**Chapter 16**

**16-1** Given: *r* = 300/2 = 150 mm, *a* = *R* = 125 mm, *b* = 40 mm,*f* = 0.28,*F* = 2.2 kN, *θ*1 = 0°, *θ*2 = 120°, and *θa* = 90°. From which, sin*θa* = sin90° = 1.

Eq. (1



Eq. (16-3): 

*c* = 2(0.125 cos 30°) = 0.2165 m

Eq. (16-4): 

*pa*= *F/*[2.995(10−3)] = 2200*/*[2.995(10−3)]

= 734.5(103) Pafor cw rotation

Eq. (16-7): 

*pa*= 381.9(103) Pafor ccw rotation

A maximum pressure of 734.5 kPaoccurs on the RH shoe for cw rotation. *Ans.*

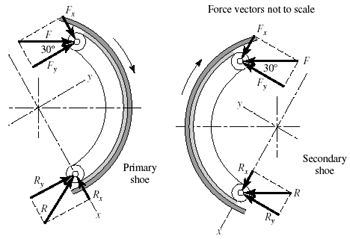
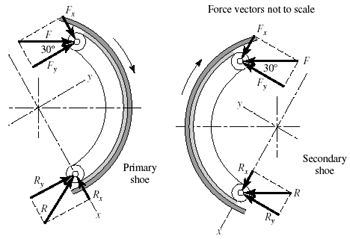
**(b)** *RH shoe*:

Eq. (16-6): 

*LH shoe*:



*T*total = 277.6 + 144.4 = 422N · m*Ans.*

** (c)**

*RH shoe*: *Fx*= 2200 sin 30° = 1100 N, *Fy*= 2200 cos 30° = 1905 N

Eqs. (16-8): 

Eqs. (16-9):  

*LH shoe*: *Fx*= 1100 N, *Fy*= 1905 N

Eqs. (16-10): 



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**16-2** Given: *r* = 300/2 = 150 mm, *a* = *R* = 125 mm, *b* = 40 mm, *f* = 0.28, *F* = 2.2 kN, *θ*1 = 15°,

*θ*2 = 105°, and *θa* = 90°. From which, sin*θa* = sin90° = 1.

Eq. (16-2): 

Eq. (16-3): 

*c* = 2(0.125) cos 30° = 0.2165 m

Eq. (16-4): 

*RH shoe*: *pa*= 2200/ [2.581(10− 3)] = 852.4 (103)Pa

= 852.4 kPa on RH shoe for cw rotation *Ans.*

Eq. (16-6): 

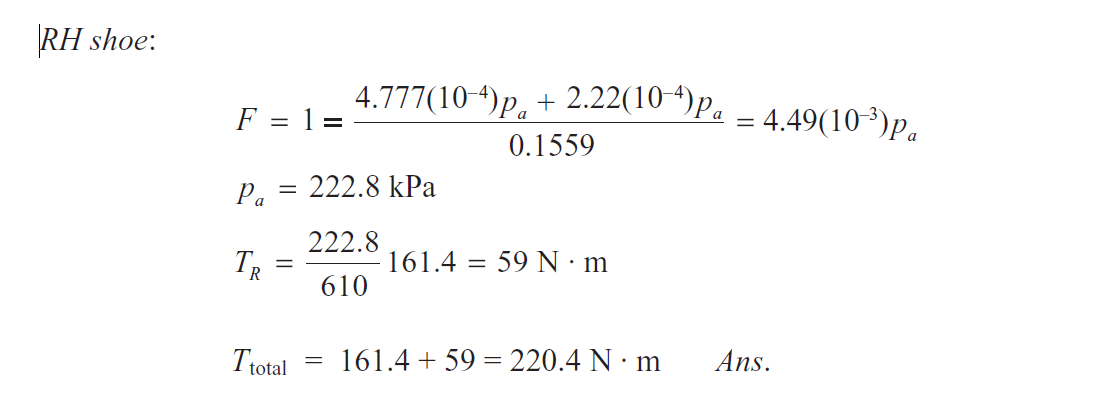
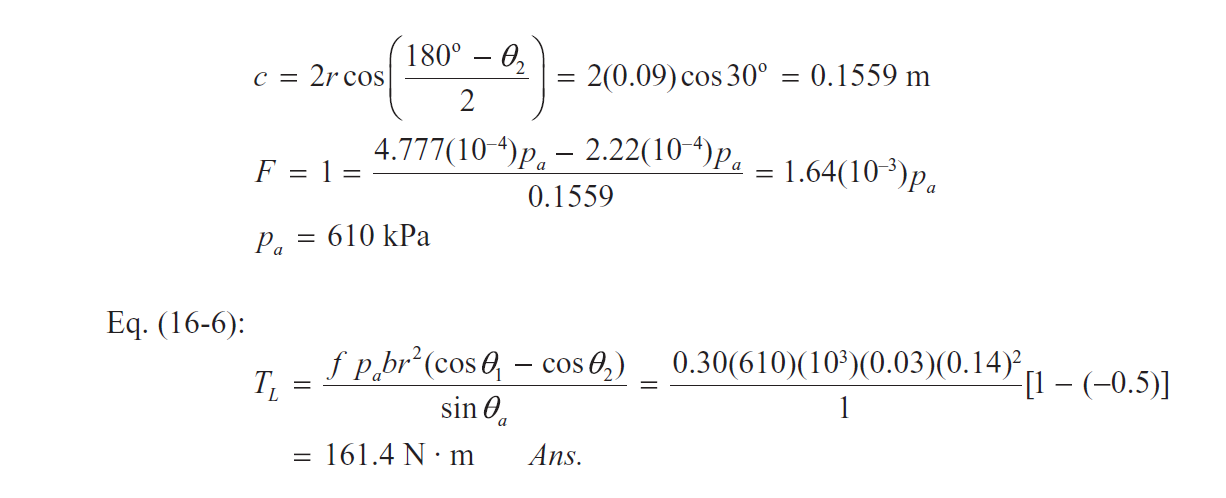
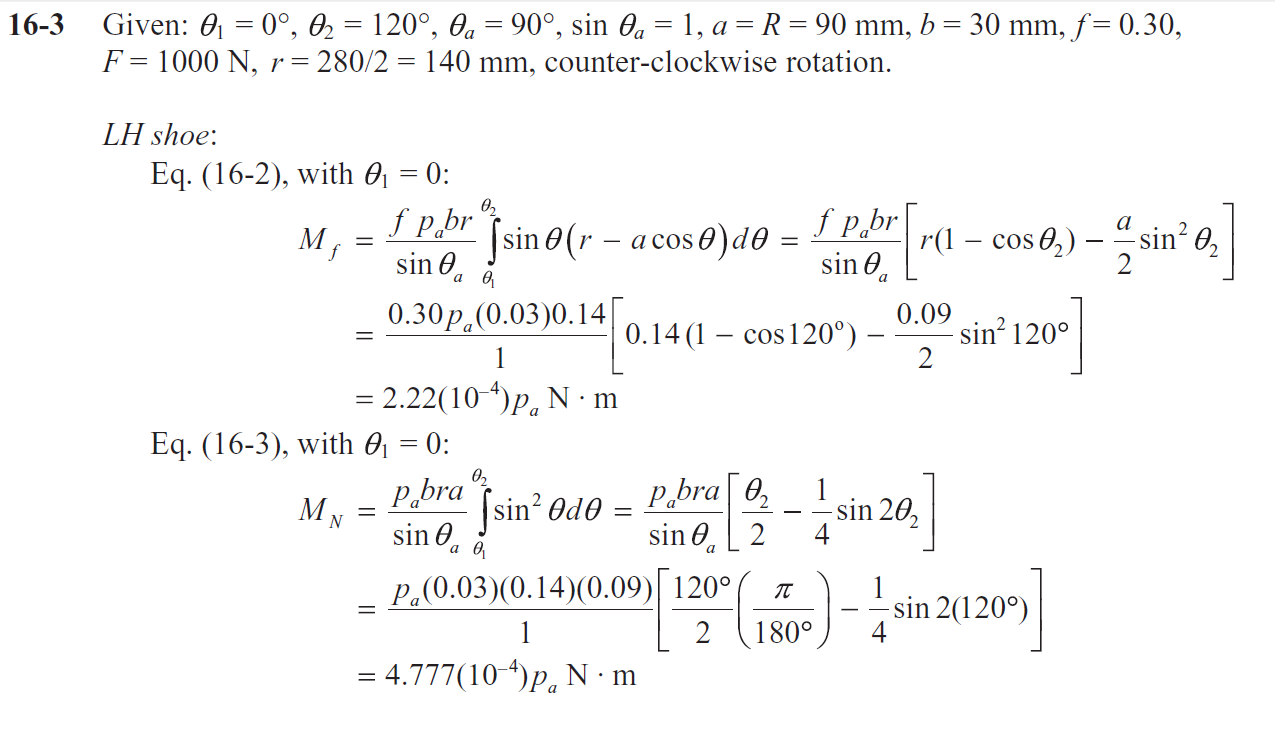
*LH shoe*:





Comparing this result with that of Prob. 16-1, a 2.6% reduction in torque is obtained byusing 25% less braking material.

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**16-4** **(a)** Given: *θ*1 = 10°, *θ*2 = 75°, *θa*= 75°, *pa*= 106 Pa, *f* = 0*.*24,*b* = 0*.*075m (shoe width), *a* = 0*.*150 m, *r* = 0*.*200 m, *d* = 0*.*050 m, *c* = 0*.*165 m*.*

Some of the terms needed are evaluated here:



Now converting to Pascals and meters, we have from Eq. (16-2),



From Eq. (16-3),



Finally, using Eq. (16-4), we have



**(b)** Use Eq. (16-6) for the primary shoe.



For the secondary shoe, we must first find *pa.*Substituting



Then

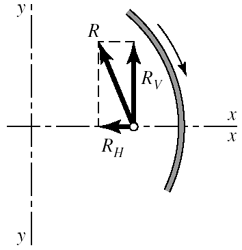


so the braking capacity is *T*total = 2(541) + 2(335) = 1750 N · m *Ans.*

**(c)** *Primary shoes*:



*Secondary shoes*:



Note from figure that +*y* for secondary shoe is opposite to

+*y* for primary shoe.

Combining horizontal and vertical components, 

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**16-5** Given: Face width *b* = 1.25 in, *F* = 90 lbf, *f* = 0.25.

Preliminaries: *θ*1 = 45° − tan−1(152*/*203) = 8*.*13°, *θ*2 = 98*.*13°, *θa*= 90°,

*a* = (152.42 + 203.22)1*/*2 = 254 mm

Eq. (16-2):



Eq. (16-3):



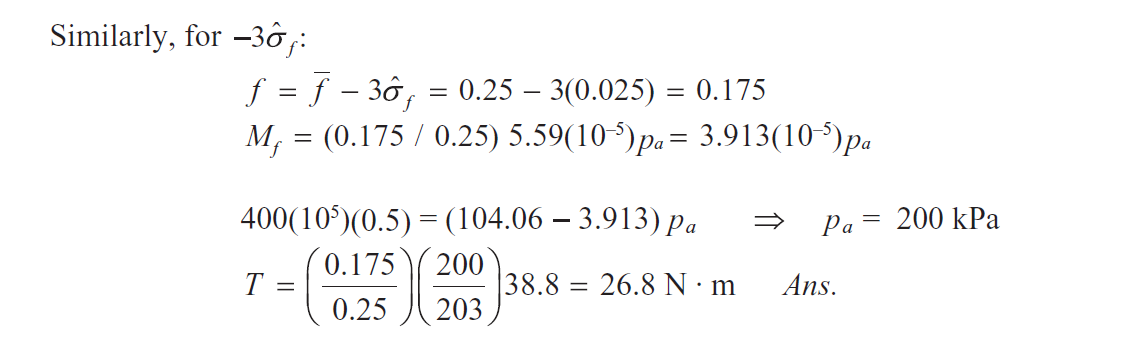
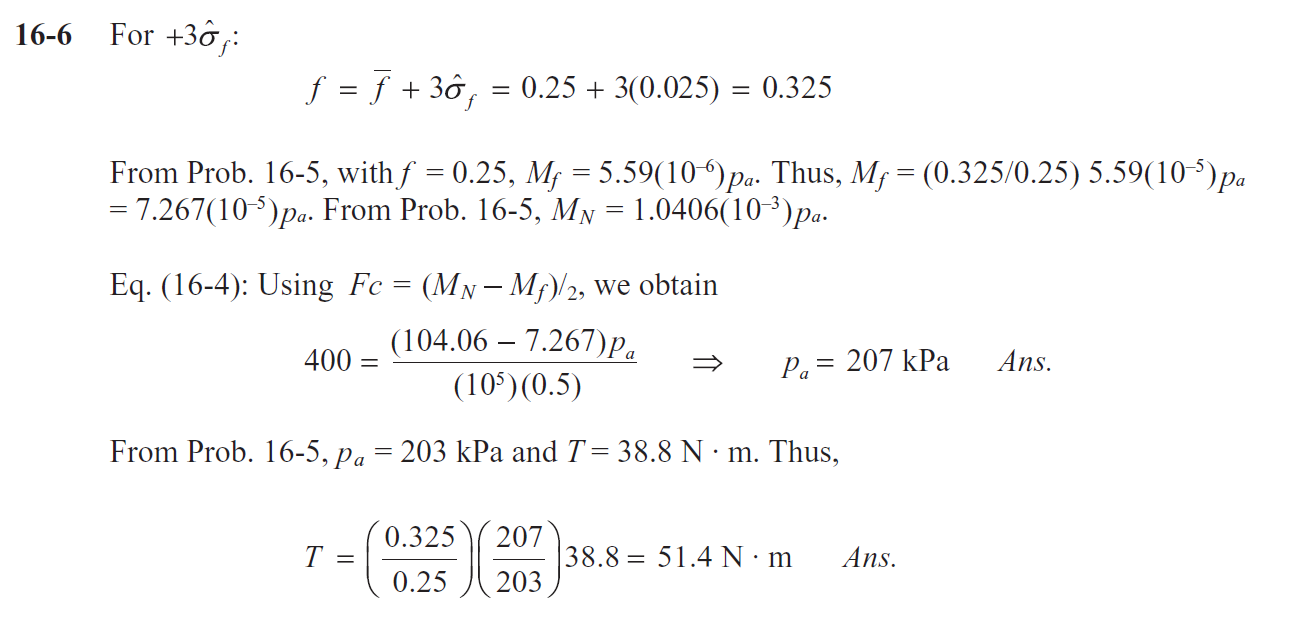
Eq. (16-4): Using *Fc*= *MN*−*Mf*, we obtain



Eq. (16-6):



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**16-7** Preliminaries: *θ*2 = 180° − 30° − tan−1(76.2*/*304.8) = 136°, *θ*1 = 20° − tan−1(76.2*/*304.8) = 6°,*θa*= 90°, sin*θa*= 1, *a* = (76.22 + 304.82)1*/*2 = 314.2 mm, *r* = 254 mm, *f* = 0*.*30, *b* = 50.8 mm , *pa* = 1034.2 kPa.

Eq. (16-2): 

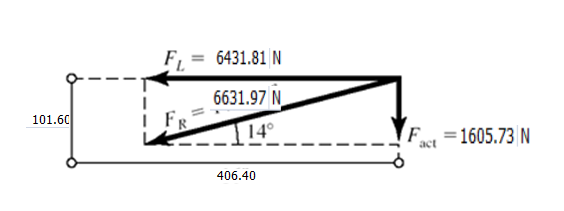
Eq. (16-3): 

*LH shoe*:

*cL*= 304.8 + 304.8 + 101.6 = 711.2 mm

Now note that *Mf*is cw and *MN*is ccw. Thus,





Eq. (16-6): 

*RH shoe*:

**

On this shoe, both *MN*and *Mf*are ccw. Also,

*cR*= (609.6 − 50.8 tan 14°) cos 14° = 579.12 mm



Thus, 

Then, 

*T*total = 1742.15 + 866.70 = 2608.86 N · m  *Ans.*

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**16-8** 

From which



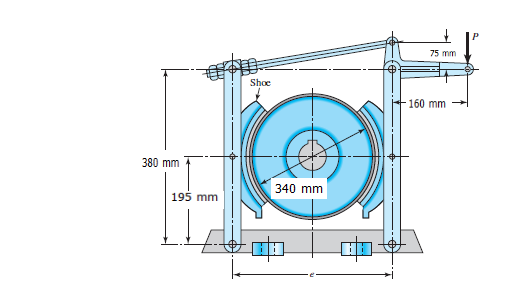
Eq. (16-15):

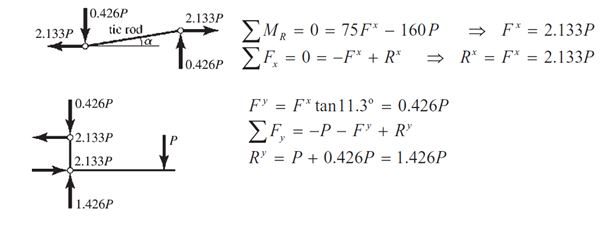
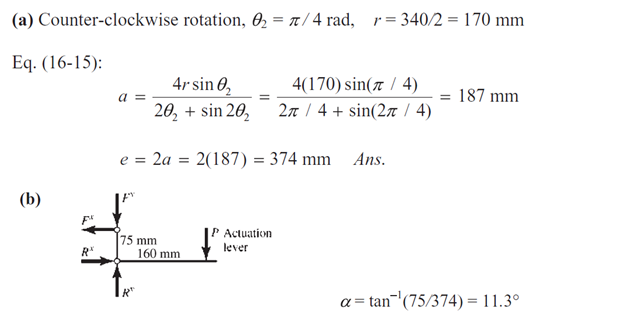


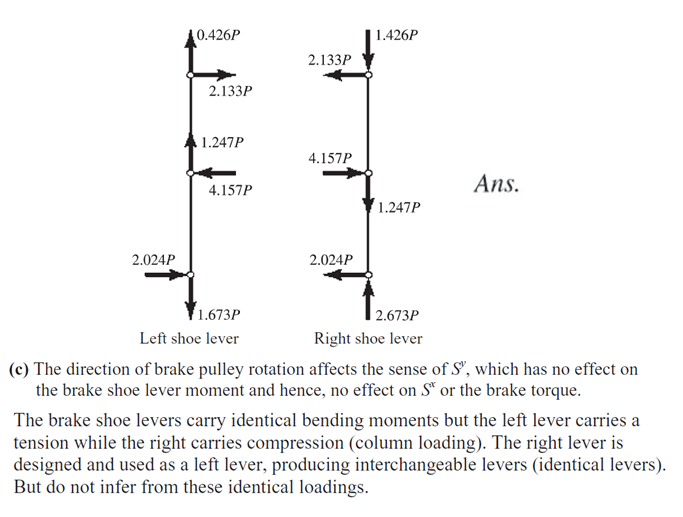
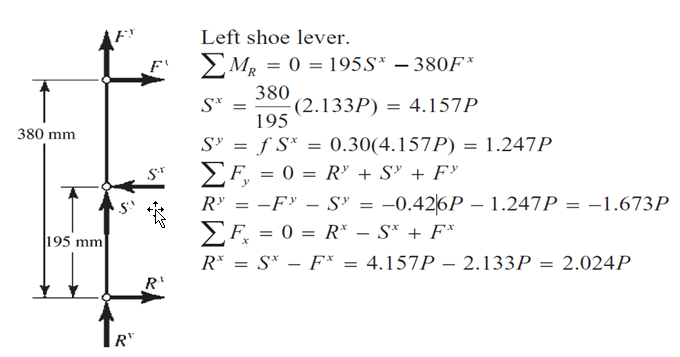
*a*differs with *a* by 100(1.170−1.209)/1.209 =− 3.23 % *Ans*.

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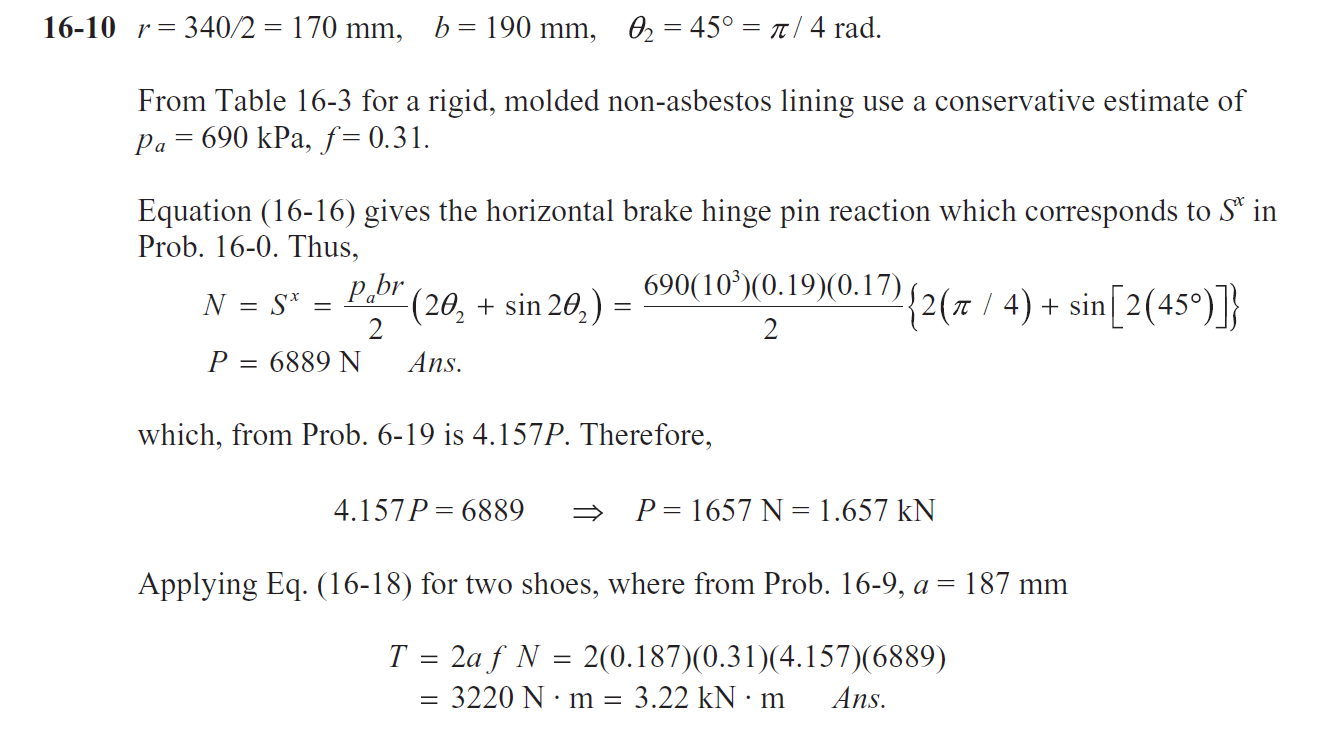
**16-9**

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**16-11** Given: *D* = 350 mm, *b* = 100 mm, *pa* = 620 kPa, *f* = 0.30, *φ* = 270°.

Eq. (16-22):



Eq. (16-19): *P*2 = *P*1 exp(−*fφ*) = 10.85 exp(−1*.*414) = 2.64 kN*Ans.*

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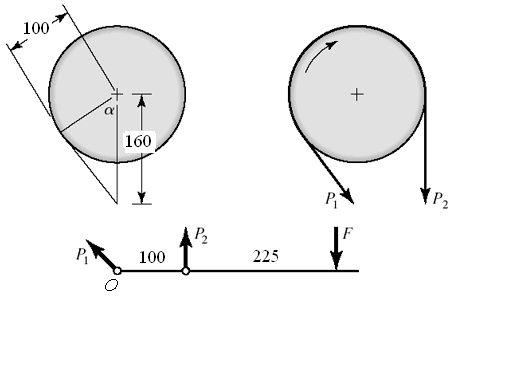
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**16-12** Given: *D* = 304.80 mm, *f* = 0*.*28, *b* = 82.55 mm, *φ*= 270°, *P*1 = 8000 N*.*

Eq. (16-22): 

26

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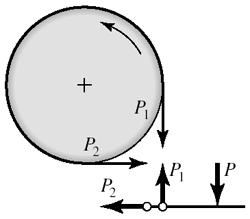


**16-13**

Σ*MO* = 0 = 100 *P*2− 325 *F*⇒*P*2 = 325(300)/100 = 975 N *Ans*.



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**16-14** **(a)** *D* = 406.4 mm, *b* = 76.2 mm

*n* = 200 rev/min

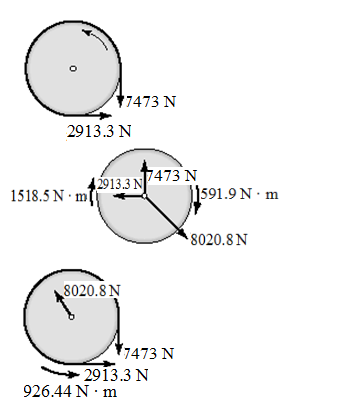
*f* = 0*.*20, *pa*= 482.63 kPa

Eq. (16-22): 



Eq. (16-14): 



****

**(b)** Force of belt on the drum:

*R* = (74732 + 2913.332)1*/*2 = 8020.8 N

Force of shaft on the drum: 7473 and 2913.33 N



Net torque on drum due to brake band:



The radial load on the bearing pair is 1803 lbf. If the bearing is straddle mounted with

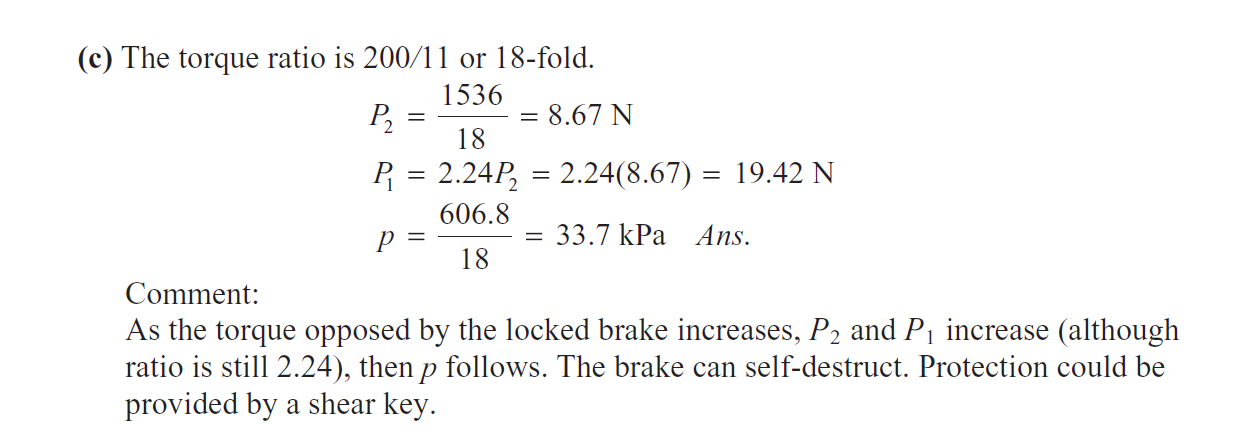
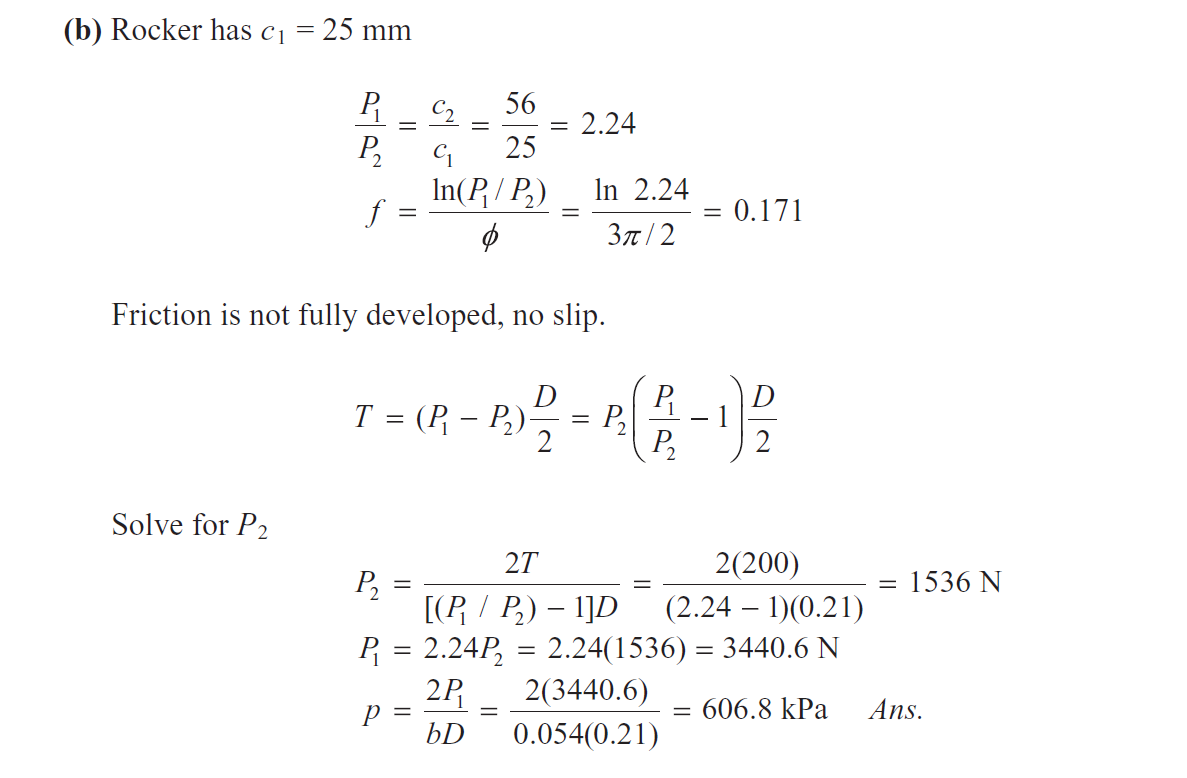
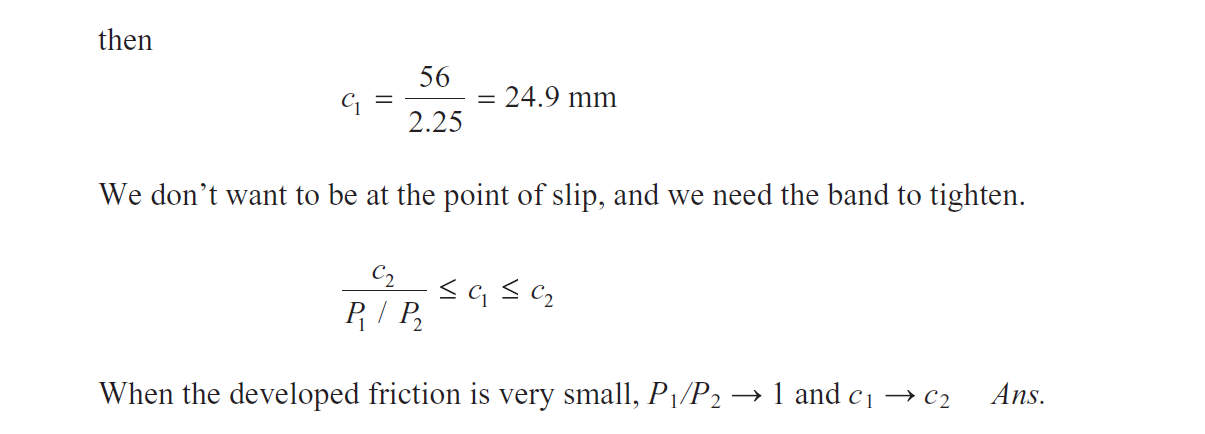
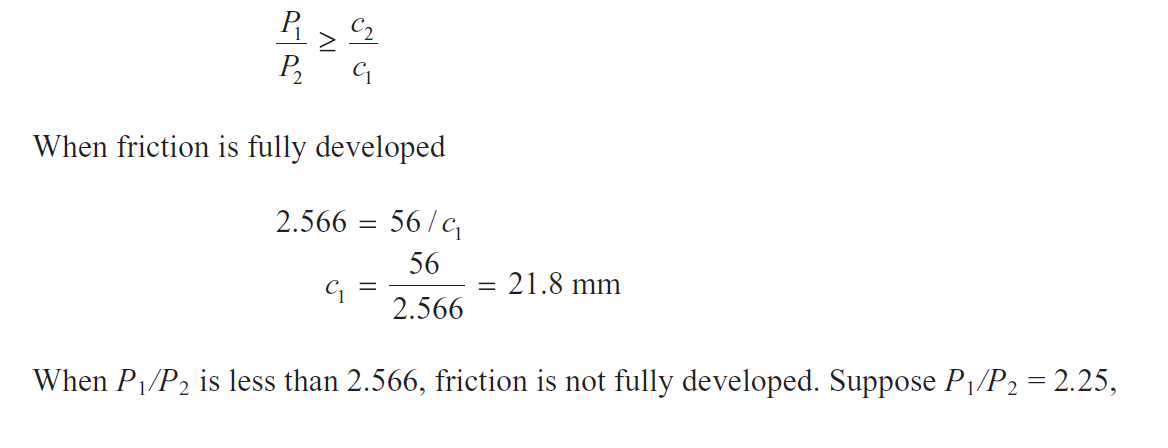
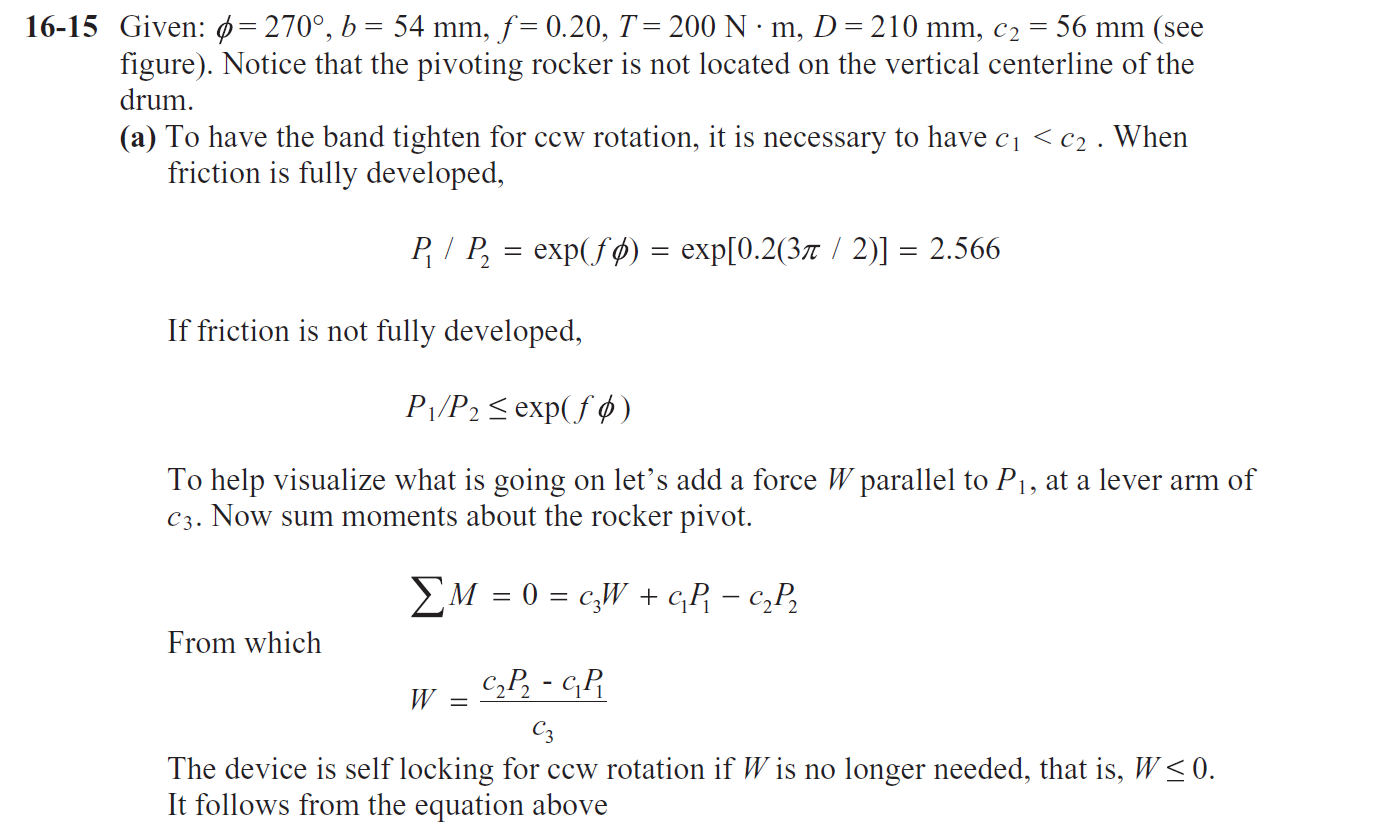
the drum at center span, the bearing radial load is 8020.8*/*2 = 4010.4 N*.*

**(c)** Eq. (16-21):





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**16-16** Given: OD = 250 mm, ID = 175 mm, *f* = 0.30, *F* = 4 kN.

**(a)** From Eq. (16-23),



Eq. (16-25):



**(b)** From Eq. (16-26),



Eq. (16-27):



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**16-17** Given: OD = 165 mm, ID = 102 mm, *f* = 0.24, *pa* = 830 MPa.

**(a)** Eq. (16-23):



Eq. (16-24) with *N* sliding planes:



**(b) **

|  |  |  |
| --- | --- | --- |
| *d*,mm | *T*,N·m |  |
| 50.8 | 586.41 |  |
| 76.2 | 764.64 |  |
| 101.6 | 804.88 | *Ans.* |
| 127 | 661.15 |  |
| 152.4 | 287.46 |  |

**(c)** The torque-diameter curve exhibits a stationary point maximum in the range of

diameter *d*. The clutch has nearly optimal proportions.

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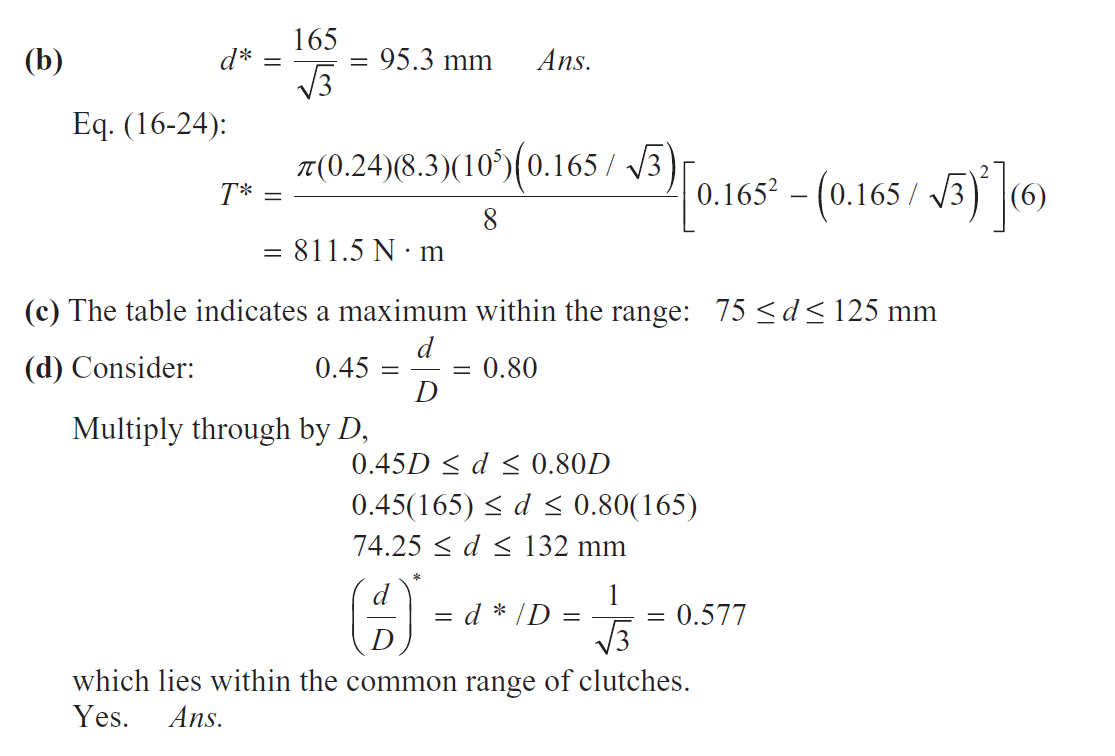
**16-18** **(a)**Eq. (16-24) with *N* sliding planes:



Differentiating with respect to *d* and equating to zero gives

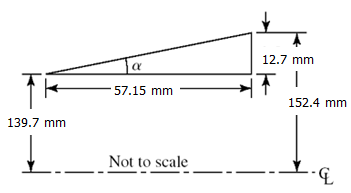


which is negative for all positive *d*. We have a stationary point *maximum*.



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**16-19** Given: *d* = 279.4 mm, *l* = 57.15 mm, *T* = 203.36 N· m, *D* = 304.80 mm, *f* = 0*.*28*.*





*Uniform wear*

Eq. (16-45):





Eq. (16-44):



*Uniform pressure*

Eq. (16-48):





Eq. (16-47):



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**16-20** *Uniform wear*

Eq. (16-34): 

Eq. (16-33): *F* = (*θ*2−*θ*1) *pari*(*ro*−*ri*)

Thus,



*Uniform pressure*

Eq. (16-38): 

Eq. (16-37): 

Thus,

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**16-21**



*Key*:



Average shear stress in key is



Average bearing stress is



Let one jaw carry the entire load.



The bearing and shear stress estimates are



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**16-22**



From Eq. (16-51),



Eq. (16-52):



Eq. (16-55): Cp =500 J/kg for steel or cast iron



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**16-23**



Eq. (16-62): *Cs* = (*ω* 2−*ω* 1) / *ω* = (*n*2−*n*1) / *n* = (260 − 240) / 250 = 0.08 *Ans*.

*ω* = 2*π* (250) / 60 = 26.18 rad/s

From Eq. (16-64):

****

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Table A-5, cast iron unit weight = 70.6 kN/m3⇒*ρ* = 70.6(103) / 9.81 = 7197 kg / m3.

Volume: *V* = *m* / *ρ* = 233.9 / 7197 = 0.0325 m3



Equating the expressions for volume and solving for *t*,



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**16-24** **(a)** The useful work performed in one revolution of the crank shaft is

*U* = 320 (103) 200 (10−3) 0*.*15 = 9.6 (103) J

Accounting for friction, the total work done in one revolution is

*U* = 9.6(103)*/*(1 − 0*.*20) = 12.0(103) J

Since 15% of the crank shaft stroke accounts for 7.5% of a crank shaft revolution, the energyfluctuation is

*E*2−*E*1 = 9.6(103) − 12.0(103)(0*.*075) = 8.70(103) J*Ans.*

**(b)** For the flywheel,



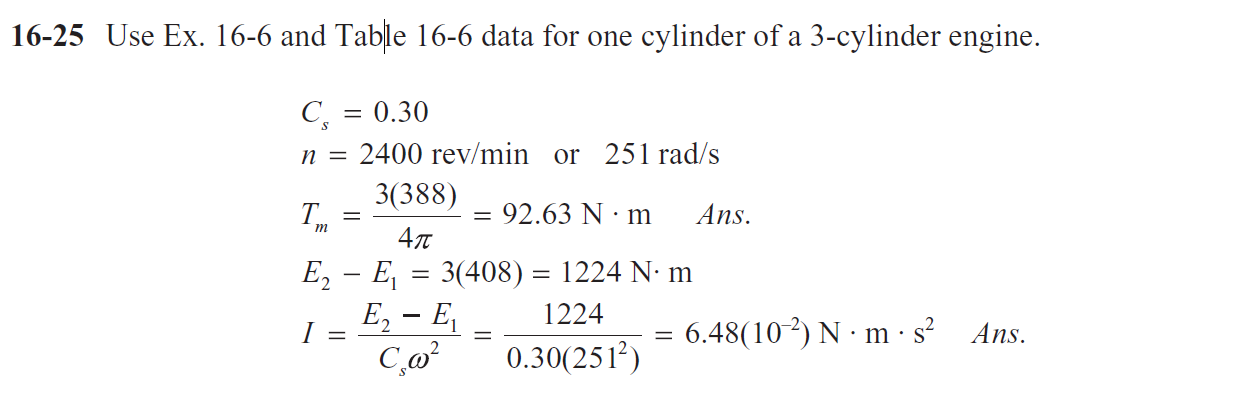
Since *Cs*= 0*.*10

Eq. (16-64): 

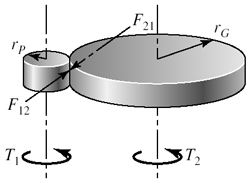
Assuming all the mass is concentrated at the effective diameter, *d*,



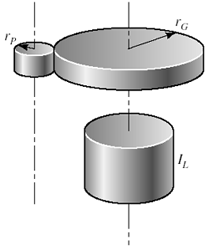
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**16-26** **(a)**

** (1)**





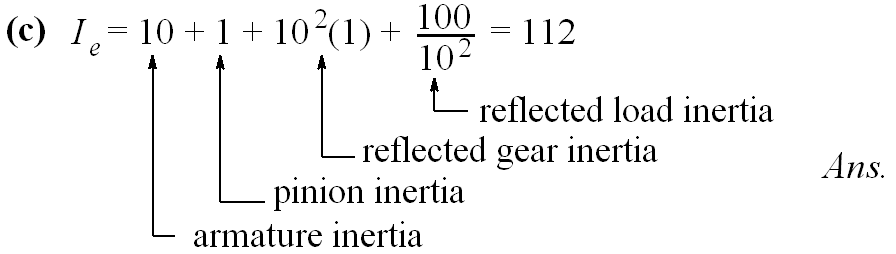
**(2)** Equivalent energy



**(3) **

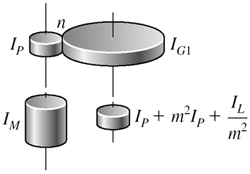
From (2) 

**(b) **

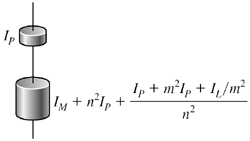


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**16-27** **(a)** Reflect *IL*, *IG*2 to the center shaft



Reflect the center shaft to the motor shaft





**(b)** For *R* = constant = *nm*, 

**(c)** For *R* = 10,

*n*6−*n*2− 200 = 0

From which

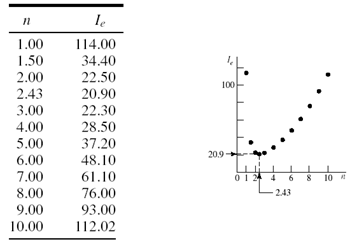


Notice that *n*\*and *m*\* are independent of *IL*.

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**16-28** From Prob. 16-27,



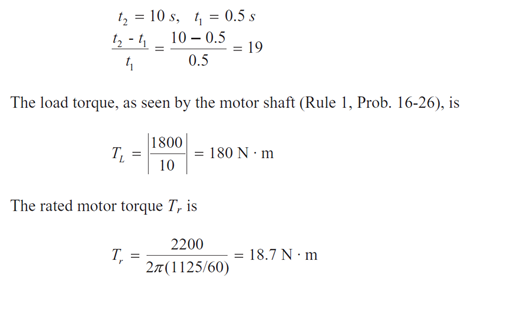


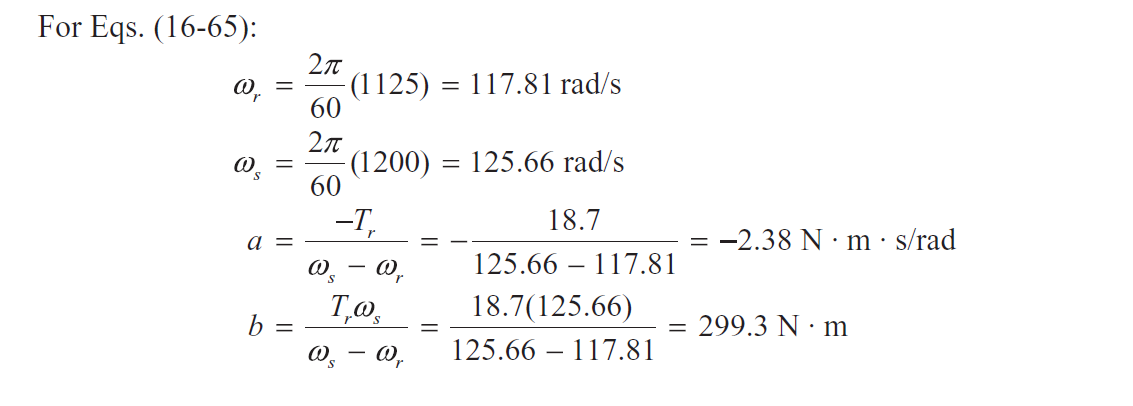
Optimizing the partitioning of a double reduction lowered the gear-train inertia to20*.*9*/*112 = 0*.*187, or to 19% of that of a single reduction. This includes the two additionalgears.

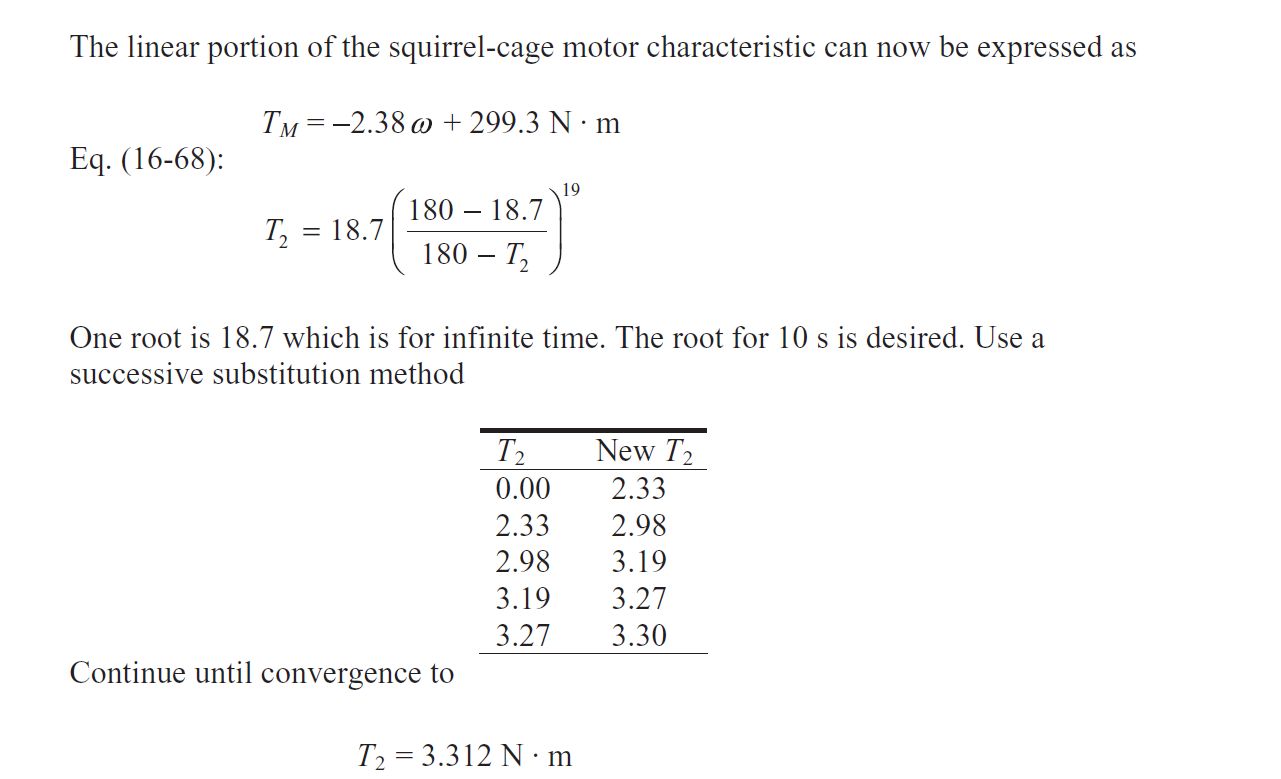
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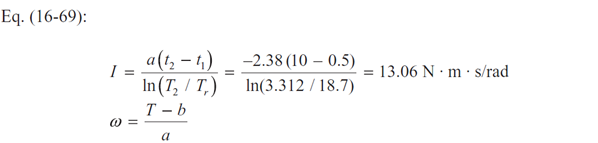
**16-29** Given : TCrankshaft =1800 Nm, Motor power = 2.2 kW and n=1125 rev/min

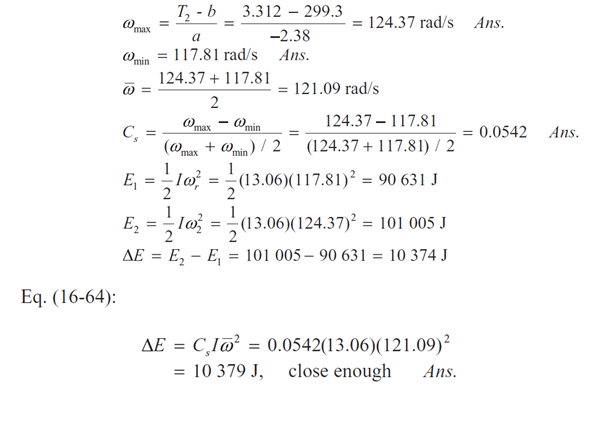
Figure 16-29 applies,

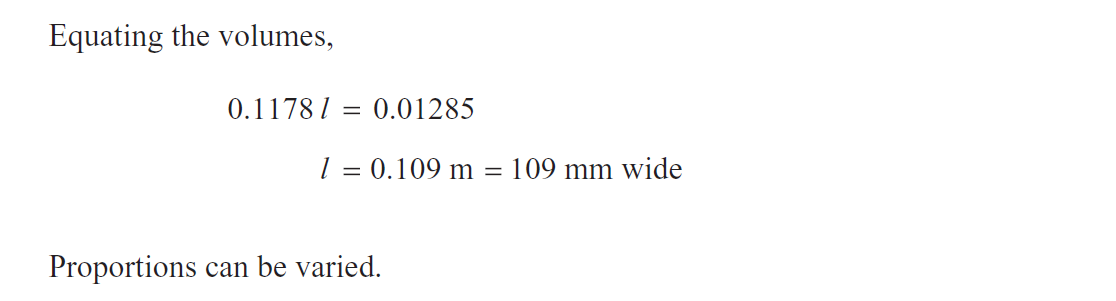
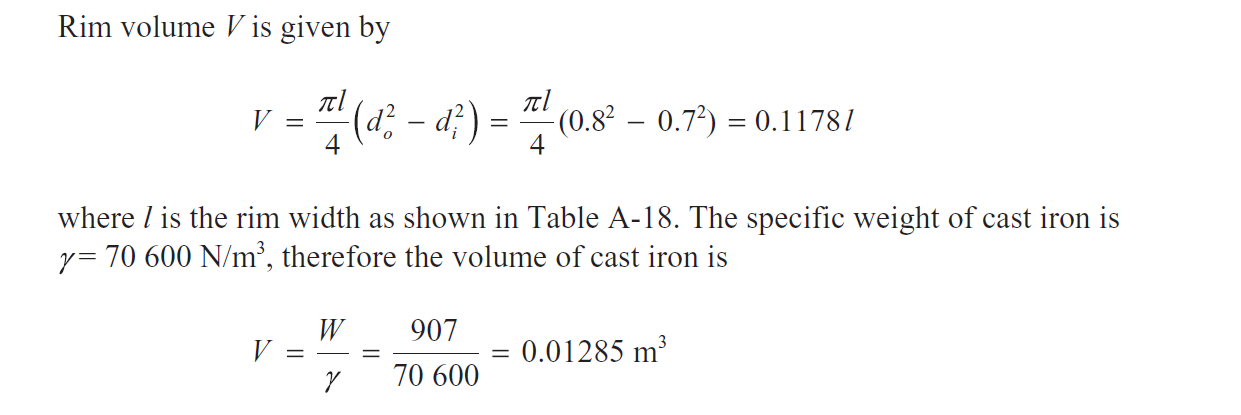
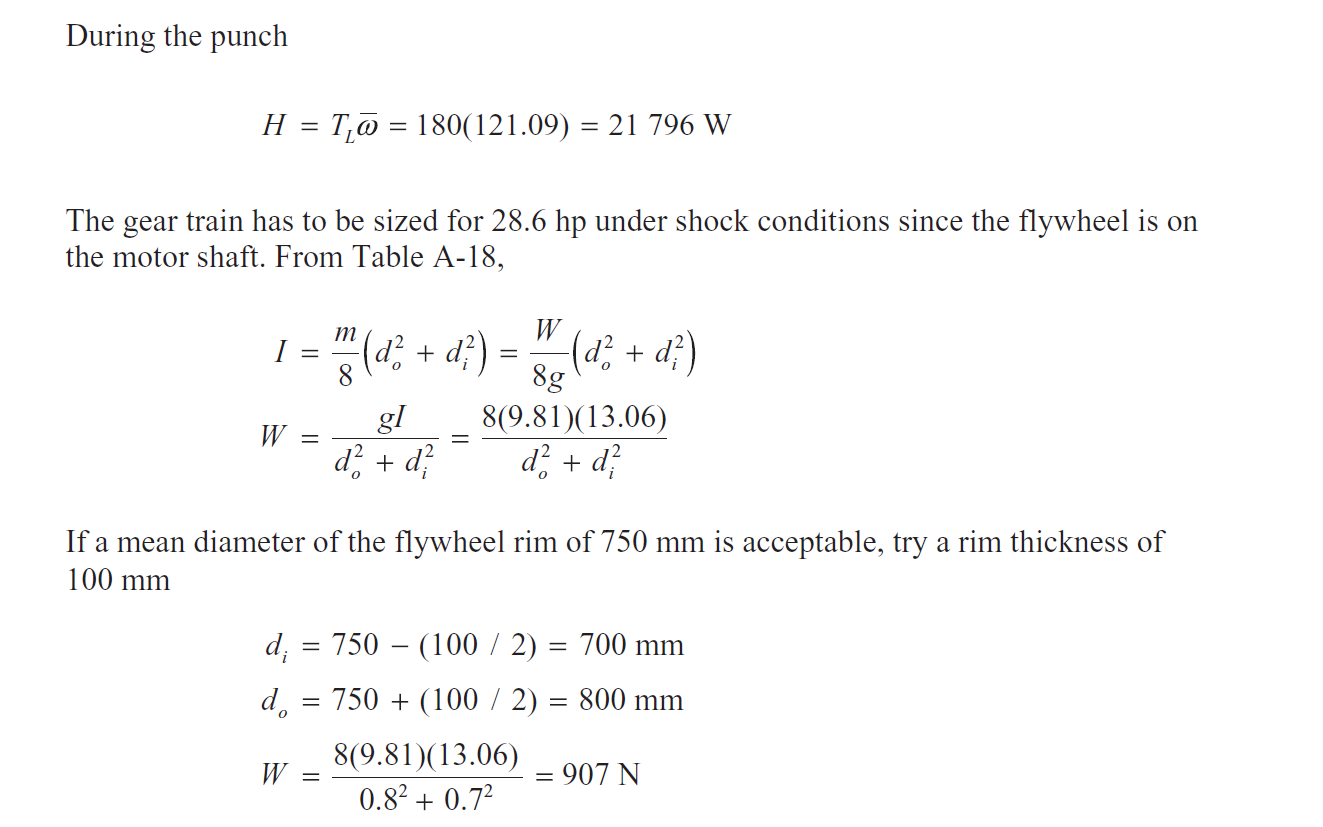




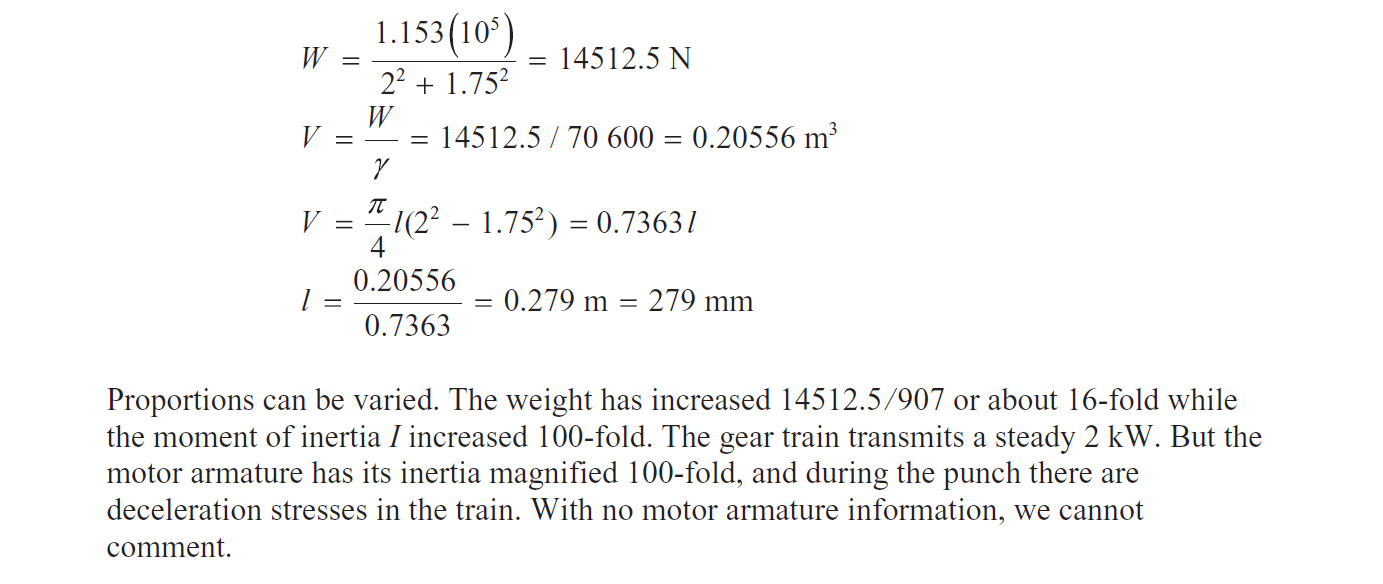
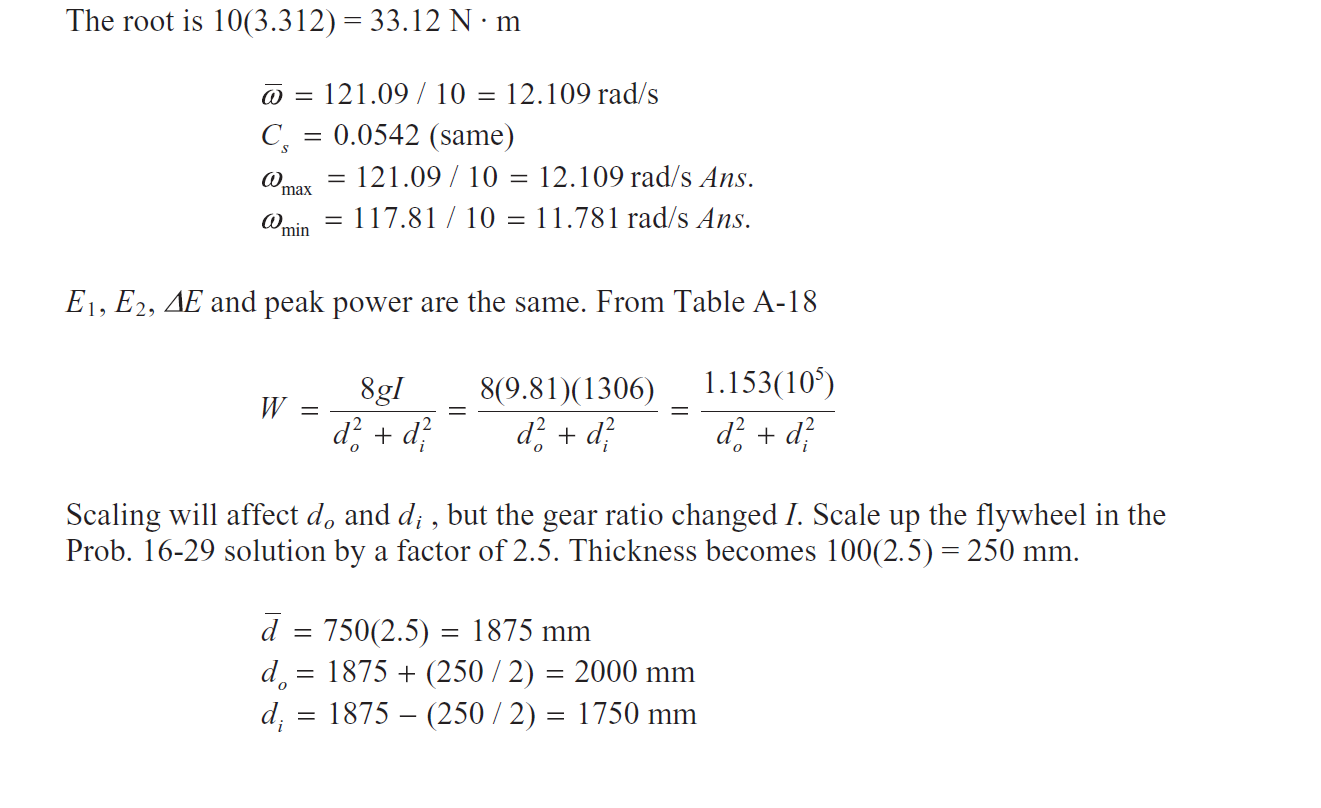
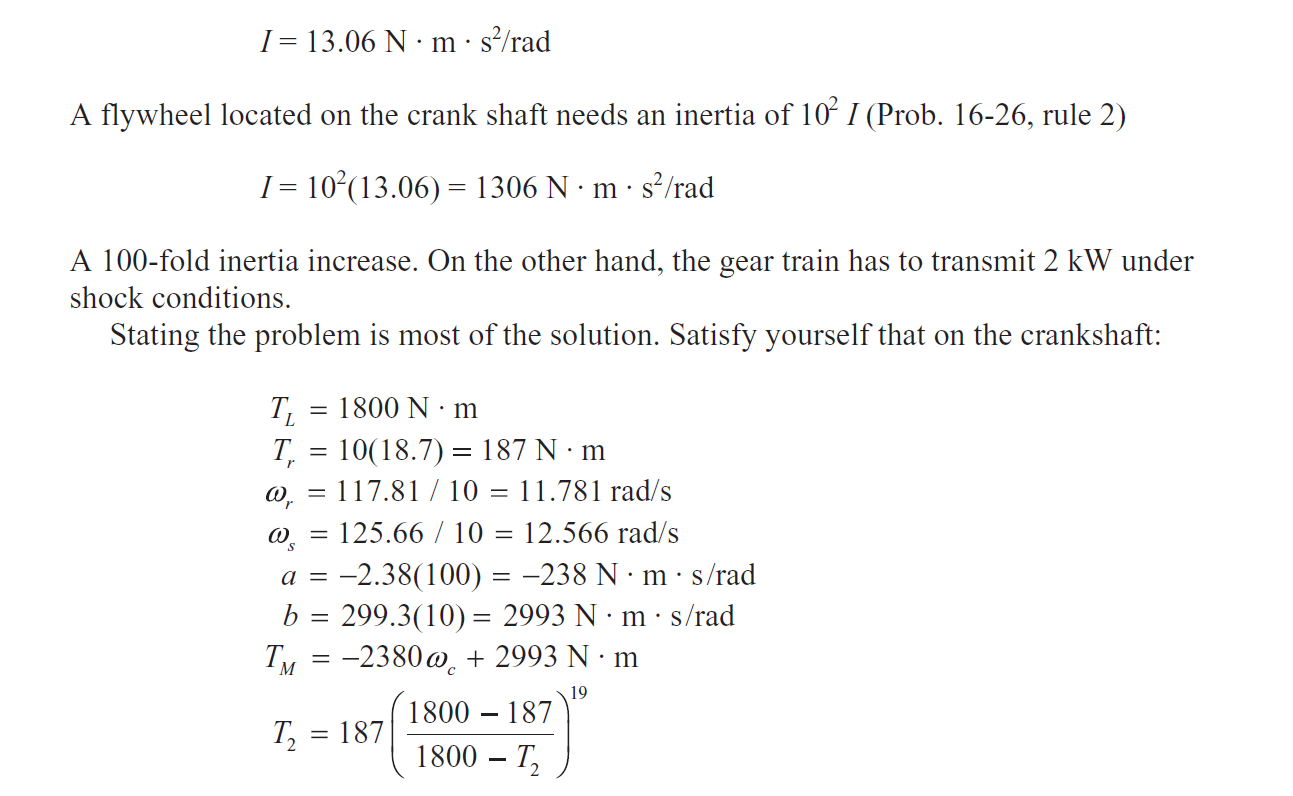
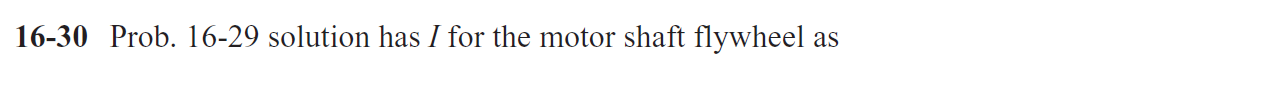








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**16-31** This can be the basis for a class discussion.