**Chapter 13**

**13-1** 







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**13-2 **

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**13-3** 







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**13-4** *Mesh:* (Use P = 3 teeth/in or 0.118 teeth/mm)











*Pinion Base-Circle:* 



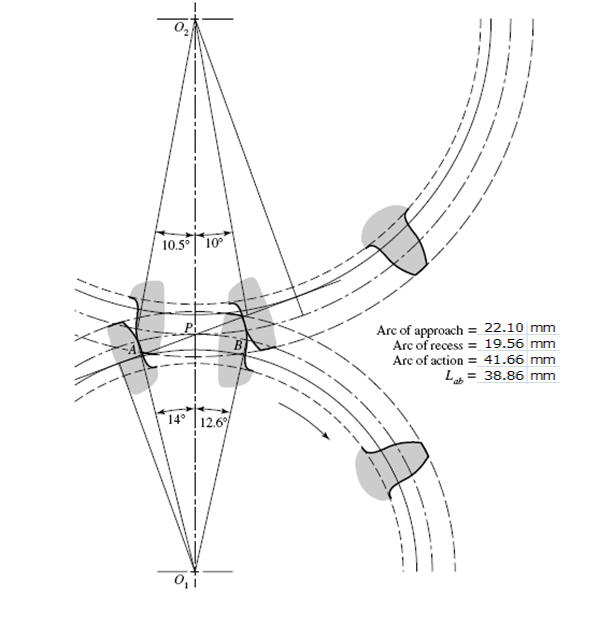
*Gear Base-Circle:* 



*Base pitch:* 

*Contact Ratio:* 

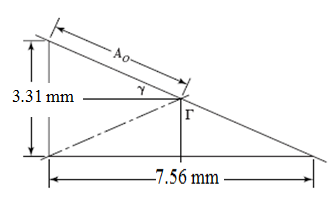
See the following figure for a drawing of the gears and the arc lengths.



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**13-5**

**(a)** 

 **(b) **

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**(c)**

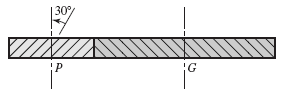
**** *Ans.*

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**(d)**From Table 13-3, 0.3*AO* = 0.3(4.128) = 1.238 mm and 10/*P* = 10/4.23 = 2.36



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**13-6**

**(a)** 





**(b)** Eq. (13-7): 

**(c)** 



**(d)** Table 13-4:

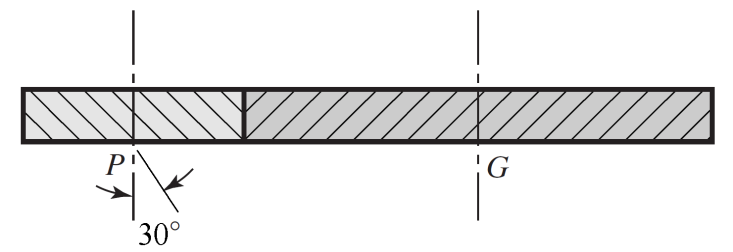








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**13-7**



**(a) **

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**(b) **

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**(c) **

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**13-8 (a)**Using Eq. (13-11) with *k* = 1, **= 20º, and *m* = 2,

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Round up for the minimum integer number of teeth.

*NP* = 15 teeth *Ans.*

**(b)**Repeating (a) with *m* = 3, *NP* = 14.98 teeth. Rounding up, *NP* = 15 teeth. *Ans.*

**(c)**Repeating (a) with *m* = 4, *NP* = 15.44 teeth. Rounding up, *NP* = 16 teeth. *Ans.*

**(d)**Repeating (a) with *m* = 5, *NP* = 15.74 teeth. Rounding up, *NP* = 16 teeth. *Ans.*

Alternatively, a useful table can be generated to determine the largest gear that can mesh with a specified pinion, and thus also the maximum gear ratio with a specified pinion. The Max *NG­*column was generated using Eq. (13-12) with *k* = 1, **= 20º, and rounding up to the next integer.

|  |  |  |
| --- | --- | --- |
| **Min *NP*** | **Max *NG*** | **Max *m* = Max *NG*/Min *NP*** |
| 13 | 16 | 1.23 |
| 14 | 26 | 1.86 |
| 15 | 45 | 3.00 |
| 16 | 101 | 6.31 |
| 17 | 1309 | 77.00 |
| 18 | unlimited | unlimited |

With this table, we can readily see that gear ratios up to 3 can be obtained with a minimum *NP* of 15 teeth, and gear ratios up to 6.31 can be obtained with a minimum *NP* of 16 teeth. This is consistent with the results previously obtained.

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**13-9** Repeating the process shown in the solution to Prob. 13-8, except with **= 25º, we obtain the following results.

**(a)**For *m* = 2, *NP* = 9.43 teeth. Rounding up, *NP* = 10 teeth. *Ans.*

**(b)**For *m* = 3, *NP* = 9.92 teeth. Rounding up, *NP* = 10 teeth. *Ans.*

**(c)**For *m* = 4, *NP* = 10.20 teeth. Rounding up, *NP* = 11 teeth. *Ans.*

**(d)**For *m* = 5, *NP* = 10.38 teeth. Rounding up, *NP* = 11 teeth. *Ans.*

For convenient reference, we will also generate the table from Eq. (13-12) for **= 25º.

|  |  |  |
| --- | --- | --- |
| **Min *NP*** | **Max *NG*** | **Max *m* = Max *NG*/Min *NP*** |
| 9 | 13 | 1.44 |
| 10 | 32 | 3.20 |
| 11 | 249 | 22.64 |
| 12 | unlimited | unlimited |

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**13-10 (a)**The smallest pinion tooth count that will run with itself is found from Eq. (13-10).



**(b)**The smallest pinion that will mesh with a gear ratio of *mG* = 2.5, from Eq. (13-11) is



The largest gear-tooth count possible to mesh with this pinion, from Eq. (13-12) is



**(c)**The smallest pinion that will mesh with a rack, from Eq. (13-13),



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**13-11** 

From Eq. (13-19), 

**(a)**The smallest pinion tooth count that will run with itself, from Eq. (13-21) is



**(b)**The smallest pinion that will mesh with a gear ratio of *m* = 2.5, from Eq. (13-22) is



The largest gear-tooth count possible to mesh with this pinion, from Eq. (13-23) is



**(c)**The smallest pinion that will mesh with a rack, from Eq. (13-24) is



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**13-12** From Eq. (13-19), 

Program Eq. (13-23) on a computer using a spreadsheet or code, and increment *NP*. The first value of *NP* that can be doubled is *NP*= 10 teeth, where*NG­*≤ 26.01 teeth. So *NG*= 20 teeth will work. Higher tooth counts will work also, for example 11:22, 12:24, etc.

Use *NP* = 10 teeth, *NG* = 20 teeth*Ans.*

Note that the given diametral pitch (tooth size) is not relevant to the interference problem.

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**13-13** From Eq. (13-19), 

Program Eq. (13-23) on a computer using a spreadsheet or code, and increment *NP*. The first value of *NP* that can be doubled is *NP* = 6 teeth, where *NG­*≤ 17.6 teeth. So *NG* = 12 teeth will work. Higher tooth counts will work also, for example 7:14, 8:16, etc.

Use *NP* = 6 teeth, *NG* = 12 teeth *Ans.*

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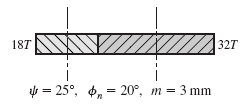
**13-14** The smallest pinion that will operate with a rack without interference is given by Eq. (13-13).



Setting *k* = 1 for full depth teeth, *NP* = 9 teeth, and solving for **,



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**13-15**

**(a)** Eq. (13-3): 

Eq. (13-16): 

Eq. (13-17): 

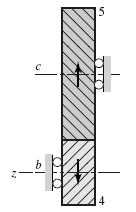
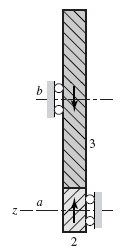
**(b)** Eq. (13-3): 

Eq. (13-19): 

**(c)** Eq. (13-2): *dp* = *mt* *Np* = 3.310 (18) = 59.58 mm *Ans.*

Eq. (13-2): *dG* = *mt* *NG* = 3.310 (32) = 105.92 mm *Ans.*

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**13-16 (a)**Sketches of the figures are shown to determine the axial forces by inspection.

The axial force of gear 2 on shaft *a* is in the negative *z*-direction. The axial force of gear 3 onshaft *b* is in the positive *z*-direction. *Ans.*

The axial force of gear 4 on shaft *b* is in the positive *z*-direction. The axial force of gear 5 on shaft *c* is in the negative *z*-direction. *Ans.*

**(b)** 

**(c) **

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**13-17**  



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**13-18** 



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**13-19 (a)**



**(b)** 





**(c)** 

\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**13-20** Applying Eq. (13-30), *e* = (*N*2 / *N*3)(*N*4 / *N*5) = 45. For an exact ratio, we will choose to factor the train value into integers, such that

*N*2 / *N*3 = 9 (1)

*N*4 / *N*5 = 5 (2)

Assuming a constant diametral pitch in both stages, the geometry condition to satisfy the in-line requirement of the compound reverted configuration is

*N*2 + *N*3 = *N*4 + *N*5 (3)

With three equations and four unknowns, one free choice is available. It is necessary that all of the unknowns be integers. We will use a normalized approach to find the minimum free choice to guarantee integers; that is, set the smallest gear of the largest stage to unity, thus *N*3= 1. From (1), *N*2­ = 9. From (3),

*N*2 + *N*3 = 9 + 1 = 10 = *N*4 + *N*5

Substituting *N*4 = 5 *N*5 from (2) gives

10 = 5 *N*5 + *N*5 = 6 *N*5

*N*5 = 10 / 6 = 5 / 3

To eliminate this fraction, we need to multiply the original free choice by a multiple of 3. In addition, the smallest gear needs to have sufficient teeth to avoid interference. From Eq. (13-11) with *k* = 1, **= 20°, and *m* = 9, the minimum number of teeth on the pinion to avoid interference is 17. Therefore, the smallest multiple of 3 greater than 17 is 18. Setting *N*3­ = 18 and repeating the solution of equations (1), (2), and (3) yields

*N*2 = 162 teeth

*N*3 = 18 teeth

*N*4 = 150 teeth

*N*5 = 30 teeth *Ans.*

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**13-21** The solution to Prob. 13-20 applies up to the point of determining the minimum number of teeth to avoid interference. From Eq. (13-11), with *k* = 1, **= 25°, and *m* = 9, the minimum number of teeth on the pinion to avoid interference is 11. Therefore, the smallest multiple of 3 greater than 11 is 12. Setting *N3­* = 12 and repeating the solution of equations (1), (2), and (3) of Prob. 13-20 yields

*N*2 = 108 teeth

*N*3 = 12 teeth

*N*4 = 100 teeth

*N*5 = 20 teeth *Ans.*

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**13-22** Applying Eq. (13-30), *e* = (*N*2 / *N*3)(*N*4 / *N*5) = 30. For an exact ratio, we will choose to factor the train value into integers, such that

*N*2 / *N*3 = 6 (1)

*N*4 / *N*5 = 5 (2)

Assuming a constant diametral pitch in both stages, the geometry condition to satisfy the in-line requirement of the compound reverted configuration is

*N*2 + *N*3 = *N*4 + *N*5 (3)

With three equations and four unknowns, one free choice is available. It is necessary that all of the unknowns be integers. We will use a normalized approach to find the minimum free choice to guarantee integers; that is, set the smallest gear of the largest stage to unity, thus *N*3= 1. From (1), *N*2­ = 6. From (3),

*N*2 + *N*3 = 6 + 1 = 7 = *N*4 + *N*5

Substituting *N*4 = 5 *N*5 from (2) gives

7 = 5 *N*5 + *N*5 = 6 *N*5

*N*5 = 7 / 6

To eliminate this fraction, we need to multiply the original free choice by a multiple of 6. In addition, the smallest gear needs to have sufficient teeth to avoid interference. From Eq. (13-11) with *k* = 1, **= 20°, and *m* = 6, the minimum number of teeth on the pinion to avoid interference is 16. Therefore, the smallest multiple of 6 greater than 16 is 18. Setting *N3­* = 18 and repeating the solution of equations (1), (2), and (3) yields

*N*2 = 108 teeth

*N*3 = 18 teeth

*N*4 = 105 teeth

*N*5 = 21 teeth *Ans.*

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**13-23** Applying Eq. (13-30), *e* = (*N*2 / *N*3)(*N*4 / *N*5) = 45. For an approximate ratio, we will choose to factor the train value into two equal stages, such that



If we choose identical pinions such that interference is avoided, both stages will be identical and the in-line geometry condition will automatically be satisfied. From Eq. (13-11) with *k* = 1, **= 20°, and, the minimum number of teeth on the pinions to avoid interference is 17. Setting *N*3*­* = *N*5 = 17, we get



Rounding to the nearest integer, we obtain

*N*2 =*N*4 = 114 teeth

*N*3 =*N*5 = 17 teeth *Ans.*

Checking, the overall train value is *e* = (114 / 17)(114 / 17) = 44.97.

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**13-24** *H* = 25 hp, *i* = 2500 rev/min

Let *ωo* = 300 rev/min for minimal gear ratio to minimize gear size.



Let 

From Eq. (13-11) with *k* = 1, **= 20°, and *m* = 2.887, the minimum number of teeth on the pinions to avoid interference is 15.

Let *N*2 = *N*4 = 15 teeth

*N*3 = *N*5 = 2.887(15) = 43.31 teeth

Try *N*3 = *N*5 = 43 teeth.



Too big. Try *N*3 = *N*5 = 44.



*N*2 = *N*4 = 15 teeth, *N*3 = *N*5 = 44 teeth*Ans.*

\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**13-25 (a)** The planet gears act as keys and the wheel speeds are the same as that of the ringgear.Thus,



**(b) **

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**(c)**The wheel spins freely on icy surfaces, leaving no traction for the other wheel.

The car is stalled. *Ans.*

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**13-26 (a)**The motive power is divided equally among four wheels instead of two.

**(b)**Locking the center differential causes 50 percent of the power to be applied to the

rear wheels and 50 percent to the front wheels. If one of the rear wheels rests on a slippery surface such as ice, the other rear wheel has no traction. But the front wheels still provide traction, and so you have two-wheel drive. However, if the rear differential is locked, you have 3-wheel drive because the rear-wheel power is now distributed 50-50.

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**13-27** Let gear 2 be first, then *nF* = *n*2 = 0. Let gear 6 be last, then *nL* = *n*6 = –12 rev/min.







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**13-28** Let gear 2 be first, then *nF* = *n*2 = 0 rev/min. Let gear 6 be last, then *nL* = *n*6 = 85 rev/min.











The positive sign indicates the same direction as*n*6.

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**13-29** *φ* = 20o, *P* = 6 teeth/in. Since *N* α*d*, then,

*N*2 + *N*3 =*N*4 +*N*5 (1)

Hour hand moves 1/12 of minute hand. Thus, *ω*5 / *ω*2 =. Now,

*ω*5 / *ω*4 = *N* 4 / *N* 5 , *ω*3 / *ω*2 = *N* 2 / *N* 3, and *ω*4 = *ω*3. Thus,



So, try *N*3 = 4 *N*2, and *N*5 = 3 *N*4. Substituting *N*5 = 3 *N*4 into Eq. (1) gives

*N*2 + *N*3 = 4*N*4⇒*N*4 = (*N*2 + *N*3)/4. Let *N*2 = 1. Then, *N*3 = 4*N*2 = 4(1) = 4,

*N*4 = (*N*2 + *N*3)/4 = (1 + 4)/4 = 5/4, *N*5 = 3 *N*4 = 3(5/4) = 15/4. Teeth must be a multiple of 4. To avoid interference for the smaller pinion use Eq. (13-11) with *m* = *N* 3 / *N* 2 = 4:



Use *N*2 = 16 teeth which is a multiple of 4. Then, *N*3 = 4(16) = 64 teeth, *N*4 = (5/4)16 = 20 teeth, and *N*5 = (15/4) 16 = 60 teeth. Thus,

*N*2 = 16 teeth, *N*3 = 64 teeth, *N*4 = 20 teeth, and *N*5 = 60 teeth  *Ans*.

\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**13-30** The geometry condition is ****.Since all the gears are meshed, they will all have the same diametral pitch. Applying *d* = *N / P*,

****

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Let gear 2 be first, *nF* = *n*2 = 320 rev/min. Let gear 5 be last, *nL* = *n*5 = 0 rev/min.







The negative sign indicates opposite of *n*2. 

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**13-31** Let *nF* = *n*2, then *nL* = *n*7 =0.















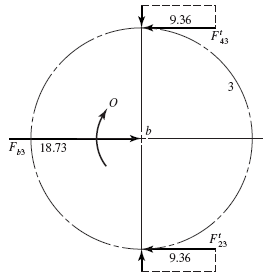
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**13-32 (a)**  

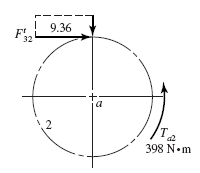


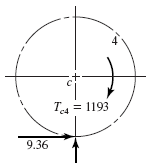
So 



So 





**(b) **

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Note: The solution is independent of the pressure angle.

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**13-33 **

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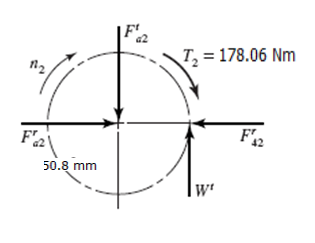
Noting that power equals torque times angular velocity, the input torque is



For 100 percent gear efficiency, the output power equals the input power, so



Next, we’ll confirm the output torque as we work through the force analysis and complete the free body diagrams.



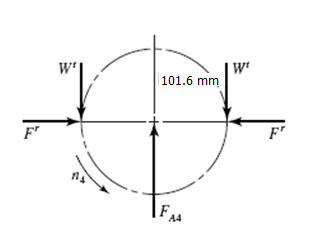
*Gear* 2

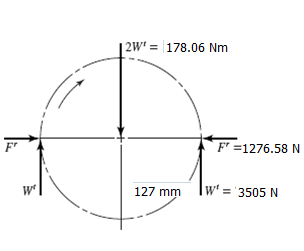




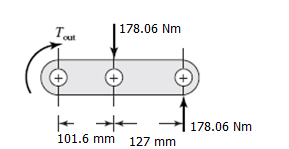
*Gear* 4





**

*Gear* 5



*Arm*



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**13-34** **(a)***φ* = 20o, *P* = 6 teeth/in (0.236 teeth/mm). Since *N* α*d*, then,

*N*2 + *N*3 =*N*4 +*N*5 (1)

*e* = 40 = 8 × 5. Then, let *N* 2 / *N* 3 = 8 and *N* 4 / *N* 5 = 5. Thus, *N* 2 = 8 *N* 3 and *N* 4 = 5 *N* 5.

Substituting *N* 4 = 5 *N* 5 into Eq. (1) gives *N*2 + *N*3 = 6*N*5⇒*N*5 = (*N*2 + *N*3)/6.

To avoid interference use Eq. (13-11) with *m* = *N* 2 / *N* 3 = 8:



Try *N* 3 = 17 teeth ⇒*N* 2 = 8 *N* 3 = 8(17) = 136 teeth, *N*5 = (136 + 17)/6 = 25.5 teeth which is unacceptable. Next, try *N* 3 = 18 teeth ⇒*N* 2 = 8 *N* 3 = 8(18) = 144 teeth,

*N*5 = (144 + 18)/6 = 27 teeth, and *N* 4 = 5 *N*5 = 5 (27) = 135 teeth. Thus,

*N* 2 = 144 teeth, *N* 3 = 18 teeth, *N* 4 = 135 teeth, and *N*5 = 27 teeth *Ans*.

**(b)** Gear diameter is *d* = *N* / *P* (with *P* = 0.236 teeth/mm), 12.7 mm wall clearance on two sides, addendum of each gear is 1/*P* = 1/0.236 mm, and each wall thickness is 19.05 mm. Thus,



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**13-35** **(a)***φ* = 25o, P=6 teeth/in (0.236 teeth/mm). Since *N* α*d*, then,

*N*2 + *N*3 =*N*4 +*N*5 (1)

*e* = 40 = 8 × 5. Let *N* 2 / *N* 3 = 8 and *N* 4 / *N* 5 = 5. Thus, *N* 2 = 8 *N* 3 and *N* 4 = 5 *N* 5.

Substituting *N* 4 = 5 *N* 5 into Eq. (1) gives *N*2 + *N*3 = 6*N*5⇒*N*5 = (*N*2 + *N*3)/6.

To avoid interference use Eq. (13-11) with *m* = *N* 2 / *N* 3 = 8:



Try *N* 3 = 11 teeth ⇒*N* 2 = 8 *N* 3 = 8(11) = 88 teeth, *N*5 = (88 + 11)/6 = 16.5 teeth which is unacceptable. Next, try *N* 3 = 12 teeth ⇒*N* 2 = 8 *N* 3 = 8(12) = 96 teeth,

*N*5 = (96 + 12)/6 = 18 teeth, and *N* 4 = 5 *N*5 = 5 (18) = 90 teeth. Thus,

*N* 2 = 96 teeth, *N* 3 = 12 teeth, *N* 4 = 90 teeth, and *N*5 = 18 teeth *Ans*.

**(b)** Gear diameter is *d* = *N* / *P* (with *P* = 0.236 teeth/mm), 12.7 mm wall clearance on two sides, addendum of each gear is 1/*P* = 1/0.236 mm, and each wall thickness is 19.05 mm. Thus,



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**13-36** **(a)***φn* = 20o, *ψ* = 45o, P=6 teeth/in (0.236 teeth/mm). Since *N* α*d*, then,

*N*2 + *N*3 =*N*4 +*N*5 (1)

*e* = 40 = 8 × 5. Let *N* 2 / *N* 3 = 8 and *N* 4 / *N* 5 = 5. Thus *N* 2 = 8 *N* 3 and *N* 4 = 5 *N* 5.

Substituting *N* 4 = 5 *N* 5 into Eq. (1) gives *N*2 + *N*3 = 6*N*5⇒*N*5 = (*N*2 + *N*3)/6.

From Eq. (13-19),



Eq. (13-22):



Try *N* 3 = 12 teeth ⇒*N* 2 = 8 *N* 3 = 8(12) = 96 teeth, *N*5 = (96 + 12)/6 = 18 teeth, and

*N* 4 = 5 *N*5 = 5 (18) = 90 teeth. Thus,

*N* 2 = 96 teeth, *N* 3 = 12 teeth, *N* 4 = 90 teeth, and *N*5 = 18 teeth *Ans*.

**(b)** Gear diameter is *d* = *N* / *P* (with *P* = 0.236 teeth/mm), 12.7 mm wall clearance on two sides, addendum of each gear is 1/*P* = 1/0.236 mm, and each wall thickness is 19.05 mm. Thus,



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**13-37** *e*≈ 40, P=6 teeth/in (0.236 teeth/mm)., *φ* = 20o.

**(a)** Minimum size is *N* 2 / *N* 3 = *N* 4 / *N* 5 =  = 6.325. To avoid interference use Eq. (13-11) with *m* = 6.325:



Let *N*3 = 16 teeth. *N*2 = 16 = 101.2. Use *N*2 = 101 teeth, *e* = (101/16)2 = 39.85 which is ok since it is between 38 and 42. Thus,

*N* 2 = *N* 4 = 101 teeth, and *N* 3 = *N*5 = 16 teeth *Ans*.

**(b)** Gear diameter is *d* = *N* / *P* (with *P* = 6 teeth/mm), 12.7 mm wall clearance on two sides, addendum of each gear is 1/*P* = 1/0.236 mm, and each wall thickness is 19.05 mm. Thus,



Comparing this to Prob. 13-34 where *Y* = 1005.4 mm, we see a large reduction in gearbox size.

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**13-38** **(a)** P=6 teeth/in (0.236 teeth/mm).,, *φ* = 20o. Since *N* α*d*, then,



Solving for *N*6 yields *N*6 = 15 teeth *Ans*.

**(b)**

**(c)** Gear 2, *n*2 = 300 rev/min,

Eq, (13-34): 

**(d)** Eq. (13-35): 

**(e)** *Fr* = *Wt* tan *φ* = 1.494 tan20o = 0.544 kN  *Ans*,

**(f)**



**(g)** 



**(h)** *ωo* = *eωi* = 0.125 (300) = 37.5 rev/min *Ans*.

**(i)** Assuming no losses, *Po* = *Pi* = 2.98 kW *Ans*.

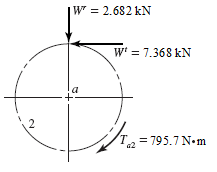
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**13-39** Given: *m* = 12 mm, *nP* = 1800 rev/min cw,

*N*2 = 18*T*, *N*3 = 32*T*, *N*4 = 18*T*,*N*5 = 48*T*

Pitch Diameters: *d*2 = 18(12) = 216mm, *d*3 = 32(12) = 384mm,

*d*4 = 18(12) = 216mm, *d*5 = 48(12) = 576mm

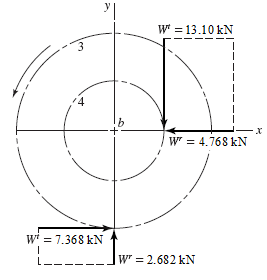


*Gear* 2

From Eq. (13-36),





*Gears* 3 *and* 4







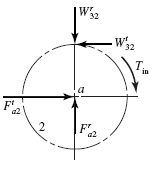
*Ans.*

\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**13-40** Given: *P* = 5.08 teeth/mm, *N*2 = 18*T*, *N*3 = 45*T*,

*H* = 23.86 kW,*n*2 = 1800 rev/min

*Gear* 2









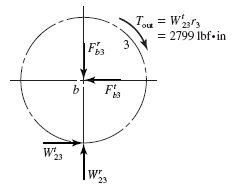








Each bearing on shaft *a* has the same radial load of *RA* = *RB* = 71.42/2 = 35.7 kN.



*Gear* 3









Each bearing on shaft *b* has the same radial load which is equal to the radial load of bearings *A* and *B*. Thus, all four bearings have the same radial load of 357 kN. *Ans.*

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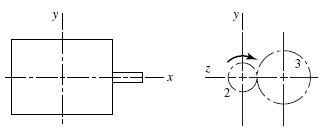
**13-41** Given: *P* = 4 teeth/in (0.16 teeth/mm), *NP* = 20*T*, *n*2 = 900 rev/min, and 200% overload.

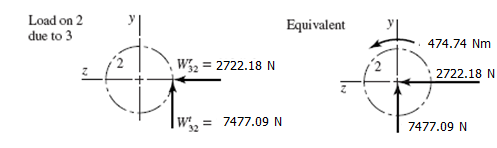




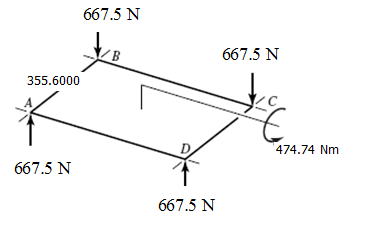








The motor mount resists the equivalent forces and torque.

The radial force due to torque is

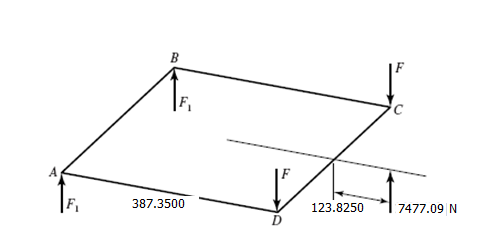


Forces reverse with rotationalsense as

torque reverses.

The compressive loads at *A* and *D* are absorbed by the base plate, not the bolts. For  the tensions in *C* and *D* are

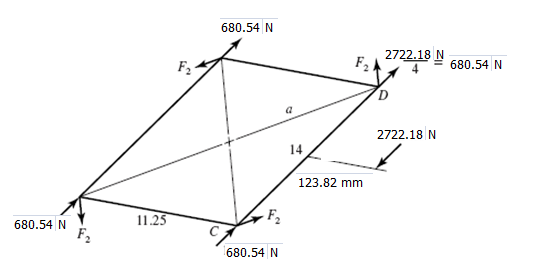




If  reverses, 387.35 mm changes to 336.55 mm, 122.30 mm changes to 73.02 mm, and the forces change direction. For *A* and *B*,



For ,









At *C* and *D*, the shear forces are:



At *A* and *B*, the shear forces are:



The shear forces are independent of the rotational sense.

The bolt tensions and the shear forces for cw rotation are,

|  |  |  |
| --- | --- | --- |
|  | Tension (N) | Shear (N) |
| A | 0 | 644.96 |
| B | 0 | 644.96 |
| C | 4932.83 | 1334.4 |
| D | 4932.83 | 1334.4 |

For ccw rotation,

|  |  |  |
| --- | --- | --- |
|  | Tension (N) | Shear (N) |
| A | 811.32 | 644.96 |
| B | 811.32 | 644.96 |
| C | 0 | 1334.4 |
| D | 0 | 1334.4 |

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**13-42 (a)** *N*2 = *N*4 = 15 teeth, *N*3 = *N*5 = 44 teeth, P=6teeth/in (0.236 teeth/mm)







**(b)** 

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**(c)** Input gears:



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Output gears:



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**(d)** 

**(e)** 

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**13-43** 

(0.394 teeth/mm)

**(a) ** *Ans.*

****  *Ans.*

**(b)** 





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**(c)** 

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**(d)** 

**(e)** 

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**13-44 (a)** For  from Eq. (13-11), with *m* = 2, *k* = 1, 



So 

**(b) **

**(c)**To transmit the same power with no change in pitch diameters, the speed and transmitted force must remain the same.

For *A*, with** = 20°,

*WtA* = *FA*cos20° = 1334.40cos20° = 1253.89 N

For *A*, with ** = 25°, same transmitted load,

*FA* = *WtA*/cos25° = 1253.89/cos25° = 1383.33 N *Ans.*

Summing the torque about the shaft axis,







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**13-45 (a)** For  from Eq. (13-11), with *m* = 5, *k* = 1, 



So 

**(b) **

**(c)**To transmit the same power with no change in pitch diameters, the speed and transmitted force must remain the same.

For *A*, with ** = 20°,

*WtA* = *FA*cos20° = 11 cos20° = 10.33 kN

For *A*, with ** = 25°, same transmitted load,

*FA* = *WtA*/cos25° = 10.33 / cos 25° = 11.40 kN *Ans.*

Summing the torque about the shaft axis,







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**13-46 (a)**Using Eq. (13-11) with *k* = 1, **= 20º, and *m* = 2,

****

Round up for the minimum integer number of teeth.

*NF* = 15 teeth, *NC* = 30 teeth *Ans.*

**(b)** 

**(c)** 

**(d)** From Eq. (13-36),

 *Ans.*

Or, we could have obtained *Wt*directly from the torque and radius,

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**13-47 (a)**Using Eq. (13-11) with *k* = 1, **= 20º, and *m* = 2,

****

Round up for the minimum integer number of teeth.

*NC* = 15 teeth, *NF* = 30 teeth *Ans.*

**(b)** 

**(c)**  *Ans.*

**(d)** From Eqs. (13-34) and (13-35),



 *Ans.*

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**13-48 (a)** Eq. (13-14):  *Ans.*

**(b)** Eq. (13-34):  *Ans.*

**(c)** Eq. (13-35):  *Ans.*

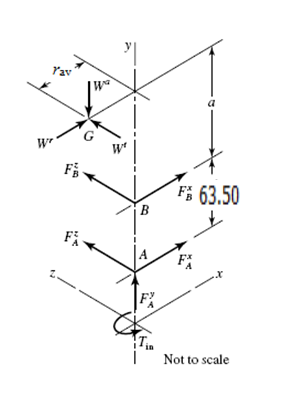
Eq. (13-38):  *Ans.*

Eq. (13-38):  *Ans.*

The tangential and axial forces agree with Prob. 3-85, but the radial force given in Prob. 3-85 is shown here to be incorrect. *Ans.*

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**13-49** 





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**W =** 571.28**i**– 285.64**j** + 1755.21**k** N

**R***AG* **=** –42.29**i** + 131.34**j**, **R***AB***=** 2.5**j**

****

Solving gives





So,









So, **F**B = −991.01**i**−3627.79**k** N *Ans.*



**F***A* = – (**F***B* + **W**)

= – (–991.01**i** – 3627.79**k** + 571.12**i** – 285.56**j** + 1753.85**k**)

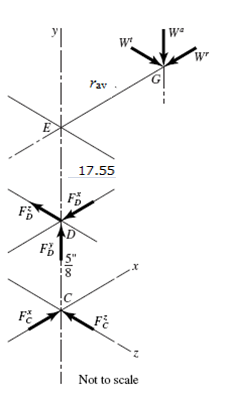
= 419.89**i** + 285.56**j** + 1873.94**k** *N Ans.*





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**13-50** P=10 teeth/in (0.394 teeth /mm)

** **

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**W** = –20.82**i** – 34.69**j** +111.20**k**

**R***DG* = 32.66**i** + 17.55**j**

**R***DC* = –15.88**j**



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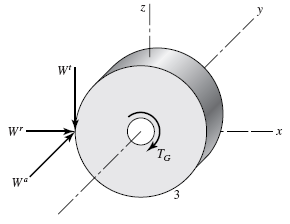
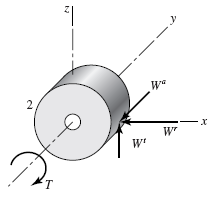
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**13-51**

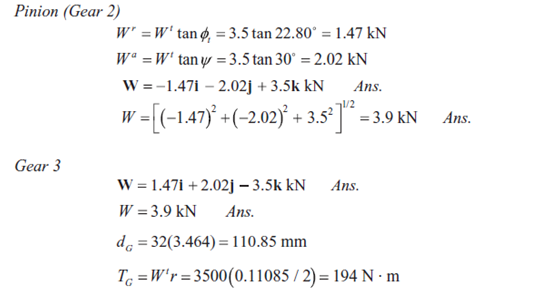
NOTE: The shaft forces exerted on the gears are not shown in the figures above.



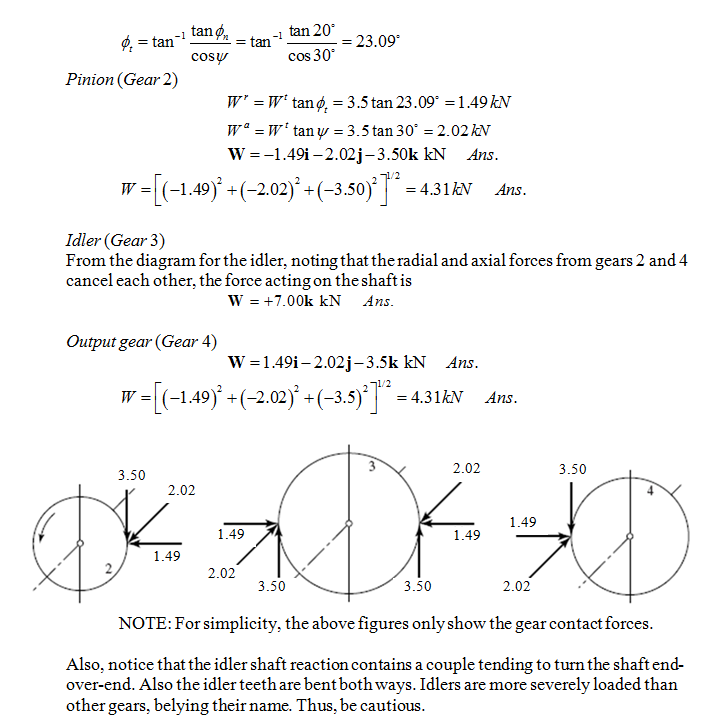




The forces on the shafts will be equal to the forces transmitted to the gears through the meshing teeth.



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**13-52** 

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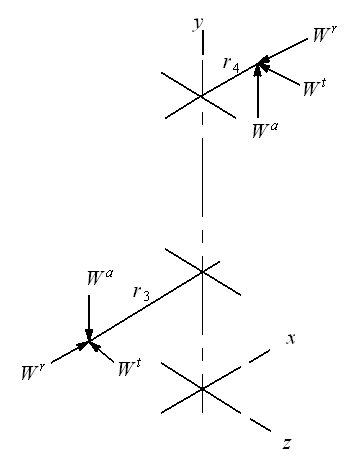
**13-53** *Gear* 3: **Using**













*Gear* 4:



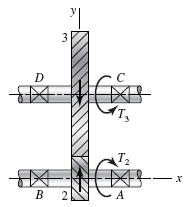








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**13-54 Use **

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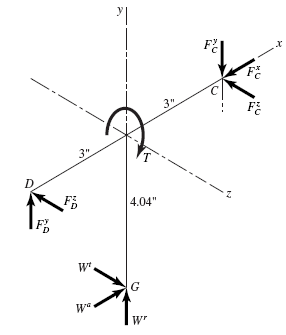
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 (1)

**** Nmm

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Substituting and solving Eq. (1) gives











Substituting and solving gives





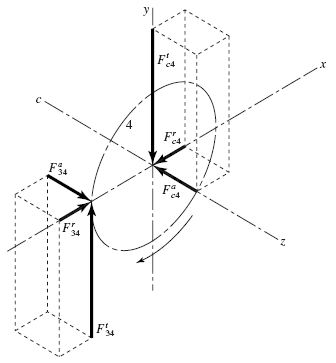
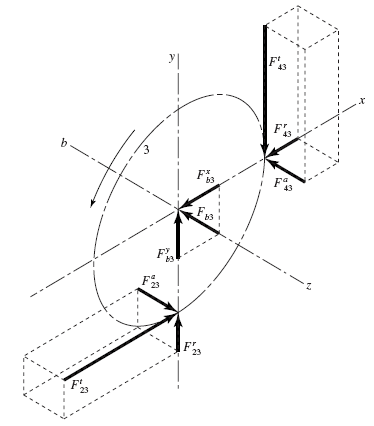


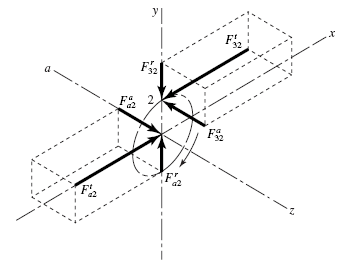




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**13-55**

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Since the transverse pressure angle is specified, we will assume the given module is also in terms of the transverse orientation.













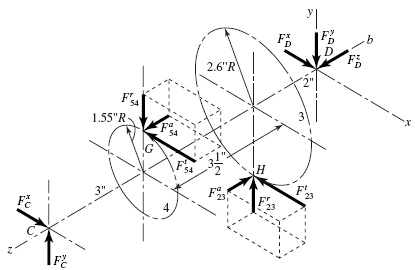


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**13-56**

**Use **

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For gears 2 and 3: 

For gears 4 and 5: 













Next, designate the points of action on gears 4 and 3, respectively, as points *G* and *H*, as shown. Position vectors are

****

****

****

Force vectors are









Now, a summation of moments about bearing *C* gives

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The terms for this equation are found to be

**** Nm

****

****

When these terms are placed back into the moment equation, the **k** terms, representing the shaft torque, cancel. The **i** and **j** terms give





Next, we sum the forces to zero.

****

Substituting, gives

****

****

Solving gives



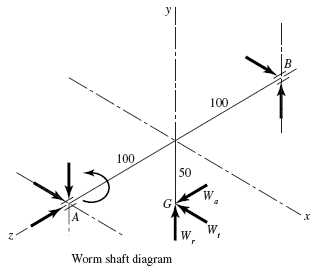




So, **F**C = 6961.12**i** + 2989.06**j** N *Ans*.

**F**D = 7161.28**i**− 1890.40**j** + 684.99**k** N *Ans*.

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**13-57 **

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In ft/min: *VS* = 3.28(3.152) = 10.33 ft/s = 620 ft/min

Use *f* = 0.043 from curve A of Fig. 13-38. Then, from the first of Eq. (13-43)







The force acting against the worm is

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Thus,*A* is the thrust bearing. *Ans.*

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Substituting and solving gives





So ****

Or 



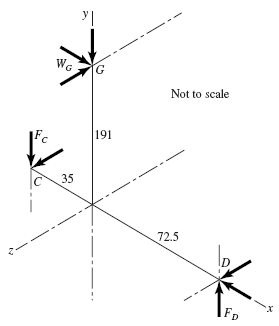


Radial 



Thrust 

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**13-58** From Prob. 13-57,

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So 

Bearing *D*takes the thrust load.

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Putting it together and solving,



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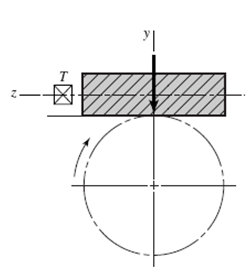
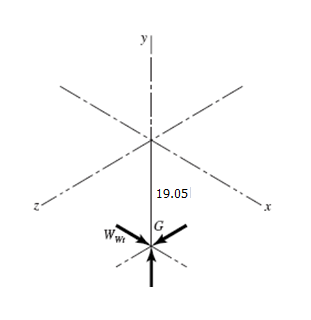
Radial 

Or 



\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_\_

**13-59**

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So ****

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**13-60** Computer programs will vary.