**Chapter 17**

**17-1** Begin with Eq. (17-10),

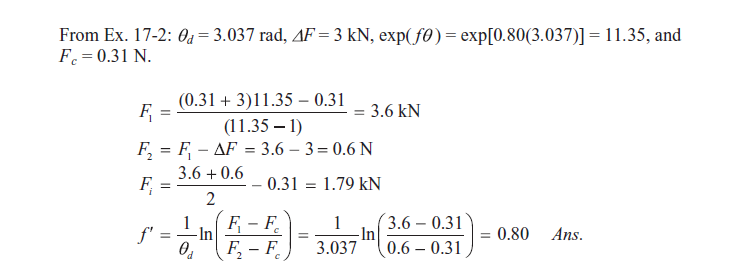


Introduce Eq. (17-9):

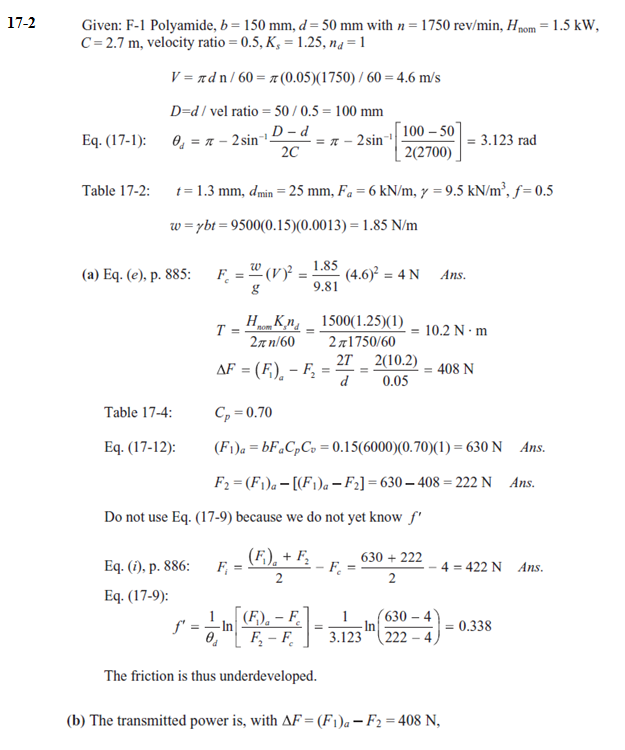


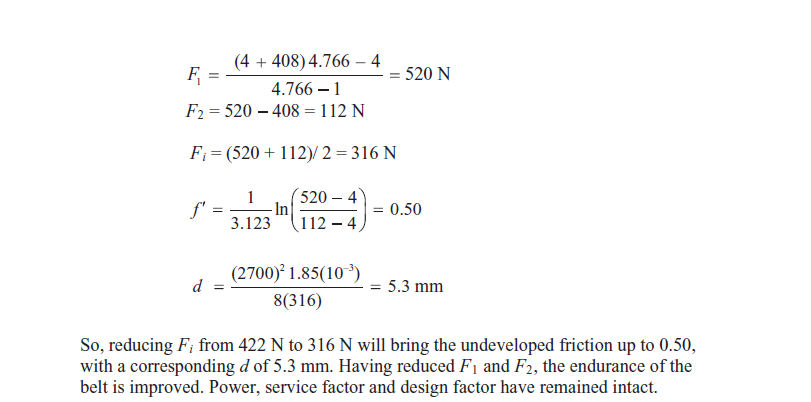
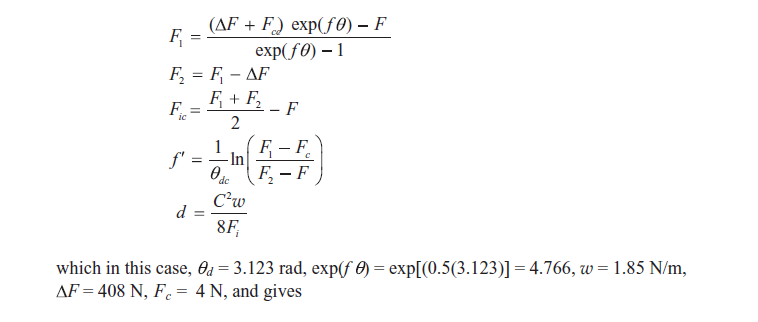
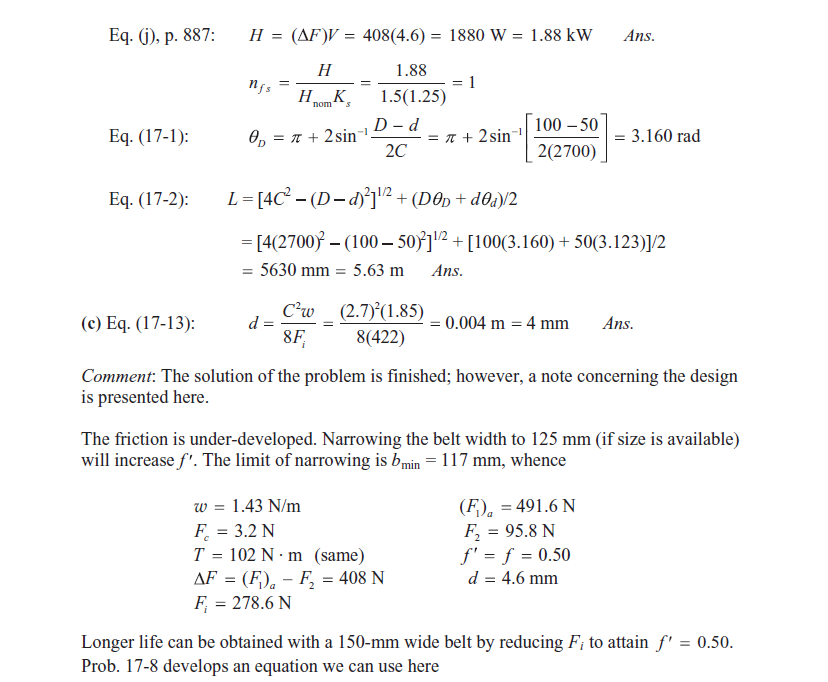
Now add and subtract 



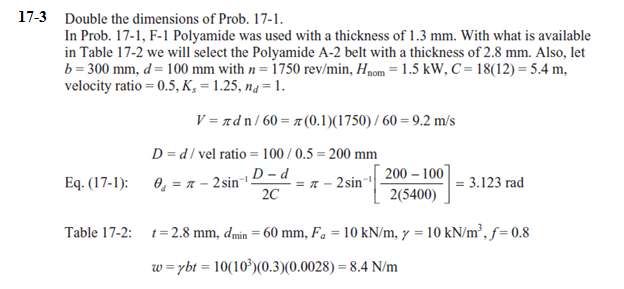


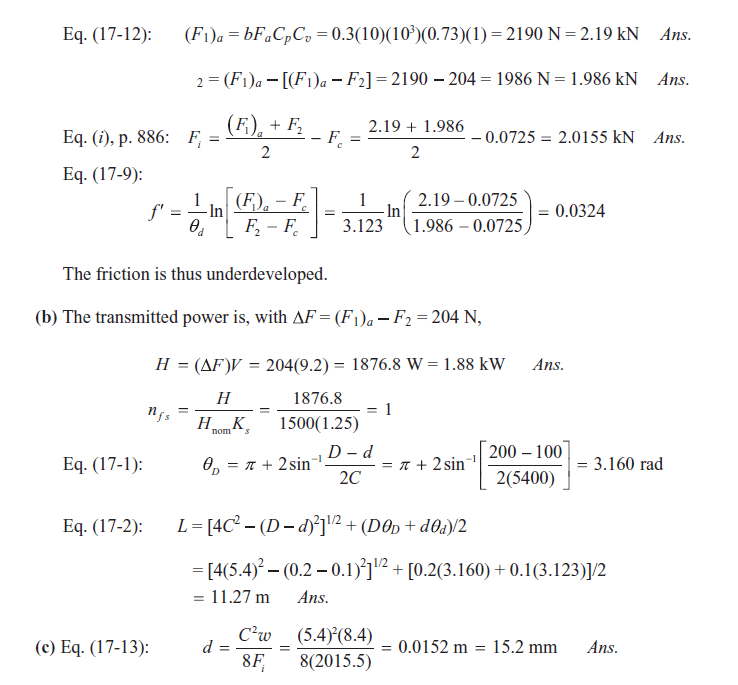
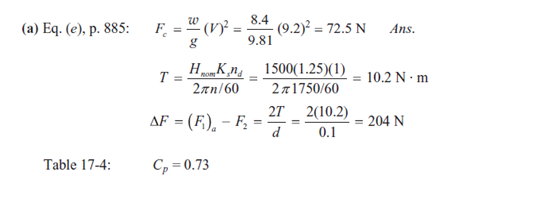
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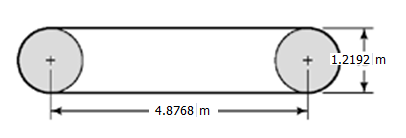
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**17-4**

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As a design task, the decision set just before Ex. 17-2 is useful.

A priori decisions:

• Function: *H*nom = 44.76 kW, *n* = 380 rev/min, *C* = 4.876 m, *Ks*= 1*.*1

• Design factor: *nd*= 1

• Initial tension: Catenary

• Belt material. Table 17-2: Polyamide A-3, *Fa* = 17512.7 N/m, *γ* = 6.6 N/m3, *f* = 0.8

• Drive geometry: *d* = *D* = 1.22 m

• Belt thickness: *t* = 3.3 mm

Design variable: Belt width.

Use a method of trials. Initially, choose *b* = 152.4 mm

Transmitted power *H*



Eq. (17-2): *L* = [4(4.876)2− (1.22 − 1.22)2]1/2 + [1.22(*π*) + 1.22(*π*)] / 2 = 13.58 m

Friction is not fully developed, so *b*min is just a little smaller than 6 in (0.1448 m ). Not having

a figure of merit, we choose the most narrow belt available (0.1524 m). We can improve the

design by reducing the initial tension, which reduces *F*1 and *F*2, thereby increasing belt life (see the result of Prob. 17-8).This will bring to 0.80



Therefore



These are small reductions since is close to *f* , but improvements nevertheless.



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**17-5** From the last equation given in the problem statement,



But 2*T/d* = 33 000*Hd/V*. Thus,



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**17-6** Refer to Ex. 17-1 for the values used below.

**(a)** The maximum torque prior to slip is,



The corresponding initial tension, from Eq. (17-9), is,



**(b)** See Prob. 17-4 statement. The final relation can be written



This is the minimum belt width since the belt is at the point of slip. The design must round up to an available width.

Eq. (17-1):



Eq. (17-2):



**(c) **



Transmitted belt power *H*



Dip: 

|  |  |  |
| --- | --- | --- |
|  | Ex. 17-1 | This Problem |
| *b* | 152.4 | 104.9 |
| *w* | 5.73 | 3.81 |
| *Fc* | 113.8 | 75.6 |
| (*F*1)*a* | 1868.1 | 1285.9 |
| *F*2 | 766.8 | 184.6 |
| *Fi* | 1203.6 | 659.6 |
|  | 0.33\* | 0.80\*\* |
| dip | 3.53 | 4.29 |

(**d**) If you only change the belt width, the parameters in the following table change as shown.

\*Friction underdeveloped

\*\*Friction fully developed

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**17-7** The transmitted power is the same.

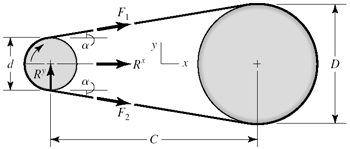
|  |  |  |  |
| --- | --- | --- | --- |
|  | *b* = 152.4 mm | *b* = 304.80 mm | *n*-Fold  Change |
| *Fc* | 114.09 | 228.18 | 2 |
| *Fi* | 1202.52 | 2957.48 | 2.46 |
| (*F*1)*a* | 1868.16 | 3736.32 | 2 |
| *F*2 | 766.84 | 2635.00 | 3.44 |
| *Ha* | 15.38 | 15.38 | 1 |
| *nfs* | 1.1 | 1.1 | 1 |
|  | 0.139 | 0.125 | 0.90 |
| dip | 8.33 | 2.90 | 0.34 |

If we relax *Fi*to develop full friction (*f* = 0*.*80) and obtain longer life, then

|  |  |  |  |
| --- | --- | --- | --- |
|  | *b* = 152.4 mm | *b* = 304.80 mm | *n*-Fold  Change |
| *Fc* | 113.87 | 228.18 | 2 |
| *Fi* | 658.75 | 658.75 | 1 |
| *F*1 | 1323.72 | 1437.59 | 1.09 |
| *F*2 | 222.40 | 336.27 | 1.51 |
|  | 0.80 | 0.80 | 1 |
| dip | 6.48 | 12.78 | 2 |

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**17-8**

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Find the resultant of *F*1 and *F*2:



From Ex. 17-2, *d* = 0.4064 m , *D* = 0.9144 m, *C* = 0.4064(0.3048) = 4.876 m, *F*1 = 4181.12 N, *F*2 = 1227.65 N



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**17-9** This is a good class project. Form four groups, each with a belt to design. Once each group agrees internally, all four should report their designs including the forces and torques on the line shaft. If you give them the pulley locations, they could design the line shaft.

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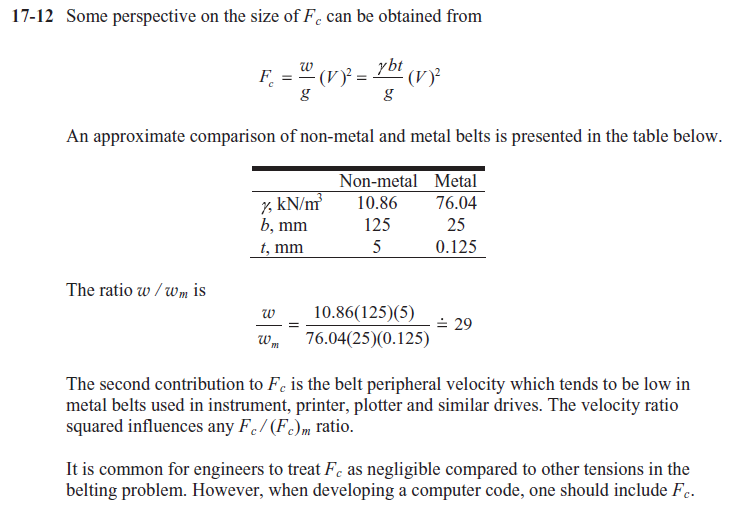
**17-10** If you have the students implement a computer program, the design problem selectionsmay differ, and the students will be able to explore them. For *Ks*= 1*.*25, *nd*= 1*.*1, *d* = 355.6 mm and *D* = 711.20 mm, a polyamide A-5 belt, 203.2 mm wide, will do (*b*min = 167 mm)

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**17-11** An efficiency of less than unity lowers the output for a given input. Since the object of the drive is the output, the efficiency must be incorporated such that the belt’s capacity is increased. The design power would thus be expressed as



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**17-13** Eq. (17-8):



Assuming negligible centrifugal force and setting *F*1 = *ab* from step 3, just before Ex. 17-3,

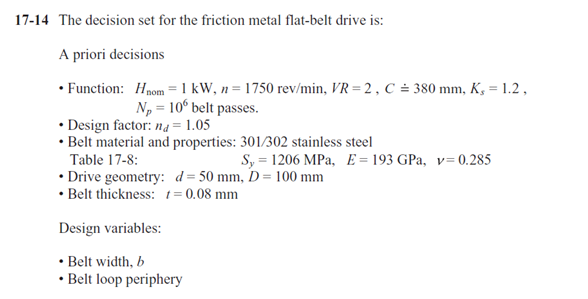


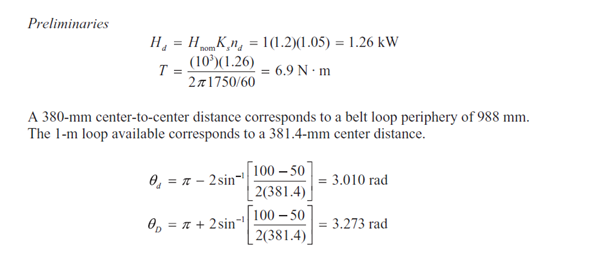
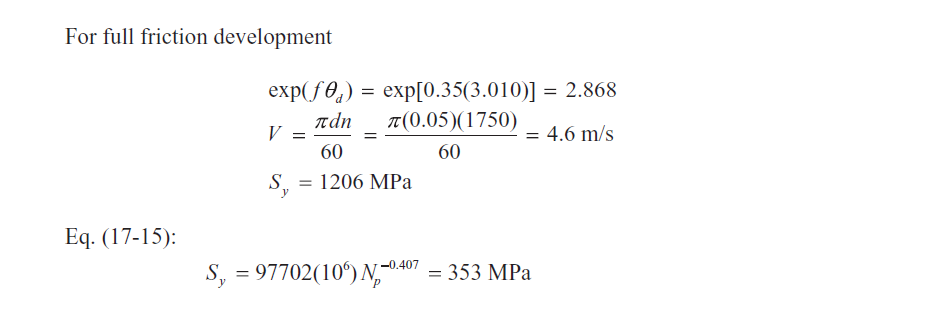
Also, 

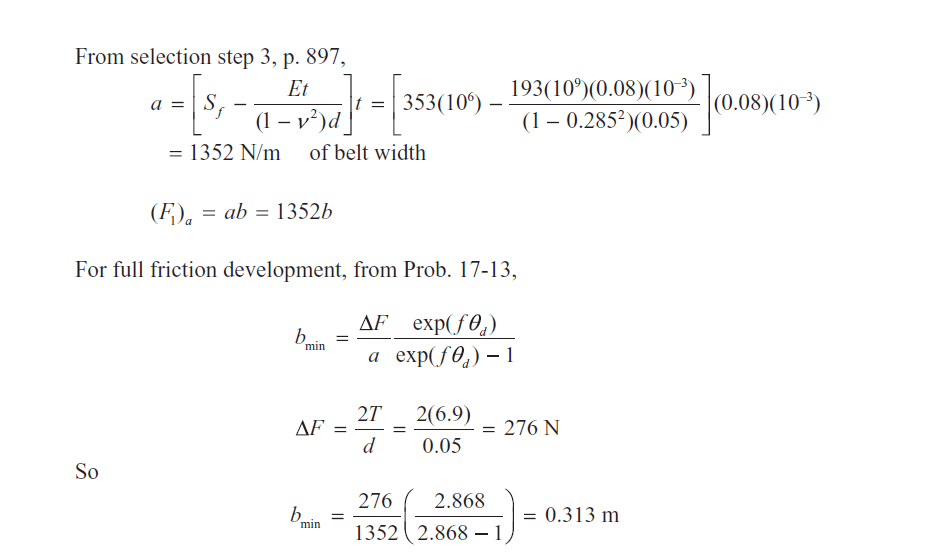


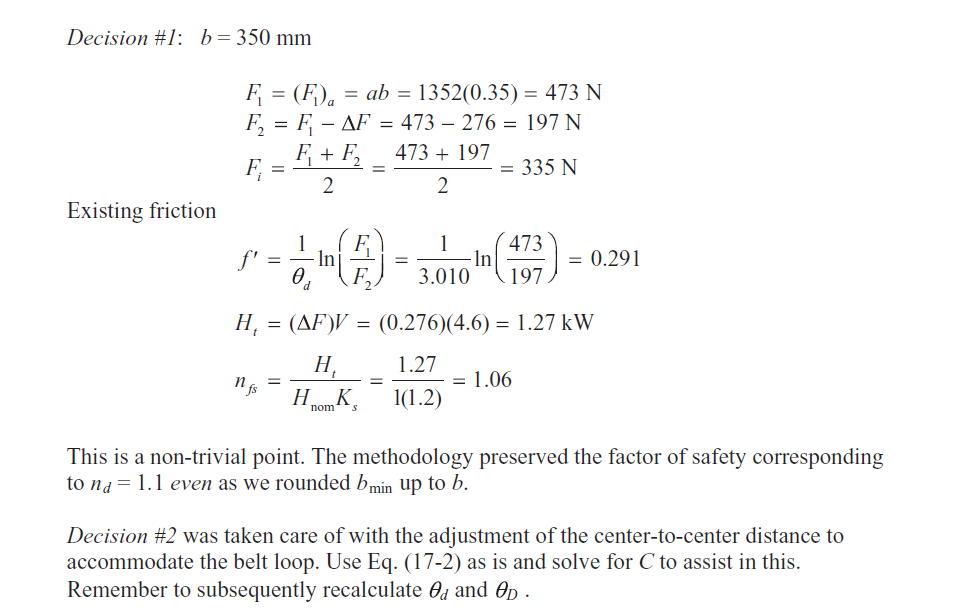
Substituting into Eq. (1), 

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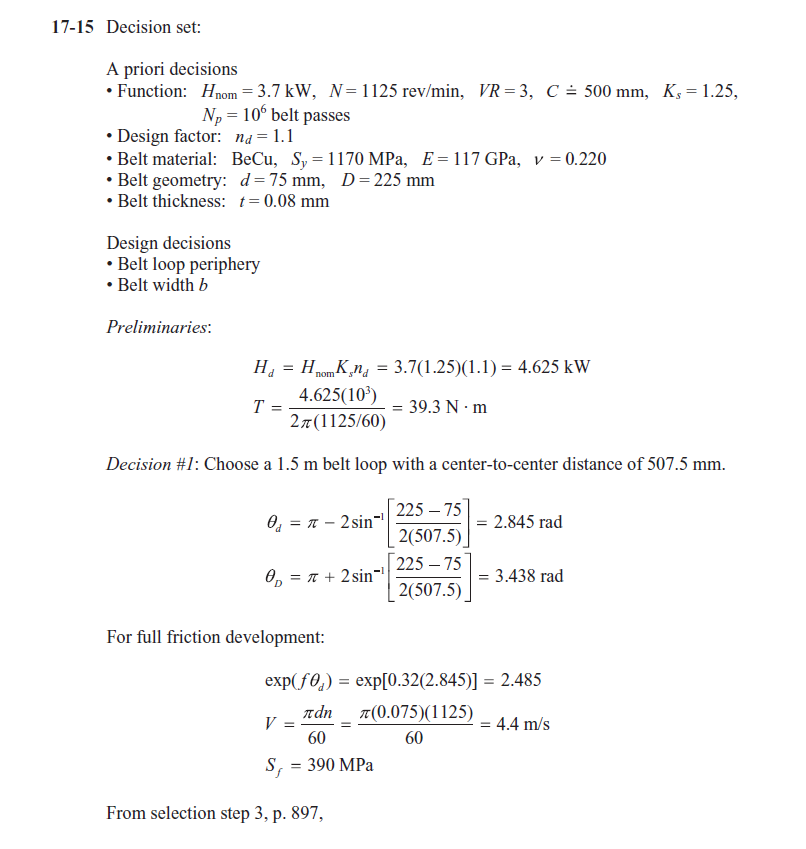


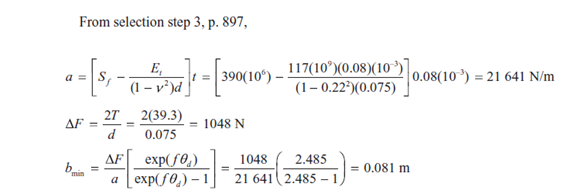
 

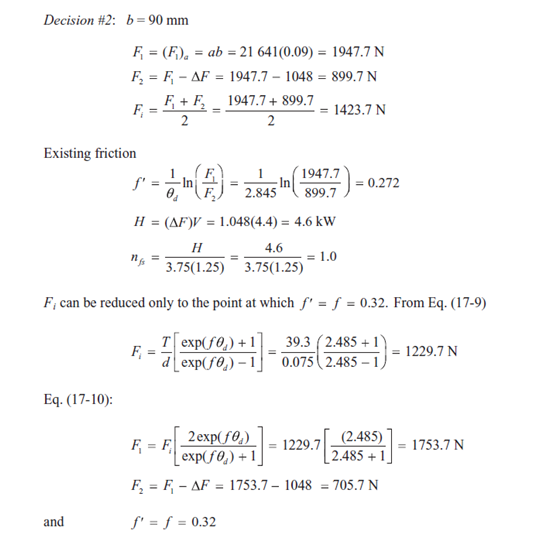




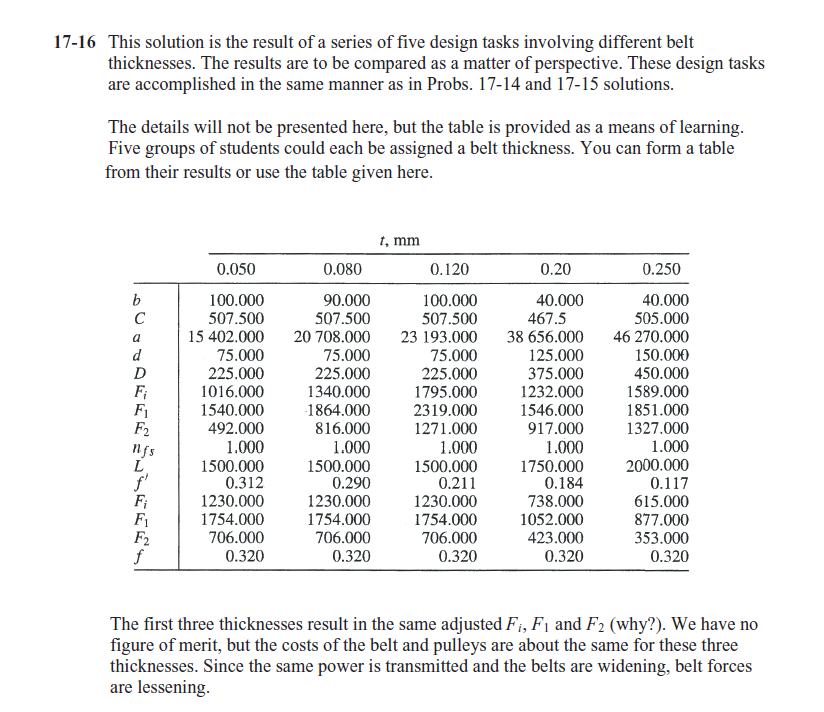
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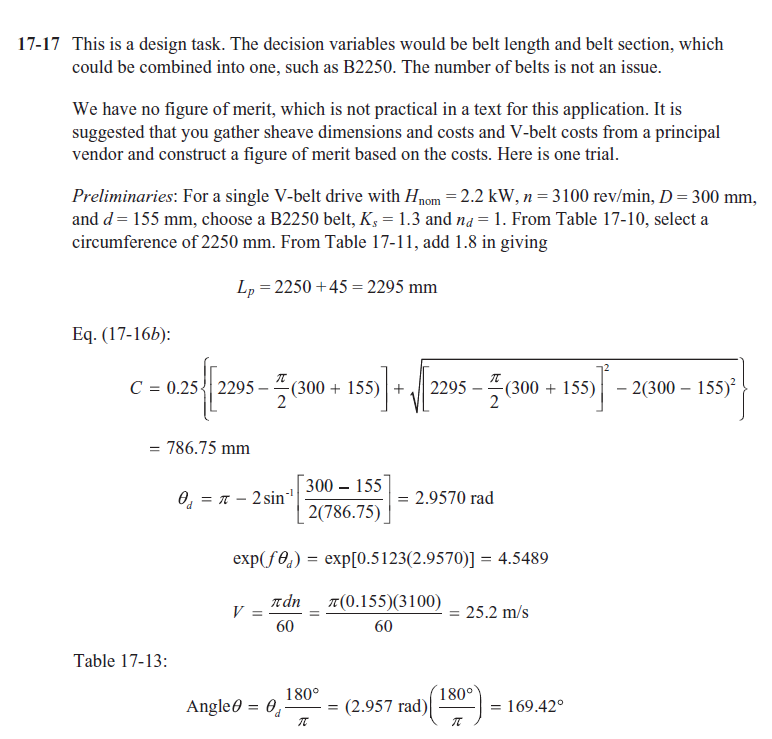


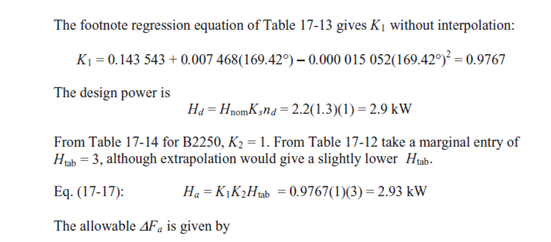


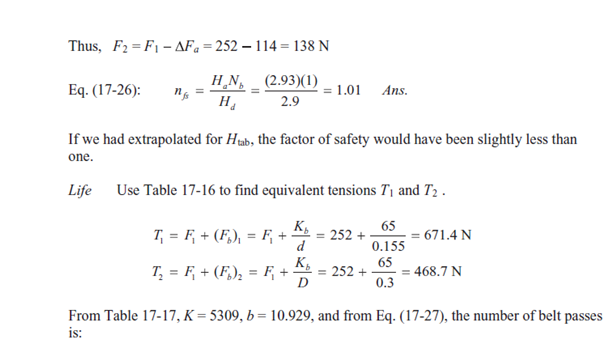
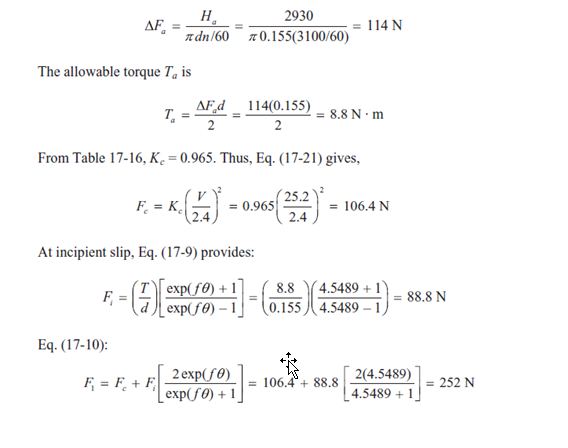
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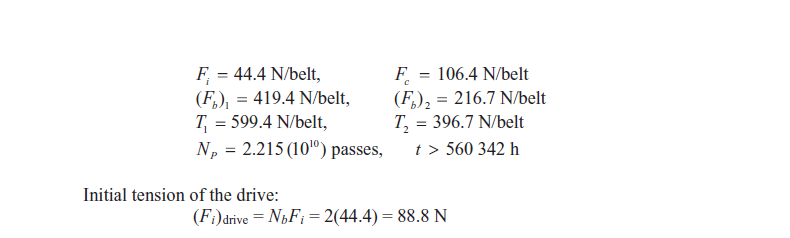
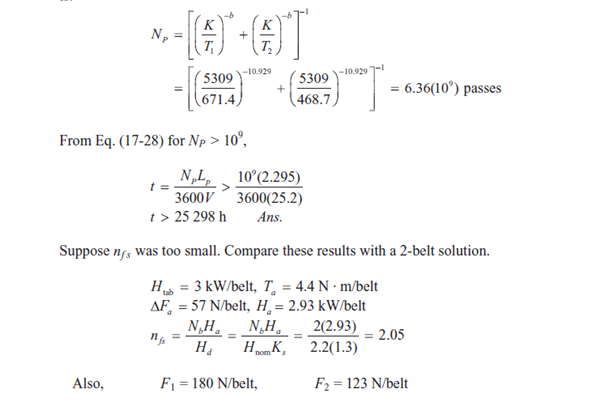


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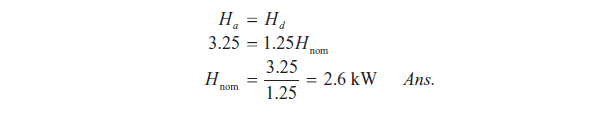
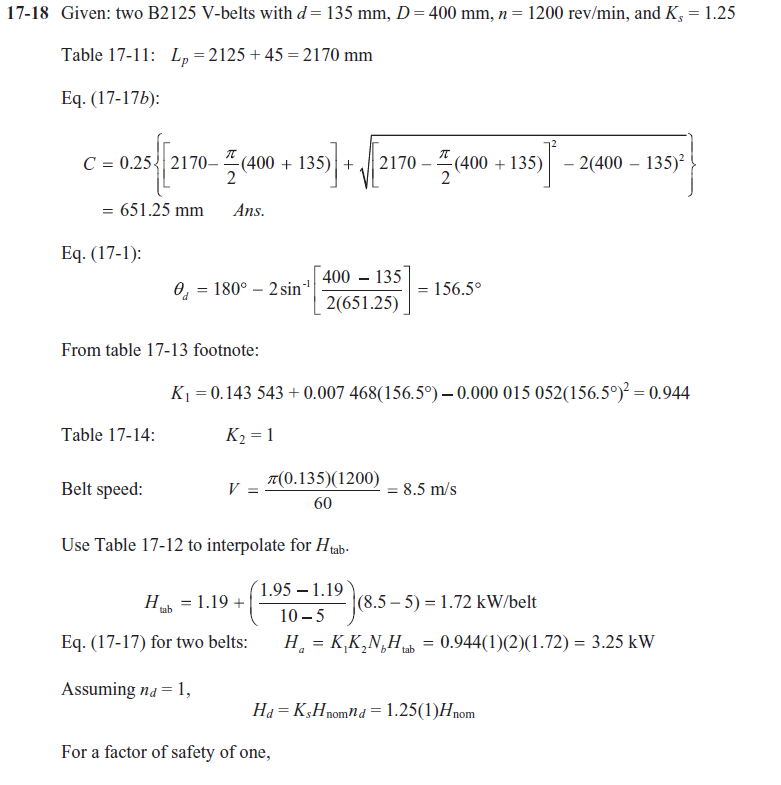




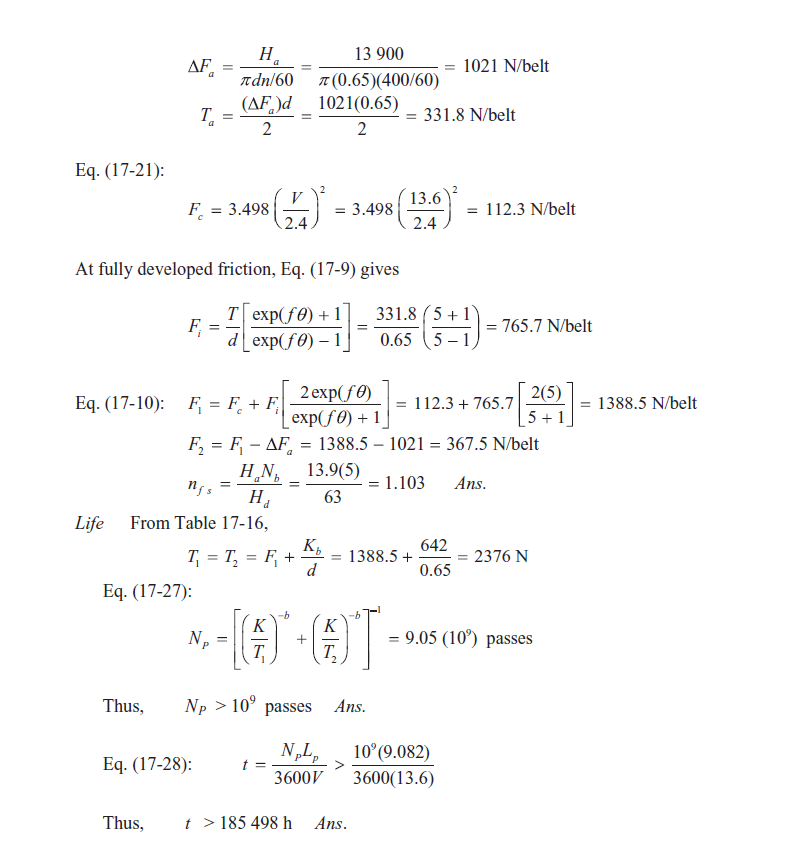
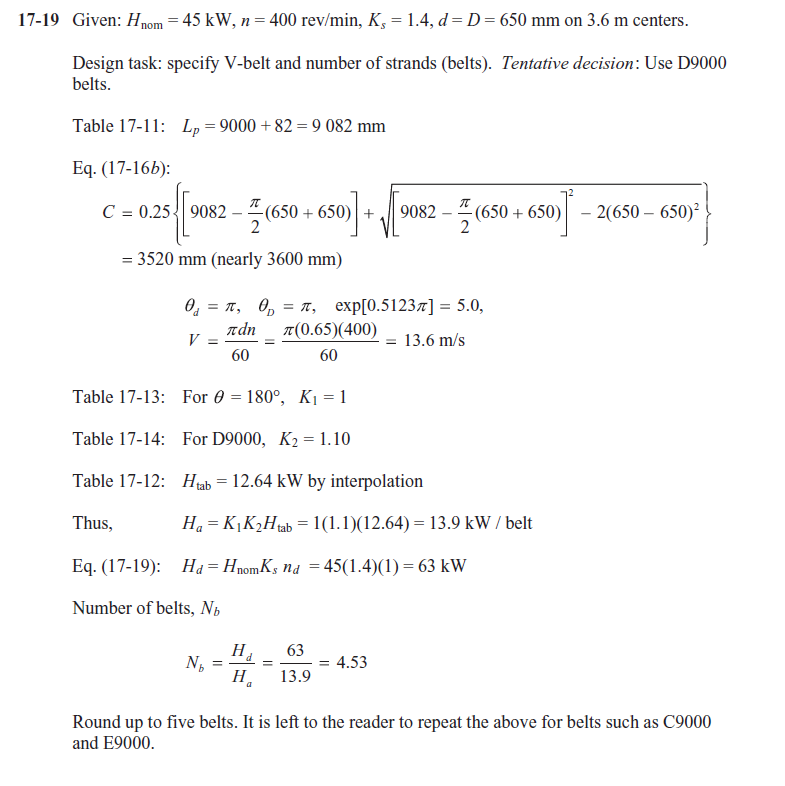


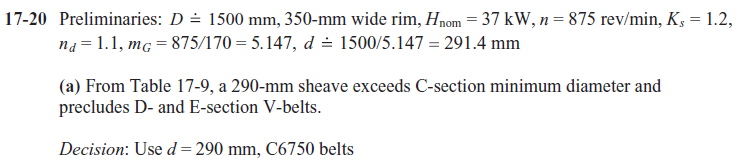


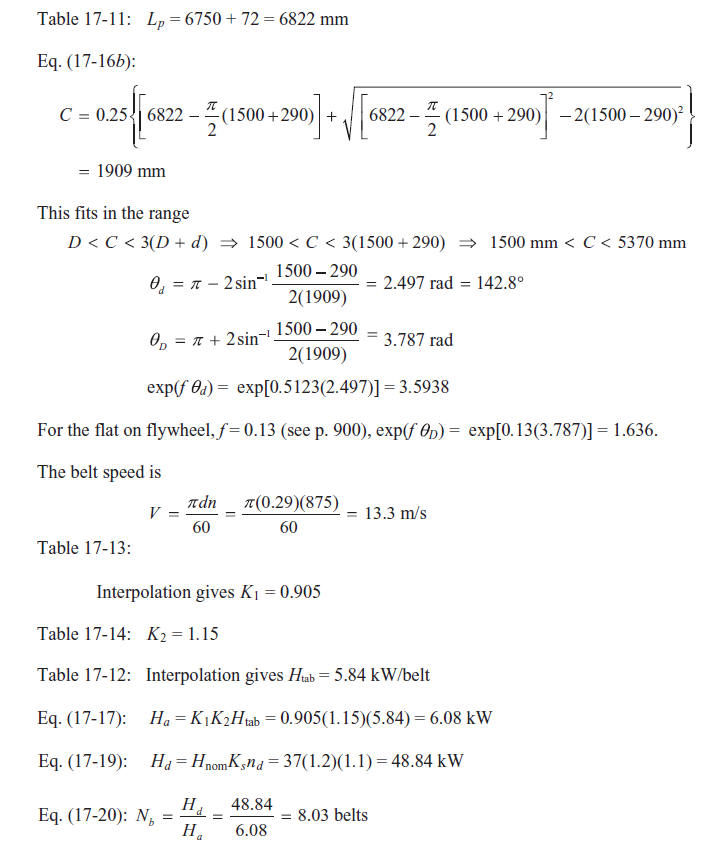
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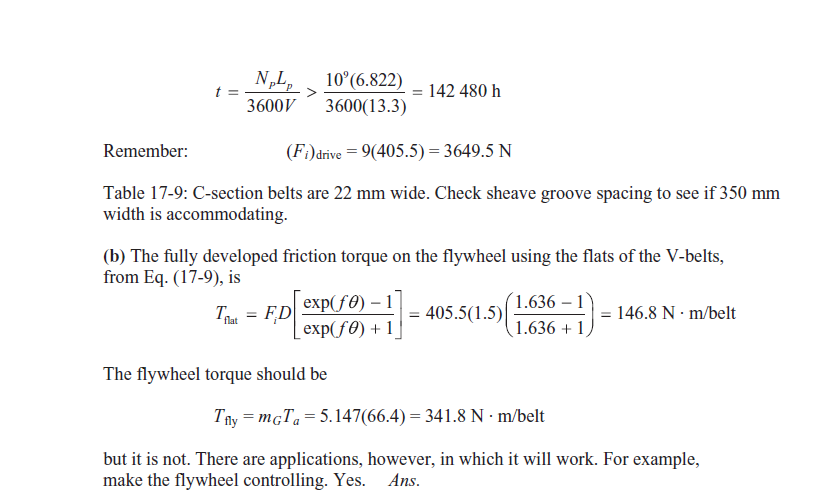
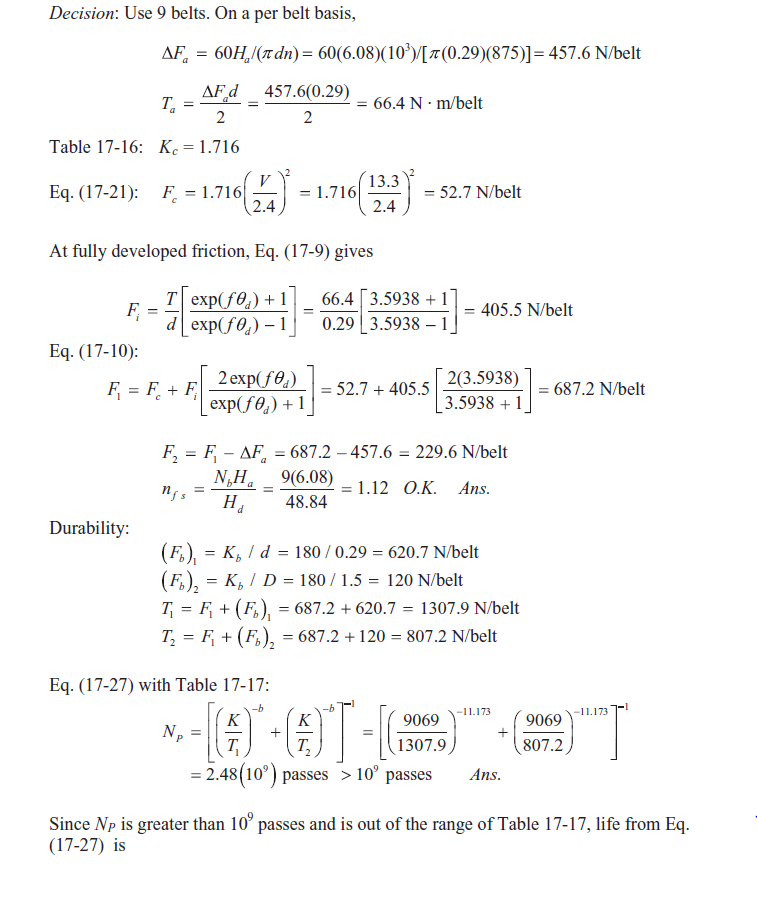


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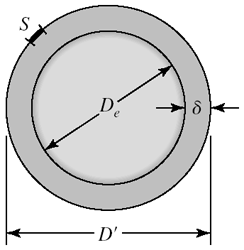






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**17-21**

** (a)**

*S* is the spliced-in string segment length

*De*is the equatorial diameter  
 is the spliced string diameter *δ*is the radial clearance

*S* + *πDe*= *π* = *π*(*De*+ 2*δ*) = *πDe*+ 2*πδ*

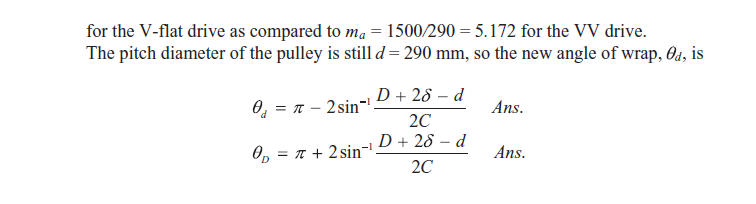
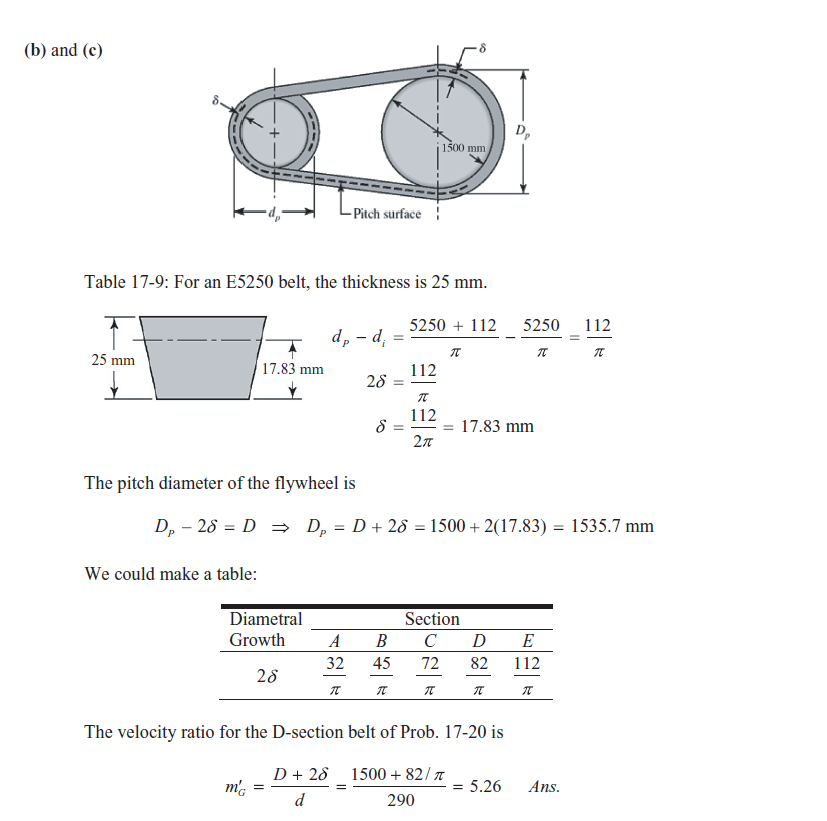
From which



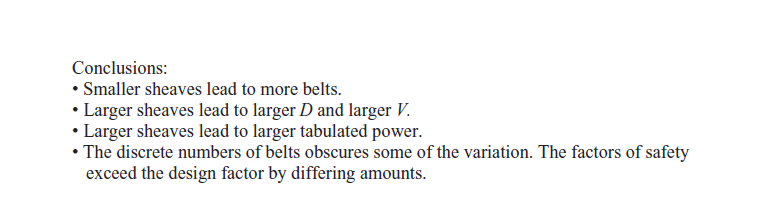
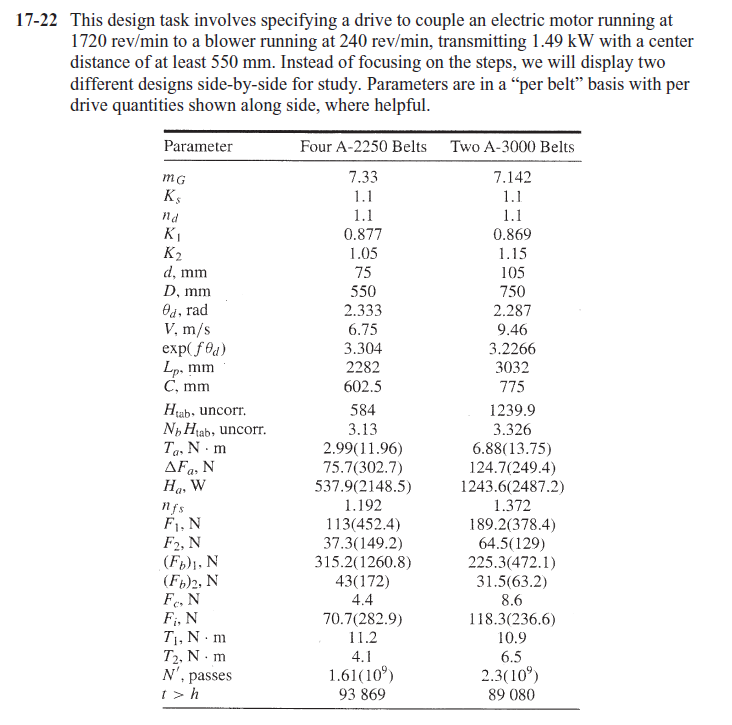
The radial clearance is thus *independent* of *De*.



This is true whether the sphere is the earth, the moon or a marble. Thinking in termsof a radial or diametral increment removes the basic size from the problem.

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**17-23** In Ex. 17-5 the selected chain was 140-3, making the pitch of this 140 chain14*/*8 = 1*.*75 in. Table 17-19 confirms.

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**17-24** **(a)** Eq. (17-32): 

Eq. (17-33): 

Equating and solving for *n*1 gives



**(b)** For a No. 60 chain, *p* = 6*/*8 = 19 mm, *N*1 = 17, *Kr*= 17

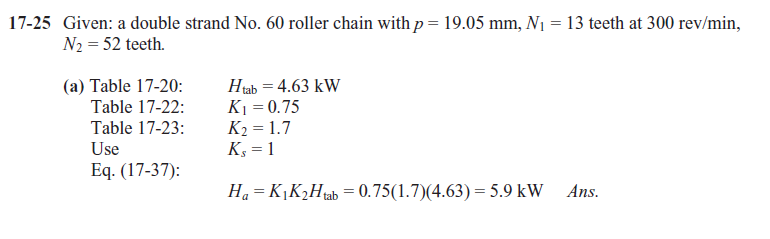


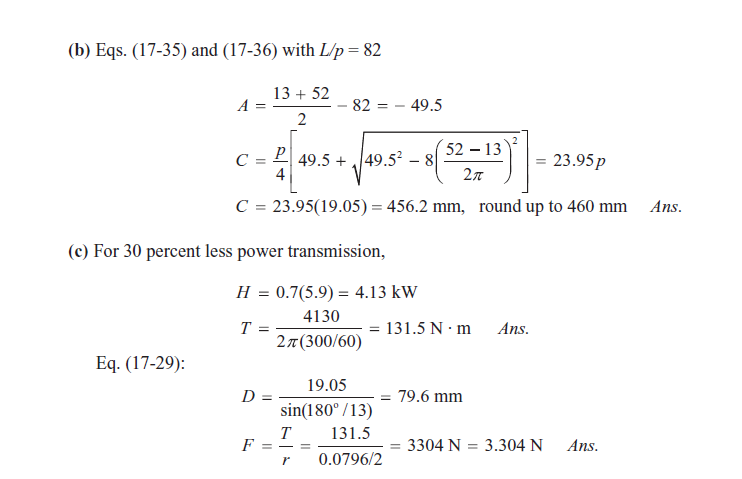
Table 17-20 confirms that this point occurs at 1200 ± 200 rev/min.

**(c)** Life predictions using Eq. (17-40) are possible at speeds greater than 1227 rev/min.

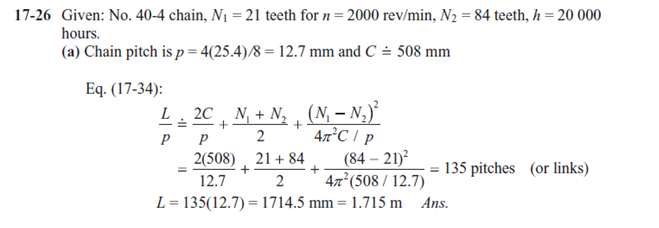
*Ans.*

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**(b)** Table 17-20: No.40 chain, at 2000 rpm, *H*tab = 5.76 kW

Table 17-22: Assuming for now the “post-extreme”, that is, failure by fatigue of the roller,



Table 17-23: Four strands, *K*2 = 3*.*3

Eq. (17-37): *Ha = K*1*K*2*H*tab = 1.37(3.3)5.76 = 26.04 kW *Ans.*

Note that this table value does not account for the differences from the assumptions for chain length and life.

**(c)** Eq. (17-32):



To compare with the tabulated value from part (b), adjust with *K*2 for the number of strands. It is not necessary to adjust with *K*1, since the number of teeth is included in Eq. (17-37).



This is larger than the table value from part (a), probably indicating that *H*2 is the limiting power in the table.

Eq. (17-33): 

Adjusting for the number of strands,

*Ha = K*2*H*2 = 3.3(7.83) = 25.86 kW *Ans.*

This compares pretty well to the adjusted table value from part (b) of 26.04 kW. Note that this still doesn’t account for the desired chain length or life being different from the table assumptions.

**(d)** Eq. (17-39):



Adjusting for the 4 strands,

 *Ans.*

This is only slightly different from the 25.89 kW from part (c). Note that accounting for the longer chain length increased the power rating slightly, while accounting for the longer life decreased the power rating slightly.

**(e)** Eq. (17-30):  *Ans.*

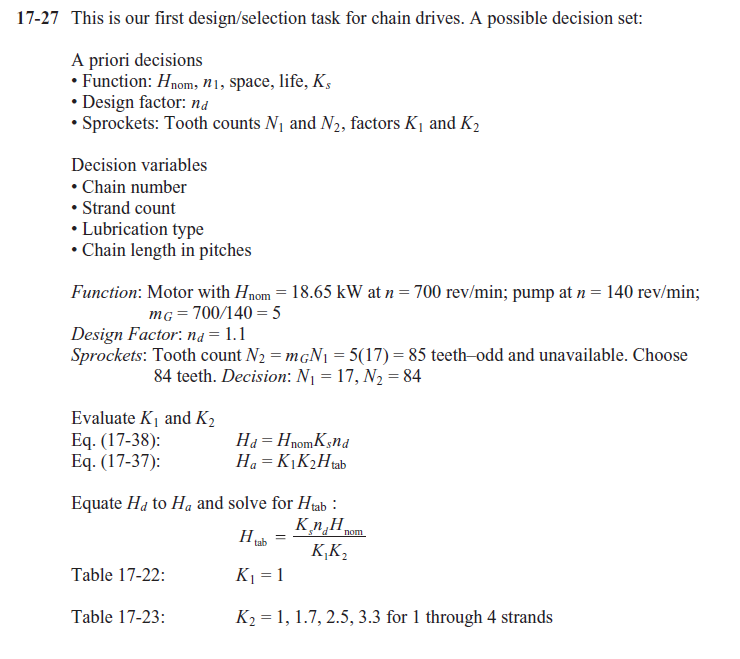
**(f)** From the standard relationship of power, torque, and speed, *H*=*T*

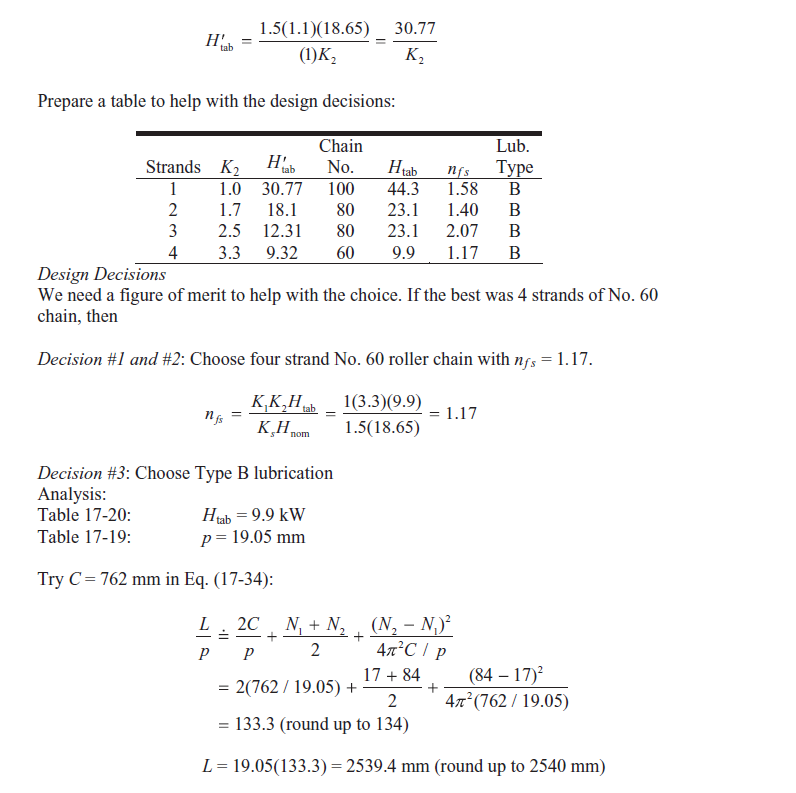
 *Ans.*

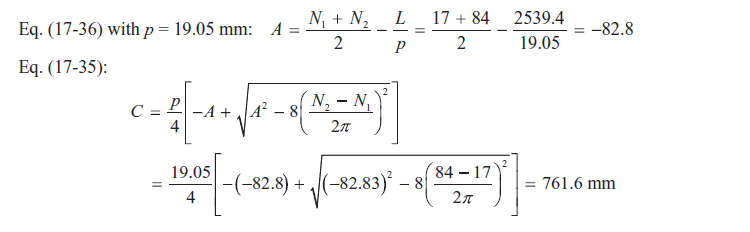
**(g)** Eq. (17-29): 

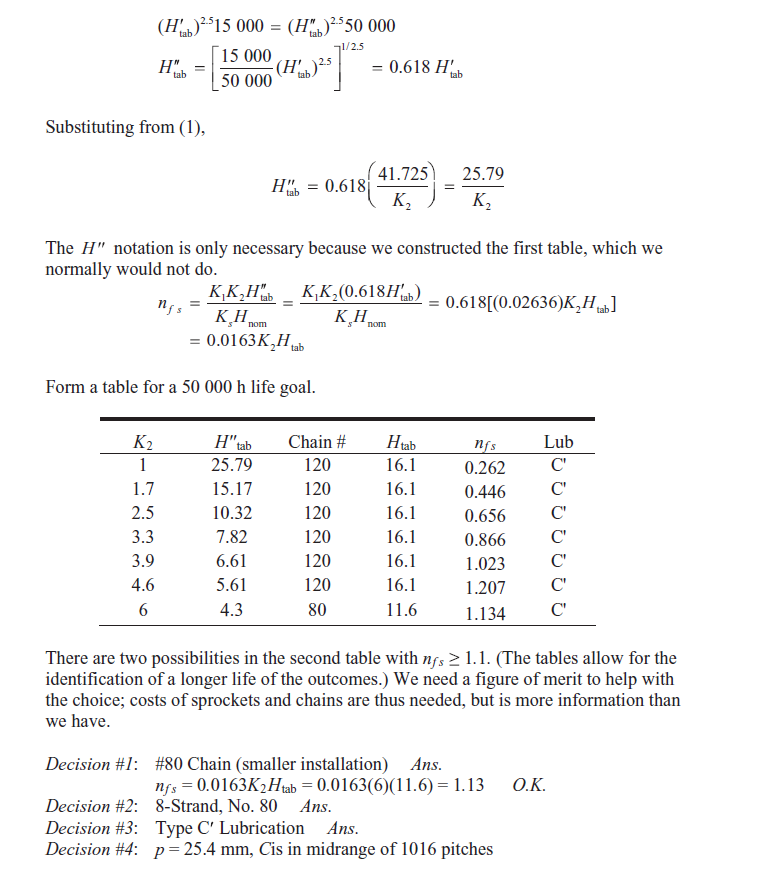
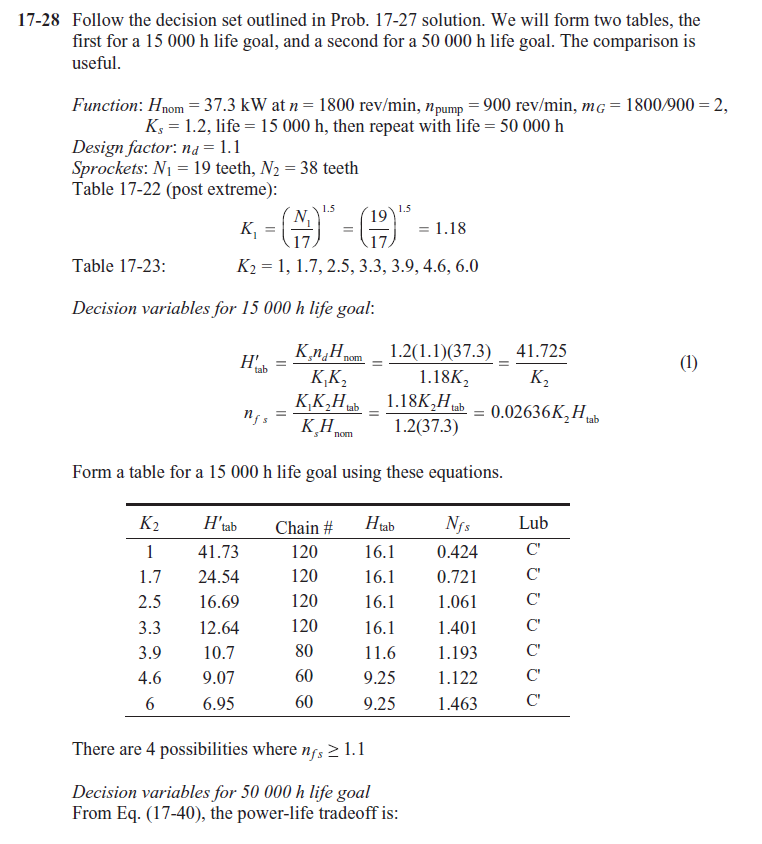


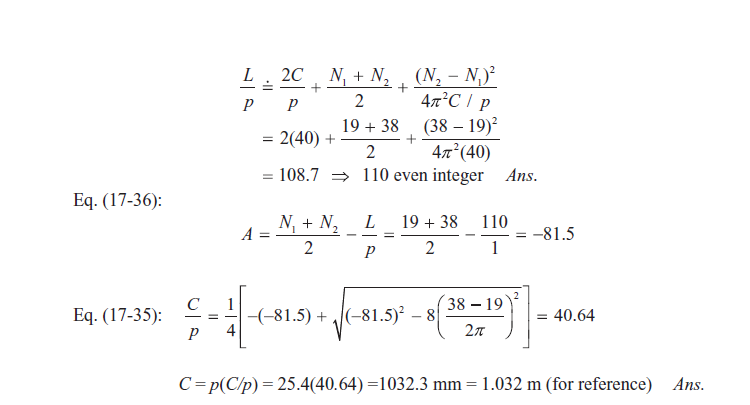
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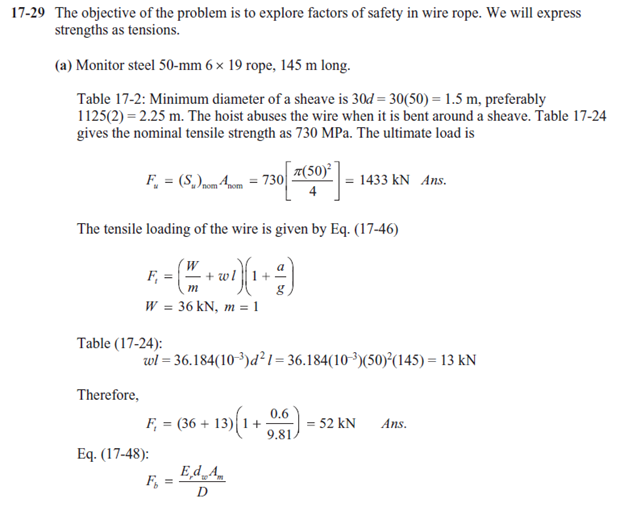


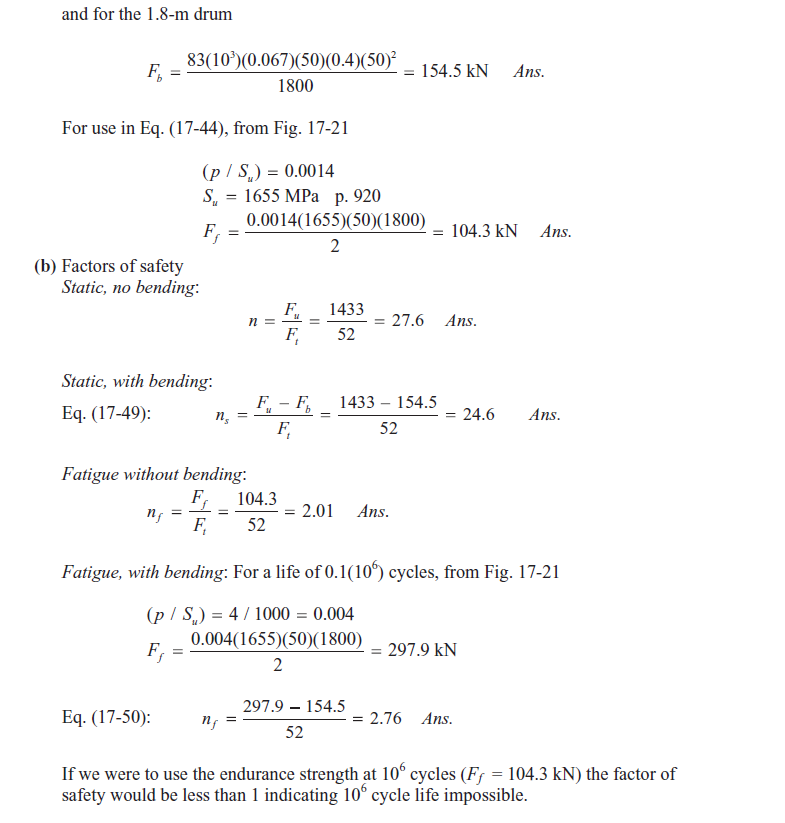
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Comments:

• There are a number of factors of safety used in wire rope analysis. They are different,

with different meanings. There is no substitute for knowing exactly which factor

of safety is written or spoken.

• Static performance of a rope in tension is impressive.

• In this problem, at the drum, we have a finite life.

• The remedy for fatigue is the use of smaller diameter ropes, with multiple ropes

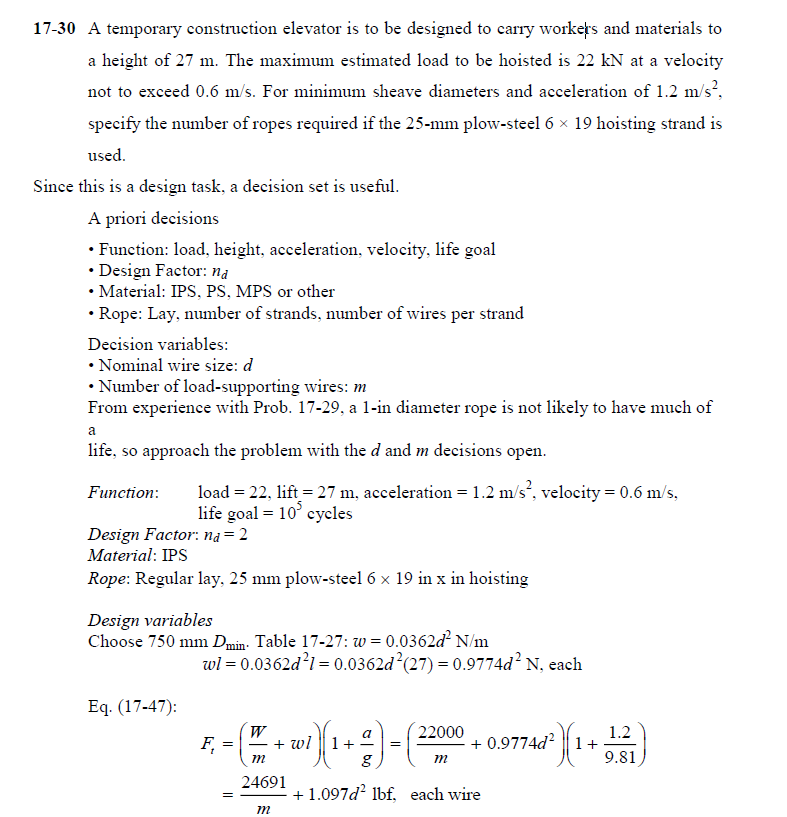
supporting the load. See Ex. 17-6 for the effectiveness of this approach. It will also

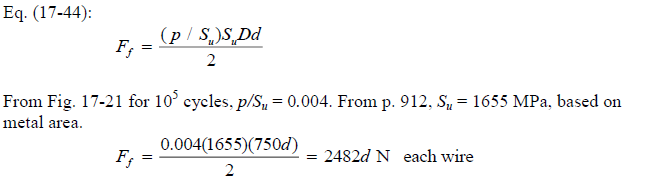
be used in Prob. 17-30.

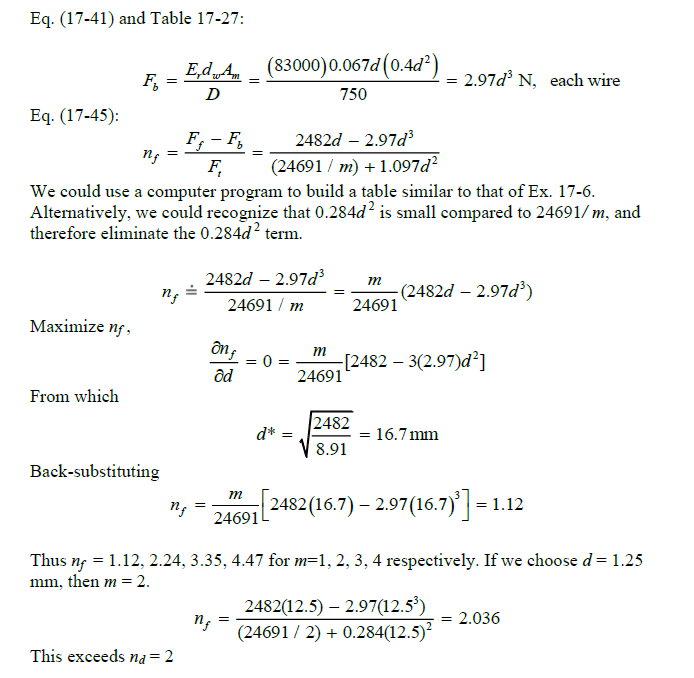
• Remind students that wire ropes do not fail suddenly due to fatigue. The outer

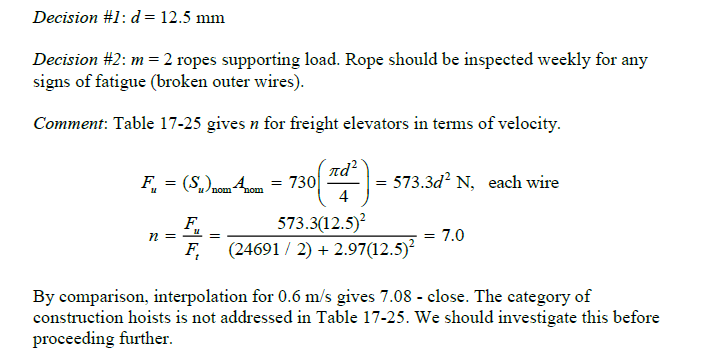
wires gradually show wear and breaks; such ropes should be retired. Periodic inspectionsprevent fatigue failures by parting of the rope.

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**17-31** A 0.61 km mine hoist operates with a 1.83 mm drum using 6 x 19 monitor-steel wire rope. The cage and load weigh 35.59 kN, and the cage is subjected to an acceleration of 0.61 m/s2 when starting.

(a) For a single-strand hoist how does the factor of safety *nf = Ff/Ft*, neglecting bending, vary with the choice of rope diameter?

(b) For four supporting strands of wire rope attached to the cage, how does the factor of safety vary with the choice of rope diameter?

Given: 610 m, 1.83 mm drum, 6 × 19 MS rope, cage and load 35.59 kN, acceleration 0.61 m/s2*.*

**(a)** Table 17-24: (*Su*)nom = 730 MPa; *Su* = 1655 MPa (p. 912);

Fig. 17-21: (*p/Su*)106 = 0*.*0014

Eq. (17-44):



Table 17-24: *wl* = 36.18*d* 2 610(10−3) = 22.07*d* 2 kN

Eq. (17-47): 



37.8

Note that bending is not included.



|  |  |  |
| --- | --- | --- |
| *d*, mm | *nf* |  |
| 0.2 | 1.094 |  |
| 0.4 | 2.041 |  |
| 0.6 | 2.751 |  |
| 0.8 | 3.212 |  |
| 1.0 | 3.462 |  |
| 1.2 | 3.555 |  |
| 1.4 | 3.544 | ← maximum *nf Ans.* |
| 1.6 | 3.468 |  |
| 1.8 | 3.355 |  |
| 2.0 | 3.223 |  |
| 2.2 | 3.083 |  |
| 2.4 | 2.944 |  |
| 2.6 | 2.808 |  |
| 2.8 | 2.679 |  |
| 3.0 | 2.557 |  |
| 0.2 | 1.094 |  |

**(b)** Try *m* = 4 strands



|  |  |  |
| --- | --- | --- |
| *d*, mm | *nf* |  |
| 0.2 | 4.082 |  |
| 0.4 | 6.424 |  |
| 0.6 | 7.110 | ← maximum *nf Ans.* |
| 0.8 | 6.936 |  |
| 1.0 | 6.445 |  |
| 1.2 | 5.888 |  |
| 1.4 | 5.358 |  |
| 1.6 | 4.883 |  |
| 1.8 | 4.468 |  |
| 2.0 | 4.108 |  |
| 0.2 | 4.082 |  |

Comparing tables, multiple ropes supporting the load increases the factor of safety, and reduces the corresponding wire rope diameter, a useful perspective.

**17-32**



From which



These results agree closely with the Prob. 17-31 solution. The small differences are due torounding in Prob. 17-31.

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**17-33** From Prob. 17-32 solution:



Solve the above equation for *m*

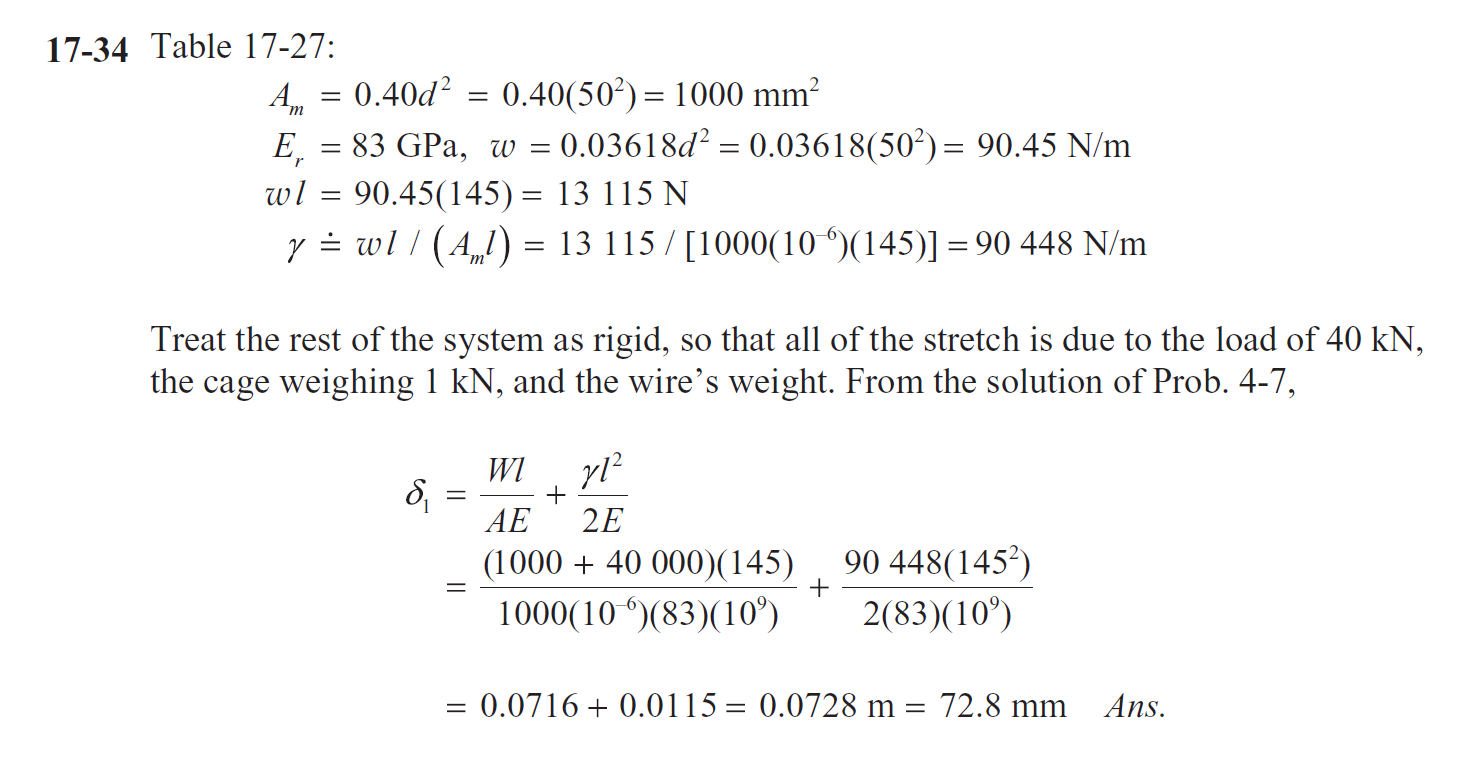


From which 

Substituting this result for *d* into Eq. (1) gives

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**17-35 to 17-38** Computer programs will vary.

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