—3.375 in.	11.9—85.7 mm.
	EXTERNAL	PRONG-LOCK®	5139*	For shafts and pins	.092—.438 in.	•
	EXTERNAL	PERMANENT SHOULDER	5590*	For shafts and pins	.250—.750	6.4—19.0 mm
	EXTERNAL	BASIC	5100	For shafts and pins	.125—10.0 in	3.2—254.0 mm.
	EXTERNAL	INVERTED	5108	For shafts and pins	.500—4.0 in.	12.7—101.6 mm.
	EXTERNAL	REINFORCED E-RING	5144	For shafts and pins	.094—.562 in.	2.4—14.3 mm.

* Non-Stocking Ring Type: Available on special order only


جداول و استانداردهای
طراحی و ماشین سازی
مولف : Reutlingen, Ulrich Fisher
مترجم : عبدا... ولی نژاد
چاپ دوازدهم با اضافات
0,05 E
E
d
Ziehstempel
Nieder-
halter
Zu-
schnitt
Ziehring
dr
D
w
s
d
S1 = π · d
F1
F2
S2 = P
a
b
x
0
1

خار فنری، خار واشری

خار فنری محور — مقایسه با DIN 471 (9.81)

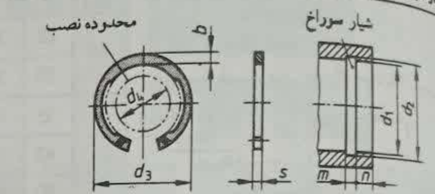


اندازه نامی d_1 mm	خار فنری s	d_3	d_4	b ≈	شیار d_2	m min	n min.
10	1	9,3	17	1,8	9,6	1,1	0,6
12	1	11	19	1,8	11,5	1,1	0,8
15	1	13,8	22,6	2,2	14,3	1,1	1,1
18	1,2	16,5	26,2	2,4	17	1,3	1,5
20	1,2	18,5	28,4	2,6	19	1,3	1,5
22	1,2	20,5	30,8	2,8	21	1,3	1,5
25	1,2	23,2	34,2	3	23,9	1,3	1,7
28	1,5	25,9	37,9	3,2	26,6	1,6	2,1
30	1,5	27,9	40,5	3,5	28,6	1,6	2,1
32	1,5	29,6	43	3,6	30,3	1,6	2,6
35	1,5	32,2	46,8	3,9	33	1,6	3
38	1,75	35,2	50,2	4,2	36	1,85	3
40	1,75	36,5	52,6	4,4	37,5	1,85	3,8
42	1,75	38,5	55,7	4,5	39,5	1,85	3,8
45	1,75	41,5	59,1	4,7	42,5	1,85	3,8
48	1,75	44,5	62,5	5	45,5	1,85	3,8
50	2,0	45,8	64,5	5,1	47,0	2,15	4,5
60	2,0	55,8	75,6	5,8	57,0	2,15	4,5
65	2,5	60,8	81,4	6,3	62,0	2,65	4,5
70	2,5	65,5	87	6,0	67,0	2,65	4,5
75	2,5	70,5	92,7	7,0	72,0	2,65	4,5
80	2,5	74,5	98,1	7,4	76,5	2,65	5,3
90	3,0	84,5	108,5	8,2	86,5	3,15	5,3

مشخصه خارفنری با $d_1 = 40$ mm و $s = 1{,}75$ mm :

خارفنری DIN 471 – 40 × 1,75

خار فنری — خار فنری سوراخ — مقایسه با DIN 472 (9.81)



اندازه نامی d_1 mm	خارفنری s	d_3	d_4	b ≈	شیار d_2	m min.	n min.
10	1	10,8	3,3	1,4	10,4	1,1	0,6
12	1	13	4,9	1,7	12,5	1,1	0,8
15	1	16,2	7,2	2	15,7	1,1	1,1
18	1	19,5	9,4	2,2	19	1,1	1,5
20	1	21,5	11,2	2,3	21	1,1	1,5
22	1	23,5	13,2	2,5	23	1,1	1,5
25	1,2	26,9	15,5	2,7	26,2	1,3	1,8
28	1,2	30,1	17,9	2,9	29,4	1,3	2,1
30	1,2	32,1	19,9	3	31,4	1,3	2,1
32	1,2	34,4	20,6	3,2	33,7	1,3	2,6
35	1,5	37,8	23,6	3,4	37	1,6	3
38	1,5	40,8	26,4	3,7	40	1,6	3
40	1,75	43,5	27,8	3,9	42,5	1,85	3,8
42	1,75	45,5	29,6	4,1	44,5	1,85	3,8
45	1,75	48,5	32	4,3	47,5	1,85	3,8
48	1,75	51,5	34,5	4,5	50,5	1,85	3,8
50	2,0	54,2	36,3	4,6	53,0	2,15	4,5
60	2,0	64,2	44,7	5,4	63,0	2,15	4,5
65	2,5	69,2	49,0	5,8	68,0	2,65	4,5
72	2,5	76,5	55,6	6,4	75,0	2,65	4,5
75	2,5	79,5	58,6	6,6	78,0	2,65	4,5
80	2,5	85,5	62,1	7,0	83,5	2,65	5,3
90	3,0	95,5	71,9	7,6	93,5	3,15	5,3

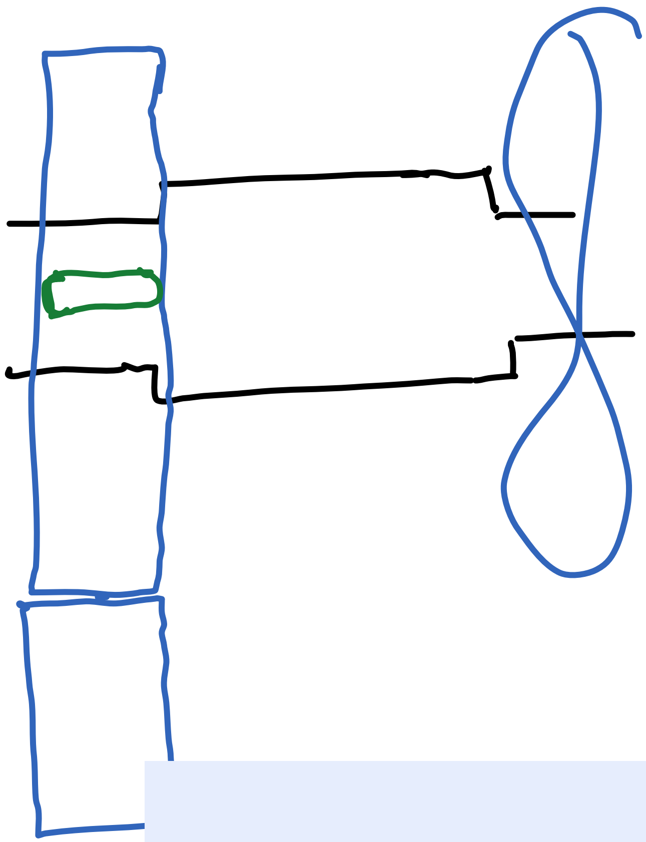
مشخصه خارفنری $d_1 = 80$ mm و $s = 2{,}5$ mm :

خارفنری DIN 472 – 80 × 2,5

خار واشری

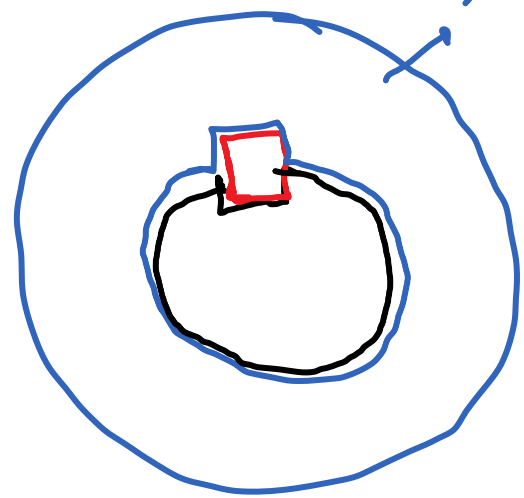
مقایسه با DIN 6799 (9.81)

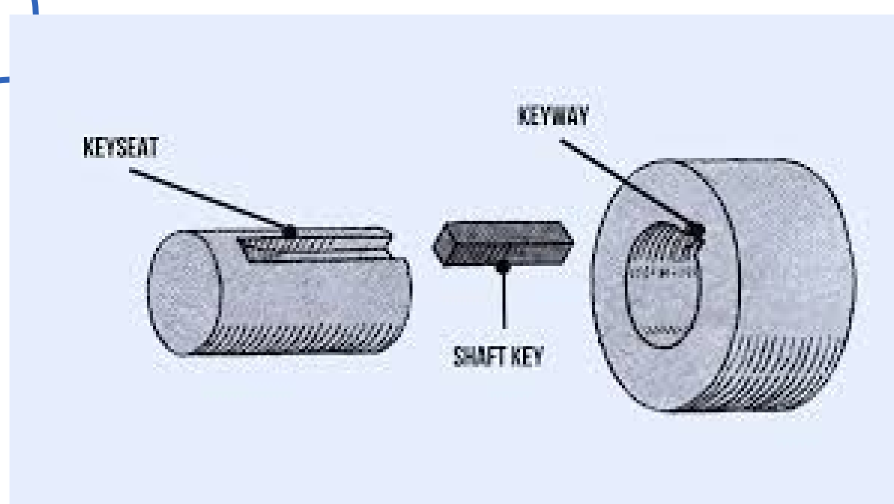
انتقال گشتاور



ورودی

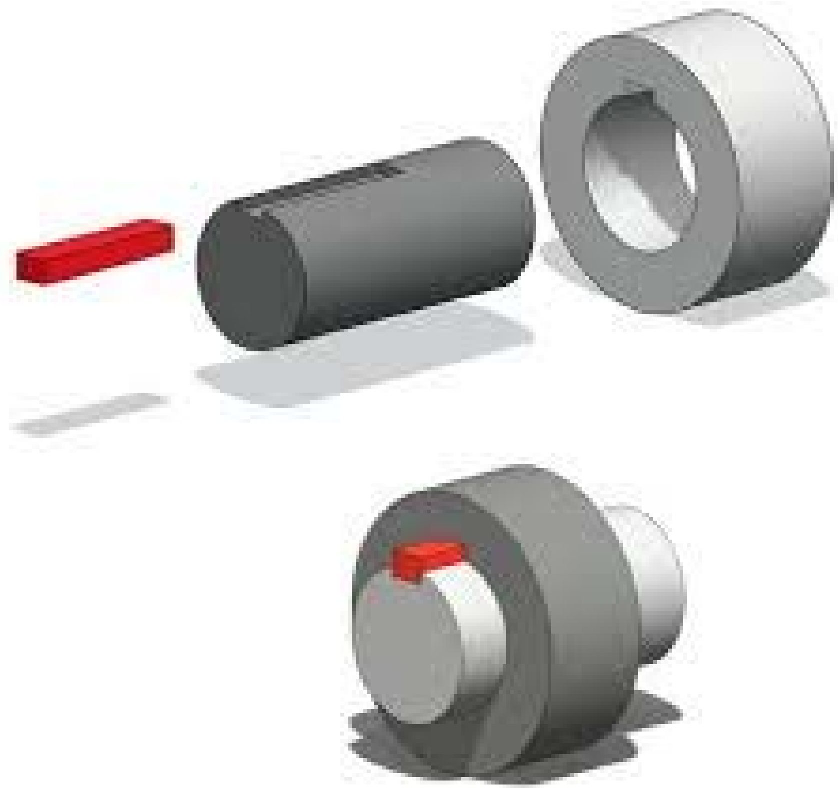
چرخدنده





خار Key

Parallel Key



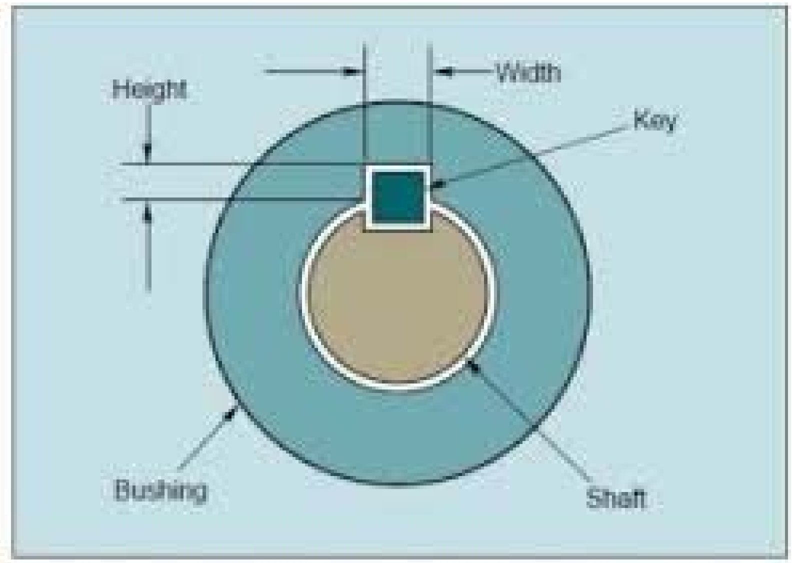
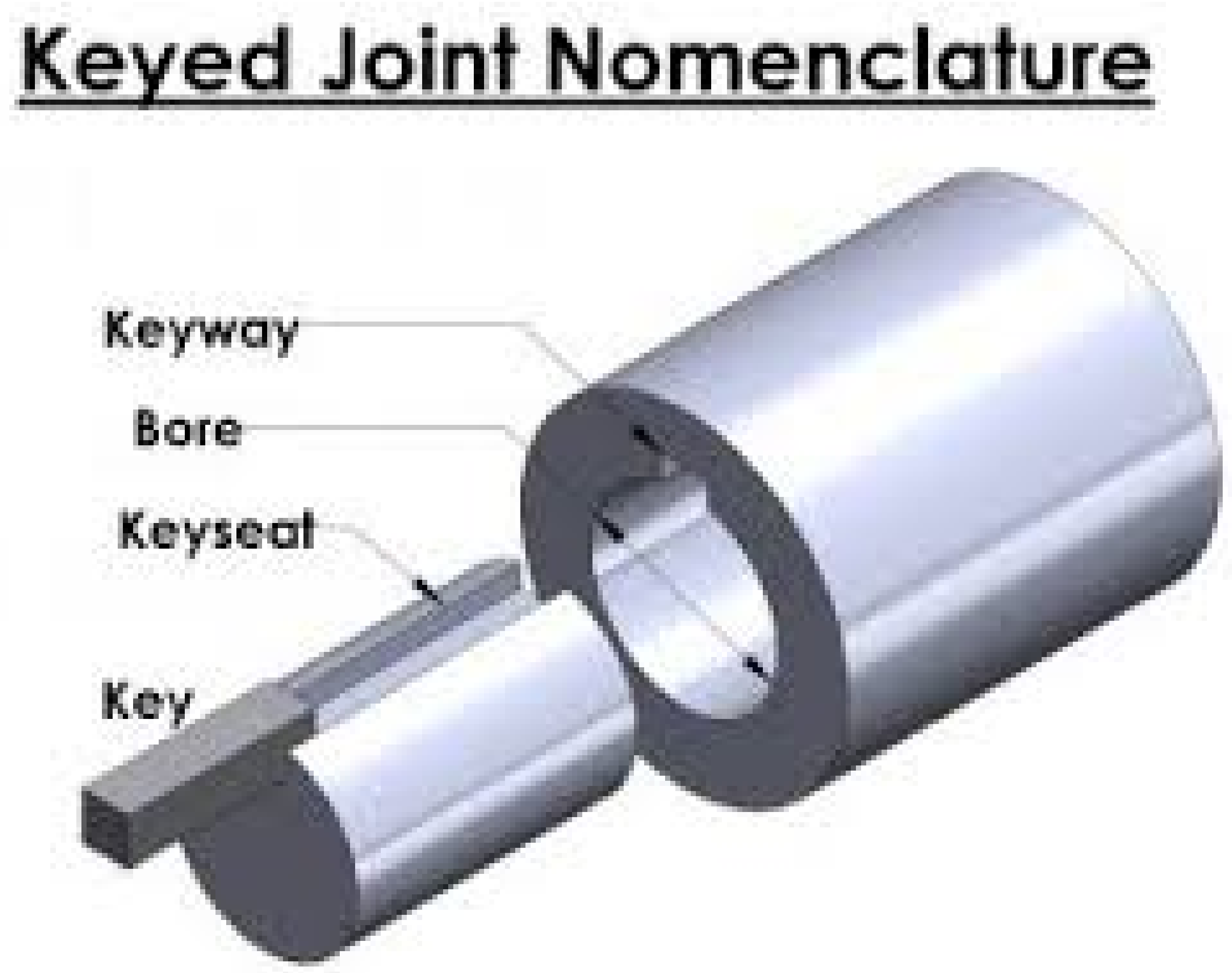


Figure 3 — Key fits snugly between bushing and shaft to lock them together radially.

جاخار روی شفت با فرز انگشتی

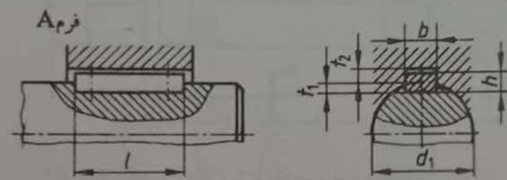
جاخار روی هاب با دستگاه کله زنی


Keyed Joint Nomenclature
Keyway
Bore
Keyseat
Key


خار مربعی
خار موازی
خار مستطیلی
خار سدل
هزارخار
خار زبانه دار
خار لغزشی
خار هلالی
خار مماس

میخ پرچهای فشنگی با محل شکست

خارهای تخت (نوع بلند) ، جای خارها — مقایسه با (8.68) DIN 6885 T1



فرم A ، با انتهای جای خار قوسی شکل

فرم B ، با انتهای جای خار تخت

تلرانس جای خار		
b عرض جای خار محور	نشیمن محکم نشیمن آزاد	P 9 N 9
b عرض جای خار توپی	نشیمن محکم نشیمن آزاد	P 9 JS 9
t_1 عمق جای خار محور	تلرانس مجاز	+ 0,1...0,2
t_2 عمق جای خار توپی	تلرانس مجاز	+ 0,1...0,2

d_1 از	6	8	10	12	17	22	30	38	44	50	58	65	75	85	95	110
d_1 تا	8	10	12	17	22	30	38	44	50	58	65	75	85	95	110	130
b	2	3	4	5	6	8	10	12	14	16	18	20	22	25	28	32
h	2	3	4	5	6	7	8	8	9	10	11	12	14	14	16	18
t_1	1,2	1,8	2,5	3	3,5	4	5	5	5,5	6	7	7,5	9	9	10	11
t_2	1	1,4	1,8	2,3	2,8	3,3	3,3	3,3	3,8	4,3	4,4	4,9	5,4	5,4	6,4	7,4
l از	6	6	8	10	14	18	20	28	36	45	50	56	63	70	80	90
l تا	20	36	45	56	70	90	110	140	160	180	200	220	250	280	320	360
طول l نامی	6,8,10,12,14,16,18,20,22,25,28,32,36,40,45,50,56,63,70,80,90,100,110,125,140,160,180 200, 220, 250, 280, 320mm															

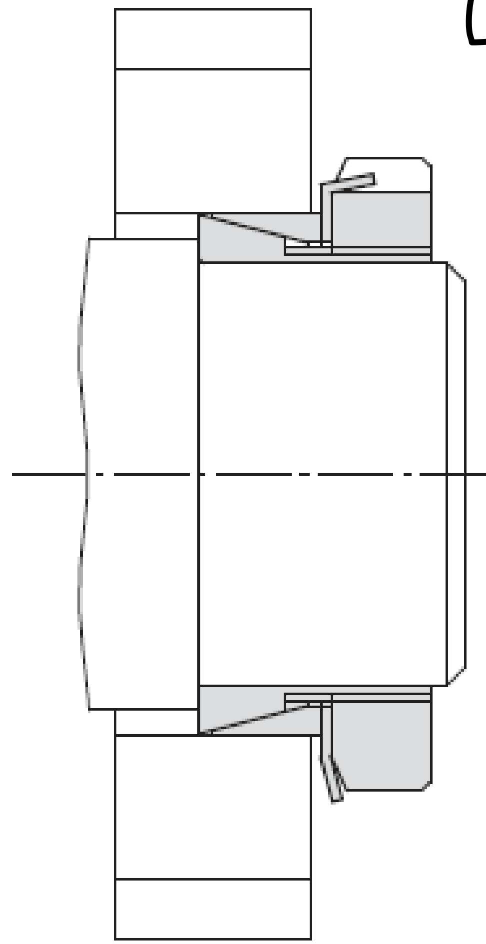
مشخصه خار تخت فرم A , b =12 mm , l =56 mm: خار تخت DIN 6885 – A 12 × 8 × 56

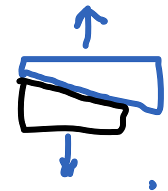
خارهای تخت ، جا خارها و میخ پرچهای فشنگی — مقایسه با (07.85) DIN 7337

فرم A، سر تخت	اندازه نامی d_1	d_1	سوراخ میخ پرچ	k

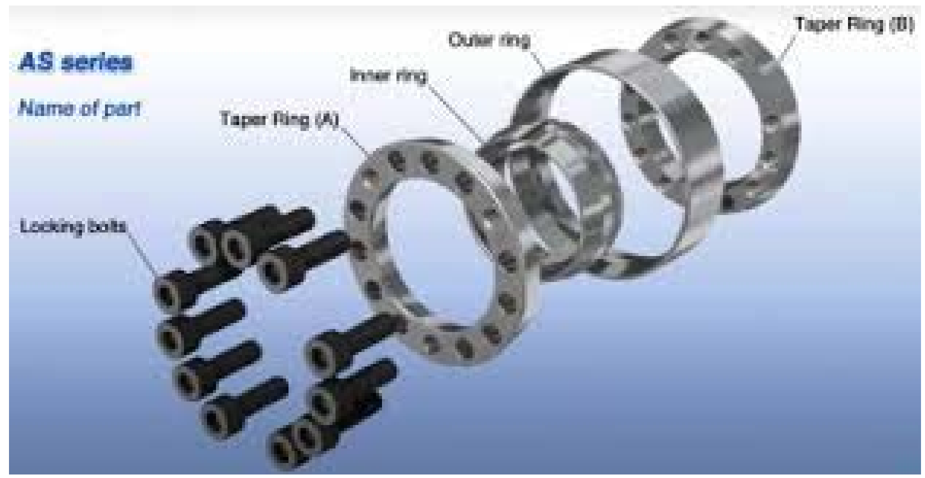
Cone clamping element

or tapered fit



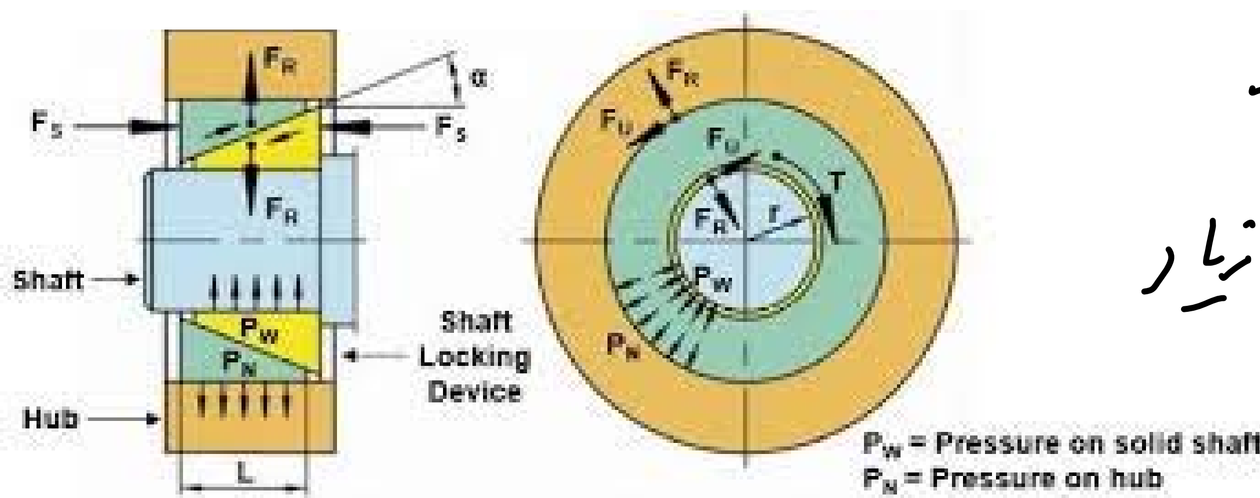


Power lock , shaft hub Locking

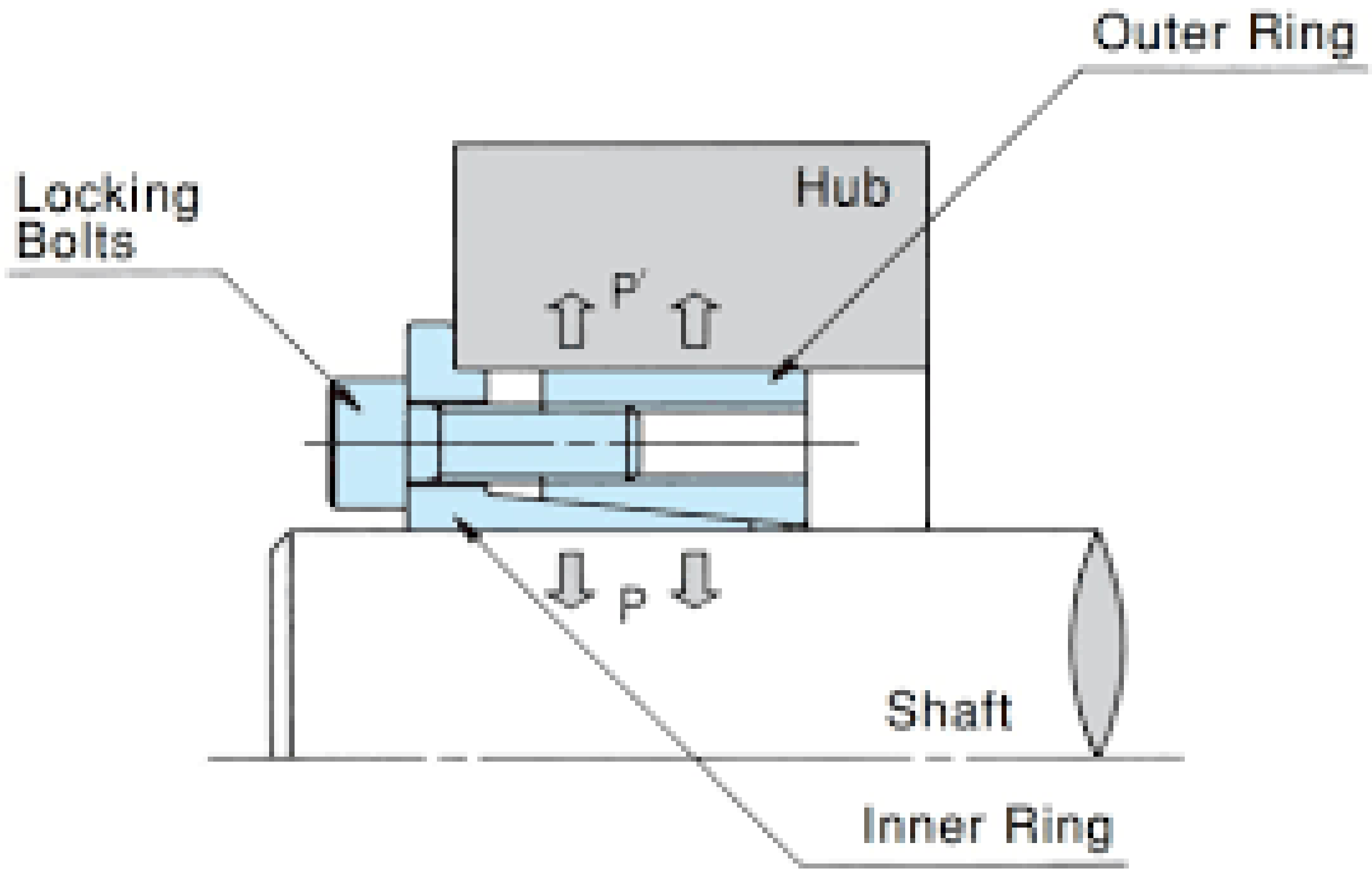


بدون نیاز به خار

تحمل بار محوری زیاد

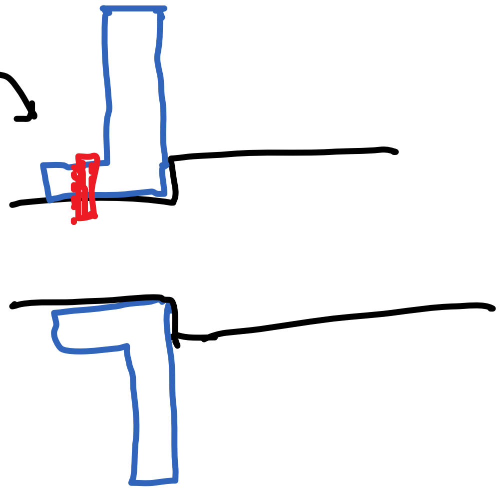


Construction and Working of Shaft Locking Devices



نیازی به خار نیست ، نیاز به جای خار نداریم ،
یعنی حذف می شود ، تمرکز تنش ناشی از جای خار را نداریم

پیچ پایین
ثابت می‌شود



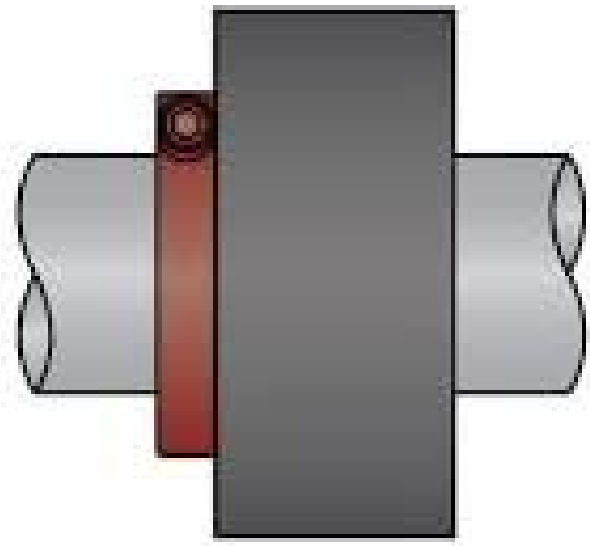
مشابه پولی کولر آبی

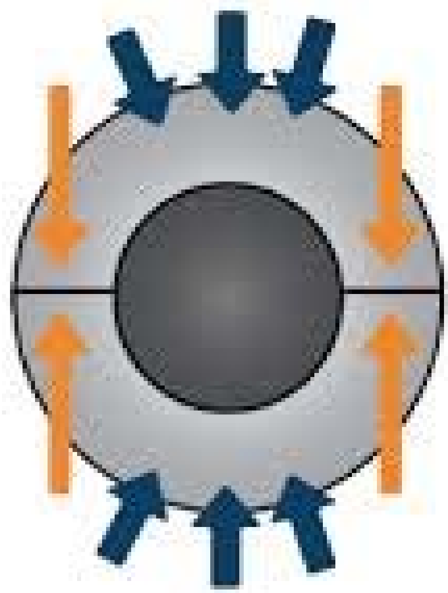
قفل کن شفت

shaft collar
set screw

بست شفت

shaft collar





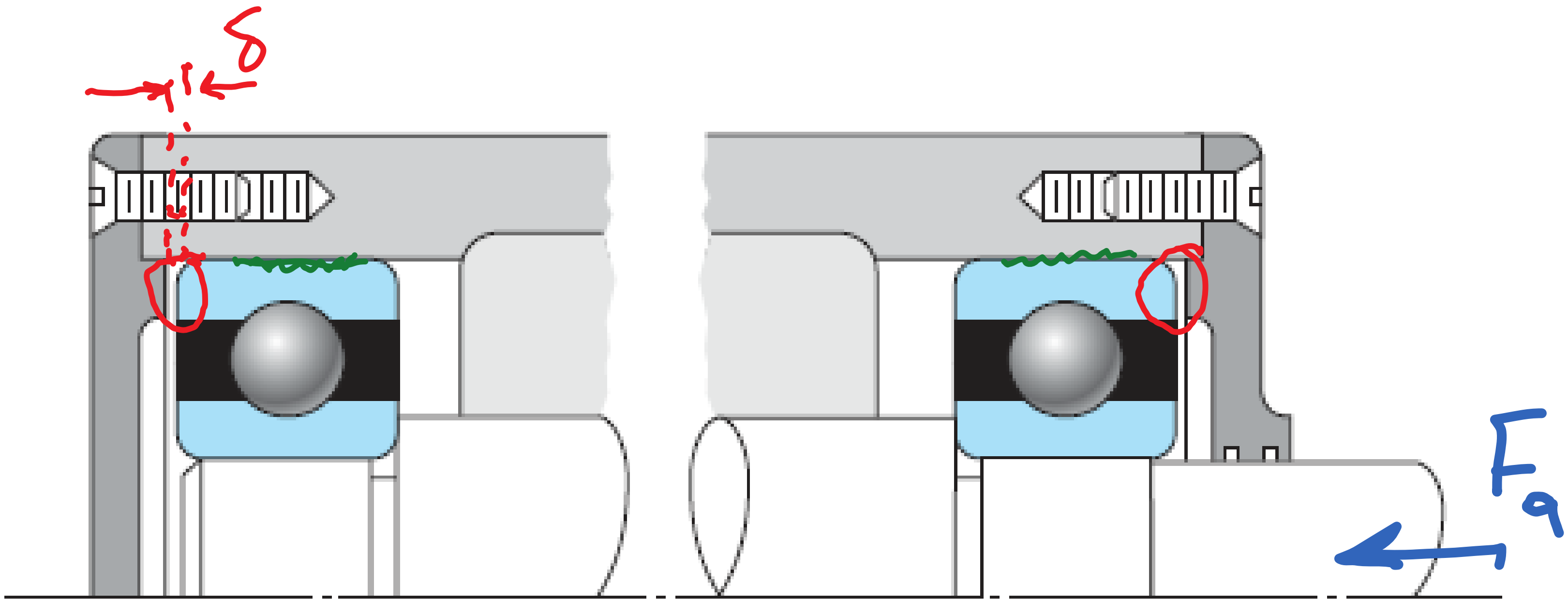
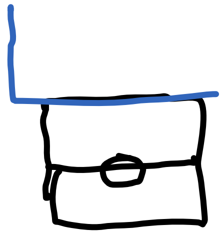
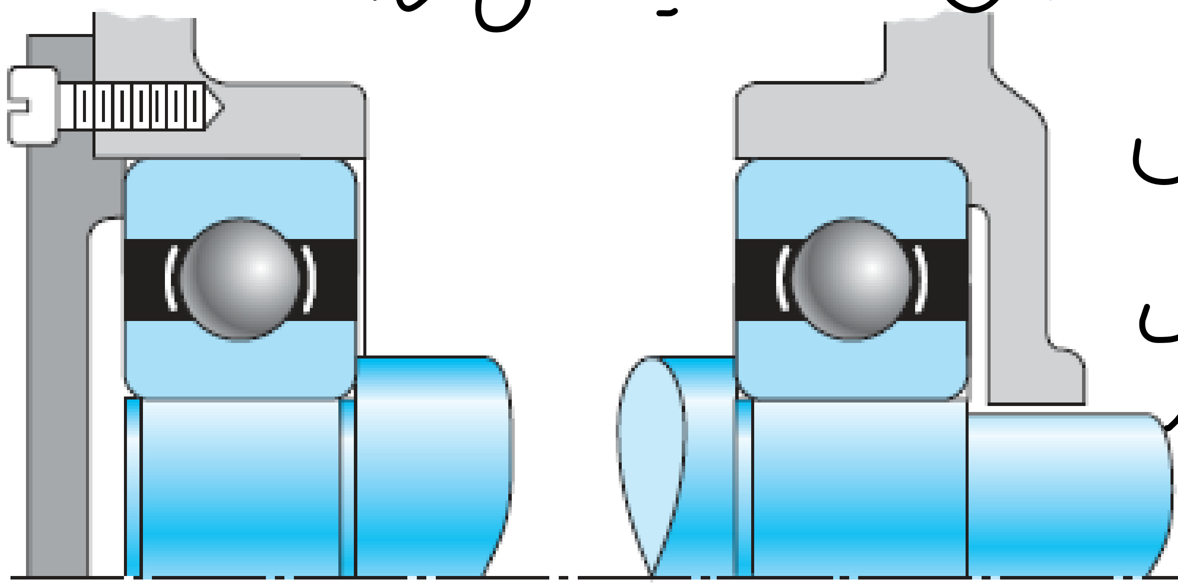


Figure 7–5

Arrangement showing bearing inner rings press-fitted to shaft while outer rings float in the housing. The axial clearance should be sufficient only to allow for machinery vibrations. Note the labyrinth seal on the right.



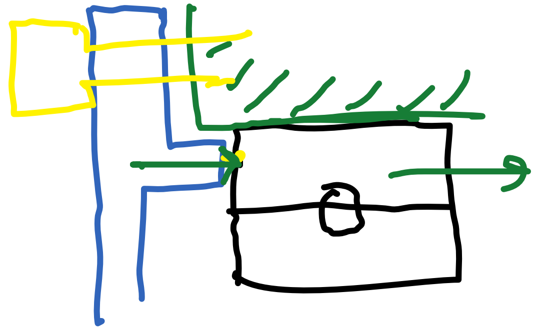
اگر مقدار پیش بار خیلی زیاد باشد
که شفت قفل می‌شود و به راحتی نمی‌چرخد



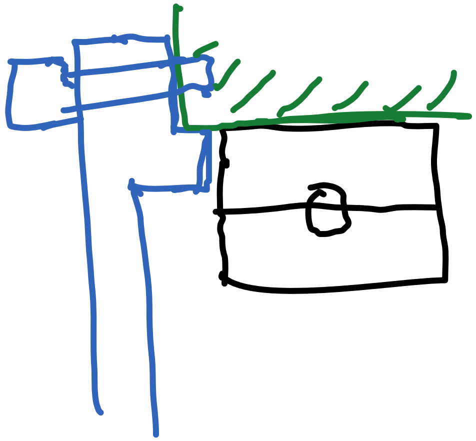
پیش بار تمامی
لقی و لرزش
را حذف می‌کند

Figure 7–6

Similar to the arrangement of Figure 7–5 except that the outer bearing rings are preloaded.

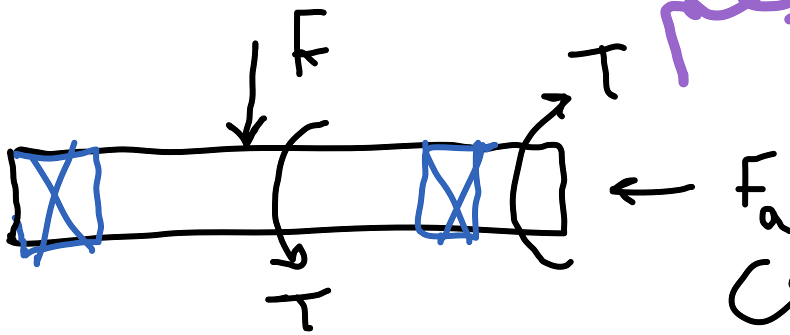


با سفت کردن پیچ درپوش،
درپوش به کنس بیرونی بیرینگ
نیرو وارد می کند



هرچه پیچ را سفت کنیم
دربوش به بیرینگ هیچ
نیرویی وارد نمی کند

جلسه شانزدهم



روی هر شفت ناشی از بارهای شعاعی (ناشی از نیروی بین چرخدنده‌ها یا تسمه یا زنجیر) ممان خمشی خواهیم داشت و به خاطر دوران شفت ممان به صورت M_a خواهد بود، شبه نوسانی خواهد بود.

اغلب از نیروی محوری F_a صرف نظر می‌شود

T ثابت و یکنواختی در طول شفت

$$M \longrightarrow \begin{cases} M_a \\ M_m \end{cases}$$

$$T \longrightarrow \begin{cases} T_a \\ T_m \end{cases}$$

$$\begin{cases} M_a \longrightarrow \sigma_a = K_f \dfrac{M_a c}{I} \\ M_m \longrightarrow \sigma_m = K_f \dfrac{M_m c}{I} \end{cases}$$

$$\begin{cases} T_a \longrightarrow \tau_a = K_{fs} \dfrac{T_a c}{J} \\ T_m \longrightarrow \tau_m = K_{fs} \dfrac{T_m c}{J} \end{cases}$$

$$c = d/2$$

$$I = \frac{\pi d^4}{64}$$

$$J = \frac{\pi d^4}{32}$$

$$\sigma'_m = \left(\sigma_m^2 + 3\tau_m^2 \right)^{1/2}$$

$$\sigma'_a = \left(\sigma_a^2 + 3\tau_a^2 \right)^{1/2}$$

با استفاده رابطه تنش
فون مایزز برای بارهای
ترکیبی

از رابطه گودمن

$$\frac{\sigma'_m}{S_{ut}} + \frac{\sigma'_a}{S_e} = \frac{1}{n}$$

بعد جایگذاری روابط و ساده سازی

$$n = \frac{\pi d^3}{16}\left(\frac{A}{S_e} + \frac{B}{S_{ut}}\right)^{-1}$$

محاسبه n

$$d = \left[\frac{16n}{\pi}\left(\frac{A}{S_e} + \frac{B}{S_{ut}}\right)\right]^{1/3}$$

محاسبه d

بخواهیم طراحی شفت انجام دهیم

$$A = \sqrt{4(K_f M_a)^2 + 3(K_{fs} T_a)^2}$$

$$B = \sqrt{4(K_f M_m)^2 + 3(K_{fs} T_m)^2}$$

M, T $(N.mm)$

S_e, S_{ut} (MPa)

d (mm)

DE-Gerber

$$\frac{1}{n} = \frac{8A}{\pi d^3 S_e}\left\{1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}}\right)^2\right]^{1/2}\right\}$$

$$d = \left(\frac{8nA}{\pi S_e}\left\{1 + \left[1 + \left(\frac{2BS_e}{AS_{ut}}\right)^2\right]^{1/2}\right\}\right)^{1/3}$$

مترتب الحنی استاتیکی (تسلیم در سیکل اول)

It is always necessary to consider the possibility of static failure in the first load cycle. A von Mises maximum stress is calculated for this purpose.

$\tau_{max} = \tau_m + \tau_a$

$$\sigma'_{max} = [(\sigma_m + \sigma_a)^2 + 3(\tau_m + \tau_a)^2]^{1/2}$$

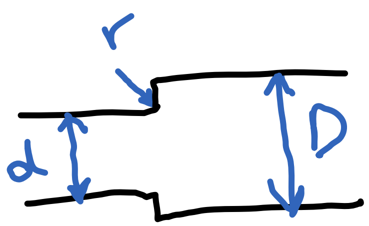
$$= \left[\left(\frac{32K_f(M_m + M_a)}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs}(T_m + T_a)}{\pi d^3}\right)^2\right]^{1/2} \qquad (7\text{–}15)$$

To check for yielding, this von Mises maximum stress is compared to the yield strength, as usual.

$$n_y = \frac{S_y}{\sigma'_{max}} \qquad (7\text{–}16)$$

EXAMPLE 7–1

At a machined shaft shoulder the small diameter d is 27.94 mm, the large diameter D is 41.91 mm, and the fillet radius is 2.79 mm. The bending moment is 142.35 N · m and the steady torsion moment is 124.28 N · m. The heat-treated steel shaft has an ultimate strength of S_{ut} = 724 MPa, a yield strength of S_y = 565 MPa, and a true fracture strength of $\tilde{\sigma}_f$ = 1069 MPa. The reliability goal for the endurance limit is 0.99.
(*a*) Determine the fatigue factor of safety of the design using each of the fatigue failure criteria described in this section.
(*b*) Determine the yielding factor of safety.



$d = 27.9\ mm$
$D = 41.9\ mm$
$r = 2.79\ mm$

$S_{ut} = 724\ M_{pa}$

$T_m = 128.28\ N.m \quad , \quad T_a = 0$

خمش دوار $\quad M_a = 142.35\ N.m \quad , \quad M_m = 0$

$M_a = 142.35 \times 10^3\ N.mm, \quad T_m = 128.28 \times 10^3\ N.mm$

(*a*) $D/d = 41.91/27.94 = 1.50$, $r/d = 2.79/27.94 = 0.10$, $K_t = 1.68$ (Figure A–15–9), $K_{ts} = 1.42$ (Figure A–15–8), $q = 0.85$ (Figure 6–26), $q_{\text{shear}} = 0.88$ (Figure 6–27).

From Equation (6–32),

$$K_f = 1 + 0.85(1.68 - 1) = 1.58$$

$$K_{fs} = 1 + 0.88(1.42 - 1) = 1.37$$

Equation (6–10): $$S'_e = 0.5(724) = 362 \text{ MPa}$$

Equation (6–18): $$k_a = 3.04(724)^{-0.217} = 0.729$$

Equation (6–19): $$k_b = \left(\frac{27.94}{7.62}\right)^{-0.107} = 0.870$$

$$k_c = k_d = k_f = 1$$

Table 6–4: $$k_e = 0.814$$

$$S_e = 0.729(0.870)0.814(361.99) = 186.75 \text{ MPa}$$

For a rotating shaft, the constant bending moment will create a completely reversed bending stress.

$$M_a = 142354.8 \text{ N} \cdot \text{mm} \qquad T_m = 124278 \text{ N} \cdot \text{mm} \qquad M_m = T_a = 0$$

For a rotating shaft, the constant bending moment will create a completely reversed bending stress.

$$M_a = 142354.8 \text{ N} \cdot \text{mm} \qquad T_m = 124278 \text{ N} \cdot \text{mm} \qquad M_m = T_a = 0$$

Applying Equations (7–6) and (7–7) for the DE-Goodman criteria gives

Equation (7–6): $$A = \sqrt{4(K_f M_a)^2} = \sqrt{4[(1.58)(142354.8)]^2} = 449271.75$$

$$B = \sqrt{3(K_{fs} T_m)^2} = \sqrt{3[(1.37)(124278)]^2} = 294814.36$$

Equation (7–7): $$n = \frac{\pi d^3}{16}\left(\frac{A}{S_e} + \frac{B}{S_{ut}}\right)^{-1} = \frac{\pi(27.94)^3}{16}\left(\frac{449271.75}{186.75} + \frac{294814.36}{723.98}\right)^{-1} = 1.52$$

Answer $n = 1.52$ DE-Goodman

MPa

دقت کنید در این روابط T و M برحسب N.mm

d برحسب mm

S_e و S_{ut} " " MPa

For comparison, consider an equivalent approach of calculating the stresses and applying the fatigue failure criteria directly. From Equations (7–4) and (7–5),

$$\sigma'_a = \left[\left(\frac{32 \cdot 1.58 \cdot 142354.8}{\pi(27.94)^3}\right)^2\right]^{1/2} = 104.91 \text{ MPa}$$

$$\sigma'_m = \left[3\left(\frac{16 \cdot 1.37 \cdot 124278}{\pi(27.940)^3}\right)^2\right]^{1/2} = 68.84 \text{ MPa}$$

Taking, for example, the Goodman failure critera, application of Equation (6–41) gives

$$\frac{1}{n} = \frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_{ut}} = \frac{104.91}{186.75} + \frac{68.84}{723.98} = 0.657$$

$$n = 1.52$$

which is identical with the previous result. The same process could be used for the other failure criteria.

(*b*) For the yielding factor of safety, determine an equivalent von Mises maximum stress using Equation (7–15).

$$\sigma'_{\max} = \left[\left(\frac{32(1.58)(142354.8)}{\pi(27.94)^3}\right)^2 + 3\left(\frac{16(1.37)(124278)}{\pi(27.94)^3}\right)^2\right]^{1/2} = 125.48 \text{ MPa}$$

Answer

$$n_y = \frac{S_y}{\sigma'_{\max}} = \frac{565.39}{125.48} = 4.50$$

روشهای کاهش تمرکز تنش در شانه شفت

نیاز به یک پله تیز

با شعاع فیلت بزرگ بیرینگ به
پشت پله نمی چسبد

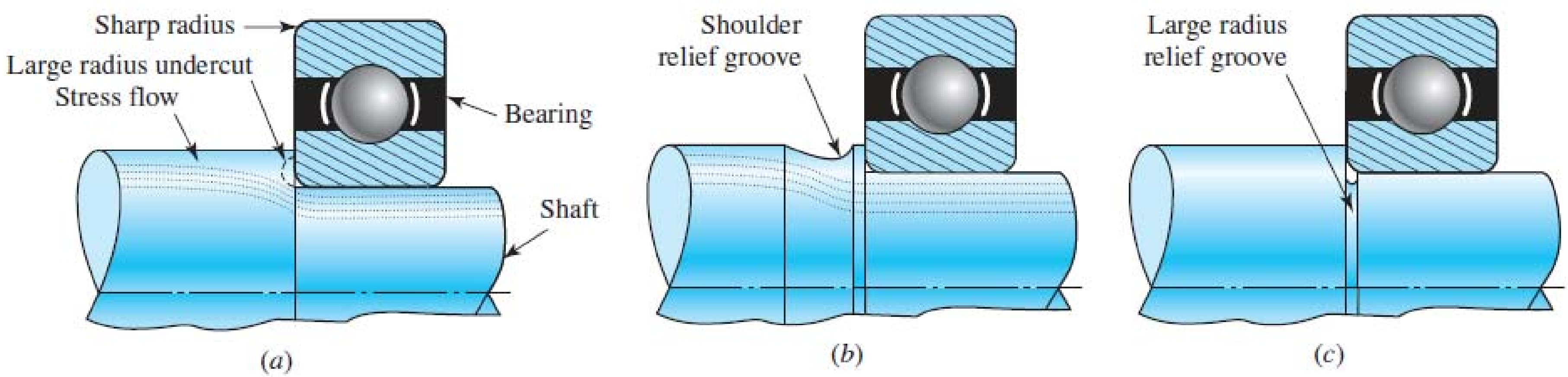
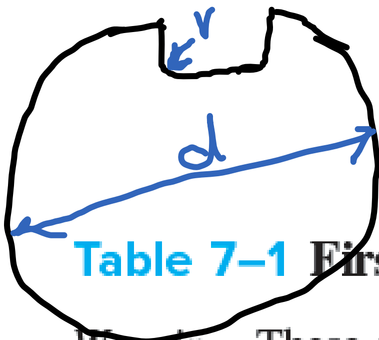


Figure 7–9

Techniques for reducing stress concentration at a shoulder supporting a bearing with a sharp radius. (*a*) Large radius undercut into the shoulder. (*b*) Large radius relief groove into the back of the shoulder. (*c*) Large radius relief groove into the small diameter.

در موقع طراحی ما هنوز مقدار d را نداریم همچنین r و D
که از روی آنها K_t و q و K_f را تخمین بزنیم



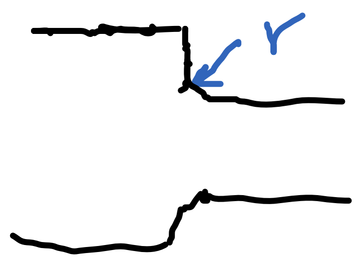
r/d = 0.02

Table 7–1 First Iteration Estimates for Stress-Concentration Factors K_t and K_{ts}

Warning: These factors are only estimates for use when actual dimensions are not yet determined. Do *not* use these once actual dimensions are available.

	Bending	Torsional	Axial
Shoulder fillet—sharp ($r/d = 0.02$) Bearing	2.7	2.2	3.0
Shoulder fillet—well rounded ($r/d = 0.1$)	1.7	1.5	1.9
End-mill keyseat ($r/d = 0.02$) فرز انگشتی	2.14	3.0	—
Sled runner keyseat فرز گرد	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

Missing values in the table are not readily available.



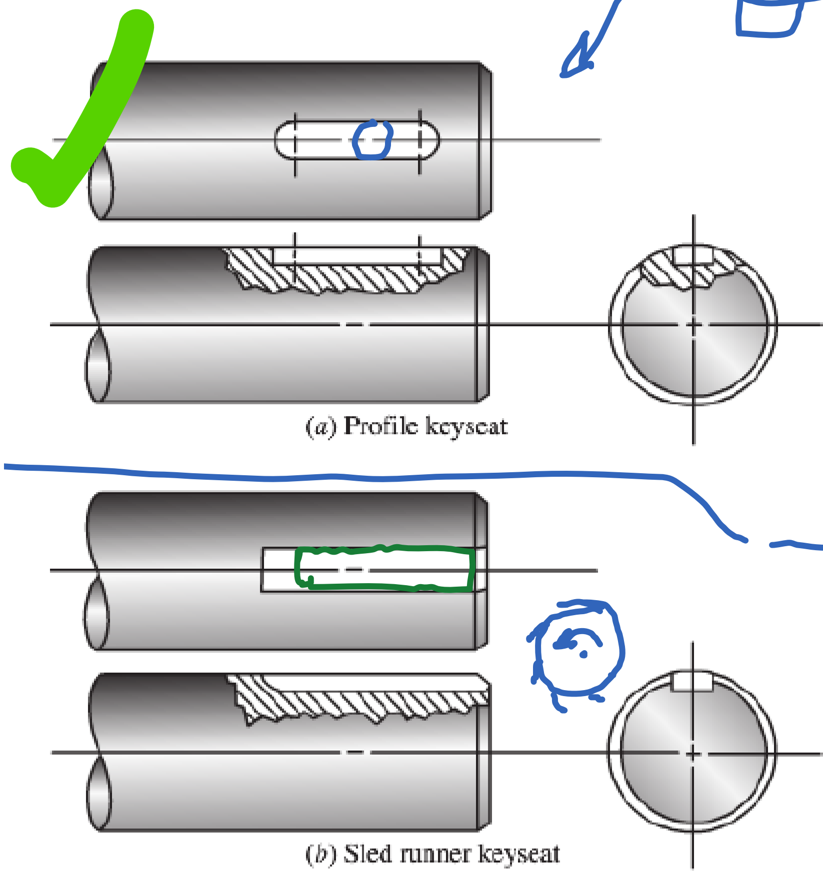
shoulder fillet

جای خار فرز انگشتی

خار فنری

End mill

فرز انگشتی



IGURE 12–7 Keyseats

Sled runner

A *key.seat* is a longitudinal groove cut into a shaft for the mounting of a key

The profile keyseat is milled into the shaft, using an end mill having a diameter equal to the width of the key. The resulting groove is flat-bottomed and has a sharp, square corner at its end.

The sled runner keyseat is produced by a circular milling cutter having a width equal to the width of the key. As the cutter begins or ends the keyseat, it produces a smooth radius.

For this reason, the stress concentration factor for the sled runner keyseat is lower than that for the profile keyseat.

جلسه هفدهم

مثال طراحی شفت

EXAMPLE 7–2

This example problem is part of a larger case study. See Chapter 18 for the full context.

A double reduction gearbox design has developed to the point that the general layout and axial dimensions of the countershaft carrying two spur gears has been proposed, as shown in Figure 7-10. The gears and bearings are located and supported by shoulders, and held in place by retaining rings. The gears transmit torque through keys. Gears have been specified as shown, allowing the tangential and radial forces transmitted through the gears to the shaft to be determined as follows.

$$W_{23}^{t} = 2402\text{ N} \qquad W_{54}^{t} = 10814\text{ N}$$

$$W_{23}^{r} = 876\text{ N} \qquad W_{54}^{r} = 3937\text{ N}$$

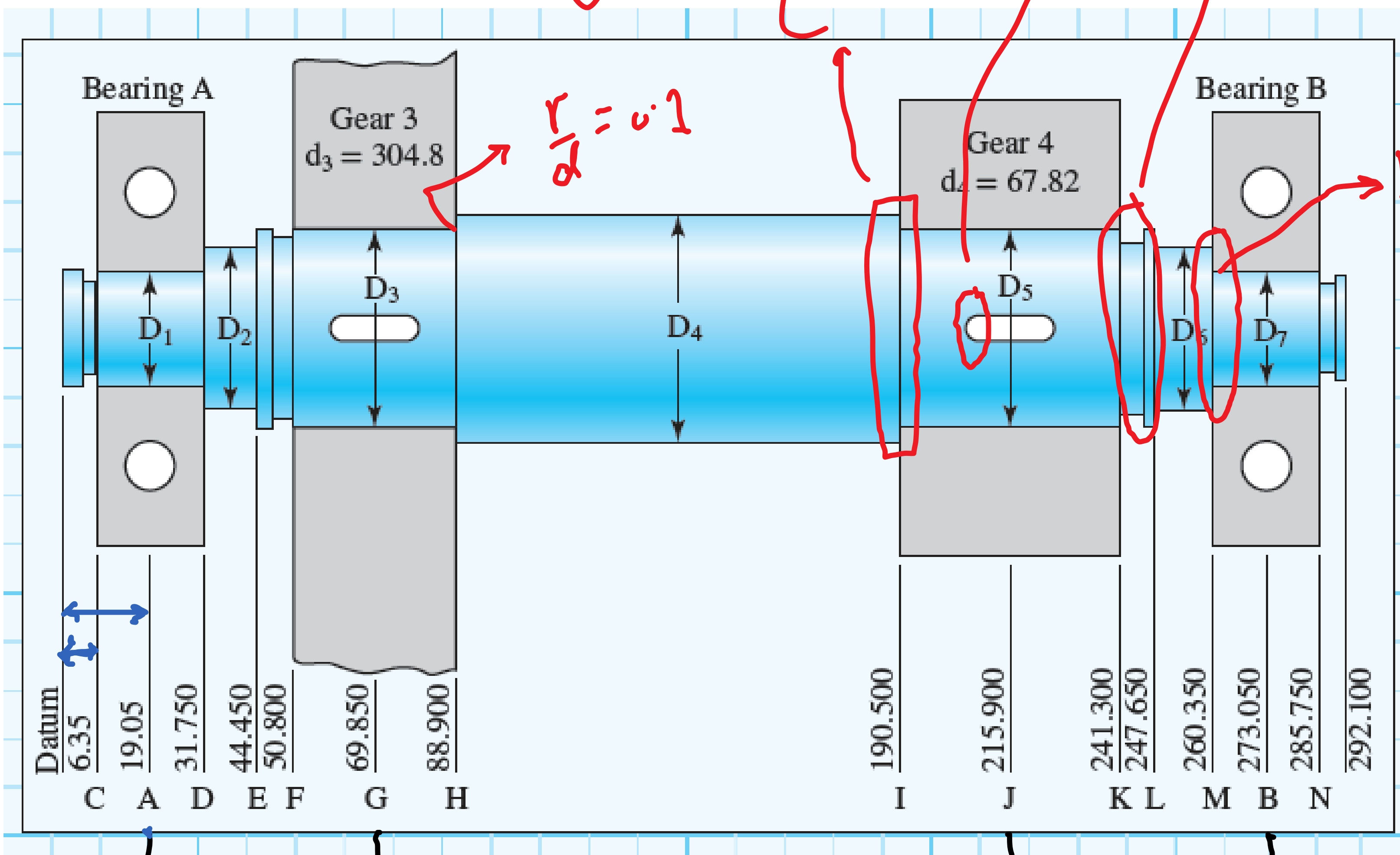
مقطع بحرانی اول

خار

شیار خار فنری

$\frac{r}{d} = 0.1$

$\frac{r}{d} = 0.02$



50.8 146.05 57.15

AG $GJ = 215.9 - 69.85$ JB

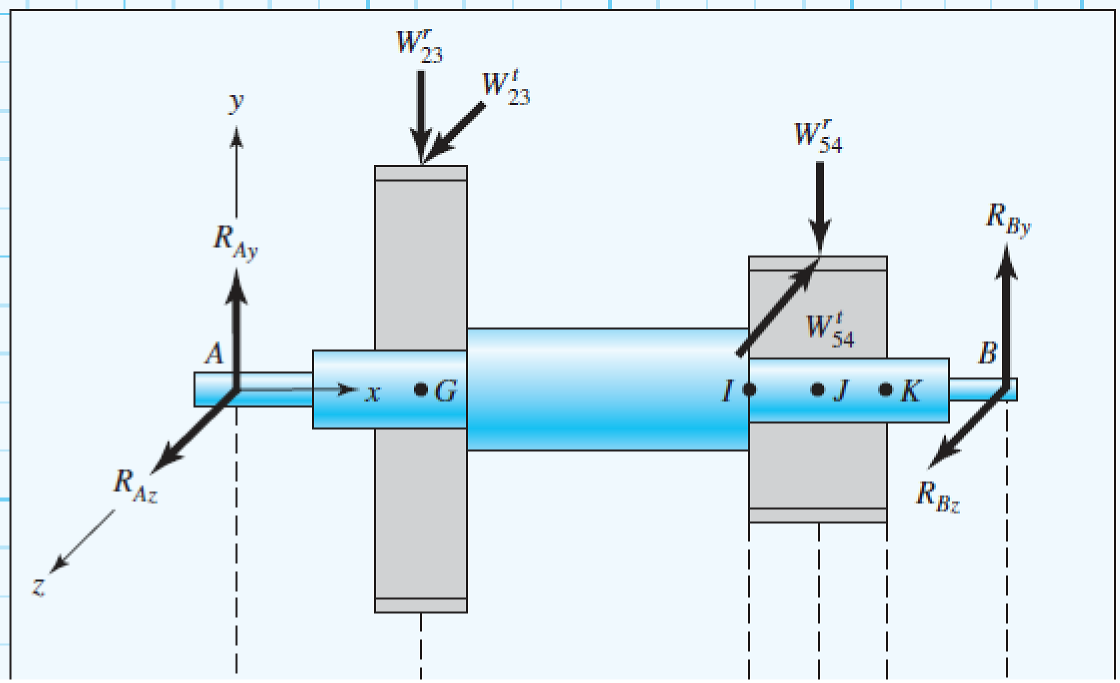
where the superscripts t and r represent tangential and radial directions, respectively; and the subscripts 23 and 54 represent the forces exerted by gears 2 and 5 (not shown) on gears 3 and 4, respectively.

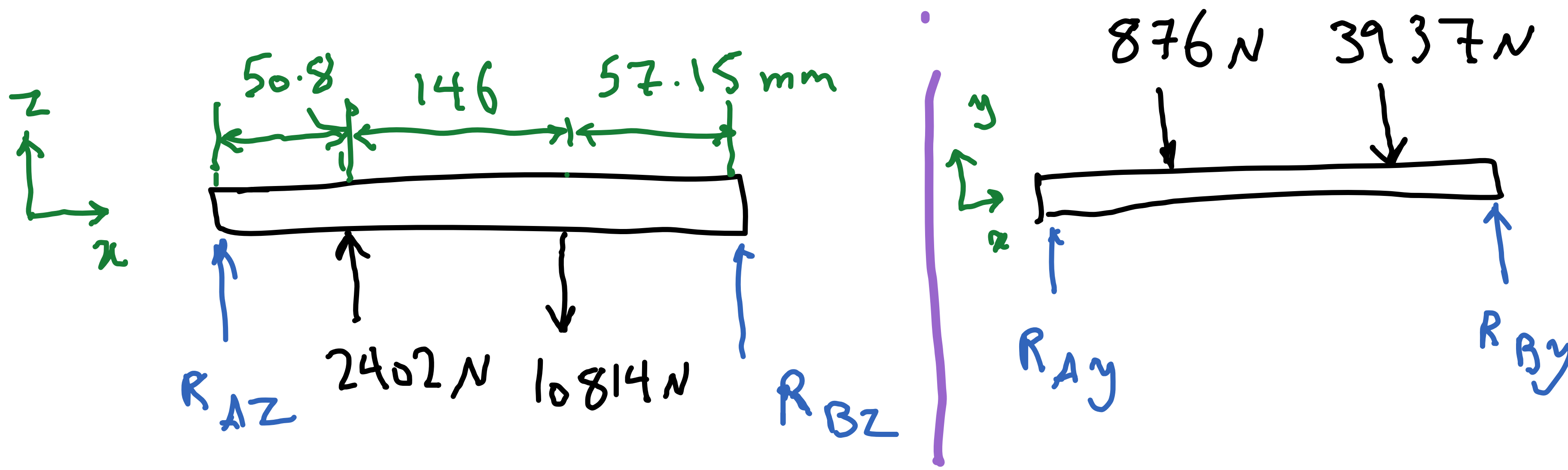
Proceed with the next phase of the design, in which a suitable material is selected, and appropriate diameters for each section of the shaft are estimated, based on providing sufficient fatigue and static stress capacity for infinite life of the shaft, with minimum safety factors of 1.5.

Solution

Perform free body diagram analysis to get reaction forces at the bearings.

$$R_{Az} = 511.54 \text{ N}$$
$$R_{Ay} = 1586.67 \text{ N}$$
$$R_{Bz} = 7900.00 \text{ N}$$
$$R_{By} = 3226.28 \text{ N}$$



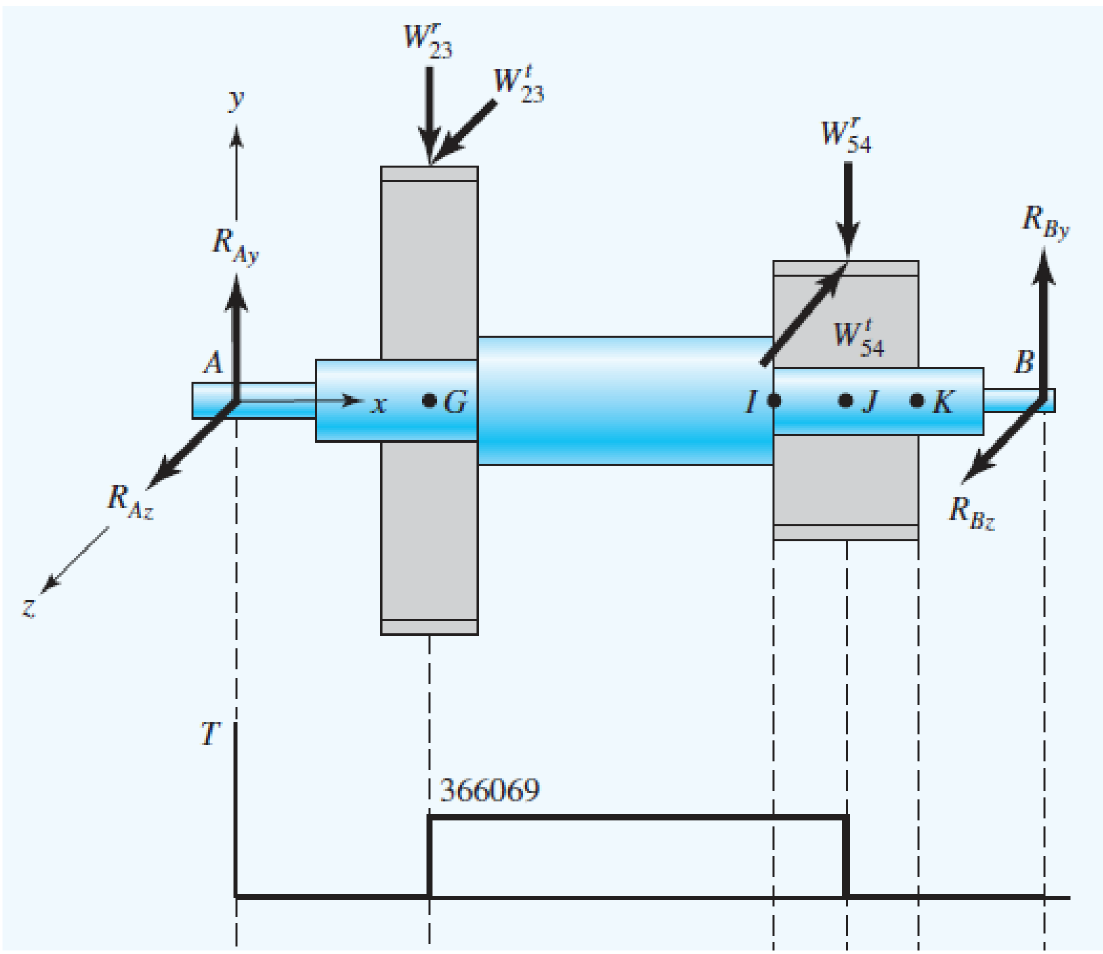


$$\Sigma M_A = 0 \Rightarrow 2402 \times 50.8 - 10814 \times 196.85 + R_{Bz} \times 254 = 0$$

$$\Rightarrow R_{Bz} = 7900\ N$$

$$\Sigma F_z = 0 \Rightarrow R_{Az} + R_{Bz} + 2402 - 10814 = 0 \Rightarrow R_{Az} = 511\ N$$

نمودار گشتاور پیچشی



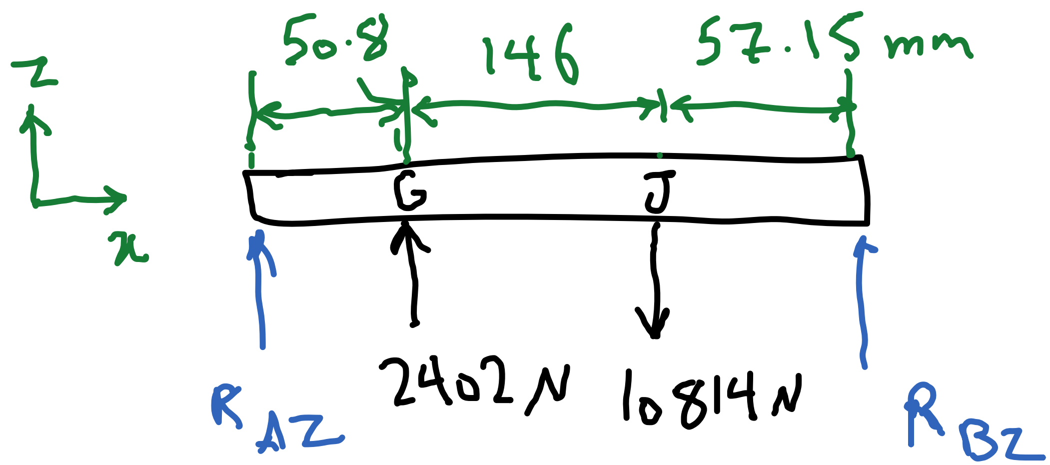
$$T = W_{23}^t \frac{d_3}{2}$$

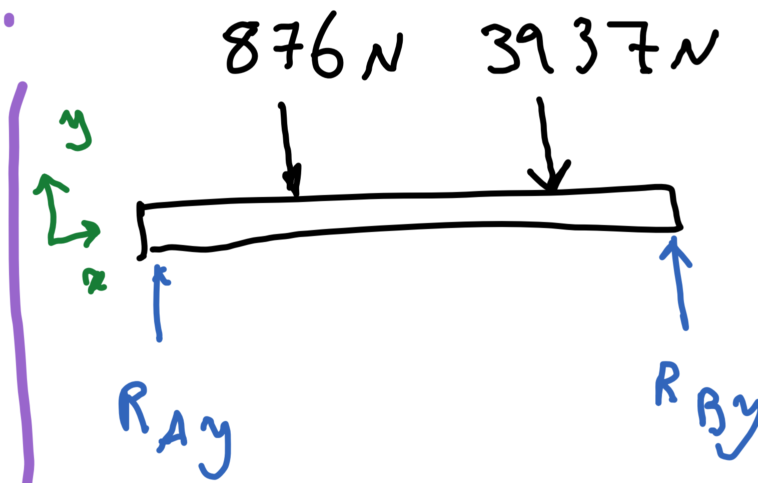
$$= 2402 \times \frac{304.8}{2}$$

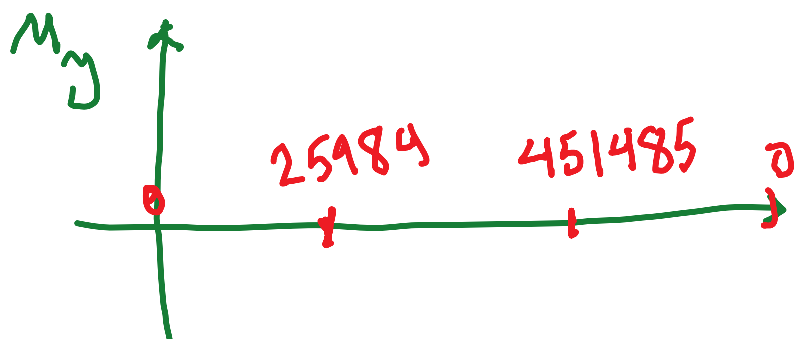
$$= 366064 \text{ N.mm}$$

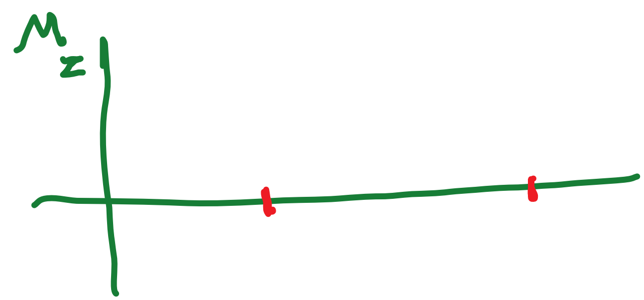
$$T = W_{54}^t \frac{d_4}{2}$$

$$= 366064 \text{ N.mm}$$



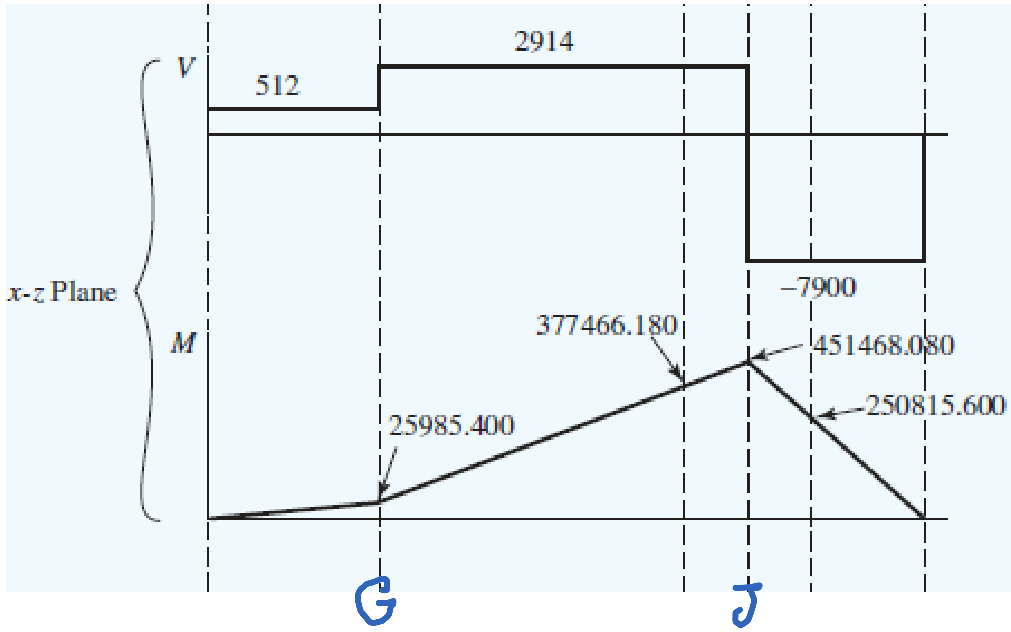


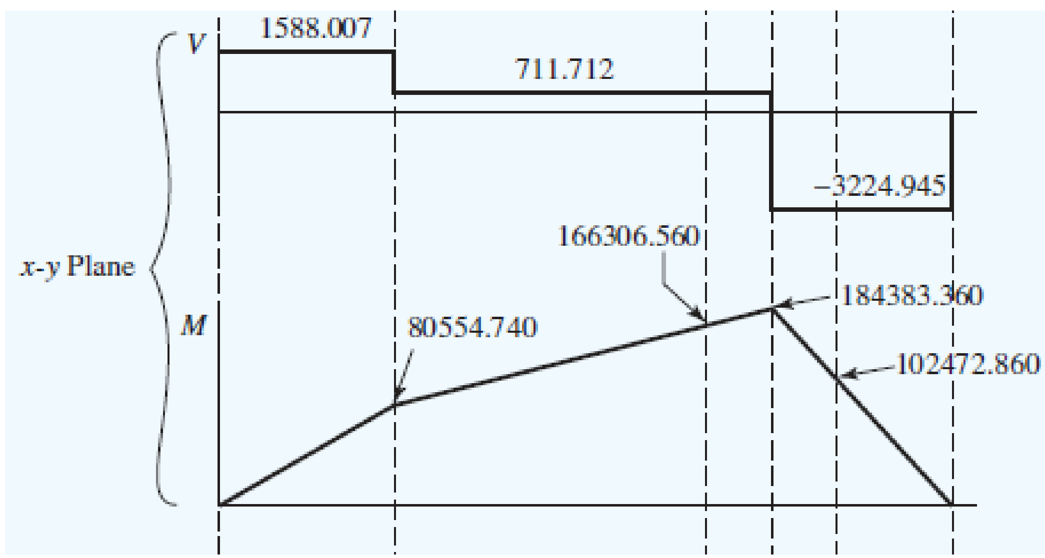




$$M_{Gy} = \overbrace{511.5}^{R_{AZ}} \times 50.8 = 25984 \, N.mm$$

$$M_{Jy} = R_{BZ} \times 57.15 = 7900 \times 57.15 = 451485$$

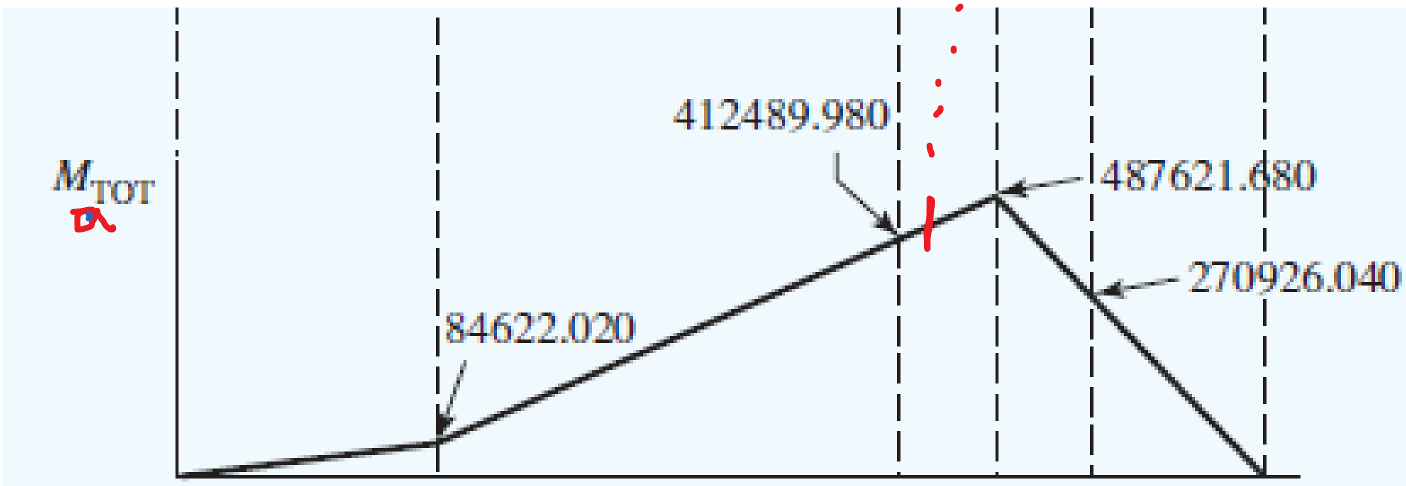

V
512
2914
-7900
x-z Plane
M
377466.180
451468.080
250815.600
25985.400
G
J


V
1588.007
711.712
-3224.945
x-y Plane
M
166306.560
184383.360
80554.740
102472.860

$$M_t = \sqrt{M_y^2 + M_z^2}$$

end of the keyway 1

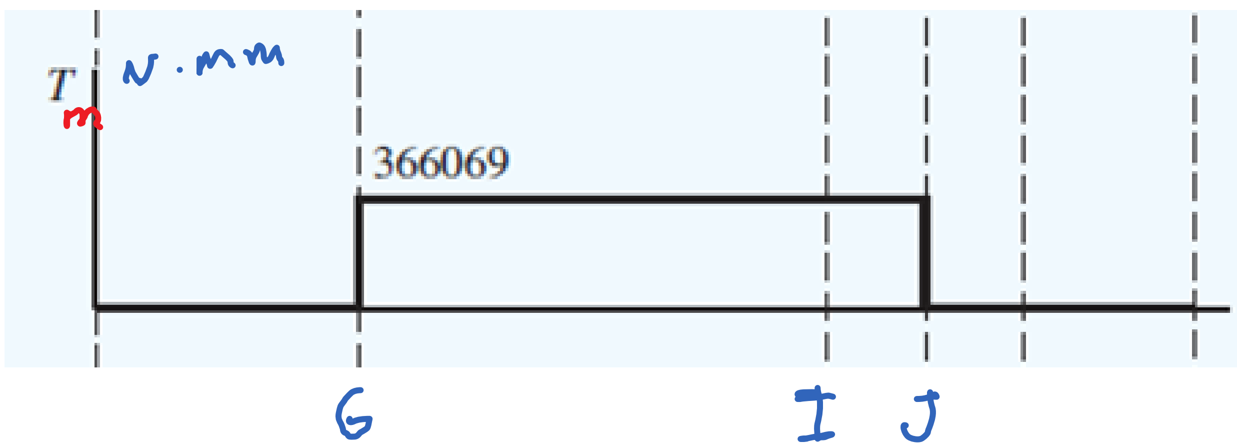
$M_m = 0$ (M_a) نمودار گشتاور خمشی



T_m

$T_a = 0$

گشتاور پیچشی



مراحل طراحی شفت

۱- محاسبه گشتاور خمشی و گشتاور پیچشی در طول شفت

۲- محاسبه تنش در نقاط بحرانی

نقطه‌ای که بیشترین گشتاور خمشی را داریم و همزمان گشتاور پیچشی هم داشته باشیم.

شروع از نقطه I

۳- تخمین اولیه مقدار K_f و K_{fs} (جدول ۱-۷)

۴- اگر قطر شفت را تعیین نکرده بود باید یک حدس انتخاب کنیم برای شروع AISI 1020 مناسب.

۵- حدس اولیه برای K_b در $K_b=0.8$ تا 0.9 و محاسبه S_e

۶- محاسبه حداقل قطر شفت با استفاده از رابطه گودمن یا (Eq 7-8) و گرد کردن آن عدد به اعداد روند و استاندارد و بزرگتر ببرید.

Start with point I, where the bending moment is high, there is a stress concentration at the shoulder, and the torque is present.

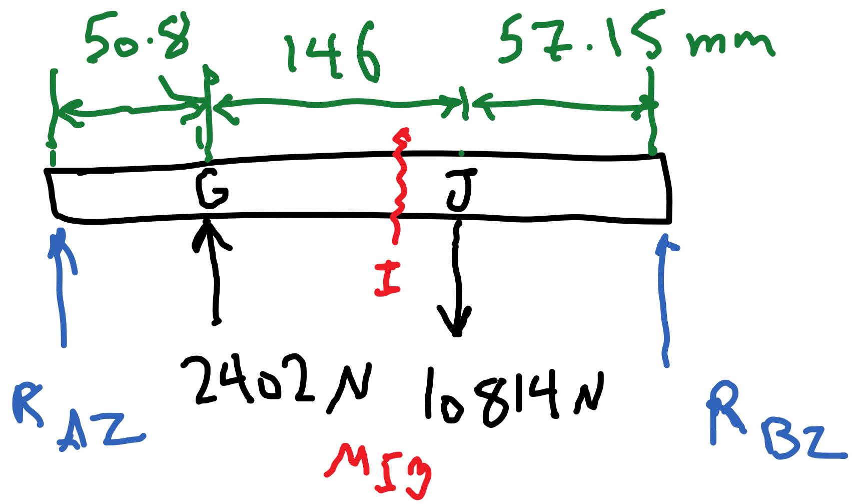
$$\text{At } I,\ M_a = 412489.98\ \text{N}\cdot\text{mm},\ T_m = 366069.07\ \text{N}\cdot\text{mm},\ M_m = T_a = 0$$

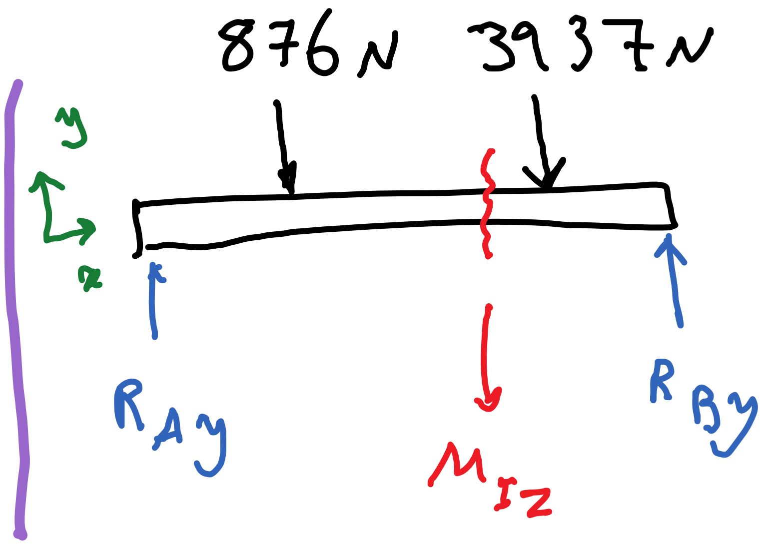
شروع طراحی از جایی است که بیشترین گشتاور خمشی را داریم.

بیشترین گشتاور خمشی در نقطه J می باشد

به دلیل ضریب تمرکز تنش در I (به واسطه وجود پله)

از I شروع می شود.





IJ = 25.4 فاصله

$$M_{Iy} = R_{Bz}(57.15+25.4) - 10814 \times 25.4$$

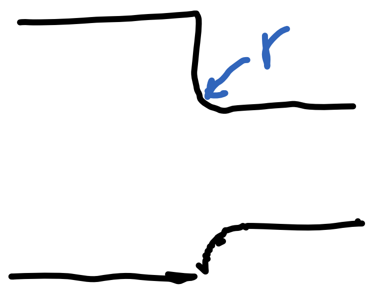
$$M_{Iz} = R_{By}(\sim + \sim) - 3937 \times 25.4$$

$$M_I = \sqrt{M_{Iy}^2 + M_{Iz}^2} = 412489 \text{ N.mm}$$

٣

Assume generous fillet radius for gear at *I*.

From Table 7-1, estimate $K_t = 1.7$, $K_{ts} = 1.5$. For quick, conservative first pass, assume $K_f = K_t$, $K_{fs} = K_{ts}$.



برای r/d در شفت فیلت پله

$r/d = 0.02 \rightarrow$ برای بیرینگ

$r/d = 0.1 \rightarrow$ برای چرخدنده، یا پولی

یا چرخ زنجیر

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

$$K_{fs} = 1 + q_s(K_{ts} - 1)$$

ماکزیمم مقدار K_f ، می تواند برابر K_t باشد. $(q=1)$

پس در بدبینانه ترین حالت می توانیم

$$K_f = K_t$$

$$K_{fs} = K_{ts}$$

برای تعیین q نیاز به r و S_{ut} داریم.

۴ و ۵

Choose inexpensive steel, 1020 CD, with $S_{ut} = 470$ MPa. For S_e,

Equation (6–18)	$k_a = aS_{ut}^b = 3.04(470)^{-0.217} = 0.8$
Guess $k_b = 0.9$. Check later when d is known.	
	$k_c = k_d = k_e = 1$
Equation (6–17)	$S_e = (0.8)(0.9)(0.5)(470) = 169.18$ MPa

در اینجا فرض کرده است که جنس شفت را هم طراح انتخاب می کند. در مثلی که ما می دهیم اغلب جنس را مشخص می کنیم. k_b را در اینجا ۰.۹ فرض کرده است.

ما اغلب $k_b = 0.8$ می گیریم برای شروع

For first estimate of the small diameter at the shoulder at point I, use the DE-Goodman criterion of Equation (7-8). This criterion is good for the initial design, since it is simple and conservative. With $M_m = T_a = 0$, Equations (7-6) and (7-8) reduce to

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

$$d = \left\{ \frac{16(1.5)}{\pi} \left(\frac{2(1.7)(412489.98)}{169.18} + \frac{\{3[(1.5)(366069.07)]^2\}^{1/2}}{470} \right) \right\}^{1/3}$$

$$d = 42.35 \text{ mm}$$

All estimates have probably been conservative, so select the next standard size below 1.69 in and check, $d = 41.275$ mm.

$$\begin{cases} d = 42 \\ d = 43 \\ d = 45 \end{cases}$$

ما اگر به عدد 42.35 می رسیدیم ←

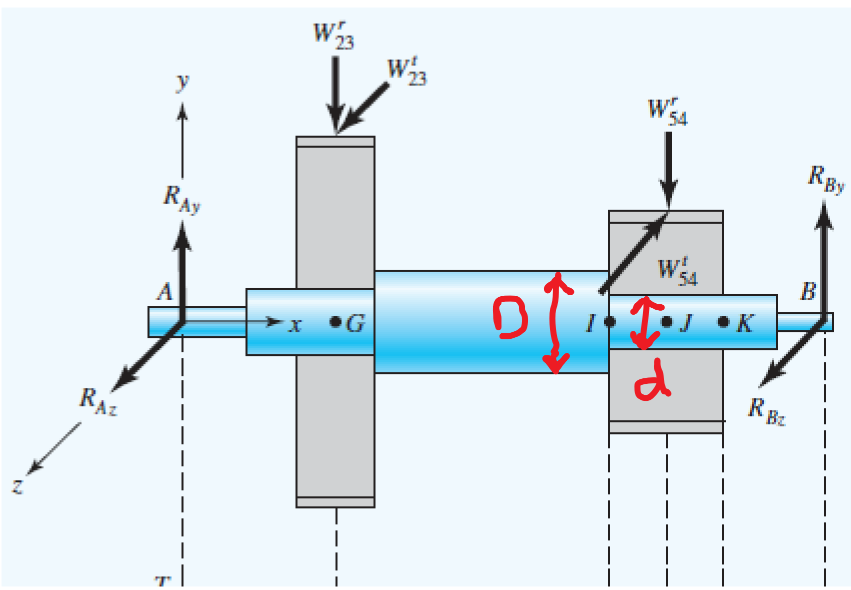
اغلب اعداد را به سمت بالا گرد می کنیم

۷- تعیین D (قطر بزرگ پله) و r (شعاع فیلت)

A typical D/d ratio for support at a shoulder is $D/d = 1.2$, thus, $D = 1.2(41.275) = 49.5300$ mm. Increase to $D = 50.8$ mm. A nominal 2-in. cold-drawn shaft diameter can be used. Check if estimates were acceptable.

$$D/d = 50.8/41.275 = 1.23$$

Assume fillet radius $r = d/10 = 4.1275$ mm, $\quad r/d = 0.1$



$$\frac{D}{d} = 1.2 \Rightarrow D = 1.2d = 49.5$$

$$\rightarrow D = 50.8 = 2''$$

ما در طراحی در سیستم متریک

$$D = 50\ mm$$

$$r/d = 0.1 \Rightarrow r = 4\ mm$$

۷- تعیین D (قطر بزرگ پله) و r (شعاع فیلت)

$D/d = 1.2$, $\Rightarrow D$

r طبق شعاع فیلت فرض شود ($r/d = 0.1$ یا $r/d = 0.02$)

۸- چک کردن مجدد ضریب اطمینان با مقادیر انتخاب شده برای d ، D ، r ، S_{ut} و محاسبه مجدد K_t ، K_{fs} ، S_e

۹- محاسبه ضریب اطمینان از رابطه 7-7 یا رابطه گودمن

ضریبی

۸- چک کردن مجدد ضرایب با مقدار انتخاب شده برای

d ، D ، r ، S_{ut} ، و محاسبه مجدد K_f ، K_{fs} ، S_e

$$K_t = 1.6 \text{ (Fig. A–15–9)}, \; q = 0.82 \text{ (Fig. 6–26)}$$

Equation (6–32)

$$K_f = 1 + 0.82(1.6 - 1) = 1.49$$

$$K_{ts} = 1.35 \text{ (Fig. A–15–8)}, \; q_s = 0.85 \text{ (Fig. 6–27)}$$

$$K_{fs} = 1 + 0.85(1.35 - 1) = 1.30$$

$$k_a = 0.801 \text{ (no change)}$$

Equation (6–19)

$$k_b = \left(\frac{41.275}{7.62}\right)^{-0.107} = 0.835$$

$$S_e = (0.801)(0.835)(0.5)(470.000) = 157.12 \text{ MPa}$$

۹- محاسبه ضریب ایمنی از رابطه 7-7 یا رابطه گودمن

$$n=\frac{\pi d^3}{16}\left(\frac{A}{S_e}+\frac{B}{S_{ut}}\right)^{-1}$$

→ Eq 7-7

Equation (7–4) $\sigma'_a=\dfrac{32K_fM_a}{\pi d^3}=\dfrac{32(1.49)412489.98(1000)}{\pi(41.275)^3}=89149.95\text{ kPa}$

Equation (7–5) $\sigma'_m=\left[3\left(\dfrac{16K_{fs}T_m}{\pi d^3}\right)^2\right]^{1/2}=\dfrac{\sqrt{3}(16)(1.30)366069.07(1000)}{\pi(41.275)^3}$

$=59585.40\text{ kPa}$

Using Goodman criterion

$$\frac{1}{n_f}=\frac{\sigma'_a}{S_e}+\frac{\sigma'_m}{S_{ut}}=\frac{89149.95/157.12}{1000}+\frac{59585.40/470}{1000}=0.696$$

$$n_f=1.44$$

Note that we could have used Equation (7-7) directly.

Check yielding.

$$n_y=\frac{S_y}{\sigma'_{max}}>\frac{S_y}{\sigma'_a+\sigma'_m}=\frac{392997.900}{89149.9464+59585.4043}=2.64$$

$n_f=1.44\simeq 1.5$

اگر ضریب ایمنی بدست آمده مساوی یا بزرگتر از مقدار خواسته شده درست است بعد که با همان اعداد بدست آمده جلو می‌رویم. اگر ضریب ایمنی از مقدار مورد نظر کمتر شد دو راه داریم: یا جنس قطعه را تغییر دهیم جهت افزایش S_{ut} یا قطر را افزایش دهیم.

$$n_f < n_{\text{خواسته شده}} \Rightarrow \begin{cases} S_{ut} \uparrow \\ d \uparrow \end{cases}$$

۱۰ - اگر ضریب ایمنی پایین آمد از مقدار مورد نظر کمتر بود باید جنس را جهت افزایش S_{ut} عوض کنیم یا قطر شفت را افزایش دهیم.

۱۱- محاسبه ضریب ایمنی در محل جای خار و محل شیار خارفنری

12- نهایتاً طراحی نهایی شفت و تعیین قطر برای قسمتهای مختلف شفت.

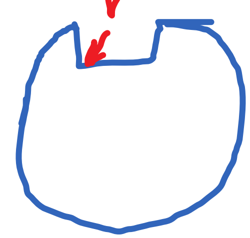
۱۱- محاسبه ضریب ایمنی در محل جای خار و محل شیار خار فنری

Also check this diameter at the end of the keyway, just to the right of point I, and at the groove at point K. From moment diagram, estimate M at end of keyway to be $M = 423675$ N · mm.

Fig 7-1

Assume the radius at the bottom of the keyway will be the standard $r/d = 0.02$, $r = 0.02d = 0.02(41.275) = 0.8255$ mm.

$r = 0.8$ mm



$$K_t = 2.14 \text{ (Table 7–1)}, \; q = 0.65 \text{ (Figure 6–26)}$$

$$K_f = 1 + 0.65(2.14 - 1) = 1.74$$

$$K_{ts} = 3.0 \text{ (Table 7–1)}, \; q_s = 0.71 \text{ (Figure 6–27)}$$

$$K_{fs} = 1 + 0.71(3 - 1) = 2.42$$

$$\sigma'_a = \frac{32K_f M_a}{\pi d^3} = \frac{32(1.74)(423675 \text{ N} \cdot \text{mm})}{\pi(41.275)^3} = 106849.00 \text{ kPa}$$

$$\sigma'_m = \sqrt{3}(16)\frac{K_{fs}T_m}{\pi d^3} = \frac{\sqrt{3}(16)(2.42)(366069.07)}{\pi(41.275)^3} = 111134.24 \text{ kPa}$$

$$\frac{1}{n_f} = \frac{\sigma'_a}{S_e} + \frac{\sigma'_m}{S_{ut}} = \frac{106849.00}{157(1000)} + \frac{111134.2416}{470(1000)} = 0.919$$

$$n_f = 1.09$$

$S_{ut} = 690$ MPa — 1050 ← تغییر جنس شفت

The keyway turns out to be more critical than the shoulder. We can either increase the diameter or use a higher strength material. Unless the deflection analysis shows a need for larger diameters, let us choose to increase the strength. We started with a very low strength and can afford to increase it some to avoid larger sizes. Try 1050 CD with $S_{ut} = 690$ MPa.

Recalculate factors affected by S_{ut}, i.e., $k_a \rightarrow S_e$; $q \rightarrow K_f \rightarrow \sigma'_a$

$$k_a = 3.04(690)^{-0.217} = 0.736, \quad S_e = 0.736(0.835)(0.5)(690) = 211.91 \text{ MPa}$$

$$q = 0.72, K_f = 1 + 0.72(2.14 - 1) = 1.82$$

$$\sigma'_a = \frac{32(1.82)(423675)}{\pi(41.275)^3} = 111746 \text{ kPa}$$

$$\frac{1}{n_f} = \frac{111746}{211.91(1000)} + \frac{111134.2416}{690(1000)} = 0.689$$

$$n_f = 1.45$$

(K) محاسبه ضریب اطمینان در محل شیار خستگی

$T_m = 0 , \quad M_a = 270926 \text{ N.mm}$

Check at the groove at K, since K_t for flat-bottomed grooves are often very high. From the torque diagram, note that no torque is present at the groove. From the moment diagram, $M_a = 270926 \text{ N} \cdot \text{mm}$,

$M_m = T_a = T_m = 0$. To quickly check if this location is potentially critical, just use $K_f = K_t = 5.0$ as an estimate, from Table 7-1.

$$\sigma_a = \frac{32K_f M_a}{\pi d^3} = \frac{32(5)(270926(1000))}{\pi(41.275)^3} = 196227 \text{ kPa}$$

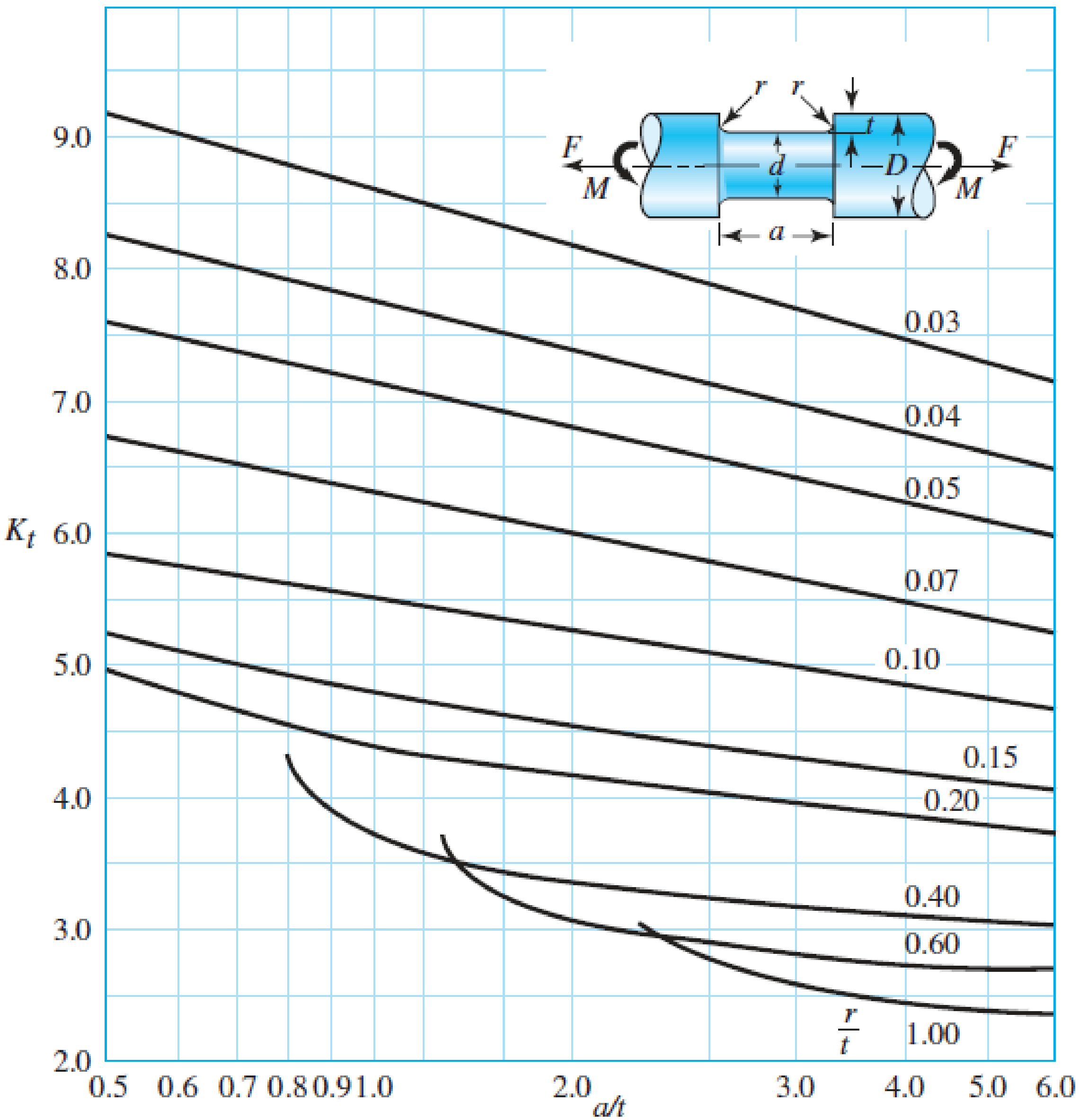
$$n_f = \frac{S_e}{\sigma_a} = \frac{211.91(1000)}{196227} = 1.08$$

table 7-1 $\longrightarrow K_t = 5$

برای چک اولیه

$K_t = K_{f_{max}}$

A-15-16


r
r
t
F
M
d
D
F
M
a
9.0
8.0
7.0
K_t 6.0
5.0
4.0
3.0
2.0
0.03
0.04
0.05
0.07
0.10
0.15
0.20
0.40
0.60
1.00
$\frac{r}{t}$
0.5 0.6 0.7 0.8 0.9 1.0 2.0 3.0 4.0 5.0 6.0
a/t

This is low. We will look up data for a specific retaining ring to obtain K_f more accurately. With a quick online search of a retaining ring specification using the website www.globalspec.com, appropriate groove specifications for a retaining ring for a shaft diameter of 41.275 mm are obtained as follows: width, $a = 1.727$ mm; depth, $t = 1.219$ mm; and corner radius at bottom of groove, $r = 0.254$ mm. From Figure A-15-16, with $r/t = 0.254/1.219 = 0.208$, and $a/t = 1.727/1.219 = 1.42$

$$K_t = 4.3,\ q = 0.65 \text{ (Figure 6–26)}$$

$$K_f = 1 + 0.65(4.3 - 1) = 3.15$$

$$\sigma_a = \frac{32K_f M_a}{\pi d^3} = \frac{32(3.15)(270926)}{\pi(41.275)^3} = 123427 \text{ kPa}$$

$$n_f = \frac{S_e}{\sigma_a} = \frac{211(1000)}{123427} = 1.71$$

محاسبه در نقطه M

$r/d = 0.02$ به خاطر بیرینگ

Quickly check if point M might be critical. Only bending is present, and the moment is small, but the diameter is small and the stress concentration is high for a sharp fillet required for a bearing. From the moment diagram, $M_a = 108347.82$ N · mm, and $M_m = T_m = T_a = 0$.

Estimate $K_t = 2.7$ from Table 7-1, $d = 25.4$ mm, and fillet radius r to fit a typical bearing.

$$r/d = 0.02, \; r = 0.02(25.4) = 0.5080$$

$$q = 0.7 \text{ (Figure 6–26)}$$

$$K_f = 1 + (0.7)(2.7 - 1) = 2.19$$

$$\sigma_a = \frac{32K_f M_a}{\pi d^3} = \frac{32(2.19)(108347.82)}{\pi(25.4)^3} = 147490.23 \text{ kPa}$$

$$n_f = \frac{S_e}{\sigma_a} = \frac{211.91(1000)}{147490.23} = 1.44$$

۱۲- نوبت طراحی نهایی شفت رسیدن قطر برای مقطع‌های مختلف شفت.

This location is more critical than perhaps anticipated. The estimate for stress concentration is likely on the high side, so we will choose to continue and recheck after a specific bearing is selected.

With the diameters specified for the critical locations, fill in trial values for the rest of the diameters, taking into account typical shoulder heights for bearing and gear support.

$$D_1 = D_7 = 25.4 \text{ mm}$$

قطر استاندارد بیرینگ موجود باشد.

$$D_2 = D_6 = 35.56 \text{ mm}$$

$$D_3 = D_5 = 41.275 \text{ mm}$$

$$D_4 = 50.8 \text{ mm}$$

The bending moments are much less on the left end of the shaft, so D_1, D_2, and D_3 could be smaller. However, unless weight is an issue, there is little advantage to requiring more material removal. Also, the extra rigidity may be needed to keep deflections small.

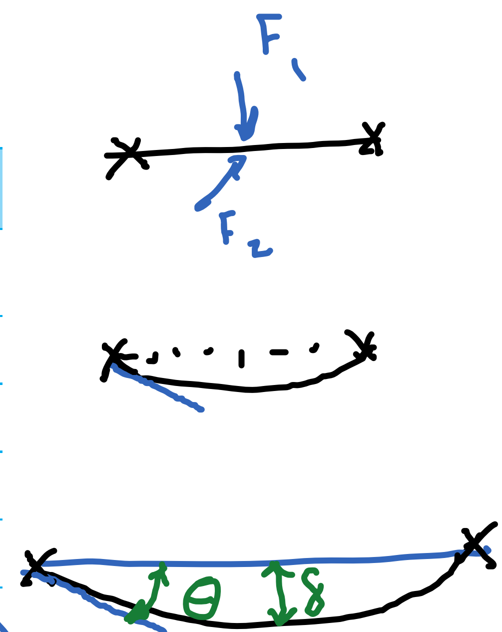
$$D/d = 1.2$$

جلسه هجدهم بررسی خیز و شیب

Table 7–2 Typical Maximum Ranges for Slopes and Transverse Deflections

Slopes	
Tapered roller	0.0005–0.0012 rad
Cylindrical roller	0.0008–0.0012 rad
Deep-groove ball	0.001–0.003 rad
Spherical ball	0.026–0.052 rad
Self-align ball	0.026–0.052 rad
Uncrowned spur gear	<0.0005 rad

Transverse Deflections	
Spur gears with $P < 10$ teeth/cm	0.25 mm
Spur gears with $11 < P < 19$	0.125 mm
Spur gears with $20 < P < 50$	0.075 mm



رینگ داخلی بلبرینگ
نسبت به رینگ خارجی
یک مقدار بازی یا لقی داره

EXAMPLE 7–3

This example problem is part of a larger case study. See Chapter 18 for the full context.

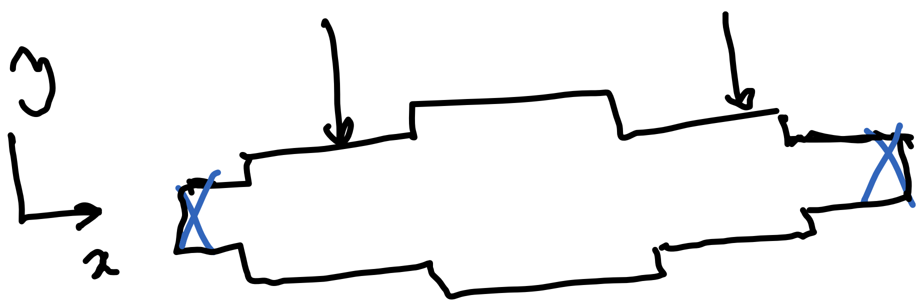
In Example 7-2, a preliminary shaft geometry was obtained on the basis of design for stress. The resulting shaft is shown in Figure 7-10, with proposed diameters of

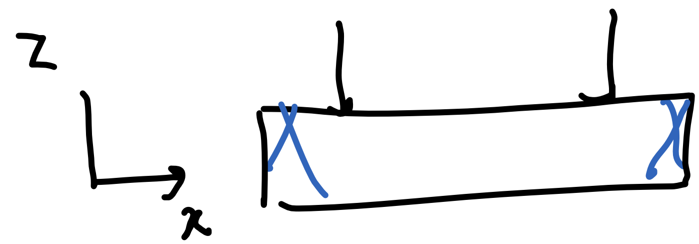
$$D_1 = D_7 = 25.4 \text{ mm}$$
$$D_2 = D_6 = 35.56 \text{ mm}$$
$$D_3 = D_5 = 41.275 \text{ mm}$$
$$D_4 = 50.8 \text{ mm}$$

Check that the deflections and slopes at the gears and bearings are acceptable. If necessary, propose changes in the geometry to resolve any problems.

Solution

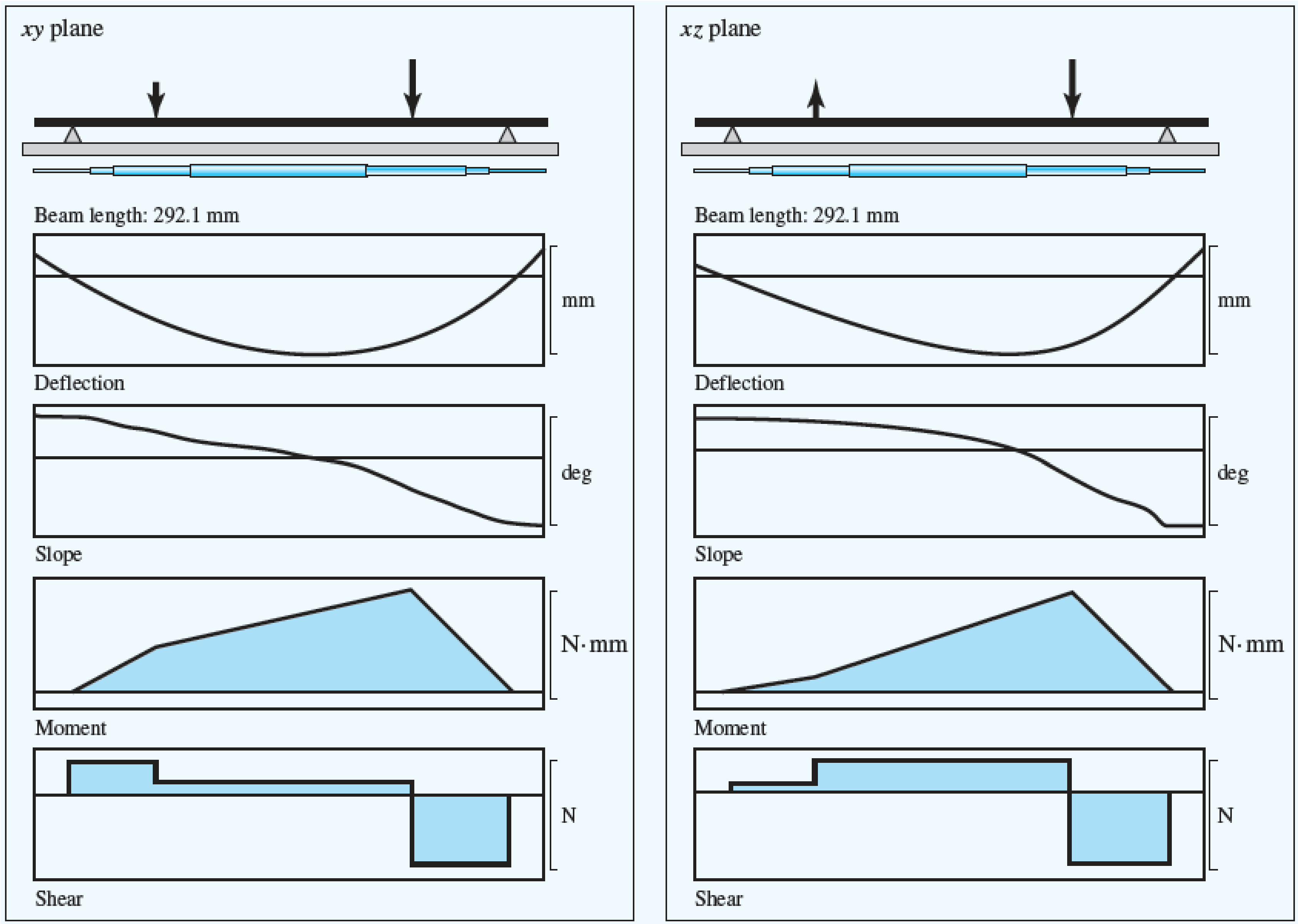
A simple planar beam analysis program will be used. By modeling the shaft twice, with loads in two orthogonal planes, and combining the results, the shaft deflections can readily be obtained. For both planes, the material is selected (steel with $E = 207$ GPa), the shaft lengths and diameters are entered, and the bearing locations are specified. Local details like grooves and keyways are ignored, as they will have insignificant effect on the deflections. Then the tangential gear forces are entered in the horizontal xz plane model, and the radial gear forces are entered in the vertical xy plane model. The software can calculate the bearing reaction forces, and numerically integrate to generate plots for shear, moment, slope, and deflection, as shown in Figure 7-11.





در حالت خیز ها که مهم نیست ، قطر مهم است :

قطر کوچک ممکن است موجب افزایش شیب و خیز در بیرینگ شود



Source: Beam 2D Stress Analysis, Orand Systems, Inc.

Table 7–3 **Slope and Deflection Values at Key Locations**

Point of Interest	*xz* Plane	*xy* Plane	Total
Left bearing slope	0.02263 deg	0.01770 deg	0.02872 deg 0.000501 rad
Right bearing slope	0.05711 deg	0.02599 deg	0.06274 deg 0.001095 rad
Left gear slope	0.02067 deg	0.01162 deg	0.02371 deg 0.000414 rad
Right gear slope	0.02155 deg	0.01149 deg	0.02442 deg 0.000426 rad
Left gear deflection	0.0007568 in	0.0005153 in	0.0009155 in
Right gear deflection	0.0015870 in	0.0007535 in	0.0017567 in

The deflections and slopes at points of interest are obtained from the plots, and combined with orthogonal vector addition, that is, $\delta = \sqrt{\delta_{xz}^2 + \delta_{xy}^2}$. Results are shown in Table 7-3.

The deflections and slopes at points of interest are obtained from the plots, and combined with orthogonal vector addition, that is, $\delta = \sqrt{\delta_{xz}^2 + \delta_{xy}^2}$. Results are shown in Table 7-3.

Whether these values are acceptable will depend on the specific bearings and gears selected, as well as the level of performance expected. According to the guidelines in Table 7-2, all of the bearing slopes are well below typical limits for ball bearings. The right bearing slope is within the typical range for cylindrical bearings. Since the load on the right bearing is relatively high, a cylindrical bearing might be used. This constraint should be checked against the specific bearing specifications once the bearing is selected.

The gear slopes and deflections more than satisfy the limits recommended in Table 7-2. It is recommended to proceed with the design, with an awareness that changes that reduce rigidity should warrant another deflection check.

Once deflections at various points have been determined, if any value is larger than the allowable deflection at that point, a larger shaft diameter is warranted. Since I is proportional to d^4, a new diameter can be found from

$$d_{\text{new}} = d_{\text{old}} \left| \frac{n_d y_{\text{old}}}{y_{\text{all}}} \right|^{1/4} \tag{7–17}$$

where y_{all} is the allowable deflection at that station and n_d is the design factor. Similarly, if any slope is larger than the allowable slope θ_{all}, a new diameter can be found from

$$d_{\text{new}} = d_{\text{old}} \left| \frac{n_d (dy/dx)_{\text{old}}}{(\text{slope})_{\text{all}}} \right|^{1/4} \tag{7–18}$$

Ex. 7.4

For the shaft in Example 7–3, it was noted that the slope at the right bearing is near the limit for a cylindrical roller bearing. Determine an appropriate increase in diameters to bring this slope down to 0.0005 rad.

Solution

Applying Equation (7–17) to the deflection at the right bearing gives

$$d_{\text{new}} = d_{\text{old}} \left| \frac{n_d \text{slope}_{\text{old}}}{\text{slope}_{\text{all}}} \right|^{1/4} = 25.4 \left| \frac{(1)(0.001095)}{(0.0005)} \right|^{1/4} = 30.9 \text{ mm}$$

Multiplying all diameters by the ratio

$$\frac{d_{\text{new}}}{d_{\text{old}}} = \frac{30.9}{25.4} = 1.216$$

gives a new set of diameters,

$$D_1 = D_7 = 30.9 \text{ mm}$$

$$D_2 = D_6 = 43.26 \text{ mm}$$

$$D_3 = D_5 = 50.21 \text{ mm}$$

$$D_4 = 61.8 \text{ mm}$$

Repeating the beam deflection analysis of Example 7–3 with these new diameters produces a slope at the right bearing of 0.0127 mm, with all other deflections less than their previous values.

سرعت بحرانی شفت

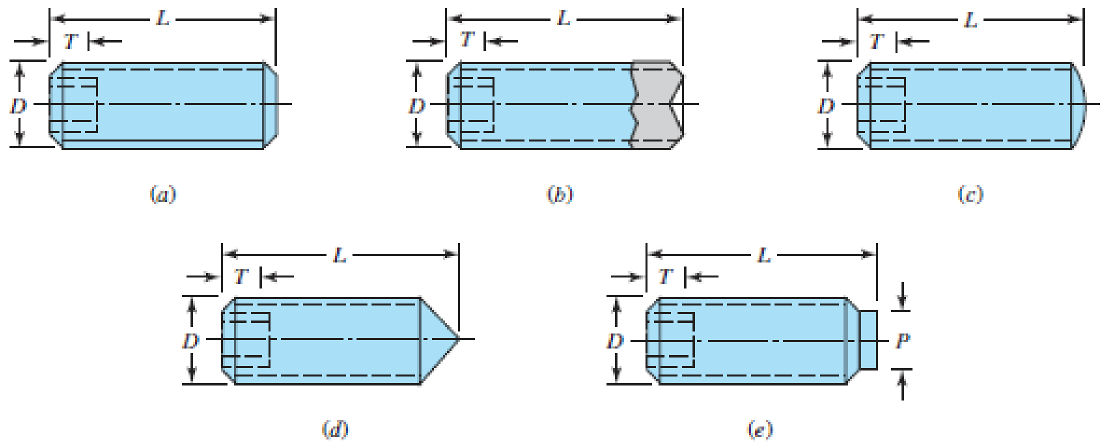
از کتاب خودتان مطالعه کنید

اجزای نگه شفت

پیچ مغزی

Figure 7–15

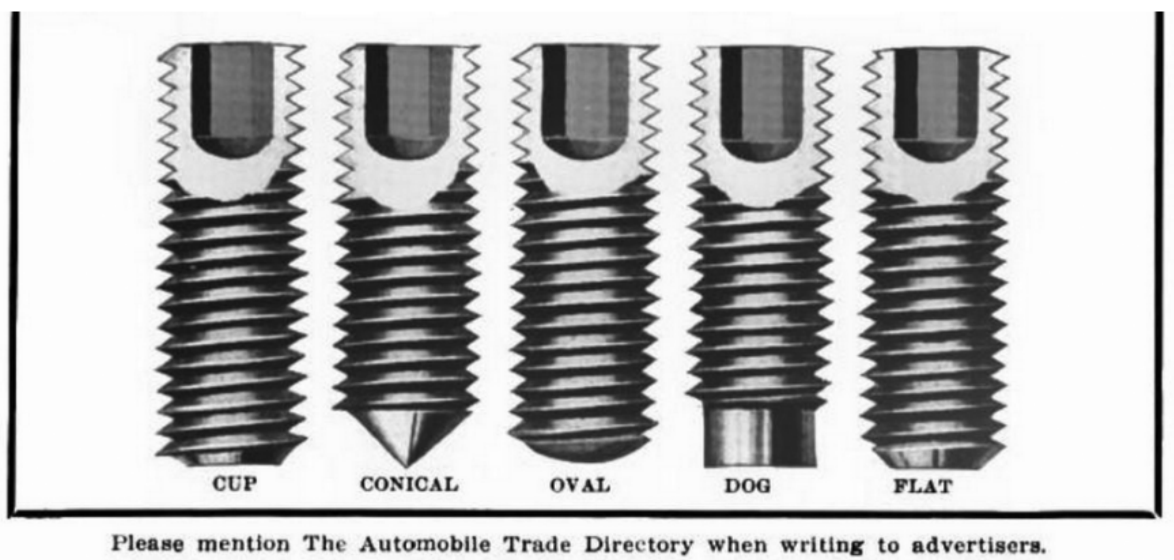
Socket setscrews: (*a*) flat point; (*b*) cup point; (*c*) oval point; (*d*) cone point; (*e*) half-dog point.



هم بار محوری

هم بار پیچشی

تحمل می کنند



رولر، غلطک، پولی

چرخ زنجیر

قفل کن شفت

کوپلینگ انعطاف پذیر

انواع مختلف خار

جهت انتقال گشتاور از شفت به هاب یا بالعکس

Figure 7–16

(*a*) Square key; (*b*) round key; (*c* and *d*) round pins; (*e*) taper pin; (*f*) split tubular spring pin. The pins in parts (*e*) and (*f*) are shown longer than necessary, to illustrate the chamfer on the ends, but their lengths should be kept smaller than the hub diameters to prevent injuries due to projections on rotating parts.

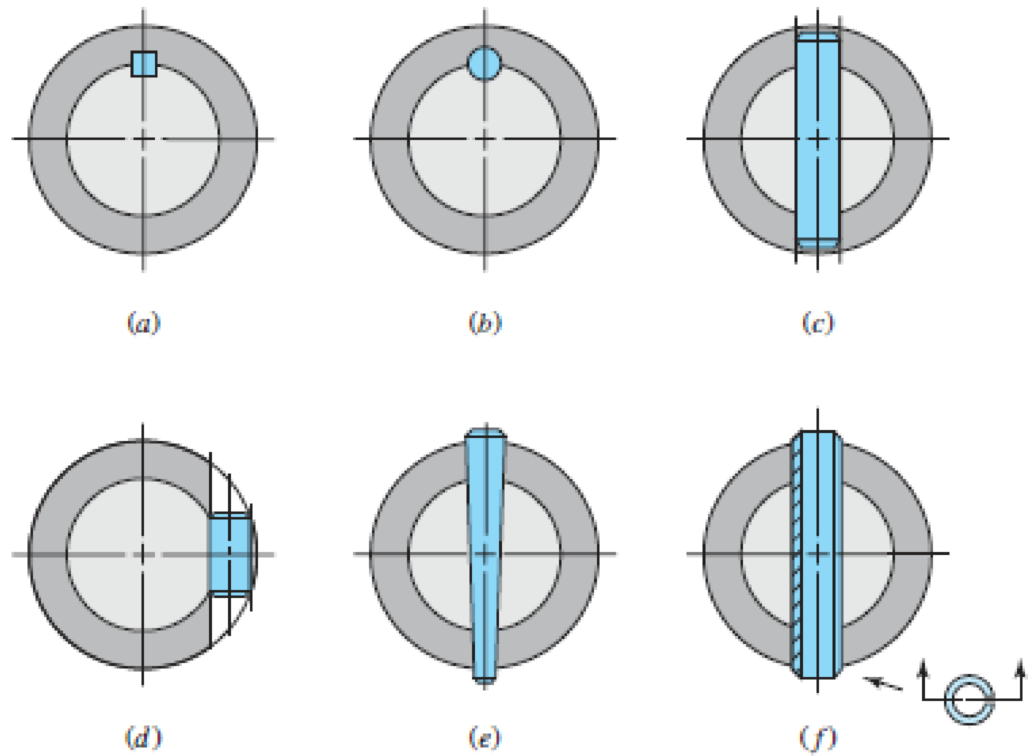


Table 7–6 Millimeter Dimensions for Some Standard Square- and Rectangular-Key Applications

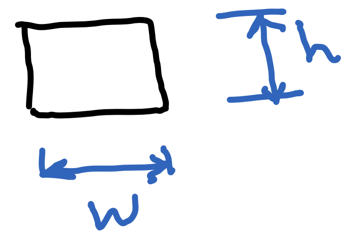
Shaft Diameter		Key Size		
Over	To (Incl.)	w	h	Keyway Depth
8	11	2	2	1
11	14	3	2	1
		3	3	1.5
14	22	5	3	1.5
		5	5	2
22	30	6	5	2
		6	6	3
30	36	8	6	3
		8	8	5
36	44	10	6	3
		10	10	5
44	58	12	10	5
		12	12	6
58	70	16	12	5.5
		16	16	8
70	80	20	12	6
		20	20	10

اندازه های استاندارد

خار مربعی یا مستطیلی

مستطیلی

مربعی



محاسبه طول خار مربعی

EXAMPLE 7–6

A UNS G10350 steel shaft, heat-treated to a minimum yield strength of 517.125 MPa, has a diameter of 36.513 mm. The shaft rotates at 600 rev/min and transmits 30 kW through a gear. Select an appropriate key for the gear, with a design factor of 1.5.

Solution

From Table 7–6, a 9.525 mm square key is selected. Choose a cold-drawn low-carbon mild steel which is generally available for key stock, such as UNS G10180, with a yield strength of 372 MPa.

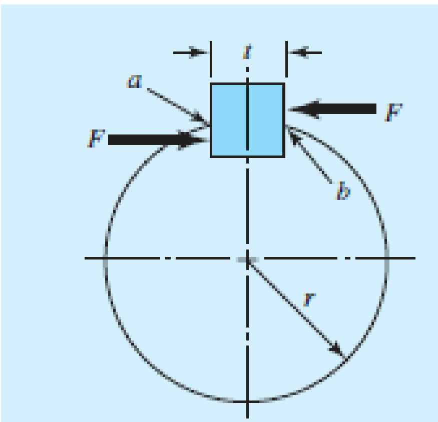


Figure 7–19

$d = 36.5\ mm$

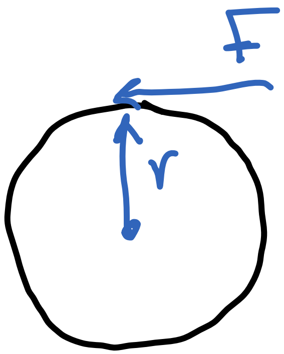
$n = 600\ rpm\ ,\ H = 30\ kW$

ضریب اطمینان $n = 1.5$

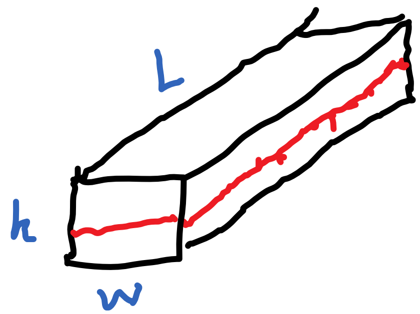
$H = T\omega \Rightarrow \underset{(N.m)}{T} = \dfrac{H\ (watt)}{\omega\ (rad/s)}$

$\omega = \dfrac{2\pi n}{60} = \dfrac{2 \times 3.14 \times 600}{60} = 62.8\ rad/s$

$$T = \frac{H}{\omega} = \frac{30 \times 10^{3}}{62.8} = 477.47 \ N \cdot m$$



$$T = F \times r \Rightarrow F = \frac{T}{r} = \frac{477.47}{\frac{36.5}{2} \times 10^{-3}} = 26 \ kN$$



$$\tau = \frac{F}{A} = \frac{F}{wL} = \frac{F}{tL}$$

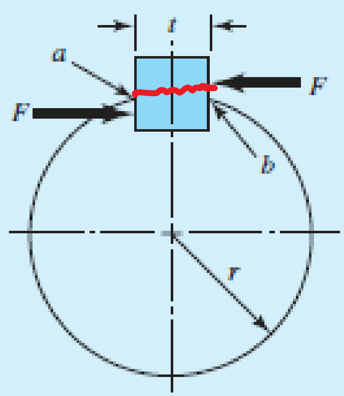


Figure 7–19

$$n = \frac{S_{sy}}{\tau} \Rightarrow n\tau = S_{sy}$$

$$\Rightarrow \frac{nF}{wL} = S_{sy} \Rightarrow L = \frac{nF}{w S_{sy}}$$

$$L = \frac{nF}{W S_{sy}} \quad , \quad S_{sy} = 0.5 S_y \leq 0.577 S_y$$

$$L = \frac{2nF}{w S_y} \quad , \quad n = 1.5 \; , \; F = 26 \text{ kN}$$

$$w = ? \quad , \quad S_y = 372 \text{ MPa}$$

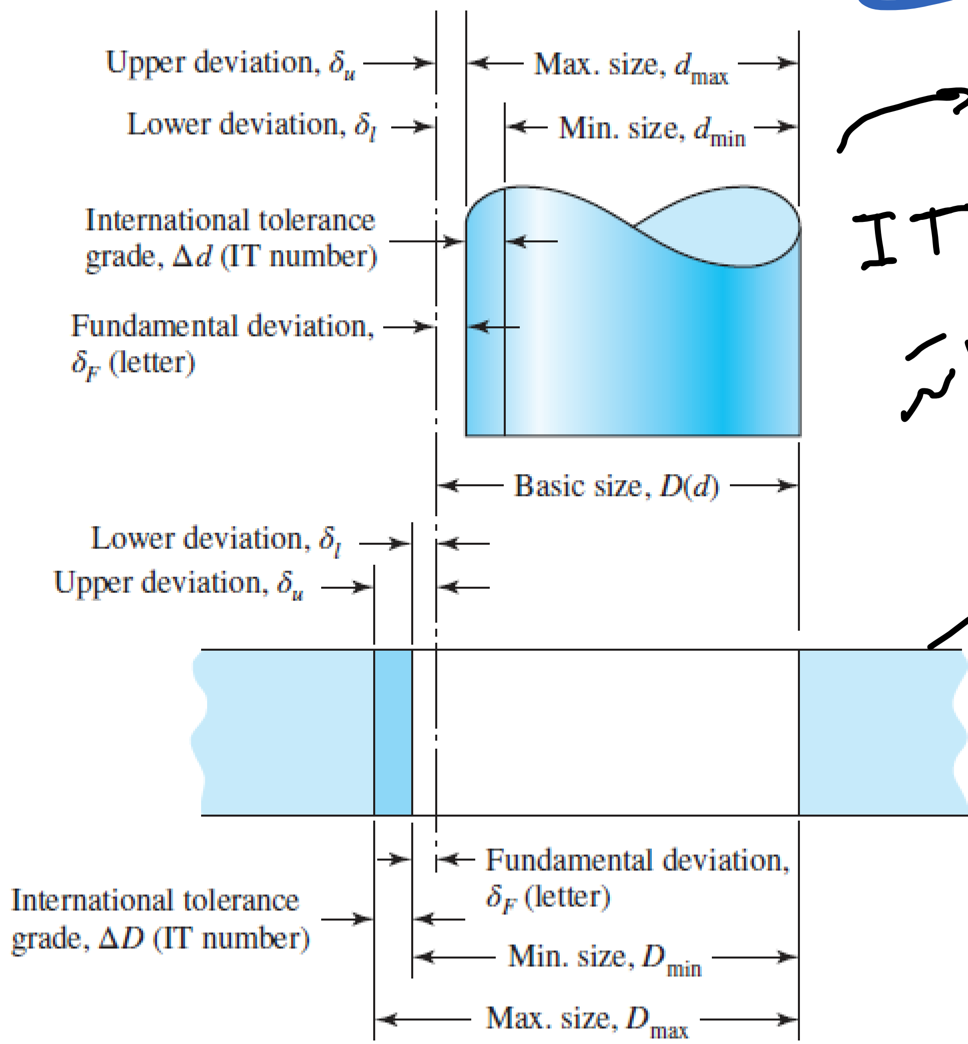
برای خارها

از جدول استاندارد خارها $10 \times 8 \; , \; \begin{cases} w = 10 \text{ mm} \\ h = 8 \text{ mm} \end{cases}$

$$L = \frac{2 \times 1.5 \times 26 \times \cancel{10^3}}{10 \times \cancel{10^{-3}} \times 372 \times \cancel{10^6}} = 0.0209 \text{ m} = 20.9 \text{ mm}$$

از جدول استاندارد $\Rightarrow \boxed{L = 22 \text{ mm}}$

تلرانس و انطباقات



Shaf

$IT = \Delta d = d_{max} - d_{min}$

IT کمتر یعنی نیز به دقت بالاتر

$IT \downarrow \equiv \Delta d \downarrow$

hole

$IT = \Delta D = D_{max} - D_{min}$

مقدار IT از جدول A-11

مقدار δ_F از جدول A-12

- ***Basic size*** is the size to which limits or deviations are assigned and is the same for both members of the fit.
- ***Deviation*** is the algebraic difference between a size and the corresponding basic size.
- ***Upper deviation*** is the algebraic difference between the maximum limit and the corresponding basic size.
- ***Lower deviation*** is the algebraic difference between the minimum limit and the corresponding basic size.

- ***Fundamental deviation*** is either the upper or the lower deviation, depending on which is closer to the basic size.
- ***Tolerance*** is the difference between the maximum and minimum size limits of a part.
- ***International tolerance grade*** numbers (IT) designate groups of tolerances such that the tolerances for a particular IT number have the same relative level of accuracy but vary depending on the basic size.
- ***Hole basis*** represents a system of fits corresponding to a basic hole size. The fundamental deviation is H.
- ***Shaft basis*** represents a system of fits corresponding to a basic shaft size. The fundamental deviation is h. The shaft-basis system is not included here.

Table 7–9 Descriptions of Preferred Fits Using the Basic Hole System

Type of Fit	Description	Symbol
Clearance	*Loose running fit:* for wide commercial tolerances or allowances on external members	H11/c11
	Free running fit: not for use where accuracy is essential, but good for large temperature variations, high running speeds, or heavy journal pressures	H9/d9
	Close running fit: for running on accurate machines and for accurate location at moderate speeds and journal pressures	H8/f7
	Sliding fit: where parts are not intended to run freely, but must move and turn freely and locate accurately	H7/g6
	Locational clearance fit: provides snug fit for location of stationary parts, but can be freely assembled and disassembled	H7/h6
Transition	*Locational transition fit:* for accurate location, a compromise between clearance and interference	H7/k6
	Locational transition fit: for more accurate location where greater interference is permissible	H7/n6
Interference	*Locational interference fit:* for parts requiring rigidity and alignment with prime accuracy of location but without special bore pressure requirements	H7/p6
	Medium drive fit: for ordinary steel parts or shrink fits on light sections, the tightest fit usable with cast iron	H7/s6
	Force fit: suitable for parts that can be highly stressed or for shrink fits where the heavy pressing forces required are impractical	H7/u6

Hole   شفت

H7/g6

IT7   IT6

جدول A-12

جدول A11

The fundamental deviations for shafts are given in Tables A–11 and A–13. For letter codes c, d, f, g, and h,

Upper deviation = fundamental deviation
Lower deviation = upper deviation − tolerance grade

For letter codes k, n, p, s, and u, the deviations for shafts are

Lower deviation = fundamental deviation
Upper deviation = lower deviation + tolerance grade

The lower deviation H (for holes) is zero. For these, the upper deviation equals the tolerance grade.

As shown in Figure 7–20, we use the following notation:

$$
\begin{aligned}
D &= \text{basic size of hole} \\
d &= \text{basic size of shaft} \\
\delta_u &= \text{upper deviation} \\
\delta_l &= \text{lower deviation} \\
\delta_F &= \text{fundamental deviation} \\
\Delta D &= \text{tolerance grade for hole} \\
\Delta d &= \text{tolerance grade for shaft}
\end{aligned}
$$

Note that these quantities are all deterministic. Thus, for the hole,

$$D_{max} = D + \Delta D \qquad D_{min} = D \tag{7–36}$$

For shafts with clearance fits c, d, f, g, and h,

$$d_{max} = d + \delta_F \qquad d_{min} = d + \delta_F - \Delta d \tag{7–37}$$

For shafts with interference fits k, n, p, s, and u,

$$d_{min} = d + \delta_F \qquad d_{max} = d + \delta_F + \Delta d \tag{7–38}$$

$D_{min} = D \quad , \quad D_{max} = D + \Delta D$

Table A–11 A Selection of International Tolerance Grades—Metric Series

(Size Ranges Are for *Over* the Lower Limit and *Including* the Upper Limit. All Values Are in Millimeters)

Basic Sizes	Tolerance Grades					
	IT6	IT7	IT8	IT9	IT10	IT11
0–3	0.006	0.010	0.014	0.025	0.040	0.060
3–6	0.008	0.012	0.018	0.030	0.048	0.075
6–10	0.009	0.015	0.022	0.036	0.058	0.090
10–18	0.011	0.018	0.027	0.043	0.070	0.110
18–30	0.013	0.021	0.033	0.052	0.084	0.130
30–50	0.016	0.025	0.039	0.062	0.100	0.160
50–80	0.019	0.030	0.046	0.074	0.120	0.190
80–120	0.022	0.035	0.054	0.087	0.140	0.220
120–180	0.025	0.040	0.063	0.100	0.160	0.250
180–250	0.029	0.046	0.072	0.115	0.185	0.290
250–315	0.032	0.052	0.081	0.130	0.210	0.320
315–400	0.036	0.057	0.089	0.140	0.230	0.360

Source: Preferred Metric Limits and Fits, ANSI B4.2-1978. See also BSI 4500.

IT0 تا IT16

δ_F نسبت به حروف (c, d, ...) وحروف مبنا از این جدول خوانده می‌شود

Table A–12 **Fundamental Deviations for Shafts—Metric Series**

(Size Ranges Are for *Over* the Lower Limit and *Including* the Upper Limit. All Values Are in Millimeters)

Basic Sizes	Upper-Deviation Letter					Lower-Deviation Letter				
	c	d	f	g	h	k	n	p	s	u
0–3	−0.060	−0.020	−0.006	−0.002	0	0	+0.004	+0.006	+0.014	+0.018
3–6	−0.070	−0.030	−0.010	−0.004	0	+0.001	+0.008	+0.012	+0.019	+0.023
6–10	−0.080	−0.040	−0.013	−0.005	0	+0.001	+0.010	+0.015	+0.023	+0.028
10–14	−0.095	−0.050	−0.016	−0.006	0	+0.001	+0.012	+0.018	+0.028	+0.033
14–18	−0.095	−0.050	−0.016	−0.006	0	+0.001	+0.012	+0.018	+0.028	+0.033
18–24	−0.110	−0.065	−0.020	−0.007	0	+0.002	+0.015	+0.022	+0.035	+0.041
24–30	−0.110	−0.065	−0.020	−0.007	0	+0.002	+0.015	+0.022	+0.035	+0.048
30–40	−0.120	−0.080	−0.025	−0.009	0	+0.002	+0.017	+0.026	+0.043	+0.060
40–50	−0.130	−0.080	−0.025	−0.009	0	+0.002	+0.017	+0.026	+0.043	+0.070
50–65	−0.140	−0.100	−0.030	−0.010	0	+0.002	+0.020	+0.032	+0.053	+0.087
65–80	−0.150	−0.100	−0.030	−0.010	0	+0.002	+0.020	+0.032	+0.059	+0.102
80–100	−0.170	−0.120	−0.036	−0.012	0	+0.003	+0.023	+0.037	+0.071	+0.124
100–120	−0.180	−0.120	−0.036	−0.012	0	+0.003	+0.023	+0.037	+0.079	+0.144
120–140	−0.200	−0.145	−0.043	−0.014	0	+0.003	+0.027	+0.043	+0.092	+0.170
140–160	−0.210	−0.145	−0.043	−0.014	0	+0.003	+0.027	+0.043	+0.100	+0.190
160–180	−0.230	−0.145	−0.043	−0.014	0	+0.003	+0.027	+0.043	+0.108	+0.210

$D = 34^{0.16}_{0.000}$ $\qquad$ $d = 34^{-0.12}_{-0.28}$

EXAMPLE 7–7

Find the shaft and hole dimensions for a loose running fit with a 34-mm basic size.

Solution

From Table 7–9, the ISO symbol is 34H11/c11. From Table A–11, we find that tolerance grade IT11 is 0.160 mm. The symbol 34H11/c11 therefore says that $\Delta D = \Delta d = 0.160$ mm. Using Equation (7–36) for the hole, we get

Answer $$D_{max} = D + \Delta D = 34 + 0.160 = 34.160 \text{ mm}$$

Answer $$D_{min} = D = 34.000 \text{ mm}$$

The shaft is designated as a 34c11 shaft. From Table A–12, the fundamental deviation is $\delta_F = -0.120$ mm. Using Equation (7–37), we get for the shaft dimensions

Answer $$d_{max} = d + \delta_F = 34 + (-0.120) = 33.880 \text{ mm}$$

Answer $$d_{min} = d + \delta_F - \Delta d = 34 + (-0.120) - 0.160 = 33.720 \text{ mm}$$

34 H11/c11 , IT11 $\Rightarrow \Delta d = \Delta D = 0.16$

$D = d = 34$ قطر مبنا

$H \rightarrow \delta_F = 0$, $c \rightarrow \delta_F = -0.12$

EXAMPLE 7–8

Find the hole and shaft limits for a medium drive fit using a basic hole size of 50 mm.

Solution

The symbol for the fit, from Table 7–9, in inch units is (50 mm)H7/s6. For the hole, we use Table A–11 and find the IT7 grade to be $\Delta D = 0.03$ mm. Thus, from Equation (7–36),

Answer
$$D_{\text{max}} = D + \Delta D = 50 + 0.03 = 50.03 \text{ mm}$$

Answer
$$D_{\text{min}} = D = 50 \text{ mm}$$

The IT6 tolerance for the shaft is $\Delta d = 0.019$ mm. Also, from Table A–12, the fundamental deviation is $\delta_F = -0.13$ mm. Using Equation (7–38), we get for the shaft that

Answer
$$d_{\text{min}} = d + \delta_F = 50 + (-0.13) = 49.87 \text{ mm}$$

Answer
$$d_{\text{max}} = d + \delta_F + \Delta d = 50 + (-0.13) + 0.019 = 49.889 \text{ mm}$$

$$D = 50^{0.03}_{0.000} \qquad d = 50^{-0.111}_{-0.13}$$

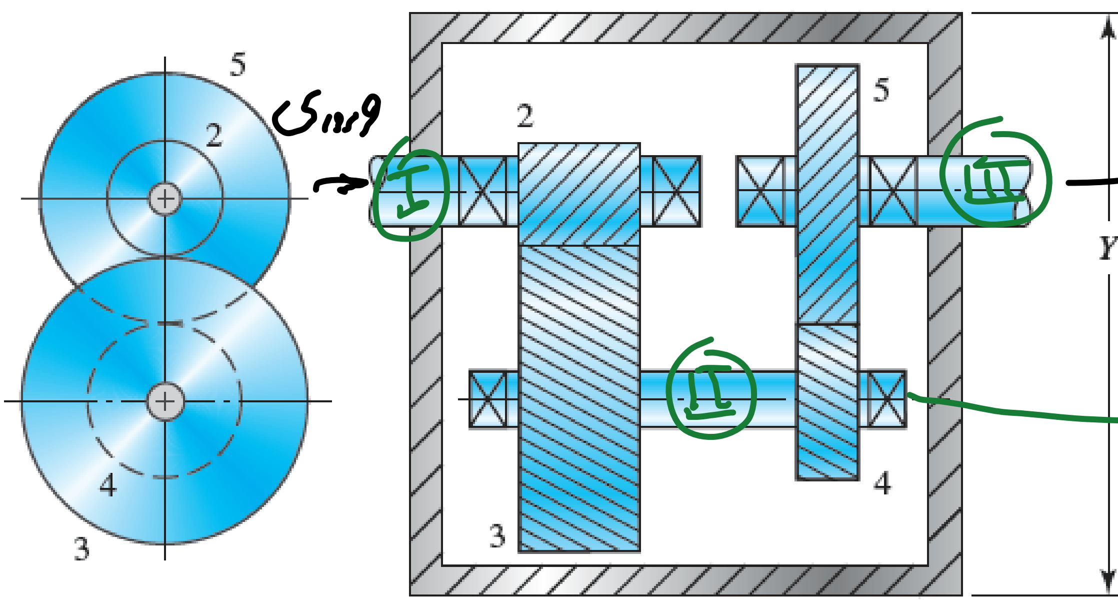
جلسه نوزدهم

Ex 18-1
فصل ۱۸ کتاب

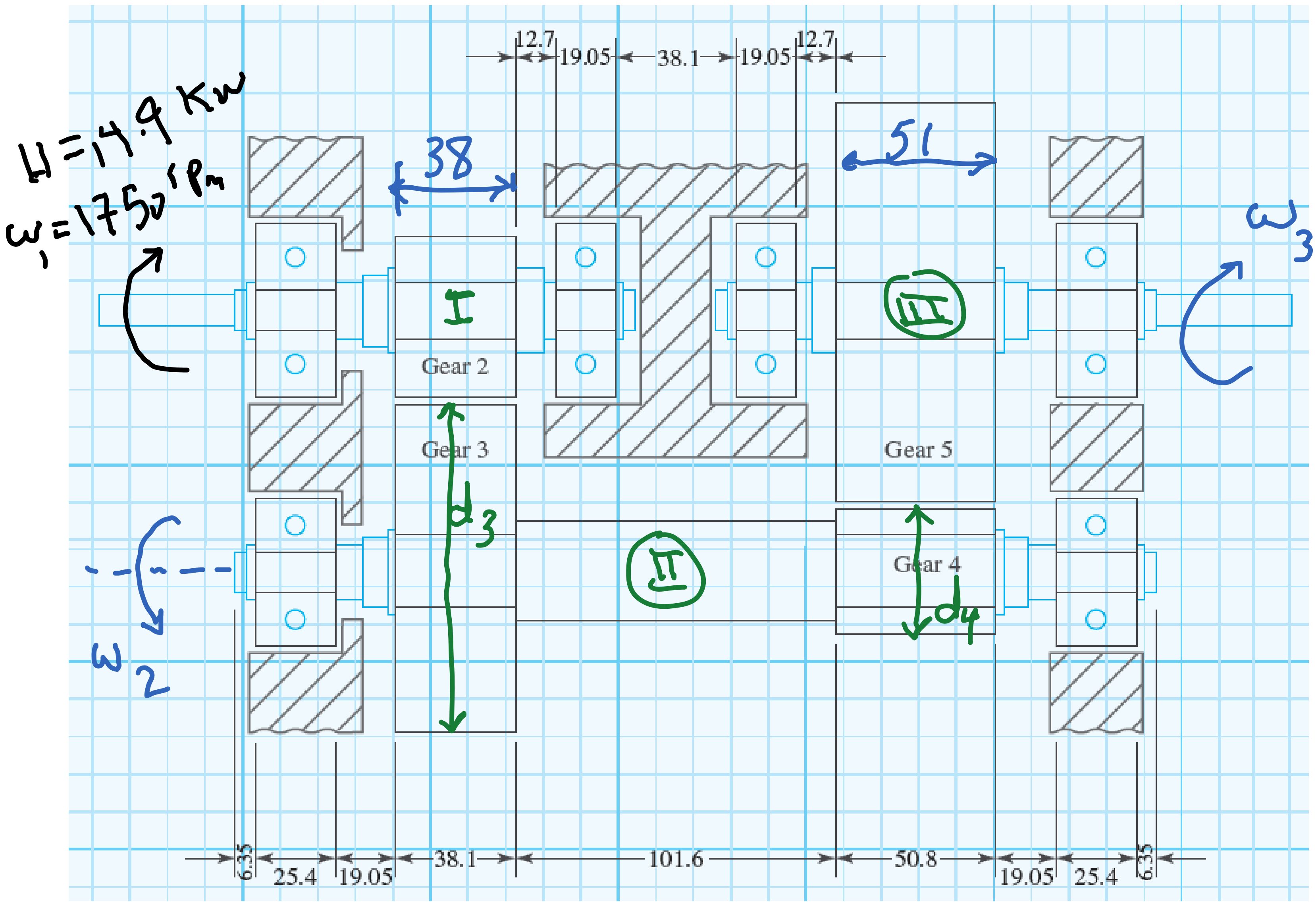
توان ورودی 14.9 KW

سرعت ورودی 1750 rpm

سرعت خروجی 82-88 rpm



همان شفتی است که در جلسه هفدهم ما طراحی کردیم


H=14.9 Kw
ω1=1750 rpm
12.7
19.05
38.1
19.05
12.7
38
51
ω3
I
Gear 2
Gear 3
d3
II
III
Gear 5
Gear 4
d4
ω2
6.35
25.4
19.05
38.1
101.6
50.8
19.05
25.4
6.35

قطر چرخ دنده

$$\left.\begin{array}{l} d_2 = d_4 = 72\,mm \quad , \quad w_2 = w_3 = 38\,mm \\ d_3 = d_5 = 324\,mm \quad , \quad w_4 = w_5 = 51\,mm \end{array}\right.$$

واقعی چرخ دنده ها

عرض چرخ دنده ها

$$n_1 = \frac{d_3}{d_2} = \frac{324}{72} = 4.5 \longrightarrow$$ نسبت تبدیل بین شفت I و II

$$n_2 = \frac{d_5}{d_4} = \frac{324}{72} = 4.5$$ III و II 〃 〃 〃

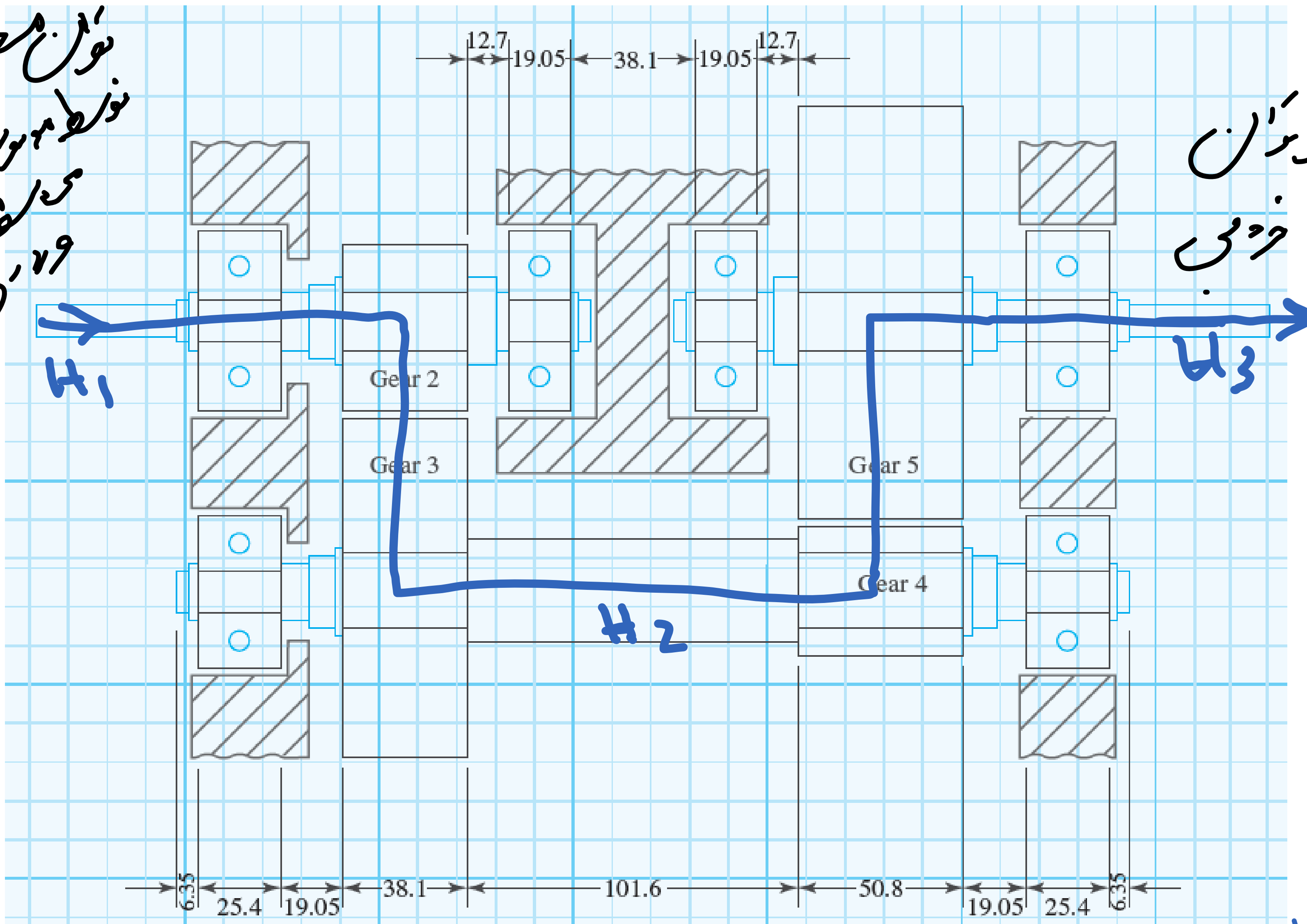
سرعت شفت II

$$\omega_1 = 1750\,rpm \,,\, \omega_2 = \frac{1750}{4.5} = 388.8\,rpm$$

$$\omega_3 = \frac{388.8}{4.5} = 86.4\,rpm$$

جریان توان

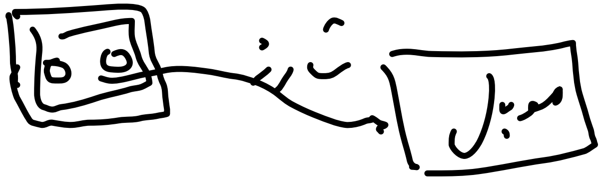
توان مصرفی توسط موتور تامین می‌شود ورودی



مصرف توان خروجی

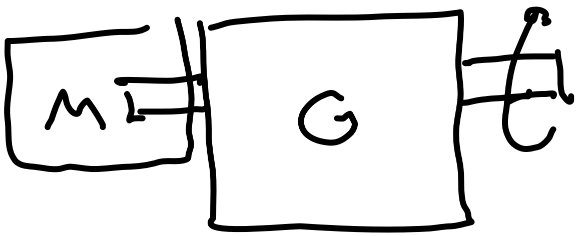
$$H_1 = H_2 = H_3$$

عدم تغییر اندازه توان



مصرف کننده توان مورد نیازش را از موتور می کشد یا تأمین می کند.

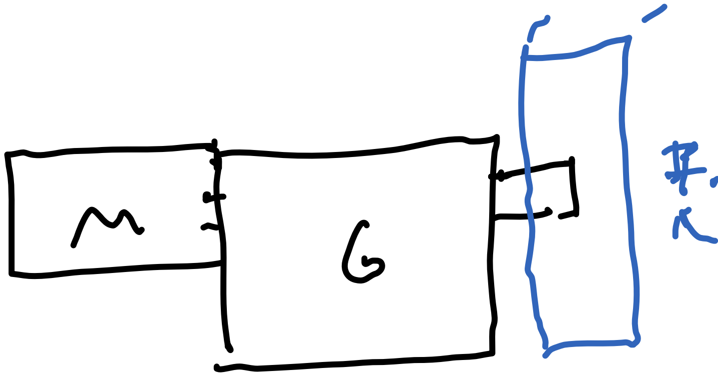
هرزگرد



سری

اگر گیربکس در خروجی به بالابر متصل نباشد، چقدر توان مصرف می کند. توان مصرفی صفر است یا با در نظر گرفتن اتلافات، 10% H

1.5 kW



سری

اگر گیربکس به یک نوار نقاله متصل شد 7.5 kW

حالتی که نوار نقاله خالی است، نیمه مصرف توان 50%

وقتی نوار نقاله پر از بار است، توان مصرفی 100% توان ورودی 15 kW

اگر مصرف‌کننده بیشتر از توان موتور توان نیاز داشته باشد چه می‌شود

$H_i = 15 kW$, $H_o = 20 kW$

موتور حداکثر می‌تواند 15kW به صورت دائم تأمین کند

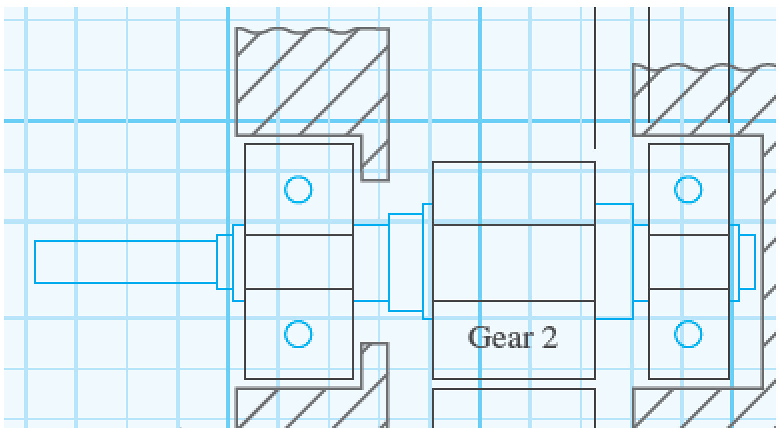
بسته به مشخصات موتور، به صورت لحظه‌ای در حد چند ثانیه هم می‌تواند تا H_{max} را تأمین کند.

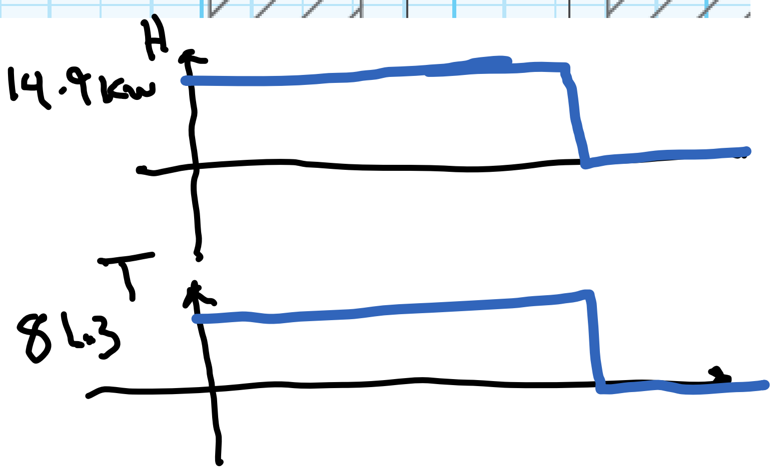
اما در طولانی مدت اگر H_i نامی $< H_o$ موتور داغ می‌کند

سپس با بار بیشتر می‌سوزد.

$$\omega_1 = 1750\, rpm \Rightarrow \omega_1 = \frac{2\pi \times 1750}{60} = 183.2\ rad/s$$

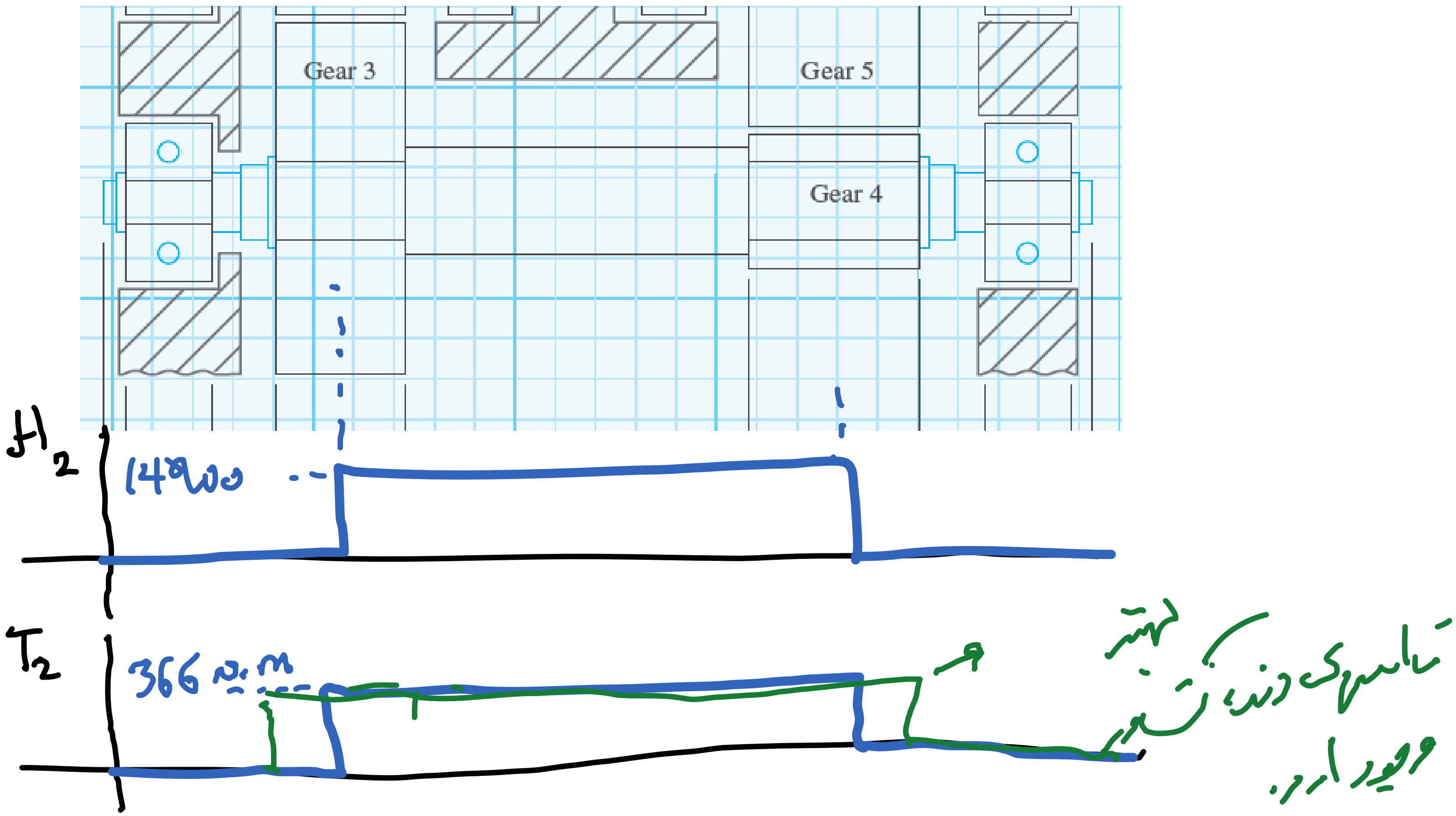
$$H_1 = T_1 \omega_1 \Rightarrow T_1 = \frac{H_1}{\omega_1} = \frac{14.9 \times 10^3}{183.2} = 81.3\ N.m$$





$$\omega_2 = 388.8 \text{ rpm} \Rightarrow \omega_2 = \frac{2\pi \times 388.8}{60} = 40.7 \text{ rad/s}$$

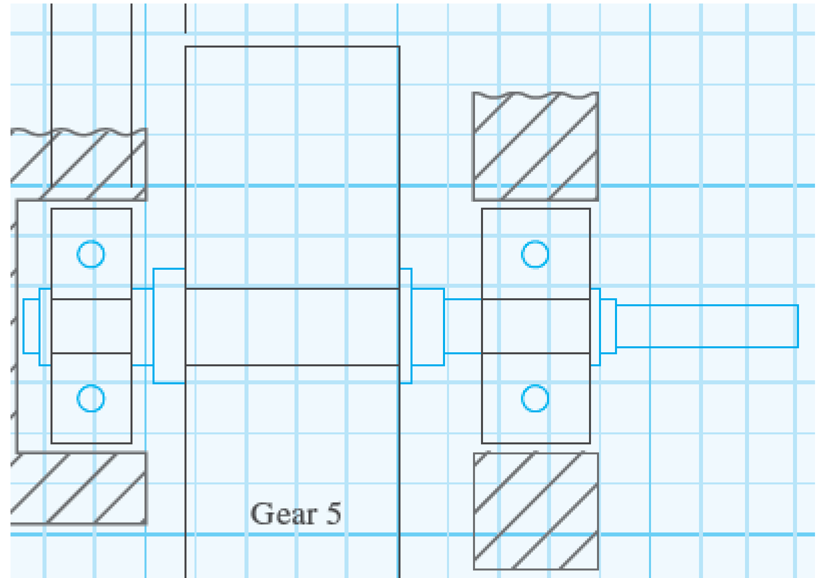
$$T_2 = \frac{H_2}{\omega_2} = \frac{14900}{40.7} = 366 \text{ N.m}$$

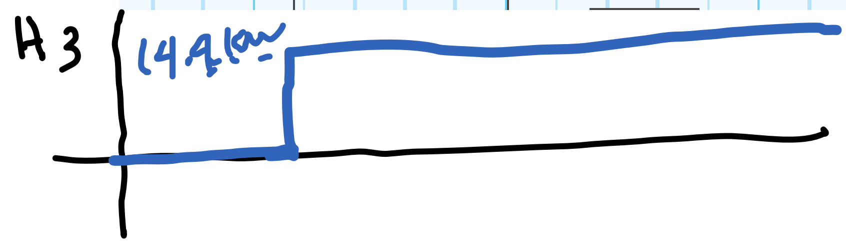


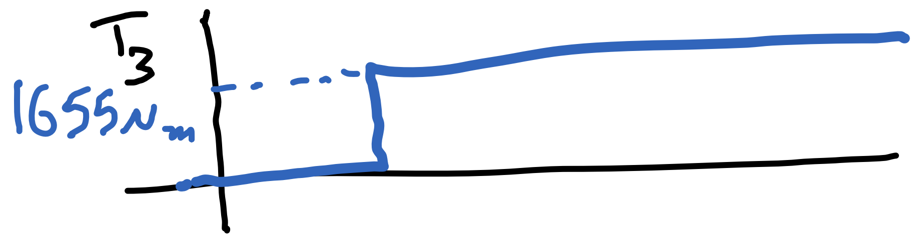
$$\omega_3 = 86.4 \text{ rpm}$$

$$\omega_3 = \frac{86.4 \times 2\pi}{60} = 9 \text{ rad/s}$$

$$T_3 = \frac{H_3}{\omega_3} = \frac{14900}{9} = 1655 \text{ N.m}$$







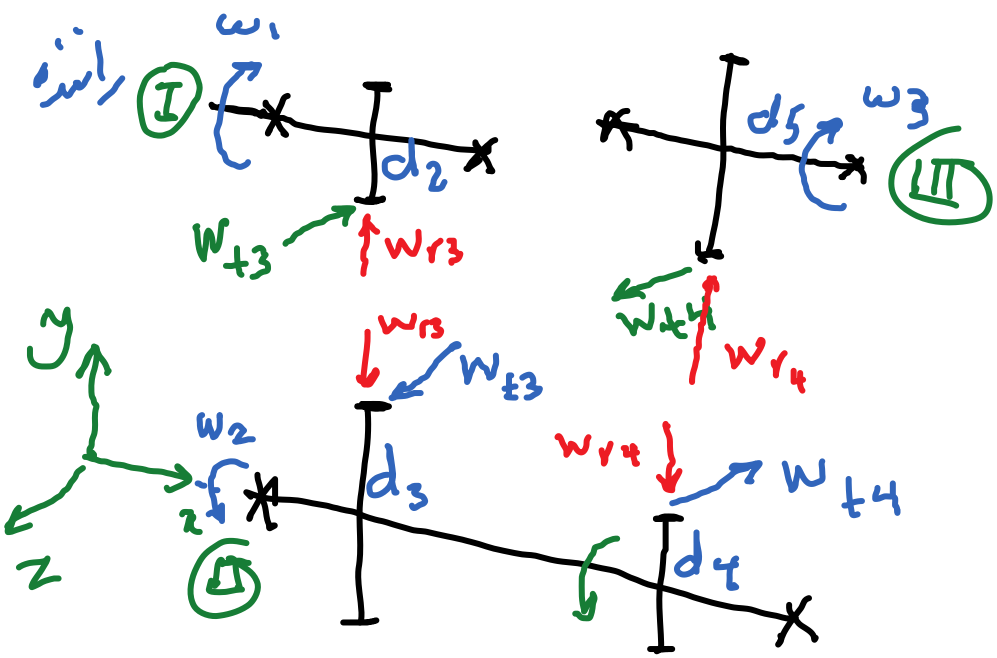
$T_3 = 1655$, $\omega_3 = 9$

$T_2 = 366$, $\omega_2 = 40.7$

$T_1 = 81.3$, $\omega_1 = 183$

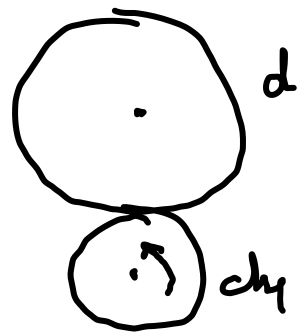
[illegible]

$$T_3 / T_2 = 4.5 \qquad \frac{T_2}{T_1} = 4.5 \qquad T_1 \omega_1 = T_2 \omega_2 = T_3 \omega_2$$



محاسبه نیروها

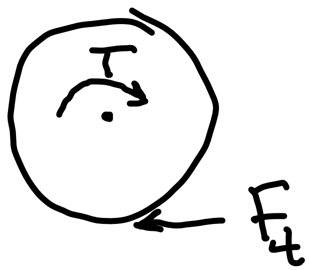
روی شفت II



d_4 رانده

d_5 راننده

$$T = F_t \times d/2$$



$$W_{t3} = \frac{T_2}{r_3} = \frac{366}{324/2 \times 10^{-3}} = 2260\ N$$

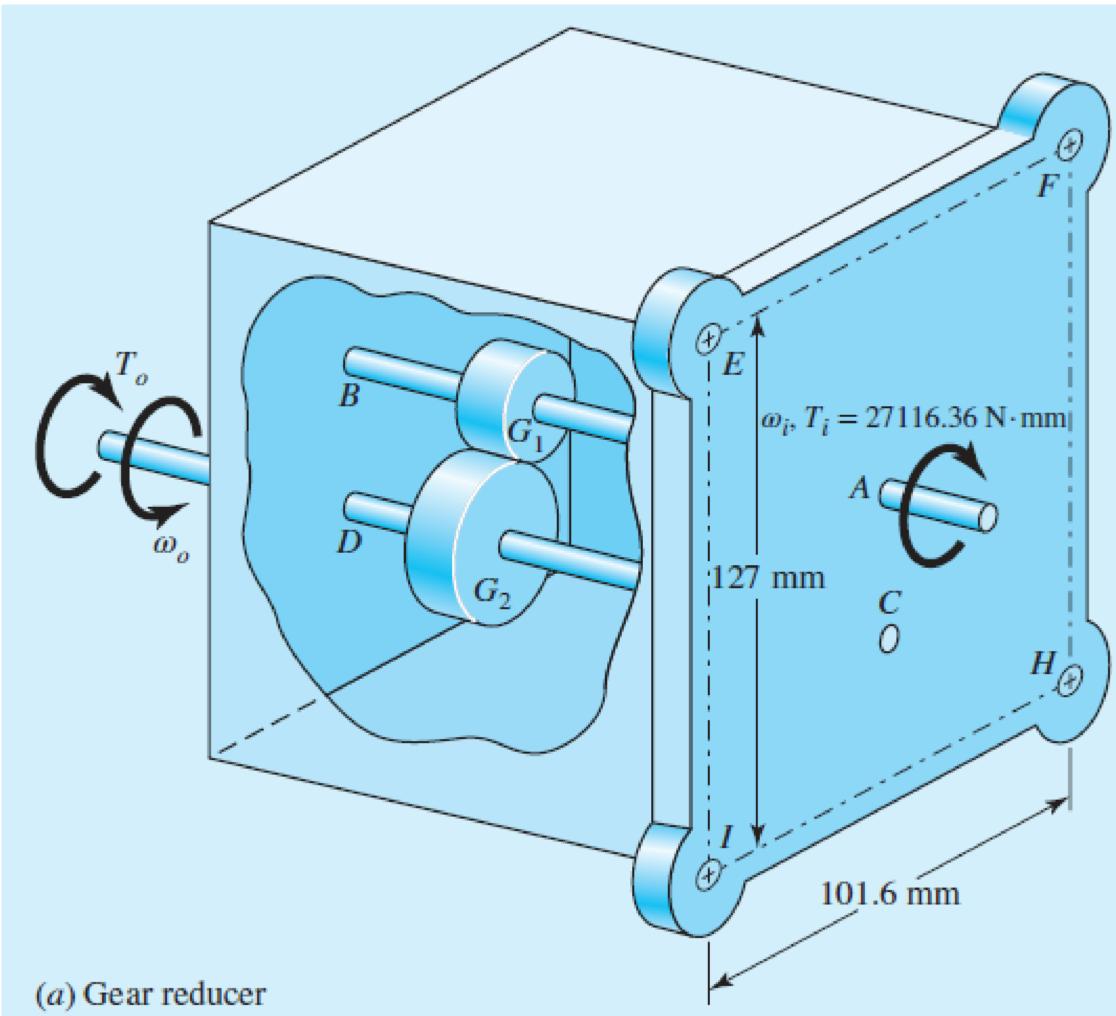
$$W_{t4} = \frac{T_2}{r_2} = \frac{366 \times 2}{72 \times 10^{-3}} = 10166\ N$$

$$W_{r3} = W_{t3}\, tg20 = 2260 \times tg20 = 822\,N$$

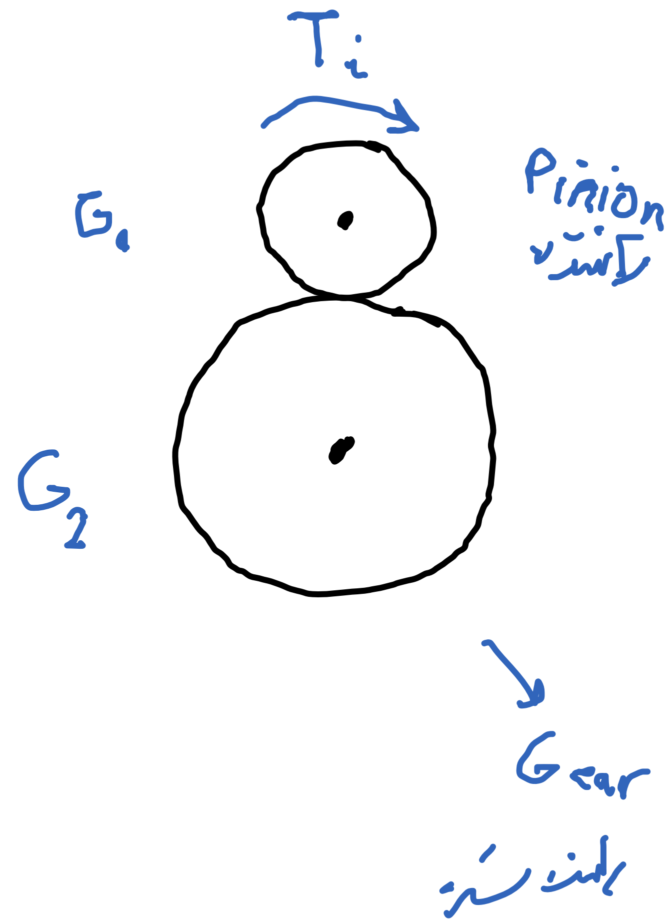
$$W_{r4} = W_{t4}\, tg20 = 10166 \times tg20 = 3700\,N$$

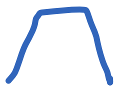
تعادل و دیاگرام آزاد

تحلیل نیرو و تنش



(a) Gear reducer





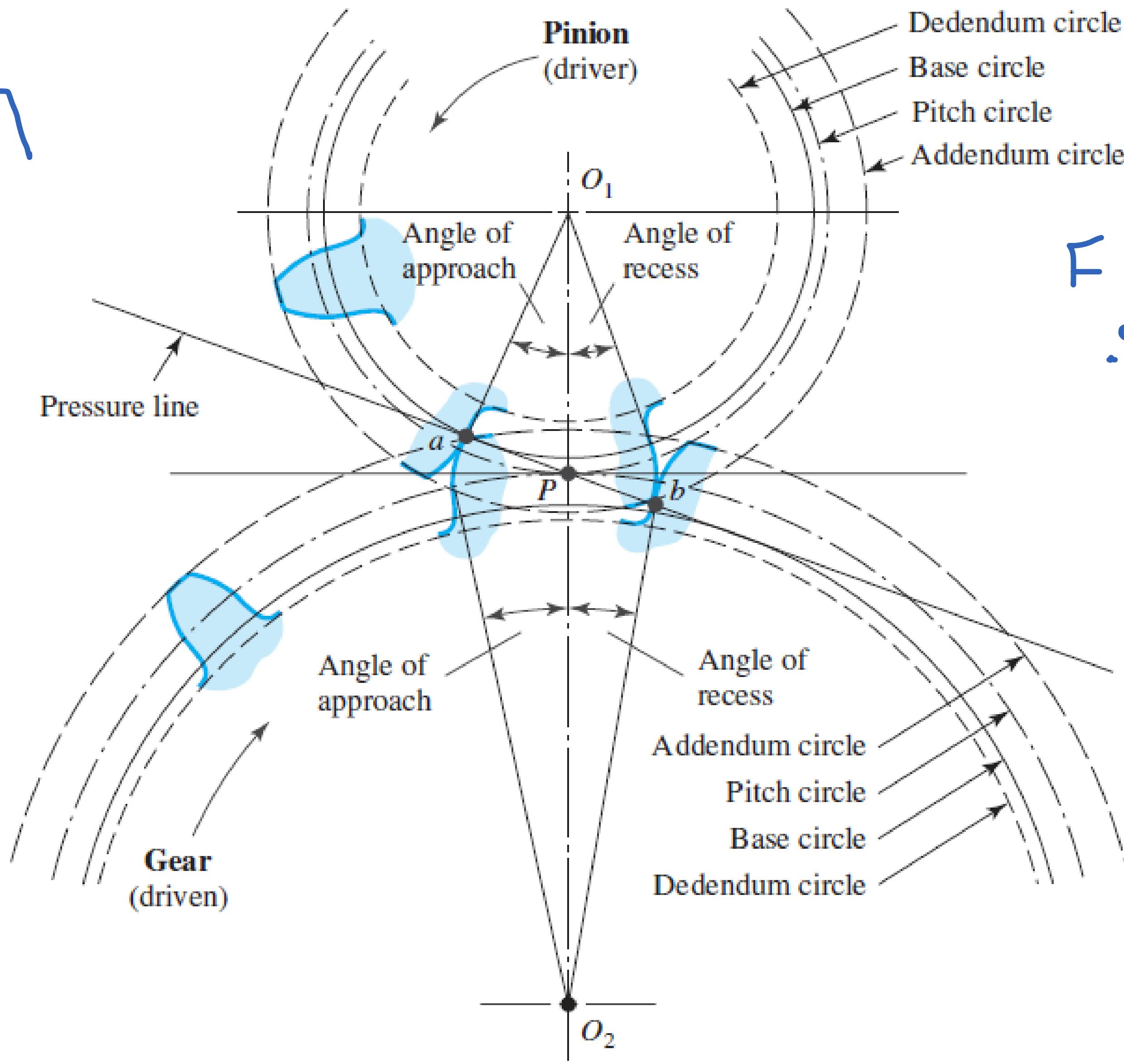


Figure 13–8

Tooth action.

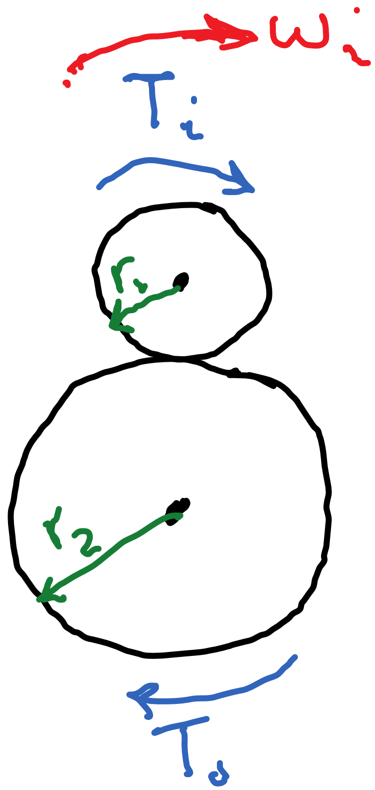
$\boxed{\varphi = 20}$

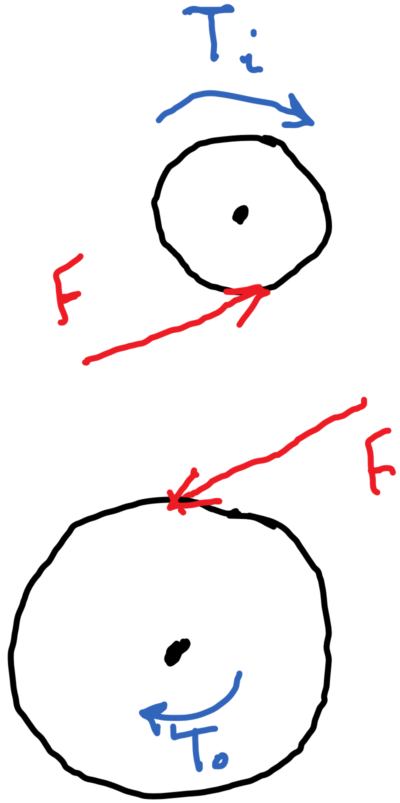
22

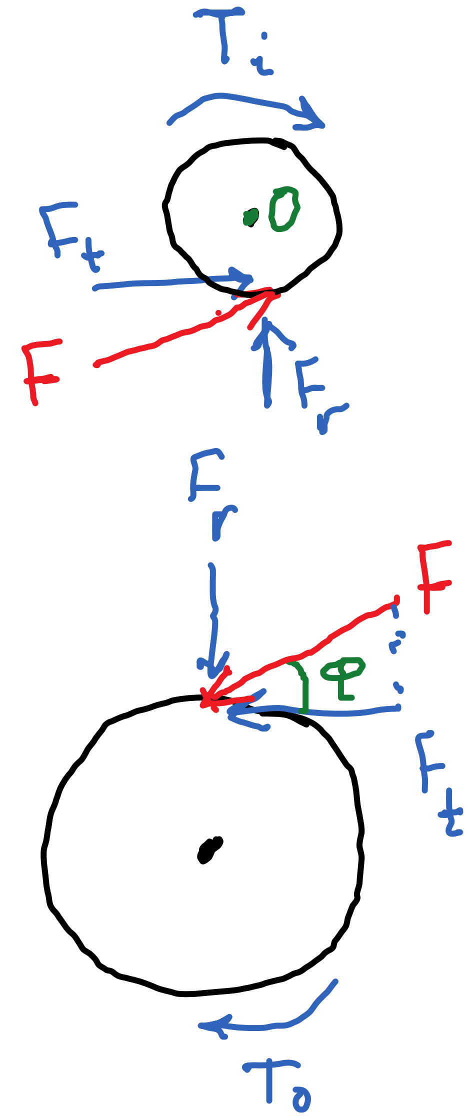
14.5

تعادل و دیاگرام آزاد

تحلیل نیرو و تنش





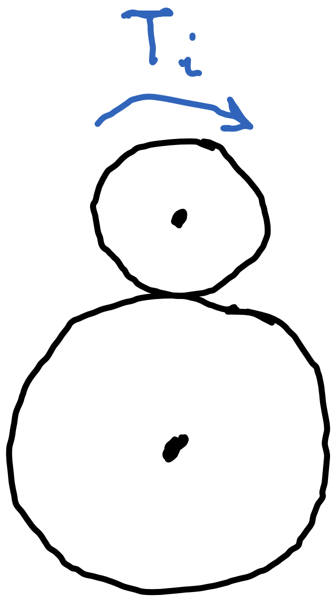


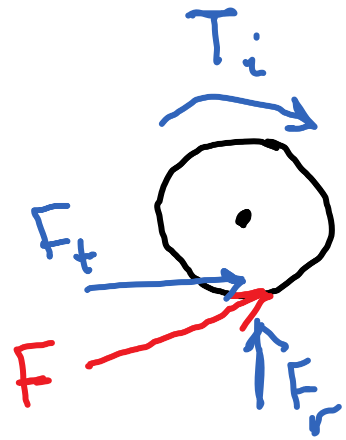
$$\sum M_o = 0 \Rightarrow F_t \times r_1 - T_i = 0$$

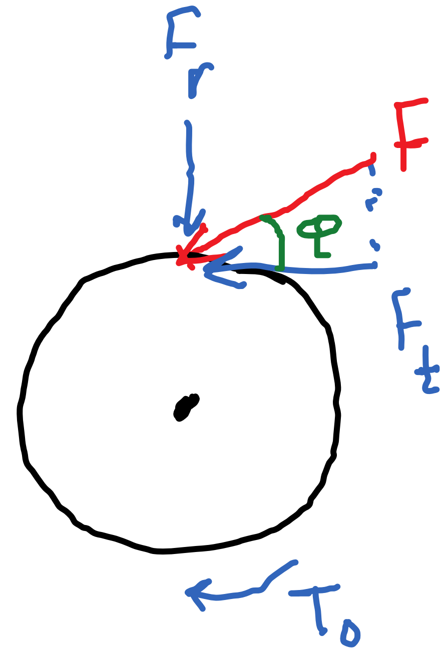
$$\Rightarrow T_i = F_t \times r_1$$

تحلیل نیرو و تنش

تعادل و دیاگرام آزاد

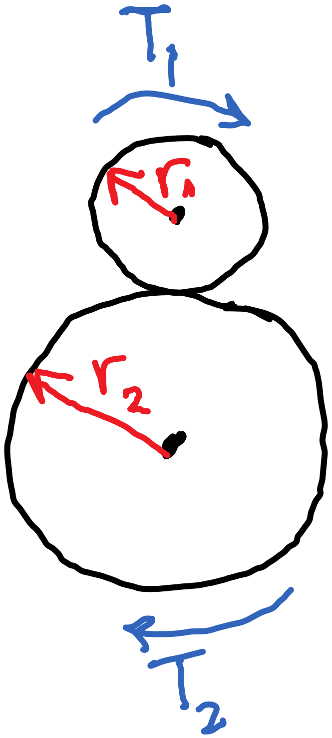






$$\left.\begin{aligned} T_i &= F_t \times r_1 \\ T_o &= F_t \times r_2 \end{aligned}\right\} \Rightarrow \frac{T_i}{T_o} = \frac{r_1}{r_2} = \frac{1}{2}$$

$$\Rightarrow T_o = 2T_i$$



$$\frac{T_2}{T_1} = \frac{r_2}{r_1} = n = \frac{N_2}{N_1}$$

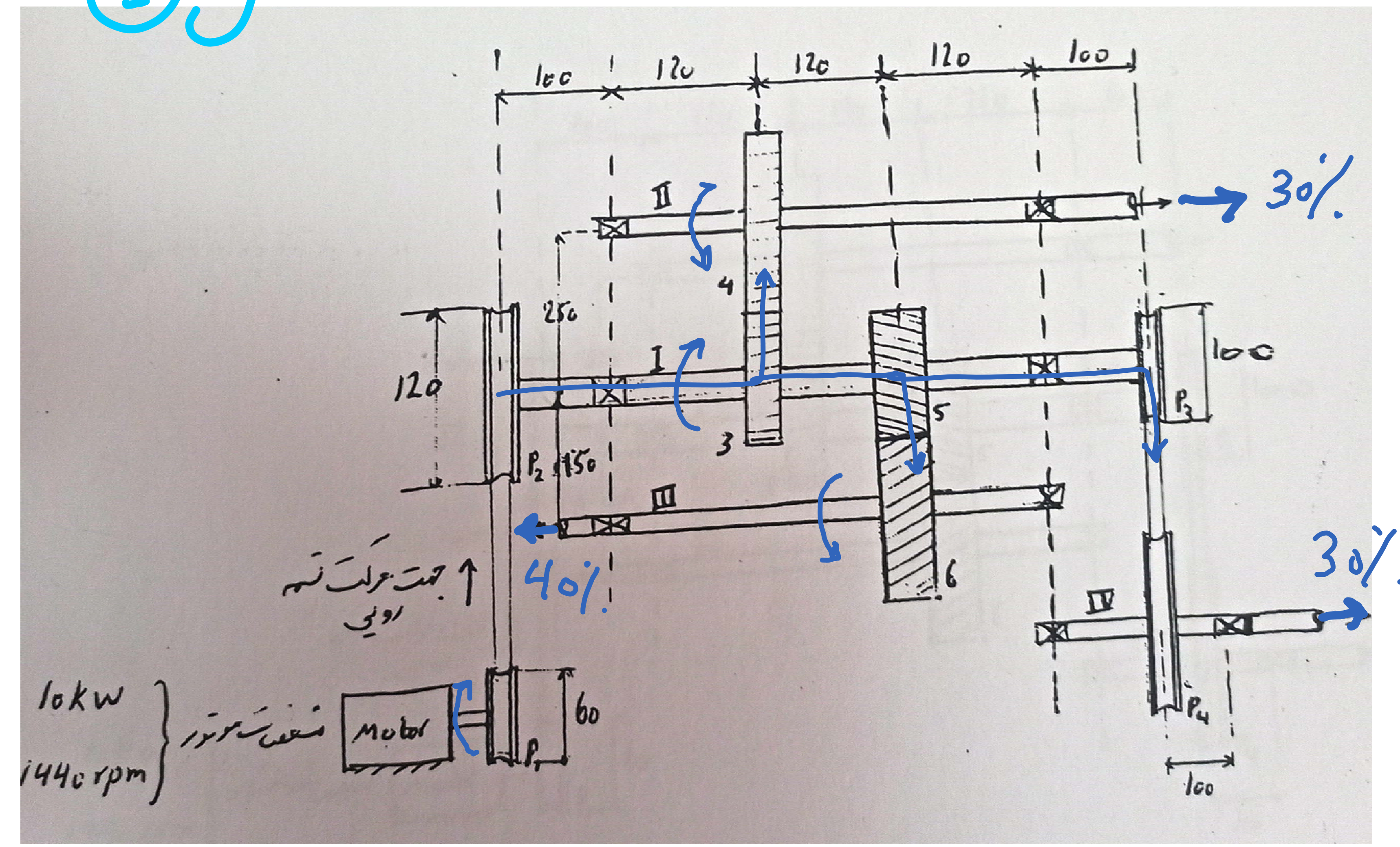
n { نسبت تبدیل گیربکس
Gear ratio

$$T_2 = n T_1$$

جلسه بیستم

مثال دیگری از طراحی شفت

مثال 1



در سیستم انتقال قدرت نشان داده شده، موتوری با توان
$10KW$ و سرعت $1440rpm$ از طریق پولی P_1 و P_2 توان را به شفت I
منتقل می نماید. شفت I از طریق چرخدنده 3 و چرخدنده 5 و
پولی P_3 به ترتیب 30%، 40% و 30% توان ورودی را به قسمتهای
مختلف انتقال می دهد. محور هم شفتها در یک صفحه می باشد.
و مصرف توان - معمورت یکنواخت و دائمی است. کلیه چرخدنده ها و پولی ها
با کمک خارهایی که با فرز انگشتی ایجاد شده اند روی شفتها سوار می شوند.
طراحی کامل شفت با ضریب اطمینان 3 و طراحی خارها با ضریب اطمینان 2.5
مورد نظر است.

جنس همه شفت‌ها از فولاد CK45 با $S_{ut} = 640\,MPa$ و $S_y = 420\,MPa$ می‌باشد. جنس خارها از فولاد با $S_y = 370\,MPa$ می‌باشد

عرض همه چرخدنده‌ها $45\,mm$ و عرض پولی‌ها $35\,mm$ می‌باشد.

همه چرخدنده‌ها را ساده با $\phi = 20^\circ$ در نظر بگیرید.

$$\frac{d_{P_2}}{d_{P_1}} = 2 \;, \quad \frac{d_{G_4}}{d_{G_3}} = 1.5 \;, \quad \frac{d_{G_6}}{d_{G_5}} = 2 \;, \quad \frac{d_{P_4}}{d_{P_3}} = 2$$

محاسبه سرعت شفتها

$$\omega_m = 1440\ rpm \Rightarrow \omega_m = \frac{2\pi \times 1440}{60} = 151\ rad/s$$

$$\frac{dP_2}{dP_1} = 2 \Rightarrow \omega_1 = \frac{\omega_m}{2} = 75.5\ rad/s$$

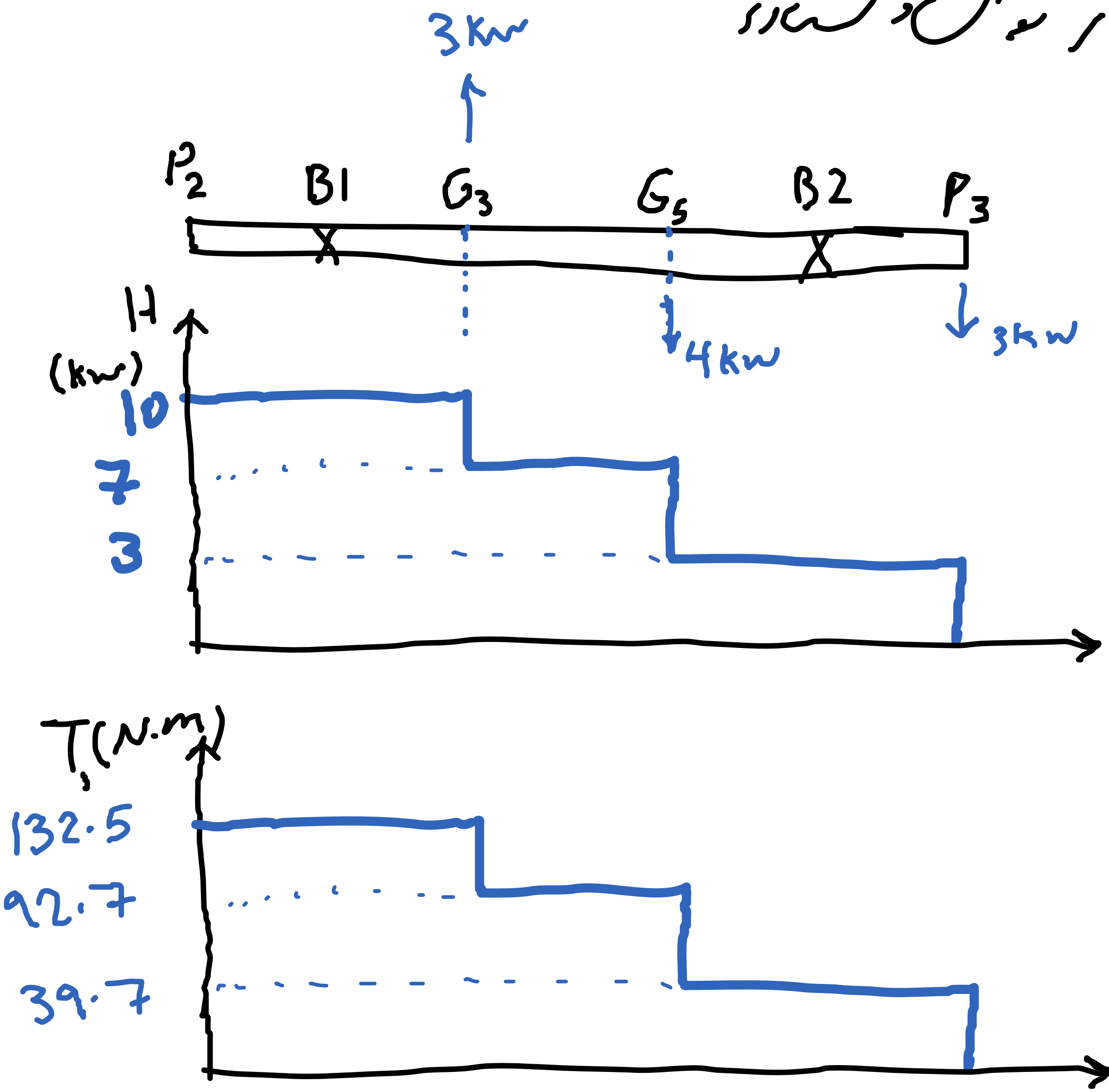
$$H_i = T_i \omega_i \Rightarrow T_1 = \frac{H}{\omega_1} = \frac{10000}{75.5} = 132.5\ N.m$$

گشتاور کل روی شفت 1

نمودار توان و گشتاور

شفت I

$\omega_1 = 75.5\ rad/s$

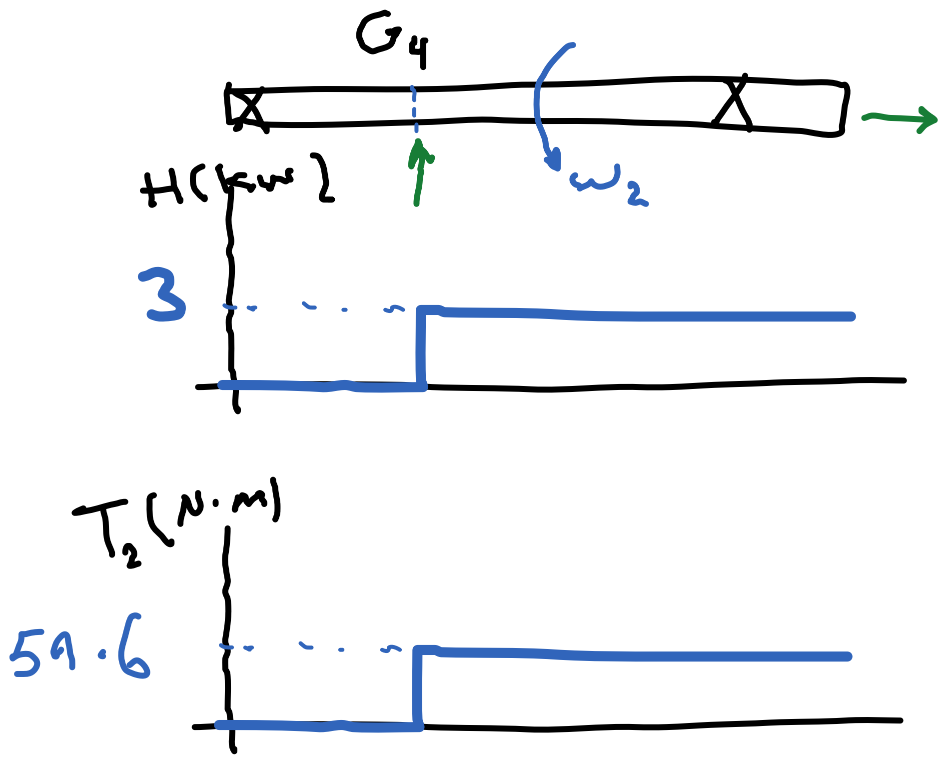


گشتاور پیچشی

$$T = \frac{7000}{75.5} = 92.7$$

$$T = \frac{3000}{75.5} = 39.7$$

شفت II

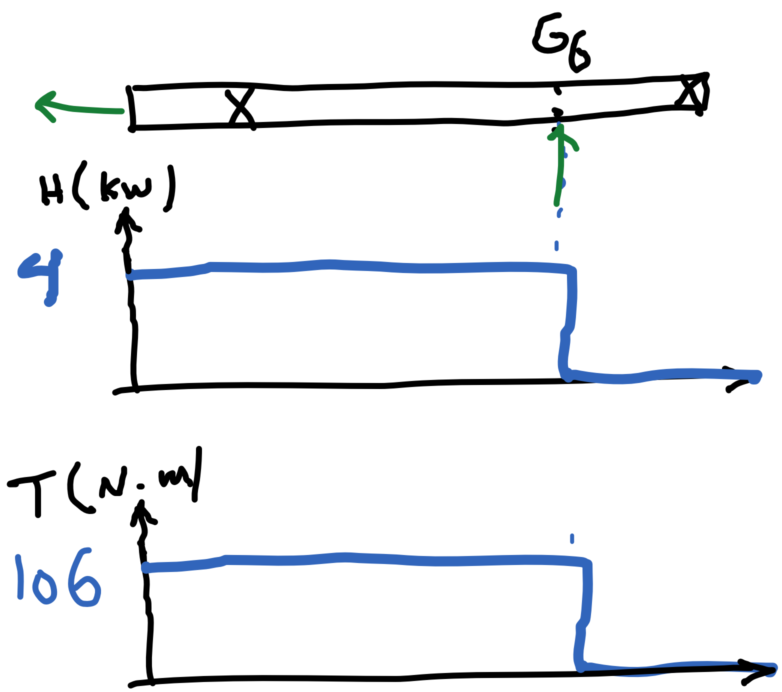


$$\frac{d_4}{d_3} = 1.5 \Rightarrow \omega_2 = \frac{\omega_1}{1.5}$$

$$\Rightarrow \omega_2 = \frac{75.5}{1.5} = 50.3\ rad/s$$

$$T_2 = \frac{3000}{50.3} = 59.6\ N.m$$

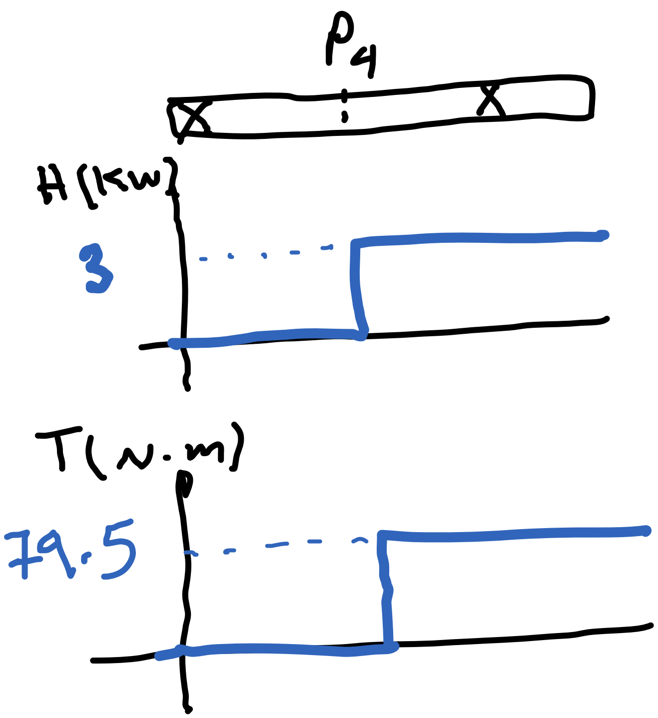
شفت III



$$\frac{d_6}{d_5} = 2 \Rightarrow \omega_3 = \frac{\omega_1}{2} = \frac{75.5}{2} = 37.75$$

$$T_3 = \frac{H}{\omega_3} = \frac{4000}{37.75} = 106 \ N.m$$

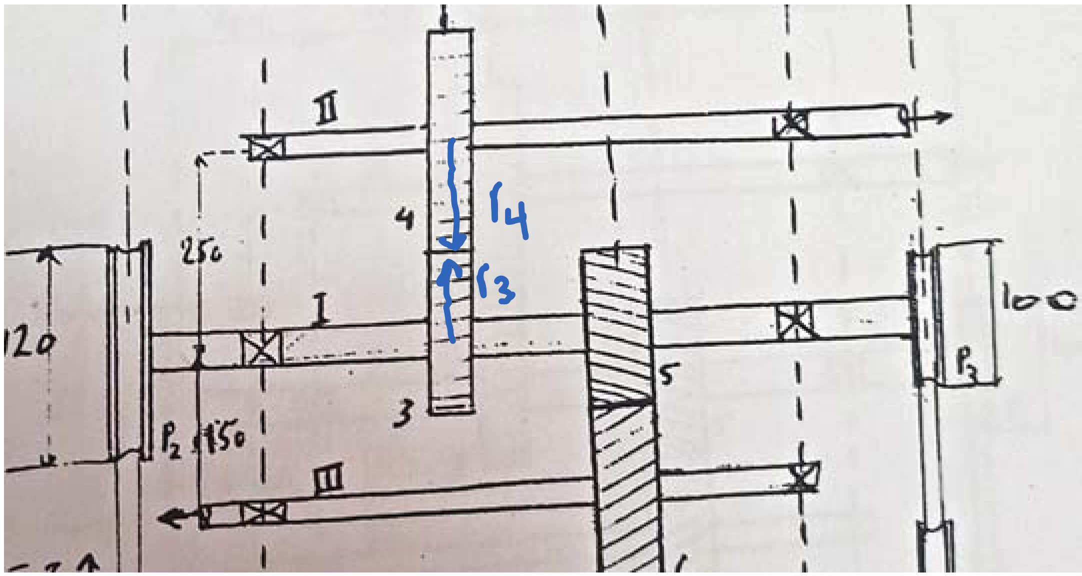
شفت IV



$$\frac{dP_4}{dP_2} = 2 \Rightarrow \omega_4 = \frac{\omega_1}{2} = \frac{75.5}{2} = 37.75$$

$$T_4 = \frac{H}{\omega_4} = \frac{3000}{37.75} = 79.5 \ N.m$$

محاسبه نیروها در پولی‌ها و چرخ‌دنده‌ها



$$F = \frac{T}{r} = \frac{T}{d/2} = \frac{2T}{d}$$

نیرو در پولی P_2

$$F_{P_2} = \frac{2T}{d_{P_2}} = \frac{2 \times 132.5}{120 \times 10^{-3}} = 2208\ N$$

$$F_{tG_3} = \frac{2T}{d_{G_3}} = \frac{2 \times 132.5}{200 \times 10^{-3}} = 1325\ N\ ,\quad F_{rG_3} = F_{tG_3}\, tg 20 = 482\ N$$

$$\begin{cases} \dfrac{d_{G_3} + d_{G_4}}{2} = 250 \Rightarrow 2.5\, d_{G_3} = 500 \Rightarrow d_{G_3} = 200\ mm \\ d_{G_4} = 1.5\, d_{G_3} \end{cases}$$

$$\begin{cases} \dfrac{d_{G5}+d_{G6}}{2} = 150 \Rightarrow 3d_{G5} = 300 \Rightarrow d_{G5} = 100\,mm \\ d_{G6} = 2d_{G5} \end{cases}$$

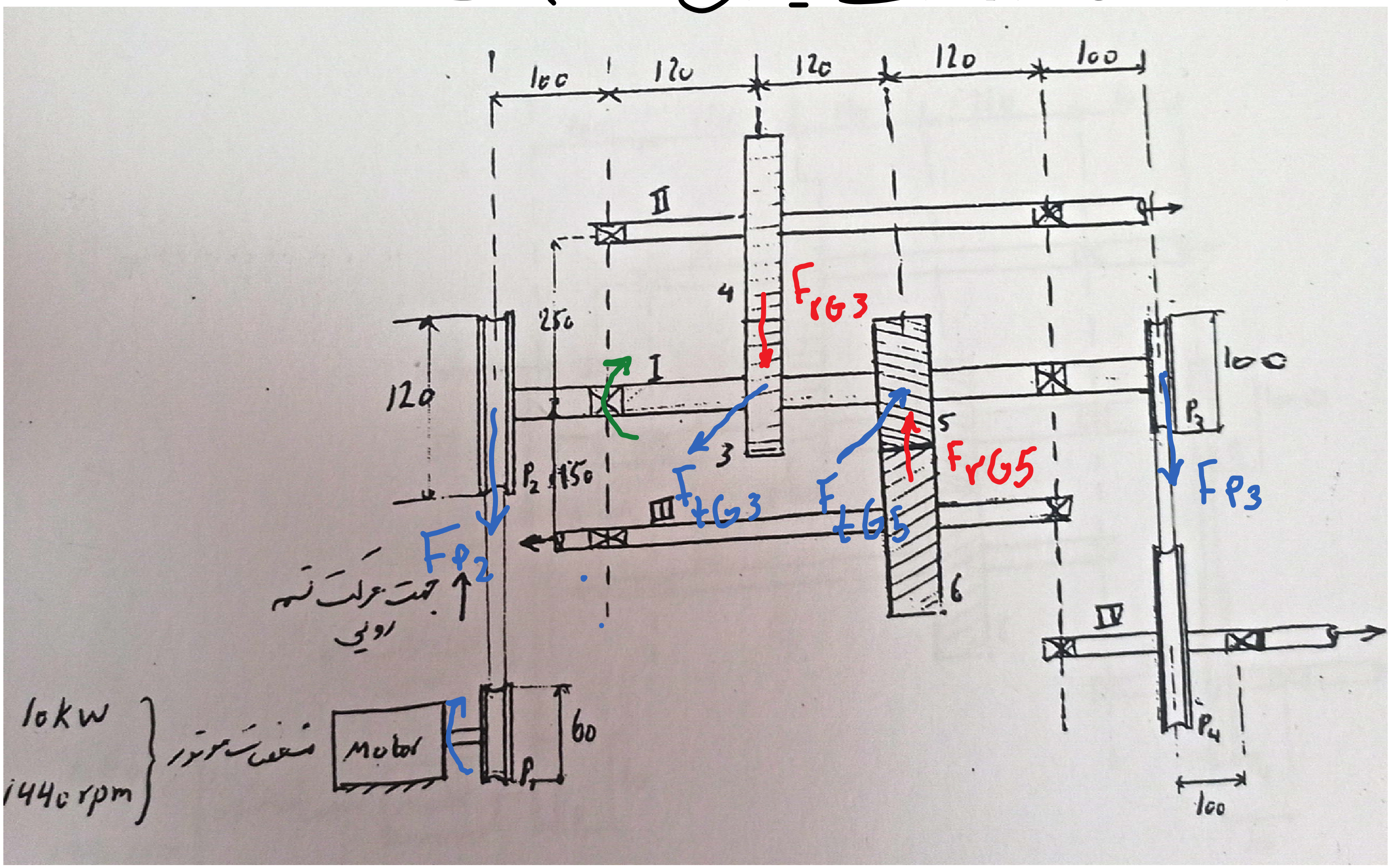
$$F_{tG5} = \frac{2T}{d_{G5}} = \frac{2\times 92.7}{100\times10^{-3}} = 1854\,N$$

$$F_{rG5} = F_{tG5}\,tg20 = 675\,N$$

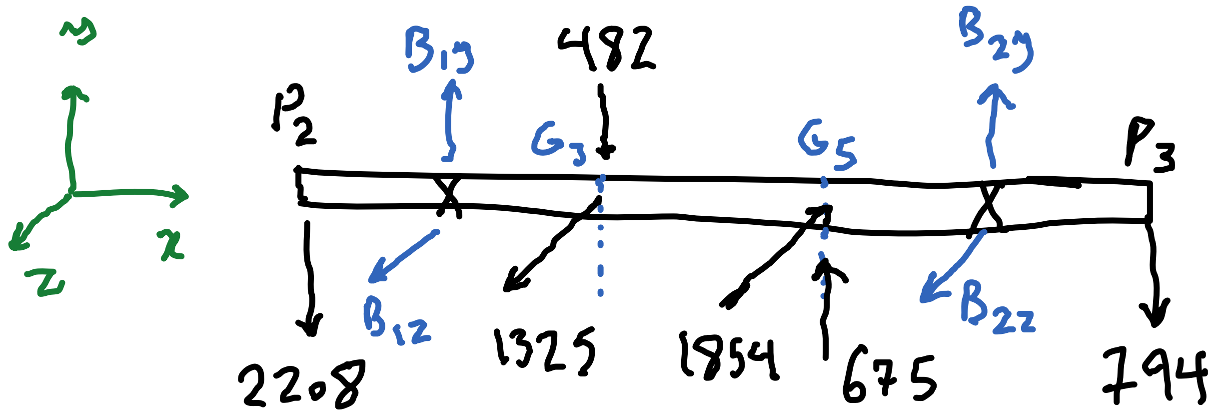
$$F_{P3} = \frac{2T}{d_{P3}} = \frac{2\times 39.7}{100\times10^{-3}} = 794\,N$$

در دنده‌های درگیر

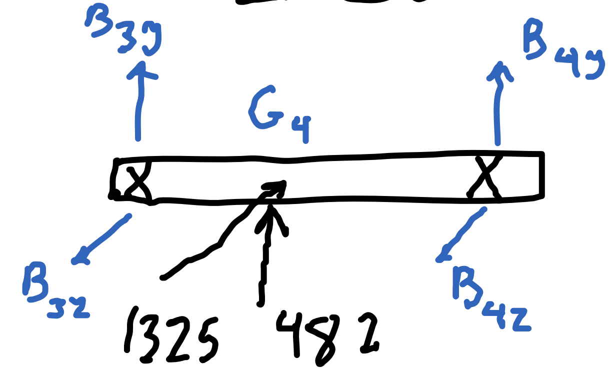
نیرویی که G_3 به G_4 وارد می کند به سمت داخل صفحه است. پس نیرویی که G_4 به G_3 وارد می کند به سمت بیرون صفحه است



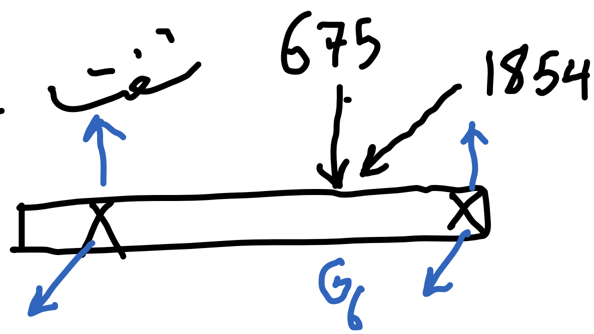
شفت I

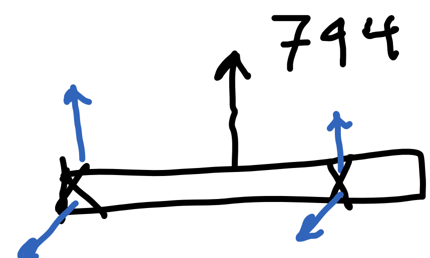


شفت II

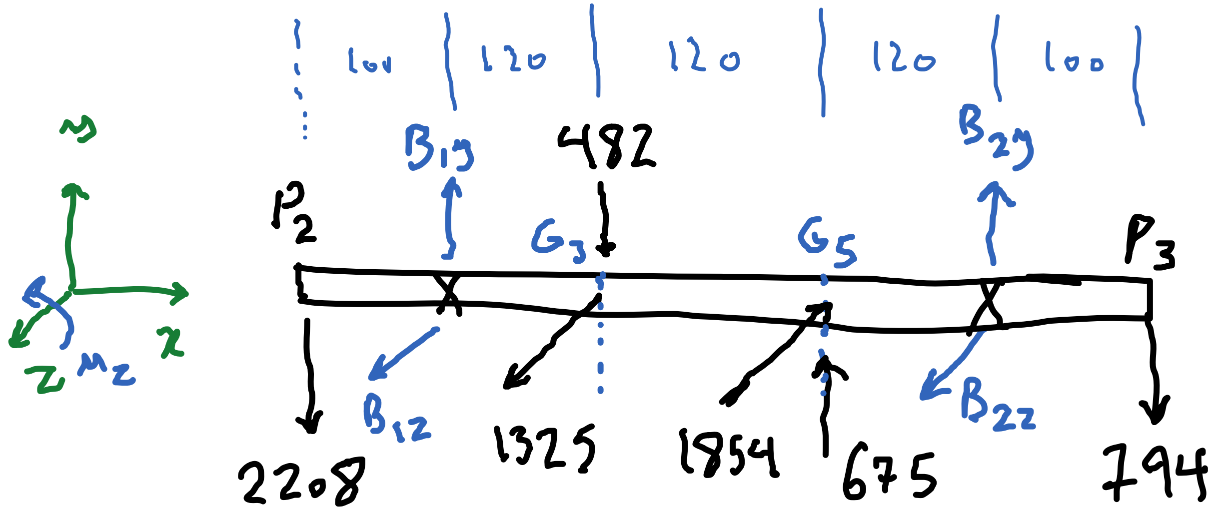


شفت III





شفت IV



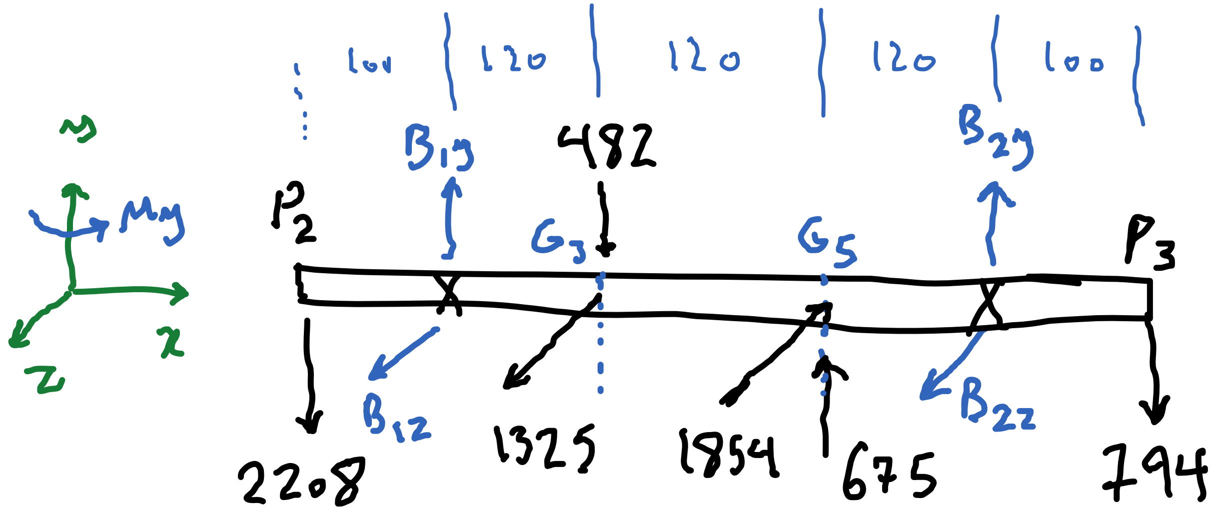
صفحه xy

$$\Sigma F_y = 0 \Rightarrow B_{1y} + B_{2y} - 2208 - 482 + 675 - 794 = 0$$

$$\Sigma M_{B1z} = 2208\times100 - 482\times120 + 675\times240 + B_{2y}\times360 - 794\times460 = 0$$

$$\Rightarrow B_{2y} = 112\ N$$

$$\Sigma F_y = 0 \Rightarrow B_{1y} = 2697\ N$$



صفحه xz

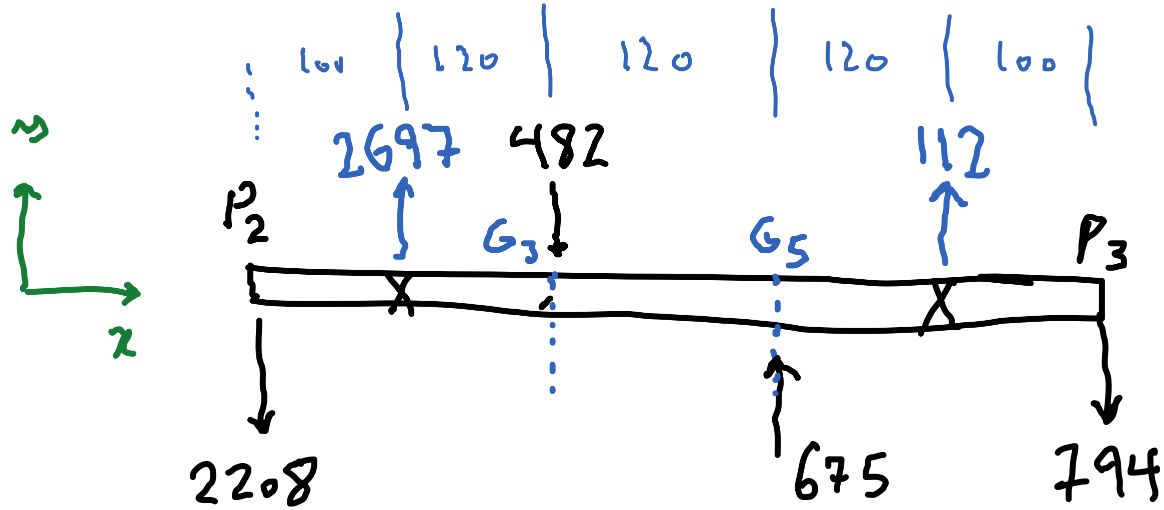
$$\sum F_z = 0 \Rightarrow B_{1z} + 1325 - 1854 + B_{2z} = 0$$

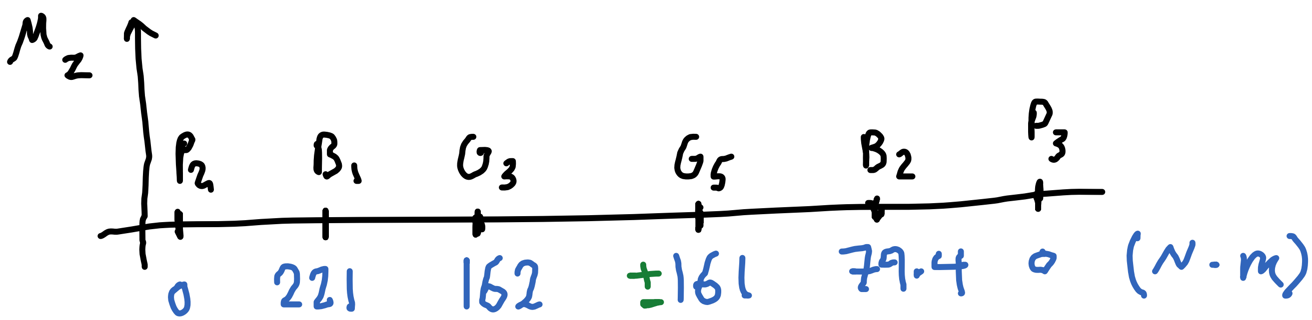
$$\sum M_{B1y} = -1325 \times 120 + 1854 \times 240 - B_{2z} \times 360 = 0$$

$$\Rightarrow B_{2z} = 794 \ N$$

$$\sum F_z = 0 \Rightarrow B_{1z} = -265$$

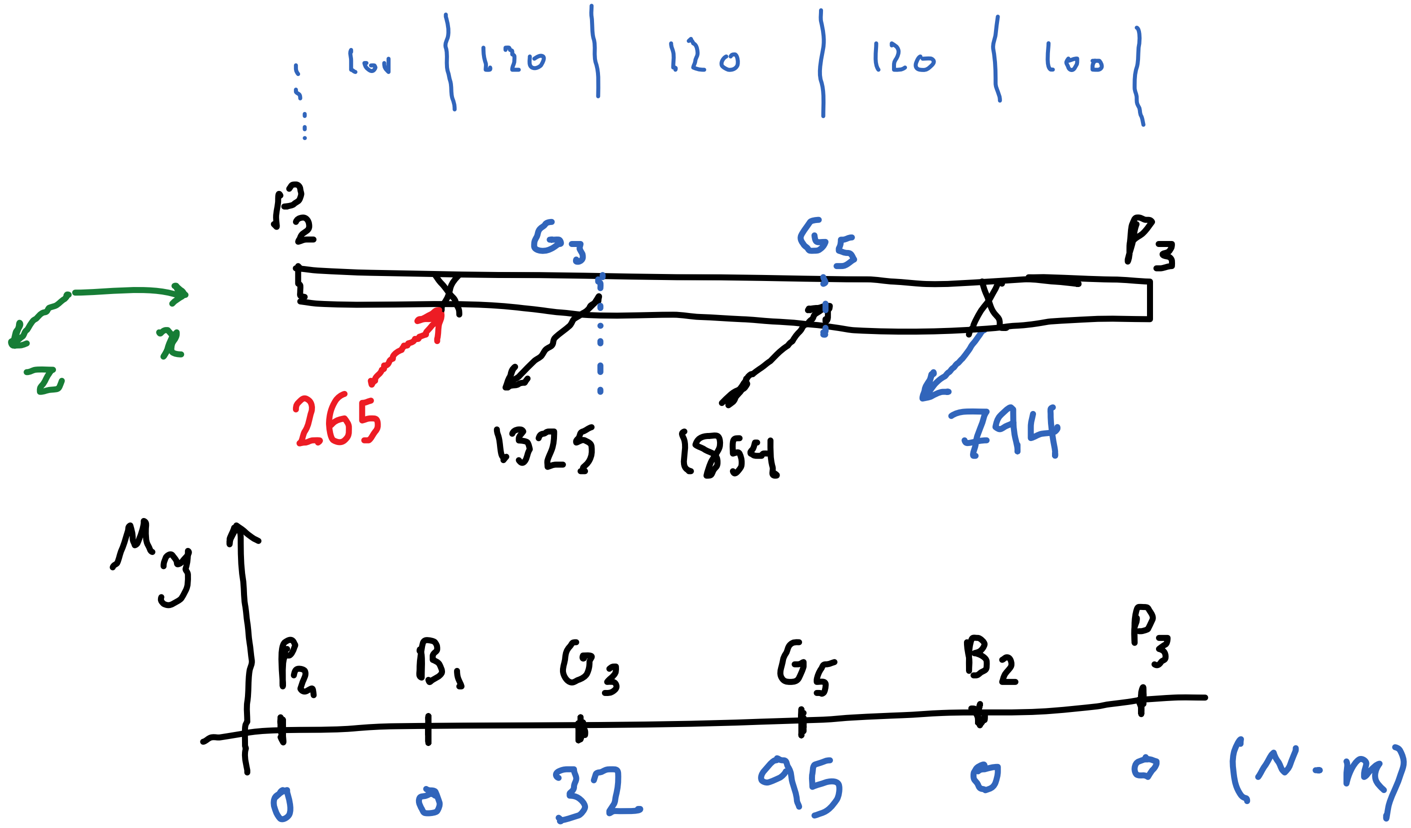
محاسبه گشتاور خمشی M_z



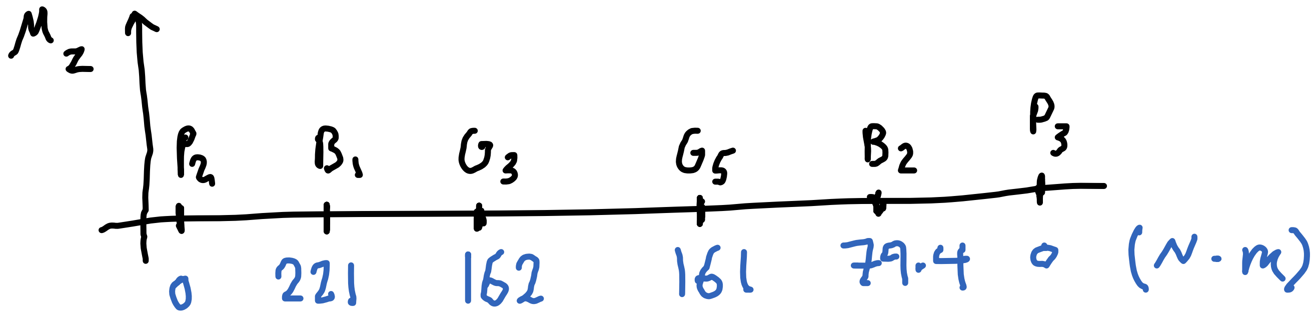


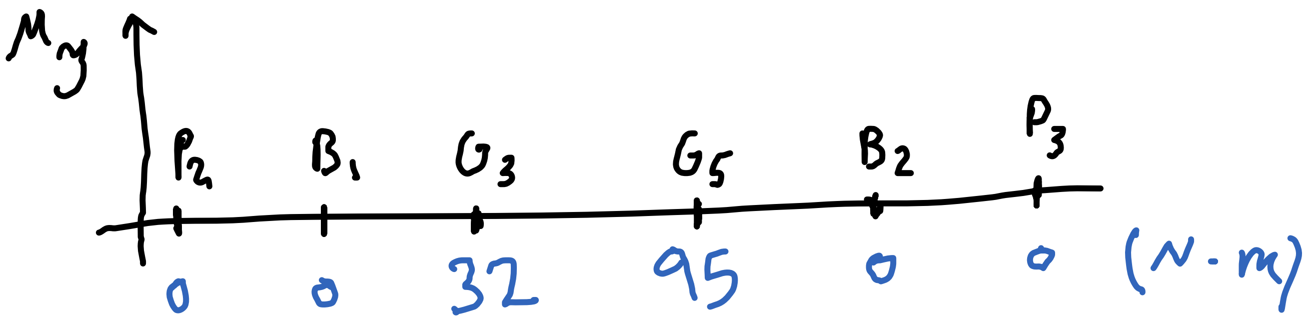
$112 \times 0.12 - 794 \times 0.22$

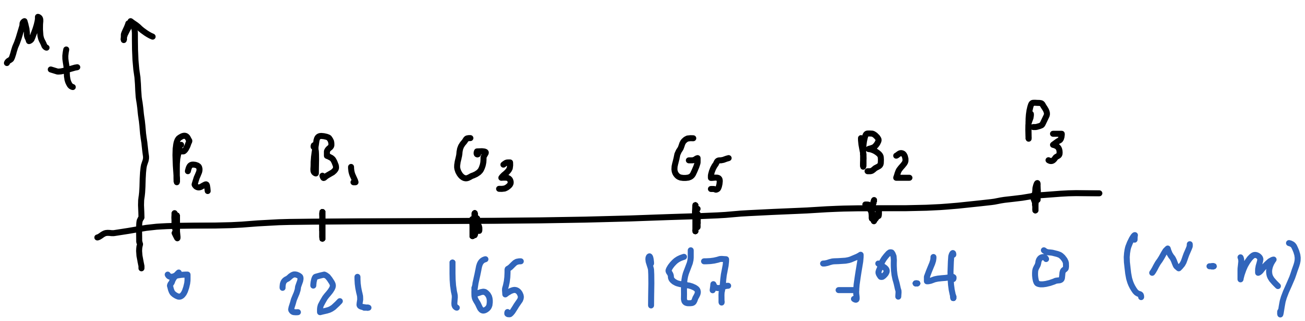
$2208 \times 0.22 - 2697 \times 0.12$


100
120
120
120
100
P2
G3
G5
P3
z
x
265
1325
1854
794
My
P2
B1
G3
G5
B2
P3
0
0
32
95
0
0
(N.m)

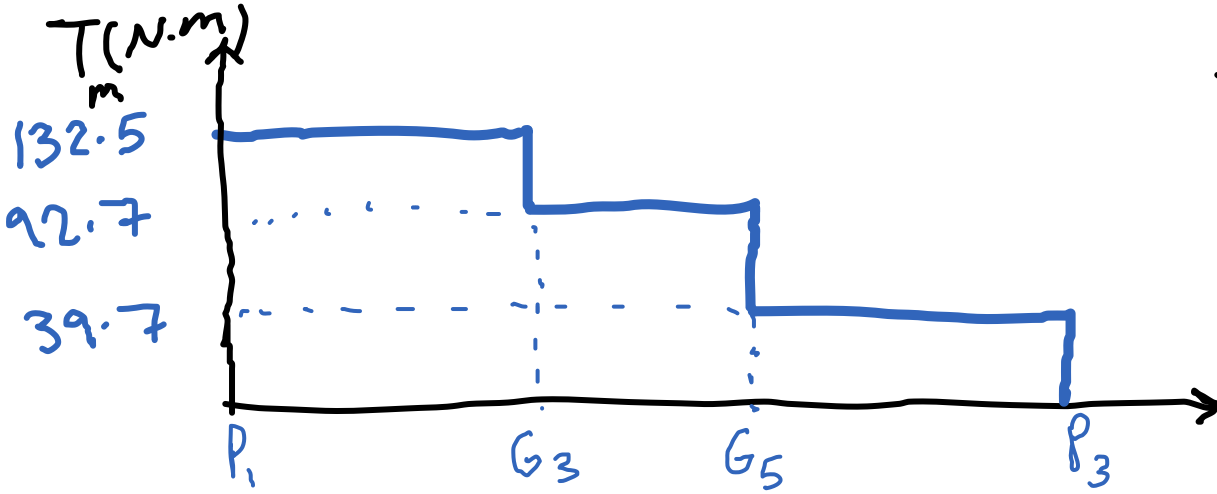
$$M_t = \sqrt{M_z^2 + M_y^2}$$



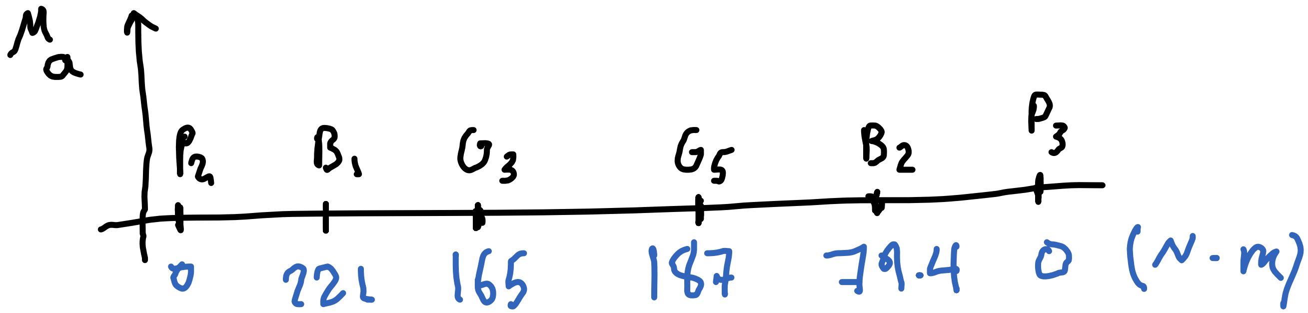




برای سختی I
از
نمودار گشتاور پیچشی



نمودار گشتاور خمشی



$$T_a = 0 \; , \; M_m = 0$$

$$P_2 \begin{cases} T_m = 132.5 \\ M_a = 0 \end{cases}$$

$$B_2 \begin{cases} T_m = 39.7 \\ M_a = 79.4 \end{cases}$$

$$B_1 \begin{cases} T_m = 132.5 \\ M_a = 221 \end{cases}$$

$$P_3 \begin{cases} T_m = 39.7 \\ M_a = 0 \end{cases}$$

$$G_3 \begin{cases} T_m = 132.5 \\ M_a = 165 \end{cases}$$

$$T_a = M_m = 0$$

$$G_5 \begin{cases} T_m = 92.7 \\ M_a = 187 \end{cases}$$

تعیین نقطه بحرانی

در تکیه‌گاه‌ها ضریب تمرکز تنش هم بیشتر است

نقطه بحرانی تکیه‌گاه B_1 می‌باشد

$$B_1 \begin{cases} T_m = 132.5 \\ M_a = 221 \end{cases}$$

$\frac{r}{d} = 0.02$, $K_t = 2.7$, $K_{ts} = 2.2 \Leftarrow$ 7-1 جدول

$$K_f = 1 + q(K_t - 1) = 1 + 0.8 \times 1.7 = 2.3$$

$$K_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.8 \times 1.2 = 1.95$$

For $S_{ut} \simeq 700\, MPa$ $\begin{cases} 0.75 < q < 0.85 \\ 0.78 < q_s < 0.9 \end{cases}$

$$K_a = a S_{ut}^{b} = 3.04(640)^{-0.217} = 0.75$$ سطح ماشین کاری

فرض $K_b = 0.8$

$$S_e = K_a K_b (0.5 S_{ut}) = 0.75 \times 0.8 \times 0.5 \times 640 = 192 \text{ MPa}$$

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

$T_a = M_m = 0$

$\sqrt{3} = 1.73$

$$d = \left\{ \frac{16n}{\pi} \left[\frac{2 K_f M_a}{S_e} + \frac{1.73 K_{fs} T_m}{S_{ut}} \right] \right\}^{1/3}$$

$$d = \left\{ \frac{16n}{\pi} \left[\frac{2 K_f M_a}{S_e} + \frac{1.73 K_{fs} T_m}{S_{ut}} \right] \right\}^{1/3}$$

$$d_{B1} = \left\{ \frac{16 \times 3}{3.14} \left[\frac{2 \times 2.3 \times \overset{N.mm}{221000}}{192} + \frac{1.73 \times 1.95 \times \overset{N.mm}{132500}}{640} \right] \right\}^{1/3}$$

$$d_{B1} = 45\ mm$$

$$B_2 \begin{cases} T_m = 39.7 \\ M_a = 79.4 \end{cases}$$

$$k_f = 2.3 ,\ k_{fs} = 1.95 \quad \Rightarrow d_{B_2} = 31.8\,mm$$

$$\frac{r}{d} = 0.1 ,\ K_t = 1.7 ,\ K_{ts} = 1.5 \leftarrow$$ جدول 7-1

$$q = 0.85 \rightarrow k_f = 1 + q(K_t - 1) = 1 + 0.85 \times 0.7 = 1.6$$

$$k_{fs} = 1 + q_s(K_{ts} - 1) = 1 + 0.87 \times 0.5 = 1.4$$

برای پله هایی که چرخدنده و پولی قرار می گیرد

$K_f = 1.6 \; , \; K_{fs} = 1.4$

این قطرها ، قطرهای کمینه هستند.
اگر مقدار حداکثر نهایی از آنها بیشتر شد ایرادی ندارد

$$P_2 \begin{cases} T_m = 132.5 \\ M_a = 0 \end{cases} \Rightarrow d_{P_1} = 19 \text{ mm}$$

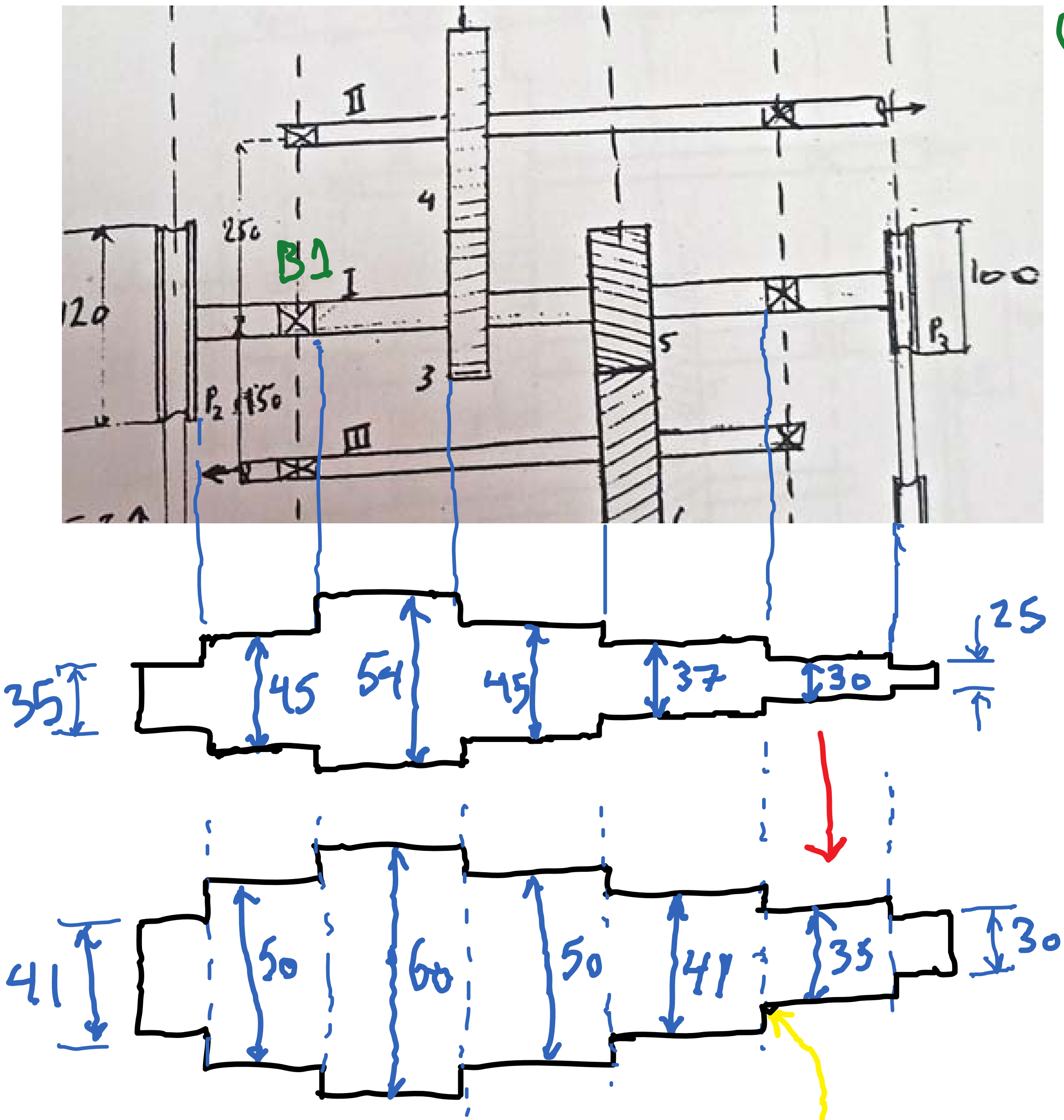
$$G_3 \begin{cases} T_m = 132.5 \\ M_a = 165 \end{cases} \Rightarrow d_{G_3} = 36.7 \text{ mm}$$

$$G_5 \begin{cases} T_m = 92.7 \\ M_a = 187 \end{cases} \Rightarrow d_{G_5} = 37.5 \text{ mm}$$

$$P_3 \begin{cases} T_m = 39.7 \\ M_a = 0 \end{cases} \Rightarrow d_{P_3} = 13 \text{ mm}$$

از B_1 شروع می‌کنیم

$d_{B_1} = 45$



$$\frac{r}{d} = 0.02 \rightarrow r = 0.02 \times 35 = 0.7$$

$45 \times 1.2 = 54$

$\frac{45}{1.2} = 37.5$

$\frac{37}{1.2} = 30.8$

$\frac{30}{1.2} = 25$

$35/1.2 = 29$

$35 \times 1.2 = 42$

$41 \times 1.2 = 49$

Table 11–2 **Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings**

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm	
				d_S	d_H
10	30	9	0.6	12.5	27
12	32	10	0.6	14.5	28
15	35	11	0.6	17.5	31
17	40	12	0.6	19.5	34
20	47	14	1.0	25	41
25	52	15	1.0	30	47
30	62	16	1.0	35	55
35	72	17	1.0	41	65
40	80	18	1.0	46	72
45	85	19	1.0	52	77
50	90	20	1.0	56	82
55	100	21	1.5	63	90
60	110	22	1.5	70	99
65	120	23	1.5	74	109
70	125	24	1.5	79	114
75	130	25	1.5	86	119
80	140	26	2.0	93	127
85	150	28	2.0	99	136
90	160	30	2.0	104	146
95	170	32	2.0	110	156

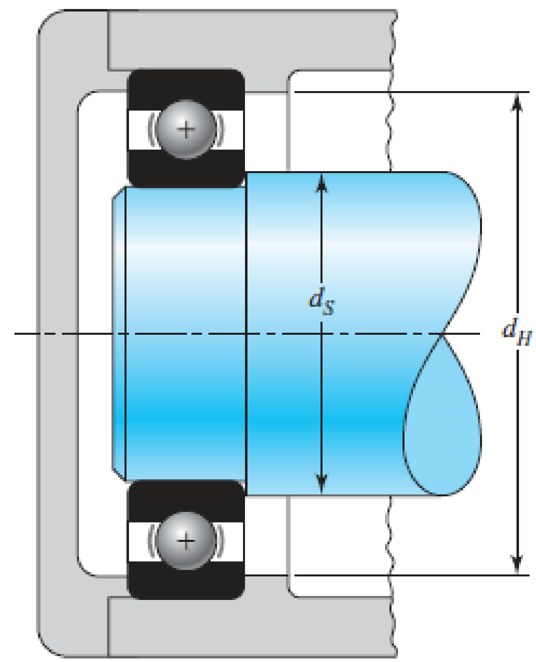


Figure 11–8

Shaft and housing shoulder diameters d_S and d_H should be adequate to ensure good bearing support.

طبق اندازه های نهایی محور باید ضریب ایمنی در نقاط مختلف چک شود به ویژه در تکیه گاهها و نقاطی که جای خار داریم. و خار فنری Retaining ring

K_t و K_{ts}... ← خار خزانگی ← جدول 7-1

طراحی خار برای پولی P_2

$S_y = 370\ \text{MPa}$

$n = 2.5$

$d_{P_2} = 41\ \text{mm}$

$T_m = 132.5\ \text{N.m}$

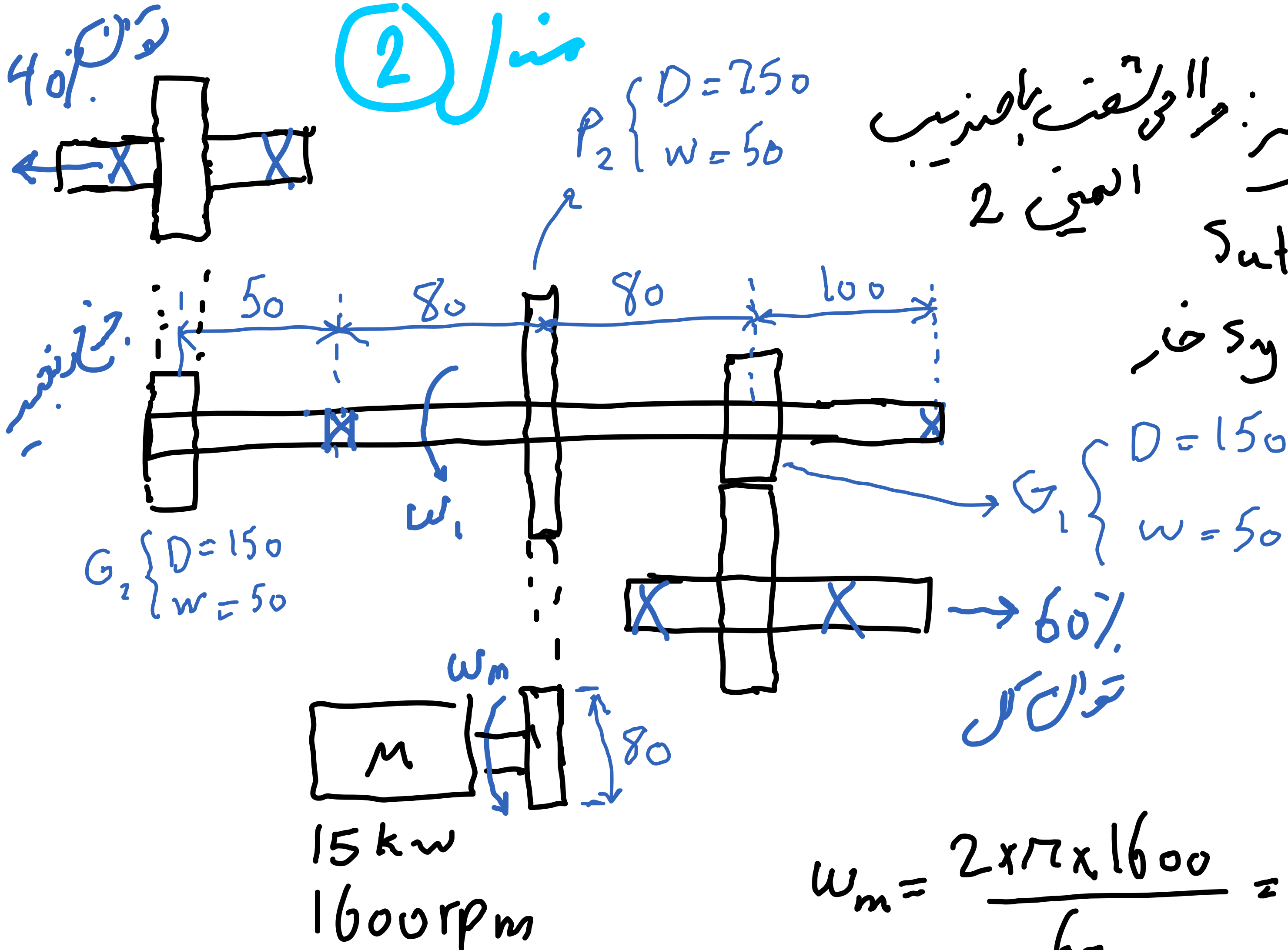
$$L = \frac{2nF}{wS_y} = \frac{2nF}{wS_y} \times \frac{d/2}{d/2} = \frac{4nT}{wdS_y}$$

$d = 41 \Rightarrow w = 12\ \text{mm}$

طول استاندارد

$$L = \frac{4 \times 2.5 \times 132.5}{12 \times 41 \times 370} = 0.007\ \text{m} = 7\ \text{mm} \Rightarrow L = 10\ \text{mm}$$

مثال 2



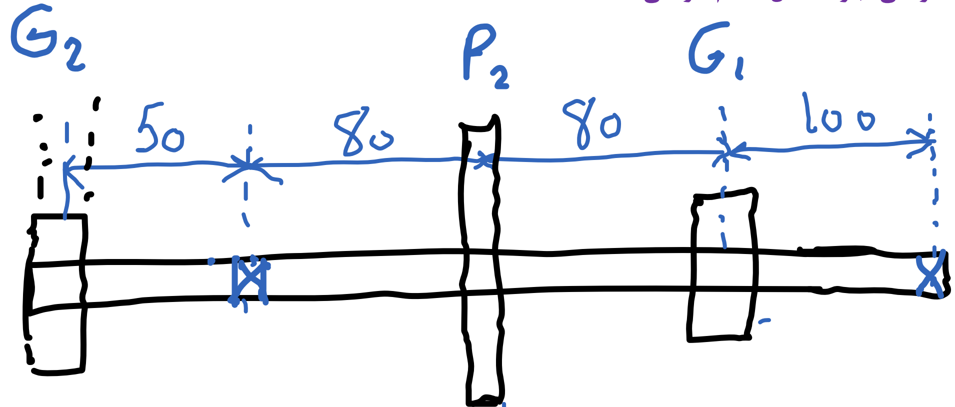
مثال دیگر: طراحی شفت با جنس
المینی 2

$S_{ut} = 640\ MPa$

خار $S_y = 370$

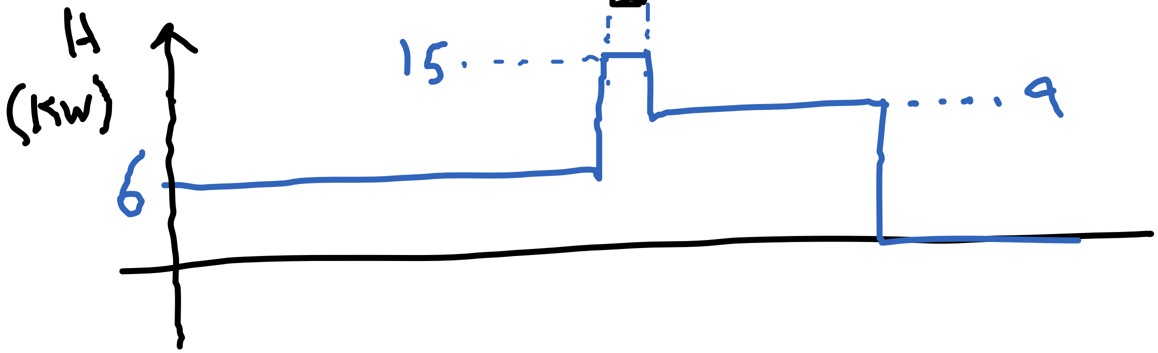
$$\omega_m = \frac{2\times\pi\times 1600}{60} = 167.5\ rad/s$$

$$\frac{\omega_m}{\omega_1} = \frac{250}{80} = 3.125 \Rightarrow \omega_1 = 53.6\ rad/s$$



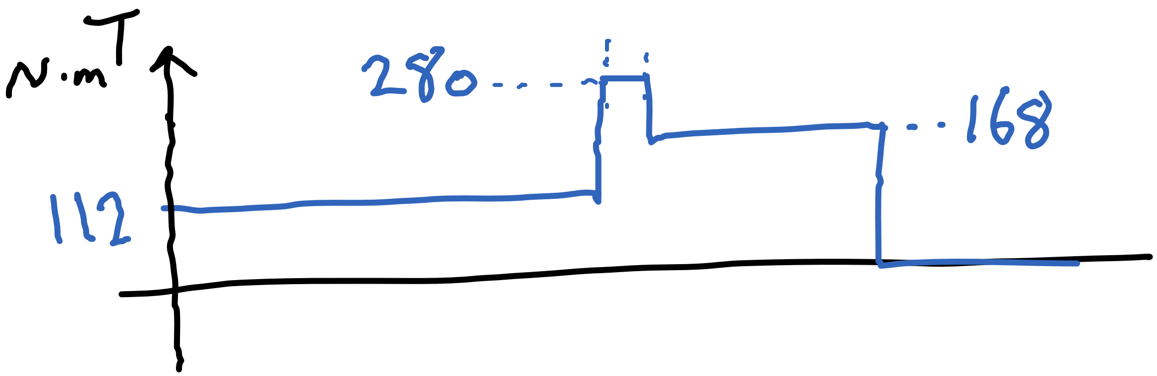
$$0.6 \times 15 = 9$$

$$0.4 \times 15 = 6$$



$$T = \frac{H}{\omega} = \frac{15000}{53.6}$$

$$T = 280 \ N.m$$



$$T = \frac{9000}{53.6} = 168$$

$$T = \frac{6000}{53.6} = 112$$

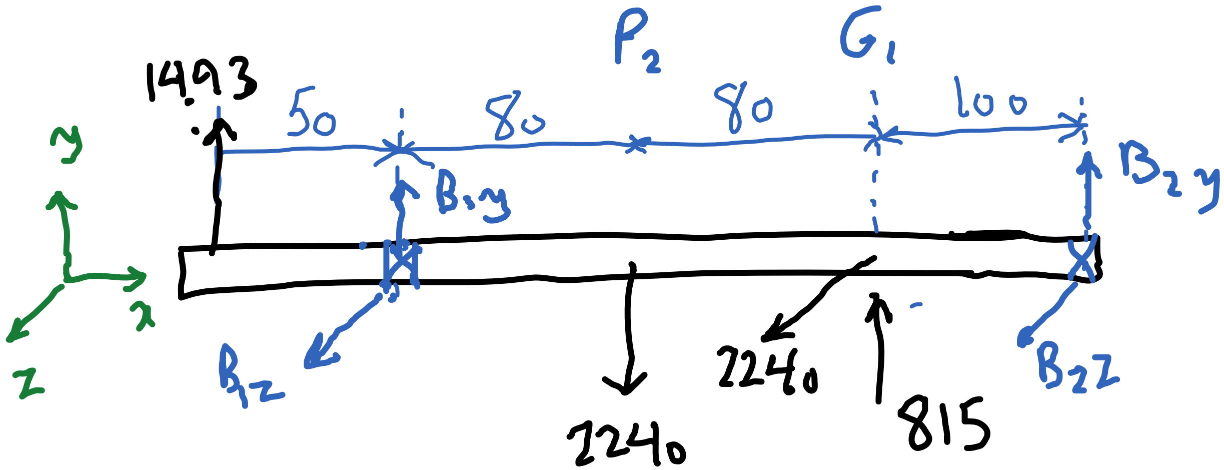
$$F_{p_2} = \frac{2T}{d_{p_2}} = \frac{2 \times 280}{250 \times 10^{-3}} = 2240\ N$$

$$F_{G_2} = \frac{2T}{d_{G_2}} = \frac{2 \times 112}{150 \times 10^{-3}} = 1493\ N$$

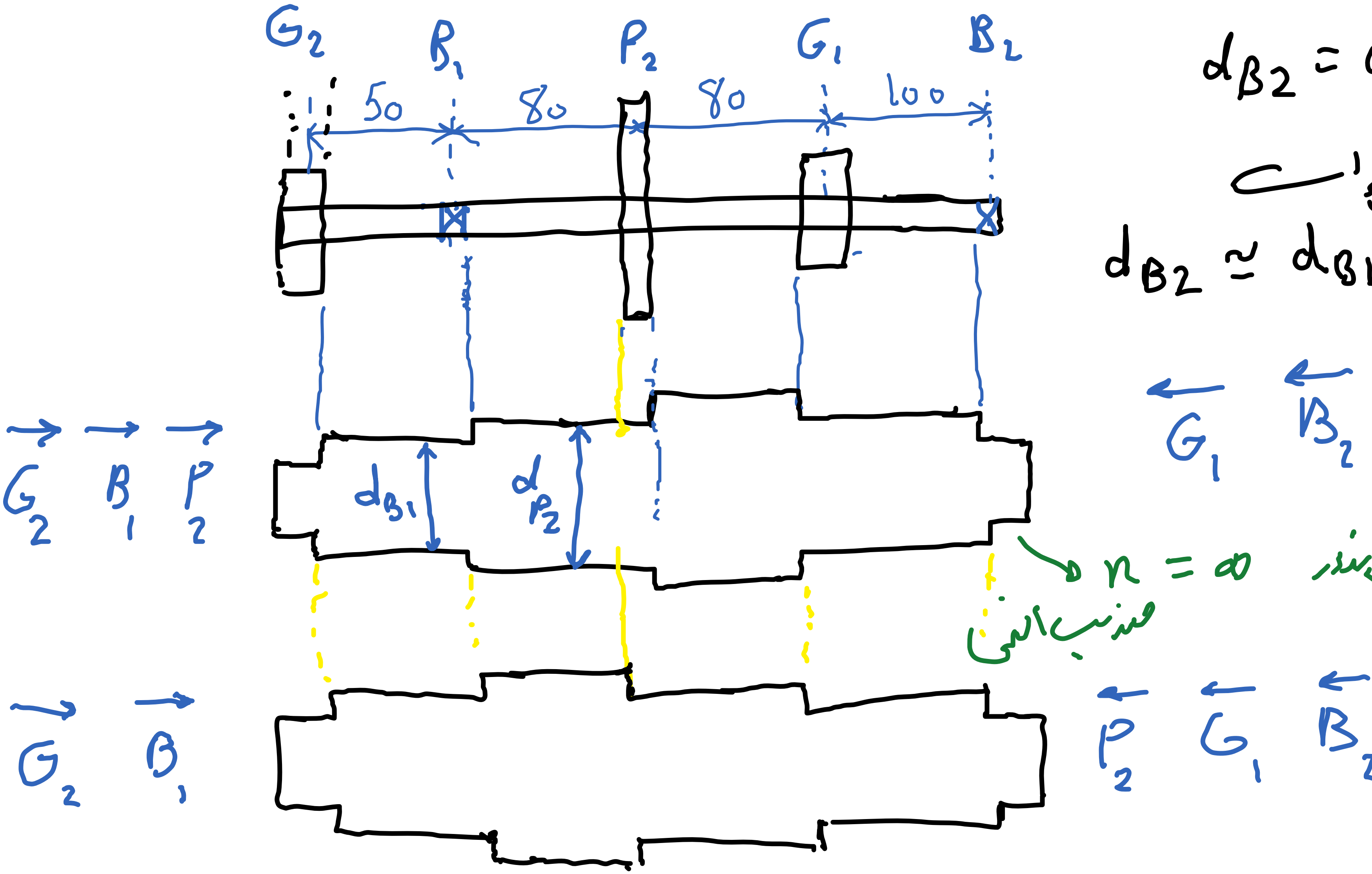
چرخ زنجیر شبیه پولی

$$F_{tG_1} = \frac{2T}{d_{G_1}} = \frac{2 \times 168}{150 \times 10^{-3}} = 2240\ N$$

$$F_{rG_1} = F_{tG_1}\, tg 20 = 815\ N$$



ادامه مسئله را خودتان حل کنید...



$d_{B2} = 0$?

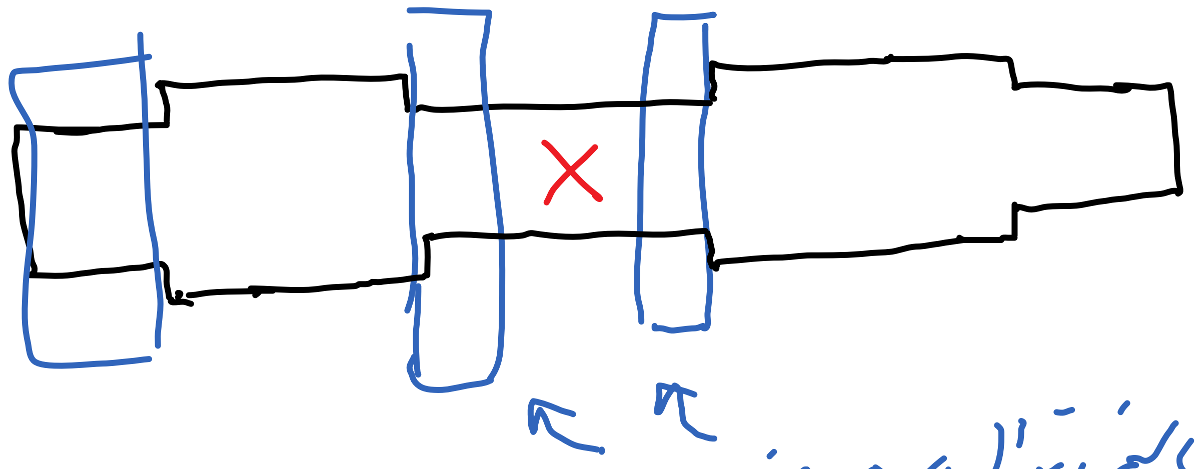
مهم است

$d_{B2} \simeq d_{B1}$

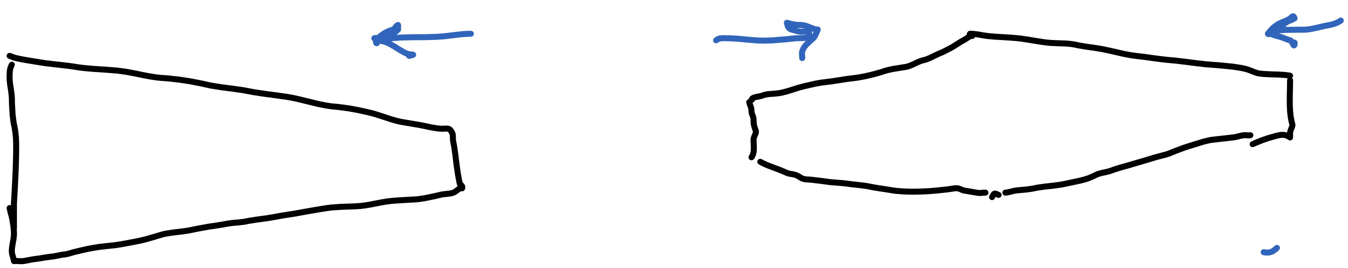
$n = \infty$ ایرادی ندارد

ضریب اطمینان

چنین طراحی کاملاً اشتباه است

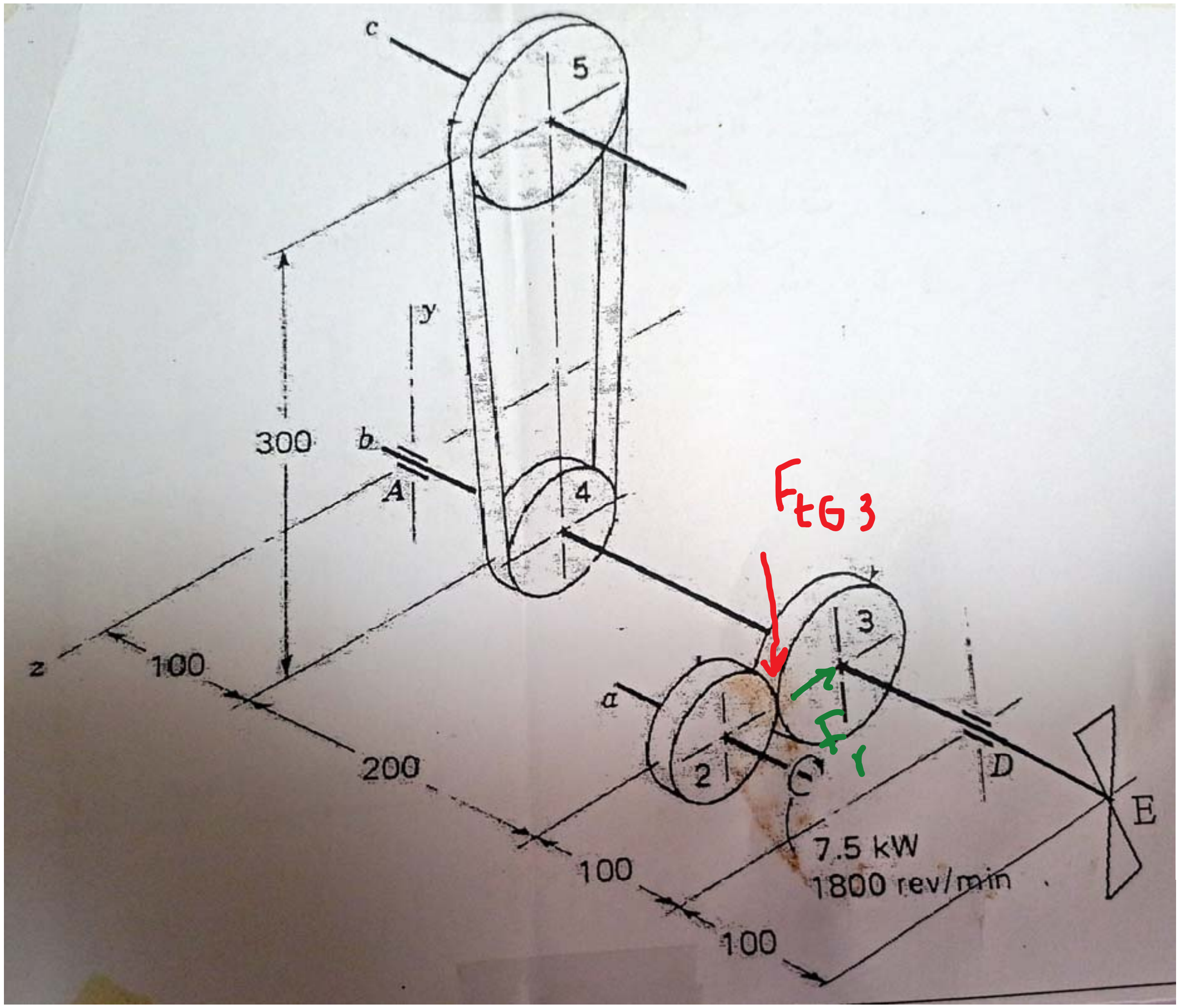


چون نمی‌توان چرخدنده‌ها را جا زد



شکل کلی شفت باید پله‌ای به بالا باشد.

مثال 3



توان موتور از طریق چرخدنده 2, 3 به شفت اصلی منتقل می‌شود. 70% کل توان از طریق چرخ تسمه 5 به مصرف می‌رسد و 30% دیگر صرف چرخاندن پره E می‌شود.

$d_{G2} = 50mm$, $d_{G3} = 100$, $d_{P4} = 80$, $\phi = 20^\circ$

$S_{ut} = 640\, MPa$, $S_y = 370$

با ضریب اطمینان 2.5 شفت را طراحی کنید

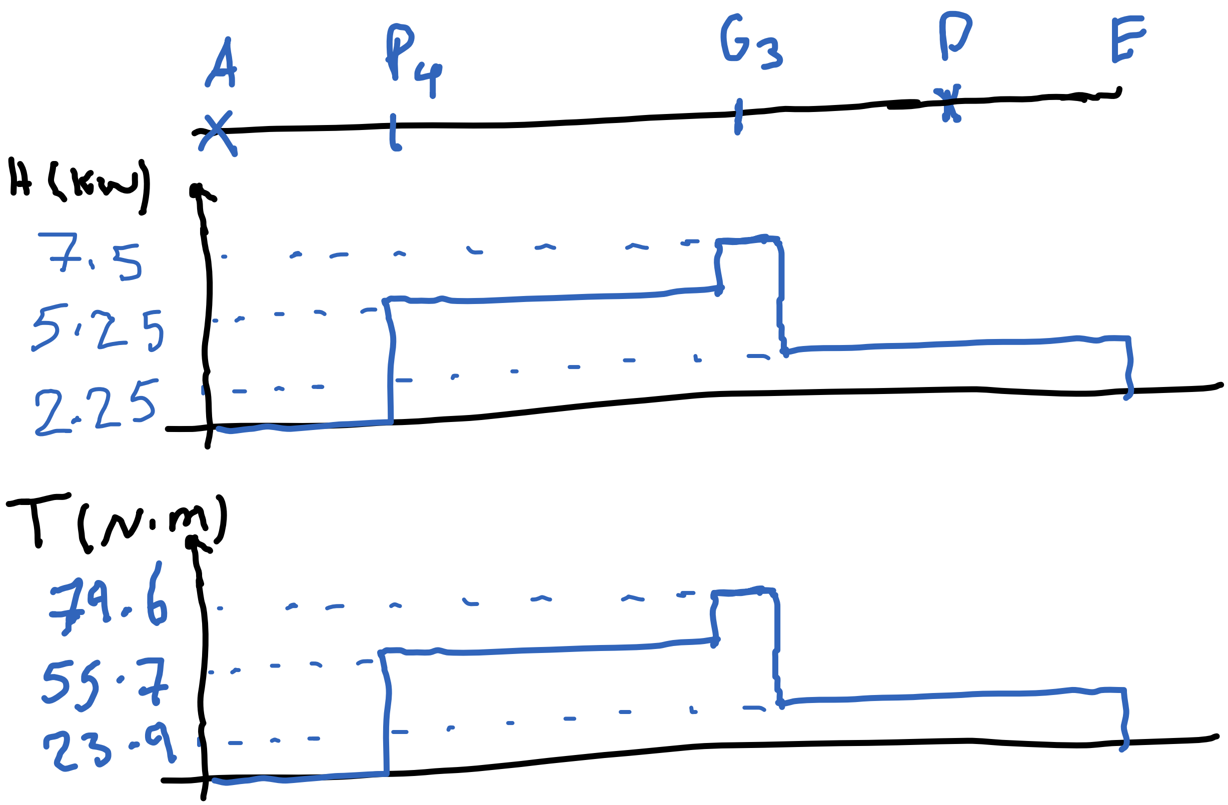
خارها را با ضریب 2 طراحی کنید

$$\omega_m = 1800\ rpm \Rightarrow \omega_1 = \frac{1800}{2} = 900\ rpm$$

$$\frac{d_{G3}}{d_{G2}} = 2$$

$$\omega_1 = \frac{2\pi \times 900}{60} = 44.2\ rad/s$$

$$T = \frac{H}{\omega_1} = \frac{7500}{44.2} = 79.6\ N.m$$



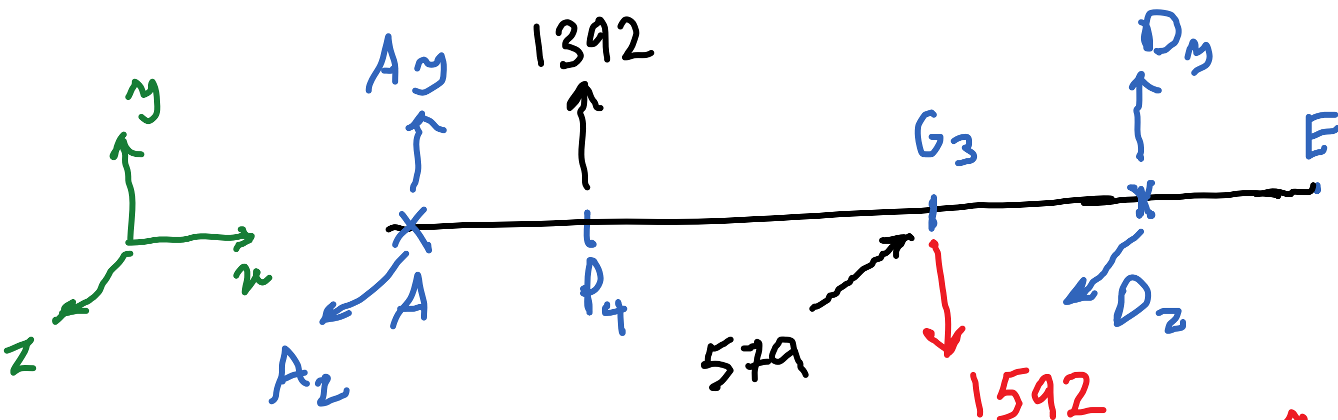
$$0.7 \times 7.5 = 5.25$$

$$0.3 \times 7.5 = 2.25$$

$$F_{G3t} = \frac{2T}{d_{G_3}} = \frac{2 \times 79.6}{100 \times 10^{-3}} = 1592 \text{ N}$$

$$F_{G3r} = F_{G3t} \, tg \, 20 = 579 \text{ N}$$

$$F_{P4} = \frac{2T}{d_{P_4}} = \frac{2 \times 55.7}{80 \times 10^{-3}} = 1392$$



دقت کنید M_a در A ، D ، E صفر است

دقت کنید F_{tG3} در راستای y است

نه Z

ادامه مسئله را خودتان حل کنید . .

تکلیف

7-1(a)

7-22 شفت a و b را طراحی کنید

7-23(a)