

CEMA  
BELT BOOK  
**FIFTH EDITION**  
CHAPTER 6

BELT TENSION, POWER, AND  
DRIVE ENGINEERING

AS REFERENCED  
OCCASIONALLY  
IN  
CEMA  
BELT BOOK  
SIXTH EDITION

# *Belt Tension, Power, and Drive Engineering*

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The earliest application engineering of belt conveyors was, to a considerable extent, dependent upon empirical solutions that had been developed by various manufacturers and consultants in this field. The belt conveyor engineering analysis, information, and formulas presented in this manual represent recent improvements in the concepts and data which have been developed over the years, using the observations of actual belt conveyor operation and the best mathematical theory.

Horsepower (*hp*) and tension formulas, incorporating successively all the factors affecting the total force needed to move the belt and its load, are presented here in a manner that permits the separate evaluation of the effect of each factor. These formulas represent the consensus of all CEMA member companies.

In recent years, CEMA member companies have developed computer programs capable of complete engineering analysis of the most complex and extensive belt conveyor systems. These programs are more comprehensive and include more extensive analysis and calculations than can be included in this manual. Although the programs are treated as proprietary information, each CEMA member company welcomes an opportunity to assist in the proper application of belt conveyor equipment. One advantage of using computer programs is the speed and accuracy with which they provide information for alternate conveyor designs.

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### Basic Power Requirements

The horsepower, *hp*, required at the drive of a belt conveyor, is derived from the pounds of the effective tension,  $T_e$ , required at the drive pulley to propel or restrain the loaded conveyor at the design velocity of the belt  $V$ , in fpm:

(1)

$$hp = \frac{T_e \times V}{33,000}$$

To determine the effective tension,  $T_e$ , it is necessary to identify and evaluate each of the individual forces acting on the conveyor belt and contributing to the tension required to drive the belt at the driving pulley.  $T_e$  is the final summarization of the belt tensions produced by forces such as:

1. The gravitational load to lift or lower the material being transported.
2. The frictional resistance of the conveyor components, drive, and all accessories while operating at design capacity.
3. The frictional resistance of the material as it is being conveyed.
4. The force required to accelerate the material continuously as it is fed onto the conveyor by a chute or a feeder.

## Belt Tension Calculations

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The basic formula for calculating the effective tension,  $T_e$ , is:

$$T_e = LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) + T_p + T_{am} + T_{ac} \quad (2)$$

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## Belt Tension Calculations

The following symbols will be used to assist in the identification and evaluation of the individual forces that cumulatively contribute to  $T_e$  and that are therefore components of the total propelling belt tension required at the drive pulley:

$A_i$  = belt tension, or force, required to overcome frictional resistance and rotate idlers, lbs (see page 91)

$C_l$  = friction modification factor for regenerative conveyor

$H$  = vertical distance that material is lifted or lowered, ft

$K_t$  = ambient temperature correction factor (see Figure 6.1)

$K_x$  = factor used to calculate the frictional resistance of the idlers and the sliding resistance between the belt and idler rolls, lbs per ft (see equation 3, page 91)

$K_y$  = carrying run factor used to calculate the combination of the resistance of the belt and the resistance of the load to flexure as the belt and load move over the idlers (see equation 4, page 94, and Table 6-2). For return run use constant 0.015 in place of  $K_y$ . See  $T_{yr}$ .

$L$  = length of conveyor, ft

$Q$  = tons per hour conveyed, tph, short tons of 2,000 lbs

$S_i$  = troughing idler spacing, ft

$T_{ac}$  = total of the tensions from conveyor accessories, lbs:

$$T_{ac} = T_{sb} + T_{pl} + T_{tr} + T_{bc}$$

$T_{am}$  = tension resulting from the force to accelerate the material continuously as it is fed onto the belts, lbs

$T_b$  = tension resulting from the force needed to lift or lower the belt, lbs (see page 116):

$$T_b = \pm H \times W_b$$

$T_{bc}$  = tension resulting from belt pull required for belt-cleaning devices such as belt scrapers, lbs

$T_e$  = effective belt tension at drive, lbs

$T_m$  = tension resulting from the force needed to lift or lower the conveyed material, lbs:

$$T_m = \pm H \times W_m$$

$T_p$  = tension resulting from resistance of belt to flexure around pulleys and the resistance of pulleys to rotation on their bearings, total for all pulleys, lbs

$T_{pl}$  = tension resulting from the frictional resistance of plows, lbs

$T_{sb}$  = tension resulting from the force to overcome skirtboard friction, lbs

$T_{tr}$  = tension resulting from the additional frictional resistance of the pulleys and the flexure of the belt over units such as trippers, lbs

$T_x$  = tension resulting from the frictional resistance of the carrying and return idlers, lbs:

$$T_x = L \times K_x \times K_t$$

$T_{yb}$  = total of the tensions resulting from the resistance of the belt to flexure as it rides over both the carrying and return idlers, lbs:

$$T_{yb} = T_{yc} + T_{yr}$$

$T_{yc}$  = tension resulting from the resistance of the belt to flexure as it rides over the carrying idlers, lbs:

$$T_{yc} = L \times K_y \times W_b \times K_t$$

$T_{ym}$  = tension resulting from the resistance of the material to flexure as it rides with the belt over the carrying idlers, lbs:

$$T_{ym} = L \times K_y \times W_m$$

$T_{yr}$  = tension resulting from the resistance of the belt to flexure as it rides over the return idlers, lbs:

$$T_{yr} = L \times 0.015 \times W_b \times K_t$$

$V$  = design belt speed, fpm

$W_b$  = weight of belt in pounds per foot of belt length. When the exact weight of the belt is not known, use average estimated belt weight (see Table 6-1)

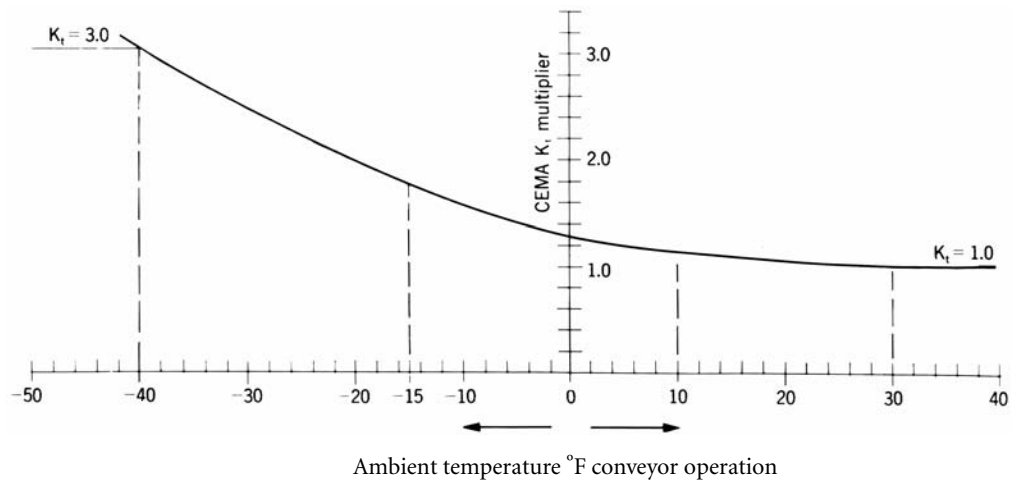
$W_m$  = weight of material, lbs per foot of belt length:

$$W_m = \frac{Q \times 2,000}{60 \times V} = \frac{33.33 \times Q}{V}$$

Three multiplying factors,  $K_t$ ,  $K_x$ , and  $K_y$ , are used in calculations of three of the components of the effective belt tension,  $T_e$ .

**$K_t$ — Ambient Temperature Correction Factor**

Idler rotational resistance and the flexing resistance of the belt increase in cold weather operation. In extremely cold weather the proper lubricant for idlers must be used to prevent excessive resistance to idler rotation.



Operation at temperatures below  $-15^{\circ}\text{F}$  involves problems in addition to horsepower considerations. Consult conveyor manufacturer for advice on special belting, greasing, and cleaning specifications and necessary design modification.

**Figure 6.1 Variation of temperature correction factor,  $K_t$ , with temperature.**

$K_t$  is a multiplying factor that will increase the calculated value of belt tensions to allow for the increased resistances that can be expected due to low temperatures. Figure 6.1 provides values for factor  $K_t$ .

$K_x$  — Idler Friction Factor

The frictional resistance of idler rolls to rotation and sliding resistance between the belt and the idler rolls can be calculated by using the multiplying factor  $K_x$ .  $K_x$  is a force in lbs/ft of conveyor length to rotate the idler rolls, carrying and return, and to cover the sliding resistance of the belt on the idler rolls. The  $K_x$  value required to rotate the idlers is calculated using equation (3).

The resistance of the idlers to rotation is primarily a function of bearing, grease, and seal resistance. A typical idler roll equipped with antifriction bearings and supporting a load of 1,000 lbs will require a turning force at the idler roll periphery of from 0.5 to 0.7 lbs to overcome the bearing friction. The milling or churning of the grease in the bearings and the bearing seals will require additional force. This force, however, is generally independent of the load on the idler roll.

Under normal conditions, the grease and seal friction in a well-lubricated idler will vary from 0.1 to 2.3 lbs/idler, depending upon the type of idler, the seals, and the condition of the grease.

Sliding resistance between the belt and idler rolls is generated when the idler rolls are not exactly at 90 degrees to the belt movement. After initial installation, deliberate idler misalignment is often an aid in training the belt. Even the best installations have a small requirement of this type. However, excessive idler misalignment results in an extreme increase in frictional resistance and should be avoided.

**Table 6-1. Estimated average belt weight, multiple- and reduced-ply belts, lbs/ft.**

Belt Width inches ( <i>b</i> )	Material Carried, lbs/ft <sup>3</sup>		
	30-74	75-129	130-200
18	3.5	4.0	4.5
24	4.5	5.5	6.0
30	6.0	7.0	8.0
36	9.0	10.0	12.0
42	11.0	12.0	14.0
48	14.0	15.0	17.0
54	16.0	17.0	19.0
60	18.0	20.0	22.0
72	21.0	24.0	26.0
84	25.0	30.0	33.0
96	30.0	35.0	38.0

1. Steel-cable belts — increase above value by 50 percent.
2. Actual belt weights vary with different constructions, manufacturers, cover gauges, etc. Use the above values for estimating. Obtain actual values from the belt manufacturer whenever possible.

Some troughing idlers are designed to operate with a small degree of tilt in the direction of belt travel, to aid in belt training. This tilt results in a slight increase in sliding friction that must be considered in the horsepower formula.

Values of  $K_x$  can be calculated from the equation:

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}, \text{ lbs tension per foot of belt length} \quad (3)$$

$A_i = 1.5$  for 6" diameter idler rolls, CEMA C6, D6

$A_i = 1.8$  for 5" diameter idler rolls, CEMA B5, C5, D5

$A_i = 2.3$  for 4" diameter idler rolls, CEMA B4, C4

$A_i = 2.4$  for 7" diameter idler rolls, CEMA E7

$A_i = 2.8$  for 6" diameter idler rolls, CEMA E6

For regenerative declined conveyors,  $A_i = 0$ .

The  $A_i$  values tabulated above are averages and include frictional resistance to rotation for both the carrying and return idlers. Return idlers are based on single roll type. *If two roll V return idlers are used, increase  $A_i$  value by 5%.* In the case of long conveyors or very high belt speed (over 1,000 fpm) refer to CEMA member companies for more specific values of  $A_i$ .

#### **$K_y$ — Factor for Calculating the Force of Belt and Load Flexure over the Idlers**

Both the resistance of the belt to flexure as it moves over idlers and the resistance of the load to flexure as it rides the belt over the idlers develop belt-tension forces.  $K_y$  is a multiplying factor used in calculating these belt tensioning forces.

Table 6-2 gives values of  $K_y$  for carrying idlers as they vary with differences in the weight/ft of the conveyor belt,  $W_b$ ; load,  $W_m$ ; idler spacing,  $S_i$ ; and the percent of slope or angle that the conveyor makes with the horizontal. When applying idler spacing,  $S_i$ , other than specified in Table 6-2, use Table 6-3 to determine a corrected  $K_y$  value.

**Example 1.** For a conveyor whose length is 800 ft and  $(W_b + W_m) = 150$  lbs/ft having a slope of 12%, the  $K_y$  value (Table 6-2) is .017. This  $K_y$  value is correct only for the idler spacing of 3.0 ft. If a 4.0-foot idler spacing is to be used, using Table 6-3 and the  $K_y$  reference values at the top of the table, the  $K_y$  of .017 lies between .016 and .018. Through interpolation and using the corresponding  $K_y$  values for 4.0-foot spacing, the corrected  $K_y$  value is .020.

**Example 2.** For a conveyor whose length is 1,000 ft and  $(W_b + W_m) = 125$  lbs/ft with a slope of 12%, the  $K_y$  value (Table 6-2) is .0165. This value is correct only for 3.5-foot spacing. If 4.5-foot spacing is needed, Table 6-3 shows that .0165 lies between .016 and .018 (reference  $K_y$ ). Through interpolation and using the corresponding  $K_y$  values for 4.5-foot spacing, the corrected  $K_y$  value is .0194.



Table 6-2. Factor  $K_y$  values.

Conveyor Length (ft)	$W_b + W_m$ (lbs/ft)	Percent Slope						
		0	3	6	9	12	24	33
		Approximate Degrees						
		0	2	3.5	5	7	14	18
250	20	0.035	0.035	0.034	0.031	0.031	0.031	0.031
	50	0.035	0.034	0.033	0.032	0.031	0.028	0.027
	75	0.035	0.034	0.032	0.032	0.030	0.027	0.025
	100	0.035	0.033	0.032	0.031	0.030	0.026	0.023
	150	0.035	0.035	0.034	0.033	0.031	0.025	0.021
	200	0.035	0.035	0.035	0.035	0.032	0.024	0.018
	250	0.035	0.035	0.035	0.035	0.033	0.021	0.018
	300	0.035	0.035	0.035	0.035	0.032	0.019	0.018
400	20	0.035	0.034	0.032	0.030	0.030	0.030	0.030
	50	0.035	0.033	0.031	0.029	0.029	0.026	0.025
	75	0.034	0.033	0.030	0.029	0.028	0.024	0.021
	100	0.034	0.032	0.030	0.028	0.028	0.022	0.019
	150	0.035	0.034	0.031	0.028	0.027	0.019	0.016
	200	0.035	0.035	0.033	0.030	0.027	0.016	0.014
	250	0.035	0.035	0.034	0.030	0.026	0.017	0.016
	300	0.035	0.035	0.034	0.029	0.024	0.018	0.018
500	20	0.035	0.033	0.031	0.030	0.030	0.030	0.030
	50	0.034	0.032	0.030	0.028	0.028	0.024	0.023
	75	0.033	0.032	0.029	0.027	0.027	0.021	0.019
	100	0.033	0.031	0.029	0.028	0.026	0.019	0.016
	150	0.035	0.033	0.030	0.027	0.024	0.016	0.016
	200	0.035	0.035	0.030	0.027	0.023	0.016	0.016
	250	0.035	0.035	0.030	0.025	0.021	0.016	0.015
	300	0.035	0.035	0.029	0.024	0.019	0.018	0.018
600	20	0.035	0.032	0.030	0.029	0.029	0.029	0.029
	50	0.033	0.030	0.029	0.027	0.026	0.023	0.021
	75	0.032	0.030	0.028	0.026	0.024	0.020	0.016
	100	0.032	0.030	0.027	0.025	0.022	0.016	0.016
	150	0.035	0.031	0.026	0.024	0.019	0.016	0.016
	200	0.035	0.031	0.026	0.021	0.017	0.016	0.016
	250	0.035	0.031	0.024	0.020	0.017	0.016	0.016
	300	0.035	0.031	0.023	0.018	0.018	0.018	0.018
800	20	0.035	0.031	0.030	0.029	0.029	0.029	0.029
	50	0.032	0.029	0.028	0.026	0.025	0.021	0.018
	75	0.031	0.029	0.026	0.024	0.022	0.016	0.016
	100	0.031	0.028	0.025	0.022	0.020	0.016	0.016
	150	0.034	0.028	0.023	0.019	0.017	0.016	0.016
	200	0.035	0.027	0.021	0.016	0.016	0.016	0.016
	250	0.035	0.026	0.020	0.017	0.016	0.016	0.016
	300	0.035	0.025	0.018	0.018	0.018	0.018	0.018

Idle spacing: The above values of  $K_y$  are based on the following idler spacing (for other spacing, see Table 6-3).

$(W_b+W_m)$ , lbs per ft	$S_i$ , ft	$(W_b+W_m)$ , lbs per ft	$S_i$ , ft
Less than 50	4.5	100 to 149	3.5
50 to 99	4.0	150 and above	3.0

Table 6-2. Factor  $K_y$  values.

Conveyor Length (ft)	$W_b + W_m$ (lbs/ft)	Percent Slope						
		0	3	6	9	12	24	33
		Approximate Degrees						
		0	2	3.5	5	7	14	18
1000	50	0.031	0.028	0.026	0.024	0.023	0.019	0.016
	75	0.030	0.027	0.024	0.022	0.019	0.016	0.016
	100	0.030	0.026	0.022	0.019	0.017	0.016	0.016
	150	0.033	0.024	0.019	0.016	0.016	0.016	0.016
	200	0.032	0.023	0.017	0.016	0.016	0.016	0.016
	300	0.033	0.021	0.018	0.018	0.018	0.018	0.018
1400	50	0.029	0.026	0.024	0.022	0.021	0.016	0.016
	75	0.028	0.024	0.021	0.019	0.016	0.016	0.016
	100	0.028	0.023	0.019	0.016	0.016	0.016	0.016
	150	0.029	0.020	0.016	0.016	0.016	0.016	0.016
	200	0.030	0.021	0.016	0.016	0.016	0.016	0.016
	300	0.030	0.019	0.018	0.018	0.018	0.018	0.018
2000	50	0.027	0.024	0.022	0.020	0.018	0.016	0.016
	75	0.026	0.021	0.019	0.016	0.016	0.016	0.016
	100	0.025	0.020	0.016	0.016	0.016	0.016	0.016
	150	0.026	0.017	0.016	0.016	0.016	0.016	0.016
	200	0.024	0.016	0.016	0.016	0.016	0.016	0.016
	300	0.022	0.018	0.018	0.018	0.018	0.018	0.018
2400	50	0.026	0.023	0.021	0.018	0.017	0.016	0.016
	75	0.025	0.021	0.017	0.016	0.016	0.016	0.016
	100	0.024	0.019	0.016	0.016	0.016	0.016	0.016
	150	0.024	0.016	0.016	0.016	0.016	0.016	0.016
	200	0.021	0.016	0.016	0.016	0.016	0.016	0.016
	300	0.020	0.018	0.018	0.018	0.018	0.018	0.018
3000	50	0.024	0.022	0.019	0.017	0.016	0.016	0.016
	75	0.023	0.019	0.016	0.016	0.016	0.016	0.016
	100	0.022	0.017	0.016	0.016	0.016	0.016	0.016
	150	0.022	0.016	0.016	0.016	0.016	0.016	0.016
	200	0.019	0.016	0.016	0.016	0.016	0.016	0.016
	300	0.018	0.016	0.016	0.016	0.016	0.016	0.016

Idler spacing: The above values of  $K_y$  are based on the following idler spacing (for other spacing, see Table 6-3).

$(W_b + W_m)$ , lbs per ft	$S_i$ , ft	$(W_b + W_m)$ , lbs per ft	$S_i$ , ft
Less than 50	4.5	100 to 149	3.5
50 to 99	4.0	150 and above	3.0

$K_y$  values in Tables 6-2 and 6-3 are applicable for conveyors up to 3,000 ft long with a single slope and a 3% maximum sag of the belt between the troughing and between the return idlers. The return idler spacing is 10 ft nominal and loading of the belt is uniform and continuous.

Equation (4) provides  $K_y$  values for the carrying idlers of belt conveyors whose length, number of slopes, and/or average belt tensions exceed the limitations specified above for the conveyors covered by Tables 6-2 and 6-3. This equation is applicable for conveyors in which the average belt tension is 16,000 lbs or less. To determine the  $K_y$  factor for use in calculating conveyors of this class, it is necessary, first, to assume a tentative value for the average belt tension. The graphical method for determining conveyor horsepower (pages 141 through 145) may be of assistance in estimating this initial tentative value of average belt tension.

After estimating the average belt tension and selecting an idler spacing, refer to Table 6-4 to obtain values for  $A$  and  $B$  for use in the following equation:

$$K_y = (W_m + W_b) \times A \times 10^{-4} + B \times 10^{-2} \tag{4}$$

By using equation (4), an initial value for  $K_y$  can be determined and an initial average belt tension can be subsequently calculated. The comparison of this calculated average belt tension with the original tentative value will determine the need to select another assumed belt tension. Recalculate  $K_y$  and calculate a second value for the average belt tension. The process should be repeated until there is reasonable agreement between the estimated and final calculated average belt tensions.

There are no tabulated  $K_y$  values or mathematical equations to determine a  $K_y$  for conveyors having an average belt tension exceeding 16,000 lbs. A reasonably accurate value that can be used for calculations is  $K_y$  equals 0.016. It is suggested that this value for  $K_y$  be considered a minimum, subject to consultation with a CEMA member company on any specific applications.

The force that results from the resistance of the belt to flexure as it moves over the idlers for the return run is calculated in the same manner as the resistance to flexure for the carrying run, except a constant value of 0.015 is used in place of  $K_y$ . The resistance of the belt flexure over idler rolls is a function of the belt construction, cover thickness and indentation by the idler rolls, type of rubber compound, idler roll diameter, temperature, and other factors. The belt flexing resistance increases at lower temperatures.

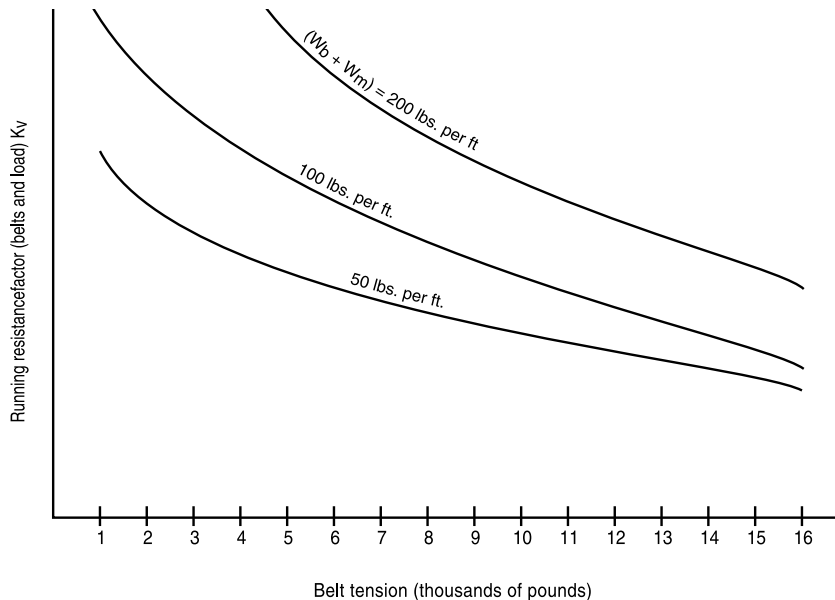


Figure 6.2 Effect of belt tension on resistance of material to flexure over idler rolls.

Table 6-3. Corrected factor  $K_y$  values when other than tabular carrying idler spacings are used.

$W_b + W_m$ (lbs/ft)	$S_p$ (ft)	Reference Values of $K_y$ for Interpolation									
		0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034
<b>Less than 50</b>	3.0	0.0160	0.0160	0.0160	0.0168	0.0183	0.0197	0.0212	0.0227	0.0242	0.0257
	3.5	0.0160	0.0160	0.0169	0.0189	0.0207	0.0224	0.0241	0.0257	0.0274	0.0291
	4.0	0.0160	0.0165	0.0182	0.0204	0.0223	0.0241	0.0259	0.0278	0.0297	0.0316
	<b>4.5</b>	<b>0.016</b>	<b>0.018</b>	<b>0.020</b>	<b>0.022</b>	<b>0.024</b>	<b>0.026</b>	<b>0.028</b>	<b>0.030</b>	<b>0.032</b>	<b>0.034</b>
	5.0	0.0174	0.0195	0.0213	0.0236	0.0254	0.0273	0.0291	0.0303	0.0329	0.0348
<b>50 to 99</b>	3.0	0.0160	0.0162	0.0173	0.0186	0.0205	0.0221	0.0239	0.026	0.0274	0.029
	3.5	0.0160	0.0165	0.0185	0.0205	0.0222	0.024	0.0262	0.0281	0.030	0.0321
	<b>4.0</b>	<b>0.016</b>	<b>0.018</b>	<b>0.020</b>	<b>0.022</b>	<b>0.024</b>	<b>0.026</b>	<b>0.028</b>	<b>0.030</b>	<b>0.032</b>	<b>0.034</b>
	4.5	0.0175	0.0193	0.0214	0.0235	0.0253	0.0272	0.0297	0.0316	0.0335	0.035
	5.0	0.0184	0.021	0.023	0.0253	0.027	0.029	0.0315	0.0335	0.035	0.035
<b>100 to 149</b>	3.0	0.0160	0.0164	0.0186	0.0205	0.0228	0.0246	0.0267	0.0285	0.0307	0.0329
	<b>3.5</b>	<b>0.016</b>	<b>0.018</b>	<b>0.020</b>	<b>0.022</b>	<b>0.024</b>	<b>0.026</b>	<b>0.028</b>	<b>0.030</b>	<b>0.032</b>	<b>0.034</b>
	4.0	0.0175	0.0197	0.0213	0.0234	0.0253	0.0277	0.0295	0.0312	0.033	0.035
	4.5	0.0188	0.0213	0.0232	0.0253	0.0273	0.0295	0.0314	0.033	0.0346	0.035
	5.0	0.0201	0.0228	0.0250	0.0271	0.0296	0.0316	0.0334	0.035	0.035	0.035
<b>150 to 199</b>	<b>3.0</b>	<b>0.016</b>	<b>0.018</b>	<b>0.020</b>	<b>0.022</b>	<b>0.024</b>	<b>0.026</b>	<b>0.028</b>	<b>0.030</b>	<b>0.032</b>	<b>0.034</b>
	3.5	0.0172	0.0195	0.0215	0.0235	0.0255	0.0271	0.0289	0.031	0.0333	0.0345
	4.0	0.0187	0.0213	0.0235	0.0252	0.0267	0.0283	0.0303	0.0325	0.0347	0.035
	4.5	0.0209	0.023	0.0253	0.0274	0.0289	0.0305	0.0323	0.0345	0.035	0.035
	5.0	0.0225	0.0248	0.0272	0.0293	0.0311	0.0328	0.0348	0.035	0.035	0.035
<b>200 to 249</b>	<b>3.0</b>	<b>0.016</b>	<b>0.018</b>	<b>0.020</b>	<b>0.022</b>	<b>0.024</b>	<b>0.026</b>	<b>0.028</b>	<b>0.030</b>	<b>0.032</b>	<b>0.034</b>
	3.5	0.0177	0.0199	0.0216	0.0235	0.0256	0.0278	0.0295	0.031	0.0327	0.0349
	4.0	0.0192	0.0216	0.0236	0.0256	0.0274	0.0291	0.0305	0.0322	0.0339	0.035
	4.5	0.021	0.0234	0.0253	0.0276	0.0298	0.0317	0.0331	0.0347	0.035	0.035
	5.0	0.0227	0.0252	0.0274	0.0298	0.0319	0.0338	0.035	0.035	0.035	0.035

To use this table to correct the value of  $K_y$  for idler spacing other than shown in bold type, apply the procedure shown in the two examples on page 91.

The resistance of the material load to flexure over idler rolls is a function of belt tension, type of material, shape of the load cross section, and idler spacing. Measurements indicate that the most important factor is belt tension, because this controls the amount of load flexure. Figure 6.2 shows this relationship for a typical idler spacing.

For a given weight per foot of belt and load, the running resistance, in pounds per ft of load, decreases with increases in belt tension. For a given belt tension, running resistance, in pounds per ft of load, increases with increases in the amount of load. However, the running resistance is not proportional to the weight of the load.

Table 6-4. *A* and *B* values for equation  $K_y = (W_m + W_b) \times A \times 10^{-4} + B \times 10^{-2}$

Average Belt Tension, lbs	Idler Spacing, ft									
	3.0		3.5		4.0		4.5		5.0	
	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>	<i>A</i>	<i>B</i>
1,000	2.150	1.565	2.1955	1.925	2.200	2.250	2.2062	2.584	2.1750	2.910
2,000	1.8471	1.345	1.6647	1.744	1.6156	1.982	1.5643	2.197	1.5429	2.331
3,000	1.6286	1.237	1.4667	1.593	1.4325	1.799	1.4194	1.991	1.4719	2.091
4,000	1.4625	1.164	1.3520	1.465	1.3295	1.659	1.3250	1.825	1.3850	1.938
5,000	1.2828	1.122	1.1926	1.381	1.1808	1.559	1.1812	1.714	1.2283	1.839
6,000	1.1379	1.076	1.0741	1.318	1.0625	1.472	1.0661	1.627	1.0962	1.761
7,000	1.0069	1.039	0.9448	1.256	0.9554	1.404	0.9786	1.549	1.0393	1.657
8,000	0.9172	0.998	0.8552	1.194	0.8643	1.337	0.8875	1.472	0.9589	1.583
9,000	0.8207	0.958	0.8000	1.120	0.7893	1.272	0.8339	1.388	0.8911	1.507
10,000	0.7241	0.918	0.7362	1.066	0.7196	1.216	0.7821	1.314	0.8268	1.430
11,000	0.6483	0.885	0.6638	1.024	0.6643	1.167	0.7375	1.238	0.7768	1.340
12,000	0.5828	0.842	0.5828	0.992	0.6232	1.100	0.6750	1.180	0.7411	1.242
13,000	0.5207	0.798	0.5241	0.938	0.5732	1.040	0.6179	1.116	0.6821	1.169
14,000	0.4690	0.763	0.4810	0.897	0.5214	0.996	0.5571	1.069	0.6089	1.123
15,000	0.4172	0.718	0.4431	0.841	0.4732	0.935	0.5179	1.006	0.5607	1.063
16,000	0.3724	0.663	0.3966	0.780	0.4232	0.875	0.4589	0.958	0.5054	1.009

A minimum  $K_y$  value of .016 should be used when tensions exceed 16,000 lbs. Refer to page 92 for further explanations.

Information similar to that in Figure 6.2 has been developed by analyzing a series of field tests on belt conveyors of different widths carrying different materials. Many investigators, both in the United States and abroad, have analyzed similar series of field tests and have obtained similar results. Although the exact expressions differ, all investigators agree that changes in belt tension affect the force required to flex the material over idler rolls to a substantially greater degree than changes in the material handled. The latter does have a noticeable effect, and thus appears to be of less importance in the overall calculation.

**Compilation of Components of  $T_e$**

The preceding pages describe the methods and provide the data for calculating factors  $K_t$ ,  $K_x$ , and  $K_y$ . These factors must be evaluated as the first step to calculating certain components of belt tension that will be summarized to determine the effective tension,  $T_e$ , required at the driving pulley.

The procedures for calculating the belt tension components are as follows:

1.  $T_x$  — from the frictional resistance of the carrying and return idlers, lbs

$$T_x = L \times K_x \times K_t$$

(References:  $K_x$  — page 90,  $K_t$  — page 89)

2.  $T_{yb}$  — from the resistance of the belt to flexure as it moves over the idlers, lbs

$$T_{yc} \text{ — for carrying idlers: } T = L \times K_y \times W_b \times K_t$$

$$T_{yr} \text{ — for return idlers: } T_{ty} = L \times 0.015 \times w_b \times K_t$$

$$T_{yb} = T_{yc} + T_{yr}$$

$$T_{yb} = L \times K_y \times W_b \times K_t + L \times 0.015 \times W_b \times K_t$$

$$= L \times W_b \times K_t (K_y + 0.015)$$

(References:  $K_y$  — page 91,  $K_t$  — page 89)

3.  $T_{ym}$  — from resistance of the material to flexure as it rides the belt over the idlers, lbs

$$T_{ym} = L \times K_y \times W_m$$

(Reference:  $K_y$  — page 91)

4.  $T_m$  — from force needed to lift or lower the load (material), lbs

$$T_m = \pm H \times W_m$$

5.  $T_p$  — from resistance of belt to flexure around pulleys and the resistance of pulleys to rotate on their bearings, lbs

Pulley friction arises from two sources. One source is the resistance of the belt to flexure over the pulleys, which is a function of the pulley diameter and the belt stiffness. The belt stiffness depends upon the ambient temperature and the belt construction.

The other source of pulley friction is the resistance of the pulley to rotate, which is a function of pillow block bearing friction, lubricant, and seal friction. The pillow block bearing friction depends upon the load on the bearings, but the lubricant and seal frictions generally are independent of load.

Since the drive pulley friction does not affect belt tension, it is not introduced into the mathematical calculation for belt tension; however, it must be included when determining the total horsepower at the motor shaft.

Table 6-5 provides conservative values for the pounds of belt tension required to rotate each of the pulleys on a conveyor. However, if a more precise value for belt tension to rotate pulleys is desired refer to **Appendix C**, page 352. Examples of belt tension and horsepower calculations shown in this book use values from Table 6-5.

$$T_p = \text{total of the belt tensions required to rotate each of the pulleys on the conveyor}$$

6.  $T_{am}$  — from force to accelerate the material continuously as it is fed onto the belt

Table 6-5. Belt tension to rotate pulleys.

Location of Pulleys	Degrees Wrap of Belt	Pounds of Tension at Belt Line
Tight side	150° to 240°	200 lbs/pulley
Slack side	150° to 240°	150 lbs/pulley
All other pulleys	less than 150°	100 lbs/pulley

**Note:** Double the above values for pulley shafts that are not operating in antifriction bearings.

When material is discharged from chutes or feeders to a belt conveyor, it cannot be assumed that the material is moving in the direction of belt travel, at belt speed, although this may be the case in some instances. Normally, the material loaded onto the belt is traveling at a speed considerably lower than belt speed. The direction of material flow may not be fully in the direction of belt travel. Therefore, the material must be accelerated to the speed of the belt in the direction of belt travel, and this acceleration requires additional effective tension.

The belt tension  $T_{am}$  can be derived from the basic equation  $F = MV_c$

where:

$$T_{am} = F = MV_c$$

$M$  = mass of material accelerated per second, slugs

$W$  = weight of material accelerated

$$= \frac{Q \times 2000}{3600}, \text{ lbs/sec}$$

$Q$  = tph

$g$  = 32.2 ft/sec<sup>2</sup>

$$M = \frac{W}{g} = \frac{Q \times 2000}{3600 \times 32.2}$$

$V_c$  = velocity change, fps

$$= \frac{V - V_o}{60}$$

$V$  = design belt speed, fpm

$V_o$  = initial velocity of material as it is fed onto belt, fpm

$$T_{am} = \frac{Q \times 2000}{3600 \times 32.2} \times \frac{V - V_o}{60}$$

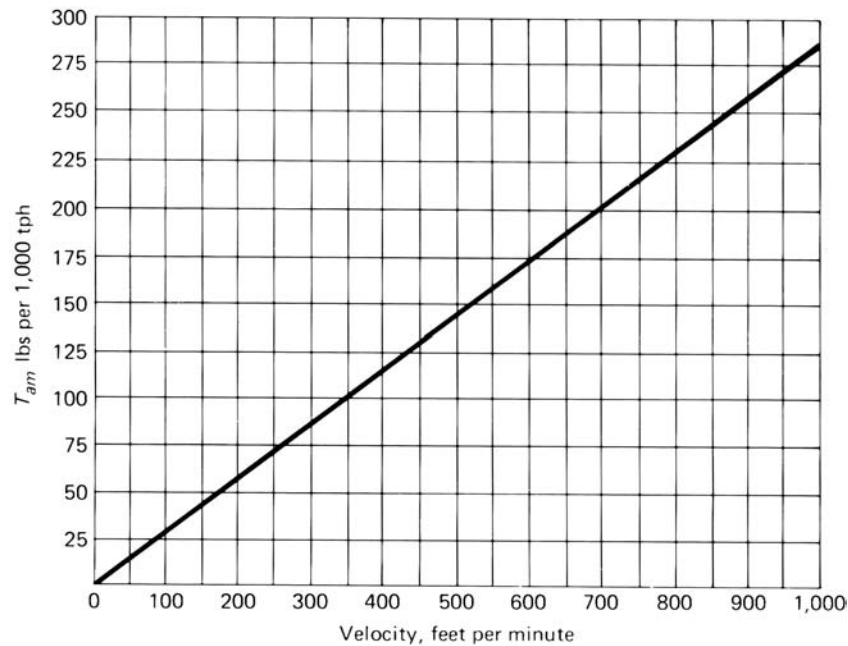
$$= 2.8755 \times 10^{-4} \times Q \times (V - V_o)$$

The graph in Figure 6.3 provides a convenient means of estimating the belt tension,  $T_{am}$ , for accelerating the material as it is fed onto the belt.

7.  $T_{ac}$  — from the resistance generated by conveyor accessories

Conveyor accessories such as trippers, stackers, plows, belt cleaning equipment, and skirtboards usually add to the effective tension,  $T_e$ . The additional belt tension requirements may come from frictional losses caused by the accessory. If the accessory lifts the conveyed material a force will be added to belt tension.

$T_{tr}$  — from trippers and stackers



To use this chart:

- Enter chart at belt velocity and read  $T_{am}$  per 1,000 tph.
- Again enter chart at material velocity in direction of belt travel and read  $T_{am}$  per 1,000 tph. This may be positive, zero, or negative.
- Subtract the second  $T_{am}$  reading from the first  $T_{am}$  reading and convert the difference from 1,000 tph to the value for the actual tonnage. This will be the  $T_{am}$  desired, lbs.

**Figure 6.3. Effective tension required to accelerate material as it is fed onto a belt conveyor.**

The additional belt pull to flex the belt over the pulleys and rotate the pulleys in their bearings can be calculated from Table 6-5 or Tables C-1 and C-2.

The force needed to lift the material over the unit can be calculated from the formula,  $T_m = H \times W_m$  lbs.

Frictional resistance of the idlers, belt, and material should be included with that of the rest of the conveyor.



$T_{pl}$  — from frictional resistance of plows

The use of a plow on a conveyor will require additional belt tension to overcome both the plowing and frictional resistances developed.

While a flat belt conveyor may be fitted with a number of plows to discharge material at desired locations, seldom is more than one plow in use at one time on one run of the belt conveyor. However, when proportioning plows are used — with each plow taking a fraction of the load from the belt — two or even three separate plows may be simultaneously in contact with the carrying run of the belt.

To approximate the amount of additional belt pull that normally will be required by well-adjusted, rubber-shod plows, the values given in Table 6-6 can be used.

**Table 6-6. Discharge plow allowance.**

Type of Plow	Additional Belt Pull per Plow, at Belt Line (lbs/in belt width)
Full V or single slant plow, removing all material from belt	5.0
Partial V or single slant plow, removing half material from belt	3.0

$T_{bc}$  — from belt-cleaning devices

Belt scraper cleaning devices add directly to the belt pull. The additional belt pull required for belt cleaning devices can vary from 2 to 14 lbs/in. width of scraper blade contact. This wide variance is due to the different types of cleaning blades and single cleaner system vs. multiple cleaner systems that are available. In lieu of data on specific cleaning system being used, use 5 lbs/in. width of scraper blade contact for each blade or scraper device in contact with the belt.

Rotary brushes and similar rotating cleaning devices do not impose appreciable belt pull, if independently driven and properly adjusted. If such devices are driven from the conveyor drive shaft, suitable additional power should be incorporated in the drive to operate them. Consult a CEMA member company for horsepower required.

$T_{sb}$  — From skirtboard friction

The force required to overcome skirtboard friction is normally larger per foot of skirtboarded conveyor than the force to move the loaded belt over the idlers. In some cases, this force can be significant. When the total conveyor length is many times that portion of the length provided with the skirtboards, the additional power requirements for the skirtboards is relatively small, and in some cases negligible. However, if a large portion of the conveyor is equipped with skirtboards, the additional belt pull

required may be a major factor in the effective tension,  $T_e$ , required to operate the conveyor.

When the spacing of the skirtboards is two-thirds the width of the belt, the depth of the material rubbing on the skirtboards will not be more than 10 percent of the belt width, providing no more than a 20-degree surcharge load is carried on 20-degree troughing idlers.

Once the cross section of the load on the belt conveyor has been determined, the skirtboard friction can be calculated by determining the total pressure of the material against the skirtboard, then multiplying this value by the appropriate coefficient of friction of the material handled. The pressure of the material against the skirtboard can be calculated by assuming that the wedge of material contained between a vertical skirtboard and the angle of repose of the material is supported equally by the skirtboard and the belt.

This results in the following formula for conveyors whereon the material assumes its natural surcharge angle:

$$P = \frac{L_b d_m h_s^2}{288} \times \frac{(1 - \sin \phi)}{(1 + \sin \phi)}$$

where:

- $P$  = total force against one skirtboard, lbs
- $L_b$  = skirtboard length, ft one skirtboard
- $d_m$  = apparent density of the material, lbs/cu ft
- $h_s$  = depth of the material touching the skirtboard, in
- $\phi$  = angle of repose of material, degrees

Combining the apparent density, coefficient of friction, the angle of repose, and the constant into one factor for one type of material, the formula can be expressed:

$$C_s = \frac{2d_m}{288} \times \left( \frac{1 - \sin \phi}{1 + \sin \phi} \right)$$

$$T = (C_s)(L_b h_s^2) = C_s L_b h_s^2$$

where:

- $T$  = belt tension to overcome skirtboard friction of two parallel skirtboards, lbs
- $C_s$  = factor for the various materials in Table 6-7

Table 6-7. Skirtboard friction factor,  $C_s$ .

Material	Factor $C_s$	Material	Factor $C_s$	Material	Factor $C_s$
Alumina, pulv. dry	0.1210	Coke, ground fine	0.0452	Limestone, pulv., dry	0.1280
Ashes, coal, dry	0.0571	Coke, lumps and fines	0.0186	Magnesium chloride, dry	0.0276
Bauxite, ground	0.1881	Copra, lumpy	0.0203	Oats	0.0219
Beans, navy, dry	0.0798	Cullet	0.0836	Phosphate rock, dry, broken	0.1086
Borax	0.0734	Flour, wheat	0.0265	Salt, common, dry, fine	0.0814
Bran, granular	0.0238	Grains, wheat, corn or rye	0.0433	Sand, dry, bank	0.1378
Cement, Portland, dry	0.2120	Gravel, bank run	0.1145	Sawdust, dry	0.0086
Cement clinker	0.1228	Gypsum, 1/2" screenings	0.0900	Soda ash, heavy	0.0705
Clay, ceramic, dry fines	0.0924	Iron ore, 200 lbs/cu ft	0.2760	Starch, small lumps	0.0623
Coal, anthracite, sized	0.0538	Lime, burned, 1/8"	0.1166	Sugar, granulated dry	0.0349
Coal, bituminous, mined	0.0754	Lime, hydrated	0.0490	Wood chips, hogged fuel	0.0095

To this skirtboard friction must be added 3 lbs for every linear foot of each skirtboard, to overcome friction of the rubber skirtboard edging, when used, with the belt. Then,

$$T_{sb} = T + 2L_b \times 3$$

$$= C_s L_b h_s^2 + 2L_b \times 3$$

$$T_{sb} = L_b (C_s h_s^2 + 6)$$

**Summary of Components of  $T_e$**

Once they have been compiled, the components of belt tension can be summarized to determine the effective tension,  $T_e$  required at the driving pulley.

$T_e$  equals the total of the following:

$$T_x, \text{ idler friction} = L \times K_x \times K_t$$

$$+ T_{yc}, \text{ belt flexure, carrying idlers} = L \times K_y \times W_b \times K_t$$

$$+ T_{yr}, \text{ belt flexure, return idlers} = L \times 0.015 \times W_b \times K_t$$

$$\text{Subtotal (A)} = LK_t(K_x + K_y W_b + 0.015 W_b)$$

$$+ T_{ym}, \text{ material flexure} = L \times K_y \times W_m$$

$$+ T_m, \text{ lift or lower} = \pm H \times W_m$$

$$\text{Subtotal (B)} = W_m(LK_y \pm H)$$

## CEMA Horsepower Formula

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$$\begin{aligned} &+ T_p, \text{ pulley resistance} \\ &+ T_{am}, \text{ accelerated material} \\ &+ T_{ac}, \text{ accessories} \\ &\quad (T_{tr} + T_{pl} + T_{bc} + T_{sb}) \\ &\quad \text{Subtotal (C)} = (T_p + T_{am} + T_{ac}) \end{aligned}$$

$$T_e = \Sigma \text{Subtotals (A), (B), and (C)}$$

$$= LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) + T_p + T_{am} + T_{ac}$$

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## CEMA Horsepower Formula

Equation 1, page 86, provides the means for calculating the horsepower (hp) required by a belt conveyor having an effective tension,  $T_e$ , at the drive pulley and a design velocity,  $V$ , of the belt, as follows:

$$\text{hp} = \frac{T_e \times V}{33,000}$$

Combining equations (1) and (2) on pages 86-87, the hp load can be expressed as follows:

$$\text{hp} = \frac{V}{33,000} [LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) + T_p + T_{am} + T_{ac}]$$

The motor that will drive a fully loaded belt conveyor without becoming overheated may not be able to accelerate the loaded conveyor from rest to the design speed. To insure adequate starting capabilities, the following conditions must exist. First, the locked rotor torque of the motor should exceed the sum of the torque required to lift the material, plus approximately twice the torque required to overcome total conveyor friction, despite any possible voltage deficiencies that may exist during the acceleration period. This may not be true for long, horizontal conveyors or for declined conveyors.

Second, the motor speed-torque curve should not drop below a line drawn from the locked rotor torque requirement to the torque of the running horsepower requirement at full speed. This is further explained in Chapter 13, "Motors and Controls."

For examples illustrating the use of the equations in determining the effective belt tension,  $T_e$ , at the drive pulley and the horsepower to operate the belt conveyor, refer to the two problems on pages 145 through 152.

It is also possible to arrive at a close approximation of the horsepower required to operate a belt conveyor by means of a graphical solution. This method, used under proper circumstances, is quick and relatively simple. Generally, a graphical solution will provide a somewhat conservative value of required horsepower. However, it must be recognized that it is impractical to incorporate all elements of belt conveyor design into a simple graphical solution. Therefore, the graphs should be used based on a

complete understanding of all aspects of the analytical method of calculating belt conveyor tension and horsepower, in order to allow for adjustment of the results to account for unusual situations. It is recommended that final design be based on calculations made by the analytical method. The graphical method of designing belt conveyors is described on pages 141-145.

### Drive Pulley Relationships

The force required to drive a belt conveyor must be transmitted from the drive pulley to the belt by means of friction between their two surfaces. The force required to restrain a downhill regenerative conveyor is transmitted in exactly the same manner. In order to transmit power, there must be a difference in the tension in the belt as it approaches and leaves the drive pulley. This difference in tensions is supplied by the driving power source. Figures 6.4 and 6.5 illustrate typical arrangements of single pulley drives.

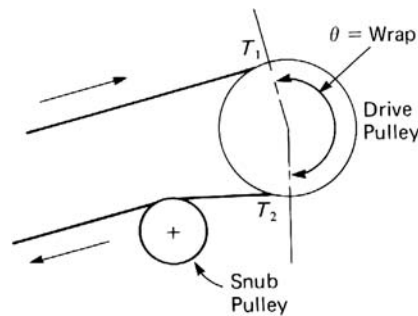


Figure 6.4 Inclined or horizontal conveyor, pulley driving belt.

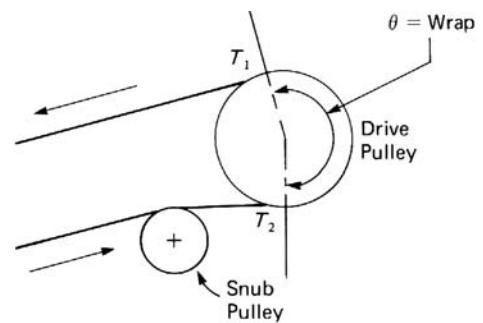


Figure 6.5 Declined conveyor. Lowering load with regeneration, belt driving pulley.

It should be noted that if power is transmitted from the pulley to the belt, the approaching portion of the belt will have the larger tension,  $T_1$ , and the departing portion will have the smaller tension,  $T_2$ . If power is transmitted from the belt to the pulley, as with a regenerative declined conveyor, the reverse is true. Wrap is used here to refer to the angle or arc of contact the belt makes with the pulley's circumference.

### Wrap Factor, $C_w$

The wrap factor,  $C_w$ , is a mathematical value used in the determination of the effective belt tension,  $T_e$ , that can be dependably developed by the drive pulley. The  $T_e$  that can be developed is governed by the coefficient of friction existing between the pulley and the belt, wrap, and the values of  $T_1$  and  $T_2$ .

The following symbols and formulas are used to evaluate the drive pulley relationships:

- $T_e = T_1 - T_2 =$  effective belt tension, lbs
- $T_1 =$  tight-side tension at pulley, lbs
- $T_2 =$  slack-side tension at pulley, lbs
- $e =$  base of naperian logarithms = 2.718
- $f =$  coefficient of friction between pulley surface and belt surface (0.25 rubber surfaced belt driving bare steel or cast iron pulley; 0.35 rubber sur-

faced belt driving rubber lagged pulley surface). Values apply to normal running calculations.

$\theta$  = wrap of belt around the pulley, radians (one degree = 0.0174 radians)

$C_w$  = wrap factor (see Table 6-8)

$$= \frac{T_2}{T_e} = \frac{1}{e^{f\theta} - 1}$$

It should be noted that the wrap factors do not determine  $T_2$  but only establish its safe minimum value for a dry belt. A wet belt and pulley will substantially reduce the power that can be transmitted from the one to the other because of the lower coefficient of friction of the wet surfaces. Various expedients, such as grooving the lagging on the pulley, lessen this problem. However, the best solution is to keep the driving side of the belt dry. If this is impractical, increasing the wrap is helpful, or providing some means of increasing the slack side tension,  $T_2$ . This can be done, for example, by increasing the counterweight in a gravity takeup.

**Wrap Factor with Screw Takeup**

When a screw takeup is used, Table 6-8 indicates an increased wrap factor. This increased wrap factor is necessary to provide sufficient slack side tension,  $T_2$ , to drive the conveyor in spite of the amount of stretch in the conveyor belt, for which the screw takeup makes no automatic provision.

**Table 6-8. Wrap factor,  $C_w$  (Rubber-surfaced belt).**

Type of Pulley Drive	$\theta$ Wrap	Automatic Takeup		Manual Takeup	
		Bare Pulley	Lagged Pulley	Bare Pulley	Lagged Pulley
Single, no snub	180°	0.84	0.50	1.2	0.8
Single with snub	200°	0.72	0.42	1.0	0.7
	210°	0.66	0.38	1.0	0.7
	220°	0.62	0.35	0.9	0.6
	240°	0.54	0.30	0.8	0.6
Dual*	380°	0.23	0.11	0.5	0.3
	420°	0.18	0.08	—	—

\*Dual values based on ideal distribution between primary and secondary drive.

For wet belts and smooth lagging, use bare pulley factor.

For wet belts and grooved lagging, used lagged pulley factor.

If wrap is unknown, assume the following:

Type of Drive	Assumed Wrap
Single—no snub	180°
Single –with snub	210°
Dual	380°

**Wrap  $\theta$  (Arc of Contact)**

So far, it has been shown that the relationship between the values known as  $T_1$ , the tight side tension (and generally the tension for which the belt must be designed and built), and  $T_2$ , the slack side tension (the minimum value that must be available for driving the belt successfully), is influenced by the angle of wrap of the belt around the drive pulley and by the coefficients of friction established by the belt and pulley surfaces as they make contact. It has been indicated that the coefficient of friction may vary when driving a rubber surfaced belt by a bare steel or cast iron pulley, or by a rubber-lagged pulley surface.

The angle of wrap of the belt around the drive pulley can be varied by the use of a snub pulley or, for larger angles of wrap, by supplying power, under the proper conditions, to more than one drive pulley.

The wrap limits for various types of pulley drives can be determined from Table 6-9.

**Table 6-9. Wrap limits.**

Type of Pulley Drive	Wrap Limits*	
	From	To
Single—no snub	180°	180°
Single—with snub	180°	240°
Dual	360°	480°

\*The above wrap angles apply to either bare or lagged pulleys.

For most cases, the belt will have an angle of wrap around the drive pulley of about 180 degrees to 240 degrees. Often, it will be necessary to arrange a drive that uses an angle of wrap greater than 180 degrees. This is accomplished by the appropriate positioning of a snub pulley, which can extend the angle of wrap to 240 degrees. However, its use is subject to the following limitations: (1) Snub pulley diameter is limited by the belt specifications; (2) In order to thread through a new or replacement belt between the pulleys, a suitable clearance between pulley rims should be allowed; (3) The departing direction of the belt from the snub pulley (plus clearance for belt fasteners, etc.) must be below the deck plate or on the underside of the carrying idlers. These limitations will be found to restrict a snubbed drive to an angle of wrap not exceeding 240 degrees, in most cases. If a greater angle of wrap is necessary, it may be necessary to use a dual-pulley drive.

**Dual-Pulley Drives**

A dual-pulley drive uses two or more separate motors, one or more driving the primary drive pulley and one or more driving the secondary drive pulley. Table 6-8, Wrap Factor  $C_w$ , shows the major increase in wrap that becomes available when using a dual-pulley drive. This increase in available wrap can often provide for lower maximum belt tension and a more efficient and lower cost conveyor design.

In any such system where two drive pulleys are involved, the secondary pulley starts out with a certain value of  $T_2$ . Contingent on its angle of wrap and the applicable friction coefficient, the secondary pulley can produce a value,  $T_3$ , such that:

$$T_1 - T_3 = T_{ep} \text{ (primary)}$$

$$T_3 - T_2 = T_{es} \text{ (secondary)}$$

$$T_{ep} \text{ (primary)} + T_{es} \text{ (secondary)} = T_e \text{ (total for conveyor)}$$

The value,  $T_3$ , for the secondary pulley, clearly is the only value available to be used as the slack side tension in the preceding primary drive. This value of  $T_3$ , added to the  $T_{ep}$  for the primary pulley, yields  $T_1$ . The sum of the secondary  $T_{es}$  and the primary  $T_{ep}$  yields a total  $T_e$  that the combined two-pulley drive can produce.

For the maximum efficiency of a two-pulley or dual drive, as described above, it is evident that the proportionate size of the two motors employed must be related appropriately to the angles of wrap and the coefficients of friction at the respective pulleys.

The ratio of tight side tension/slack side tension for each of the drive components, when multiplied together, gives the constant that would apply to the combined or total drive. Or, putting this in another way,  $T_1/T_3$  multiplied by  $T_3/T_2$  will equal  $T_1/T_2$ , provided the drive conditions are the same for both pulleys. However, if the primary drive utilizes the clean side of the belt while the secondary drive is permitted to operate on the carrying or dirty side of the belt, the friction coefficient and wrap factor for the secondary pulley will vary and the tension relationship should be investigated.

For any conveyor drive that utilizes more than one drive pulley, a snub pulley arrangement is preferable, so that both pulleys drive on the same clean side of the belt.

The following symbols and formulas will be of assistance in evaluating the drive pulley relationships for dual pulley drives:

$$T_3 = \text{belt tension between the primary and secondary drive pulleys}$$

$$C_{ws} = \text{wrap factor for the secondary drive pulley}$$

$$C_{wp} = \text{wrap factor for the primary drive pulley}$$

$$C_w = \text{the combined wrap factor for both drive pulleys}$$

$$T_{es} = \text{effective tension on the secondary drive pulley}$$

$$T_{ep} = \text{effective tension on the primary drive pulley}$$

$$T_2 = T_e C_w$$

$$T_1 = T_2 + T_e$$

$$T_1 = T_{ep} + T_3$$

$$T_3 = T_2 + T_{es}$$

$$C_{wp} = \frac{T_3}{T_{ep}}$$

$$C_{ws} = \frac{T_3}{T_{es}} - 1$$

$$T_e = T_{es} + T_{ep}, \text{ whence}$$

$$T_2 = (T_{es} + T_{ep})(C_w)$$



$$T_{es} = \frac{T_2}{C_{ws}} \text{ by definition}$$

$$T_{ep} = \frac{T_3}{C_{wp}} \text{ by definition}$$

$$T_3 = T_2 + T_{es}, \text{ whence}$$

$$T_{ep} = \frac{T_2 + T_{es}}{C_{wp}}$$

Substituting:

$$\begin{aligned} T_2 &= \left[ \frac{T_2}{C_{ws}} + \frac{(T_2 + T_{es})}{C_{wp}} \right] C_w \\ &= \left( \frac{T_2}{C_{ws}} + \frac{T_2}{C_{wp}} + \frac{T_{es}}{C_{wp}} \right) C_w \\ T_2 &= \left( \frac{T_2 C_{wp} + T_2 C_{ws} + T_{es} C_{ws}}{C_{ws} C_{wp}} \right) C_w \end{aligned}$$

solving for  $C_w$ , and noting that  $T_{es} C_{ws} = T_2$ ,

$$C_w = \frac{T_2 C_{ws} C_{wp}}{T_2 C_{wp} + T_2 C_{ws} + T_2}, \text{ and because } T_{es} C_{ws} = T_2,$$

$$C_w = \frac{C_{ws} C_{wp}}{C_{wp} + C_{ws} + 1}$$

For example, if the angles of wrap of the primary and secondary drive pulleys are 180 degrees and 220 degrees, respectively, the factors are as follows for lagged pulleys (see Table 6-8):

$$C_{ws} = 0.35 \text{ for } 220^\circ \text{ angle of wrap}$$

$$C_{wp} = 0.50 \text{ for } 180^\circ \text{ angle of wrap}$$

$$C_w = 0.095 \text{ for } 400^\circ \text{ total angle of wrap by interpolating between } 380^\circ \text{ and } 420^\circ, \text{ or:}$$

$$C_w = \frac{C_{wp}C_{ws}}{1 + C_{wp} + C_{ws}} = \frac{(0.5)(0.35)}{1 + 0.35 + 0.50} = 0.0945$$

The tensions exterior to a dual-pulley drive are the same as those for a single-pulley drive.

A part of the effective tension,  $T_e$  is taken on the primary drive pulley and a part on the secondary drive pulley. Using two motors, the ratio of  $T_{ep}$  to  $T_{es}$  is the ratio of the horsepower ratings of the two motors.

For example, if the total calculated horsepower is 250, this could be supplied, allowing for drive losses, by using a 200-horsepower primary drive and a 75-horsepower secondary drive, with a drive efficiency of 90 percent.

The primary pulley would take

$$(200/275)(250) = 182\text{hp}$$

The secondary pulley would take

$$(75/275)(250) = 68\text{hp}$$

If the belt velocity,  $V$ , is 400 fpm, then

$$T_{ep} = \frac{(182)(33,000)}{400} = 15,000 \text{ lbs}$$

$$T_{es} = \frac{(68)(33,000)}{400} = 5,625 \text{ lbs}$$

and

$$\frac{T_{ep}}{T_{es}} = \frac{15,000}{5,625} = 2.67$$

(see Problem 1, page 145.)

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## Drive Arrangements

The final selection and design of a conveyor drive arrangement is influenced by many factors, including the performance requirements, the preferred physical location, and relative costs of components and installation.

Figures 6.6A through 6.7F illustrate some of the drive combinations that have been furnished. Other arrangements may be better suited to a particular conveyor in a particular location. CEMA member companies can assist in final recommendations.

Note that the illustrated arrangements that are on a substantially downhill run are usually regenerative and are so indicated in the title.

Figure 6.6 Single-pulley/drive arrangements.

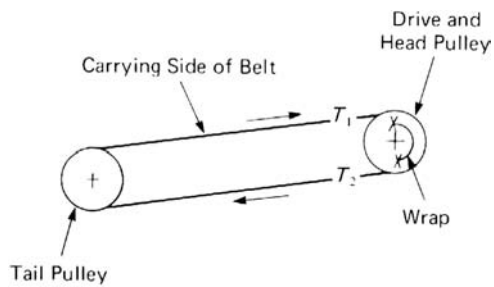


Figure 6.6A Single-pulley drive at head end of conveyor without snub pulley.

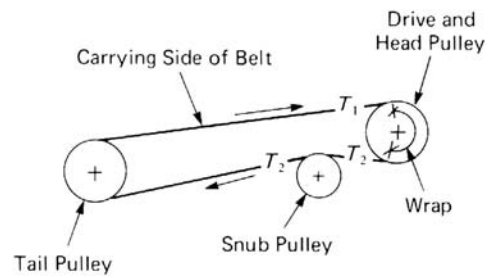


Figure 6.6B Single-pulley drive at head end of conveyor with snub pulley.

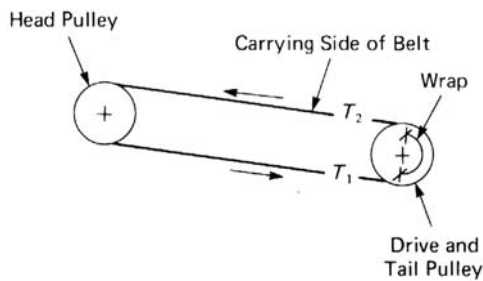


Figure 6.6C Single-pulley drive at tail end without snub pulley. Used when head end drive cannot be applied.

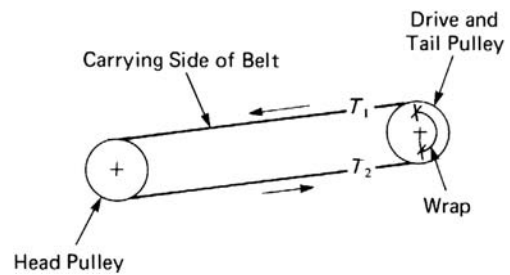


Figure 6.6D Single-pulley drive at tail end of conveyor without snub pulley; regenerative.

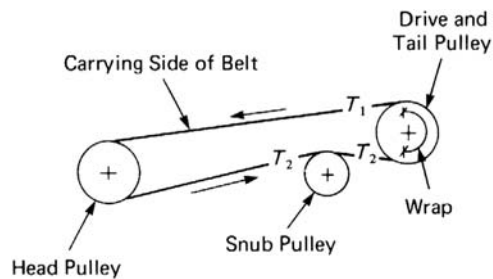


Figure 6.6E Single-pulley drive at tail end of conveyor with snub pulley; regenerative.

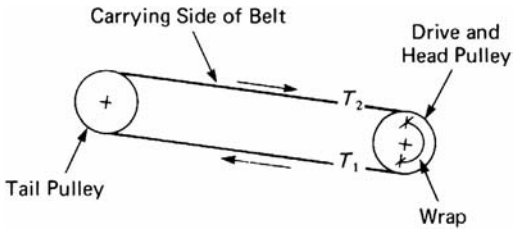


Figure 6.6F Single-pulley drive at head end of conveyor without snub pulley; regenerative.

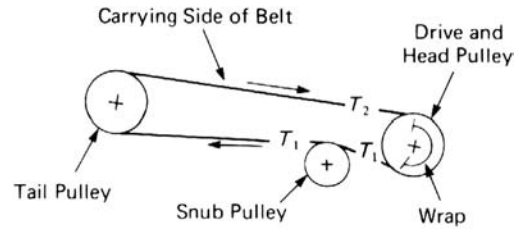


Figure 6.6G Single-pulley drive at head end of conveyor with snub pulley; regenerative.

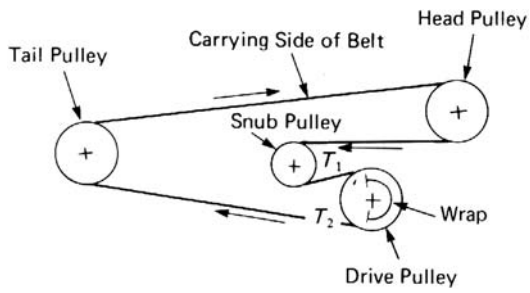


Figure 6.6H Single-pulley drive on return run.

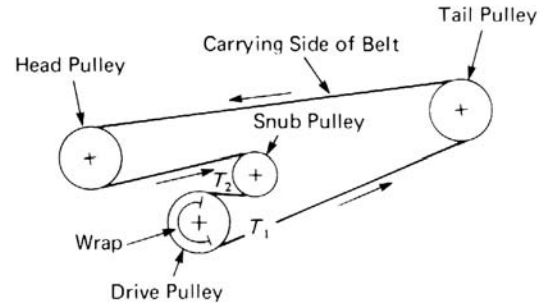


Figure 6.6I Single-pulley drive on return run; regenerative.

Figure 6.7 Dual-pulley drive arrangements.

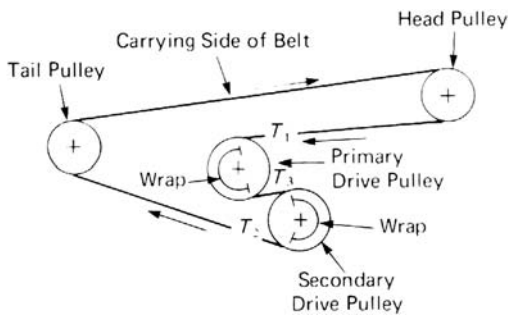


Figure 6.7A Dual-pulley drive on return run.

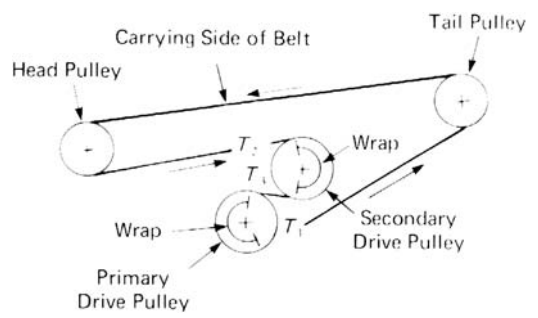


Figure 6.7B Dual-pulley drive on return run; regenerative.

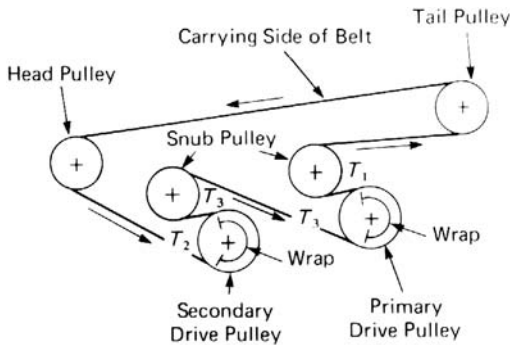


Figure 6.7C Dual-pulley drive on return run; regenerative.

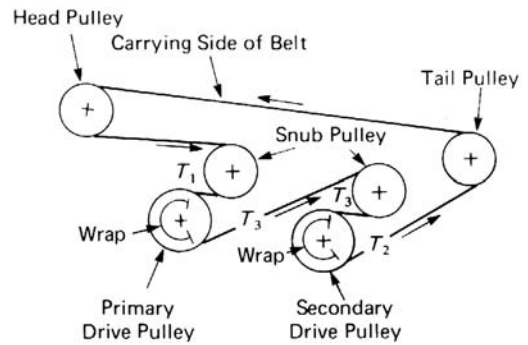


Figure 6.7D Dual-pulley drive on return run. Drive pulleys engage clean side of belt.

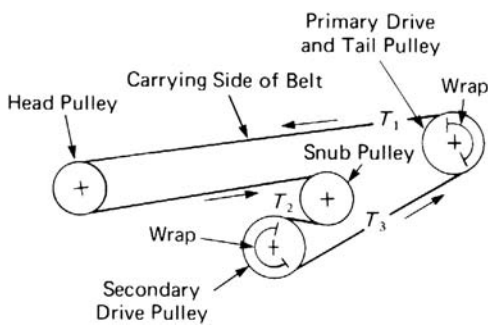


Figure 6.7E Dual-pulley drive with primary drive on tail pulley of conveyor; regenerative.

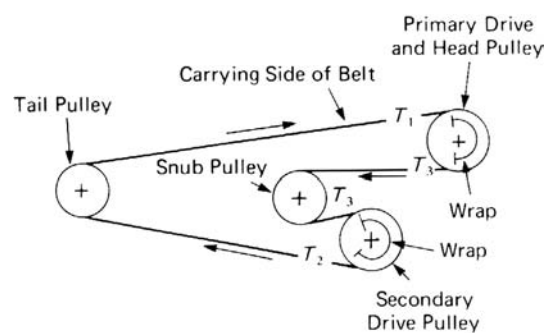


Figure 6.7F Dual-pulley drive with primary drive on head pulley of conveyor.

## Maximum and Minimum Belt Tensions

For the illustrated common conveyor profiles and drive arrangements, minimum and maximum tensions will be discussed and procedures given for calculating the belt tension at any point in the conveyor. The applicable formulas are indicated with the various profile and drive arrangements where single-pulley drives are involved. The tensions involved in multiple-pulley drives are treated separately.

### Maximum Belt Tension

*Operating Maximum Belt Tension.* The operating maximum belt tension is defined as the maximum belt tension occurring when the belt is conveying the design load from the loading point continuously to the point of discharge. Operating maximum tension usually occurs at the discharge point on horizontal or inclined conveyors and at the loading point on regenerative declined conveyors. On compound conveyors, the operating maximum belt tension frequently occurs elsewhere. Because the operating maximum belt tension must be known to select a belt, its location and magnitude must be determined. For details on belt tensions, refer to Figures 6.8 through 6.16.

Conveyors having horizontal and lowering, or horizontal and elevating, sections can have maximum tensions at points other than a terminal pulley. In this case, belt tensions can be calculated by considering the horizontal and sloping sections as separate conveyors.

*Temporary Operating Maximum Belt Tension.* A temporary operating maximum belt tension is that maximum tension which occurs only for short periods. For example, a conveyor with a profile that contains an incline, a decline, and then another incline, may generate a higher operating tension when only the inclines are loaded and the decline is empty. These temporary operating maximum belt tensions should be considered in the selection of the belt and the conveyor machinery.

### Starting and Stopping Maximum Tension

The starting torque of an electric motor may be more than  $2\frac{1}{2}$  times the motor full-load rating. Such a torque transmitted to a conveyor belt could result in starting tensions many times more than the chosen operating tension. To prevent progressive weakening of splices and subsequent failure, such starting maximum tensions should be avoided. Refer to Chapter 13. Likewise, if the belt is brought to rest very rapidly, especially on decline conveyors, the inertia of the loaded belt may produce high tensions.

The generally recommended maximum for starting belt tension is 150 percent of the allowable belt working tension. On conveyors with tensions under 75 lbs/ply in or the equivalent, the maximum can be increased to as high as 180 percent. For final design allowances, conveyor equipment or rubber belt manufacturers should be consulted.

### Minimum Belt Tension, $T_{min}$

For conveyors that do not overhaul the drive, the minimum belt tension on the carrying run will usually occur at the tail (feed) end. For conveyors that do overhaul their drive, the minimum belt tension will usually occur at the head (discharge) end. The locations and magnitude of minimum belt tensions are given in connection with the conveyor profiles and drives shown in Figures 6.8 through 6.16.

It will be seen that the minimum tension is influenced by the  $T_2$  tension required to drive, without slippage of the belt on the pulley, and by the  $T_0$  tension required to

limit the belt sag at the point of minimum tension. The minimum tension is calculated both ways and the larger value used. If  $T_0$  to limit belt sag is larger than the  $T_{min}$  produced by the  $T_2$  tension necessary to drive the belt without slippage, a new  $T_2$  tension is calculated, using  $T_0$  and considering the slope tension,  $T_b$ , and the return belt friction,  $T_{yr}$ . Formulas for calculating  $T_2$ , having  $T_0$ ,  $T_b$ , and  $T_{yr}$  are given for each of the conveyor profiles and drive arrangements.

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### **Tension Relationships and Belt Sag Between Idlers**

Chapter 5, “Belt Conveyor Idlers,” presents the basic facts on the subject of idler spacing. A major requirement, noted in Chapter 5, is that the sag of the belt between idlers must be limited to avoid spillage of conveyed material over the edges of the belt. The sag between idlers is closely related to the weight of the belt and material, the idler spacing, and the tension in the belt.

#### **Graduated Spacing of Troughing Idlers**

For belt conveyors with long centers, it is practical to vary the idler spacing so as to equalize the catenary sag of the belt as the belt tension increases.

The basic equation for the sag in a catenary can be written:

$$\text{Sag, ft} = \frac{WS_i^2}{8T}$$

where :

$W$  = weight, ( $W_b + W_m$ ), lbs/ft of belt and material

$S_i$  = idler spacing, ft

$T$  = tension in belt, lbs

The basic sag formula can also be expressed as a relation of belt tension,  $T$ , idler spacing,  $S_i$ , and the weight per foot of belt and load, ( $W_b + W_m$ ), in the form:

$y$  = vertical drop (sag) between idlers, ft

$$y = \frac{S_i^2(W_b + W_m)}{8T}$$

Experience has shown that when a conveyor belt sags more than 3 percent of the span between idlers, load spillage often results. For 3 percent sag the equation becomes:

$$\frac{S_i^2(W_b + W_m)}{8T} = \frac{3S_i}{100}$$

While pure catenary equations are used, the allowable percent sag takes into account such factors as stiffness of the belt carcass, strength of the belt span due to the “channel” shape of a troughed belt, etc.

Simplifying for minimum tension to produce various percentages of belt sag yields the following formulas:

$$\text{For 3 percent sag, } T_0 = 4.2S_i(W_b + W_m)$$

$$\text{For 2 percent sag, } T_0 = 6.25S_i(W_b + W_m)$$

$$\text{For 1½ percent sag, } T_0 = 8.4S_i(W_b + W_m)$$

See Table 6-10 for recommended belt sag percentages for various full load conditions.

The graduated spacing should be calculated to observe the following limitations: (1) A maximum of 3 percent sag should be maintained when belt is operating with a normal load. (2) A maximum of 4.5 percent sag should be maintained when the loaded belt is standing still. (3) The idler spacing should not exceed twice the suggested normal spacing of the troughing idlers listed in Table 5-2. (4) The load on any idler should never exceed the idler load ratings given in Chapter 5.

Moreover, the number of spacing variations must be based on practical considerations, such as the number of different stringer sections in the conveyor support structure, so that the fabrication cost of the support structure does not become excessive. Usually, the spacing of troughing idlers is varied in 6-inch increments.

Limiting the calculated belt sag to 3 percent of the idler spacing, at any point on the conveyor, will generally prevent spillage of the material from conveyor belts operating over 20 degrees troughing idlers.

When handling lumpy material on belts operating on 35 degrees (or deeper) troughing idlers, belt tension should be increased to reduce the percent of sag. Deep-troughed conveyor belts normally carry a relatively large cross-sectional loading and corresponding heavy weight of material per foot of length. Therefore, the material exerts a greater pressure against the side of the trough, tending to cause greater transverse belt flexure. The purpose of increasing the minimum belt tension in belts operating on idlers of greater than 20 degrees troughing angle is to keep this transverse belt flexure to an acceptable minimum and thus prevent spillage.

Similarly, when frequent surge loads are encountered or a substantial percentage of large lumps is expected, the material weight per foot of conveyor will be increased. Consideration of increased minimum belt tension at, or closely adjacent to, the loading points is recommended.



Table 6-10. Recommended belt sag percentages for various full load conditions.

Idler Troughing Angle	Material		
	All Fines	One-half the Maximum Lump Size*	Maximum Lump Size*
20°	3%	3%	3%
35°	3%	2%	2%
45°	3%	2%	1½%

\*See Figure 4.1

**Note:** Reduced load cross sections will permit an increase in the sag percentage, resulting in a decreased minimum tension. Such a choice may lead to a more economical belt selection, as the maximum tension will be reduced correspondingly.

*Slack Side Tension,  $T_2$ .* The minimum tension required to drive the belt without slippage is the product of  $T_e$  and  $C_w$ . However, the value to be used for minimum belt tension on the carrying run is either  $T_0$  (calculated as above) plus or minus the tension  $T_b$  and plus or minus the return belt friction  $T_{yr}$ , or the minimum tension to drive without slippage  $T_e \times C_w$ . By rearranging and substituting terms,

$$T_2 = T_0 \pm T_b \pm T_{yr}$$

or, by the above definition,

$$T_2 = T_e C_w$$

Use the larger value of  $T_2$ .

*Tension,  $T_b$ .* The weight of the carrying and/or return run belt for a sloped conveyor is carried on the pulley at the top of the slope. This must be considered in calculating the  $T_2$  tension, as indicated above.

$$T_b = HW_b$$

where:

$W_b$  = weight of belt, lbs/ft

$H$  = net change in elevation, ft

*Return Belt Friction Tension,  $T_{yr}$ .* The return belt friction is the belt tension resulting from the empty belt moving over the return run idlers:

$$T_{yr} = 0.015LW_bK_t$$

where :

- $L$  = length, ft, of conveyor to center of terminal pulleys  
 $K_t$  = temperature correction factor as defined on page 89

For temperatures above 32°F,  $K_t = 1.0$

*Belt Tensions for Conveyors of Marginal Decline.* Allowances made for frictional losses in a conveyor are intended as conservative assumptions. When a declined conveyor is involved, such allowances should be discounted when considered in conjunction with maximum possible regenerative tensions (or hp).

*Belt Tensions for Typical Conveyors.* When calculating tensions at any point in these conveyor profiles, the portions of the conveyor on zero slope, inclines, or declines should be considered as separate conveyors.

*Belt Tension at Any Point, X, on Conveyor Length.* In order to understand clearly the formulas for evaluating the belt tension at any point, X, on the belt conveyor length, it is necessary to establish the following nomenclature:

- $L_x$  = distance, ft, from tail pulley to point X along the conveyor  
 $H_x$  = vertical distance, ft, from tail pulley to point X  
 $T_{cx}$  = belt tension, lbs, at point X on the carrying run  
 $T_{rx}$  = belt tension, lbs, at point X on the return run  
 $T_t$  = belt tension, lbs, at tail pulley  
 $T_{hp}$  = belt tension, lbs, at head pulley  
 $T_{wcx}$  = tension, lbs, at point X on the carrying run, resulting from the weight of belt and material carried  
 $T_{fcx}$  = tension, lbs, at point X on the carrying run, resulting from friction  
 $T_{wrx}$  = tension, lbs, at point X on the return run, resulting from the weight of the empty belt  
 $T_{frx}$  = tension, lbs, at point X on the return run, resulting from friction  
 $T_{wcx} = H_x (W_b + W_m)$   
 $T_{fcx} = L_x [K_t K_x + K_y W_b] + L_x K_y W_m$   
 $T_{wrx} = H_x W_b$   
 $T_{frx} = 0.015 L_x W_b K_t$

Formulas for  $T_{cx}$  and  $T_{rx}$  are given for all the belt conveyor profiles and drives in Figures 6.8 through 6.16. [Tensions from conveyor accessories ( $T_{ac}$ ) have been omitted for clarity; however, these should be accounted for in the final design.]

### Analysis of Belt Tensions

In addition to calculation of the effective belt tension,  $T_e$ , which occurs at the drive pulley, a designer must consider the belt tension values that occur at other points of the conveyor's belt path.

Figures 6.8 through 6.16 illustrate various possible conveyor layouts and profiles and the appropriate tension analysis. Some of these examples are more commonly applied than others; the order of presentation is not intended to infer preference of design. Many of these diagrams illustrate the takeup,  $TU$ , in alternate locations. It is most unusual for a conveyor to employ more than one takeup; a preferred single location should be chosen.

Figure 6.8 Head pulley drive — horizontal or elevating.\*

$$T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_t + T_b - T_{yr}$$

Use the larger value of  $T_2$

$$T_t = T_0 \text{ or}$$

$$T_t = T_2 - T_b + T_{yr}$$

Use the larger value of  $T_t$

$$T_t = T_{min} \quad T_1 = T_{max}$$

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

$$T_{rx} = T_t + T_{wrx} - T_{frx}$$

\*With decline conveyors the  $T_e$  tension required for an empty conveyor may sometimes be greater than the  $T_e$  for the loaded conveyor.

Figure 6.8A Inclined conveyor with head pulley drive.

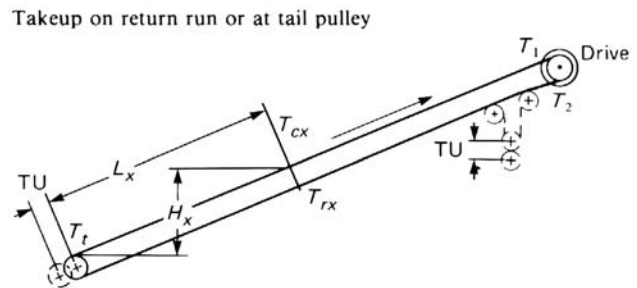


Figure 6.8B Horizontal belt conveyor with concave vertical curve, and head pulley drive.

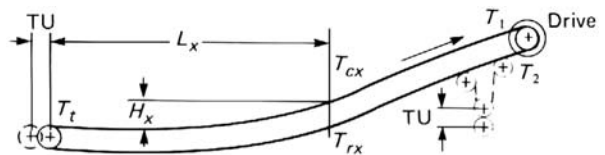
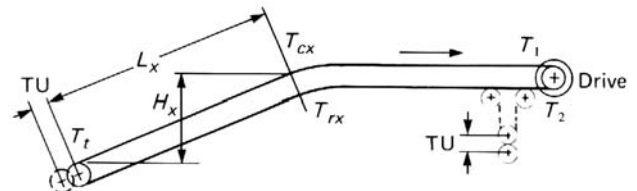


Figure 6.8C Horizontal belt conveyor with convex vertical curve, and head pulley drive.



**NOTE:** Two takeups are shown only to illustrate alternative. Two automatic takeups cannot function properly on the same conveyor.

Figure 6.9 Head pulley drive — lowering without regenerative load.\*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o - T_e \text{ or } T_2 = T_o - T_b - T_{yr}$$

Use the larger value of  $T_2$

$$T_1 = T_e + T_2$$

$$T_t = T_2 + T_b + T_{yr} \text{ or } T_t = T_o + T_b + T_{yr} - T_e \text{ or } T_t = T_o$$

$$T_{max} = T_t \text{ or } T_1$$

$$T_{min} = T_t + T_1$$

$$T_{cx} = T_t - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_t - T_{wrx} - T_{frx}$$

\*With decline conveyors the  $T_e$  tension required for an empty conveyor may sometimes be greater than the  $T_e$  for the loaded conveyor.

Figure 6.9A Declined belt conveyor with head pulley drive. Lowering without regenerative load.

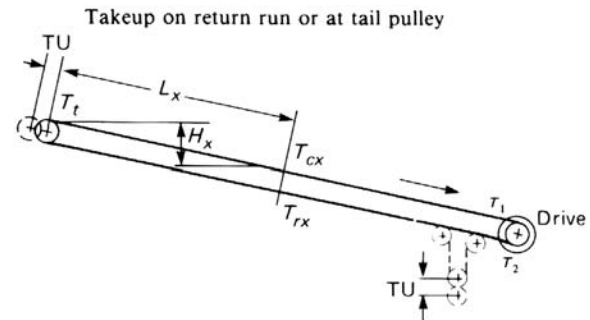


Figure 6.9B Conveyor with convex vertical curve, head pulley drive. Lowering without regenerative load.

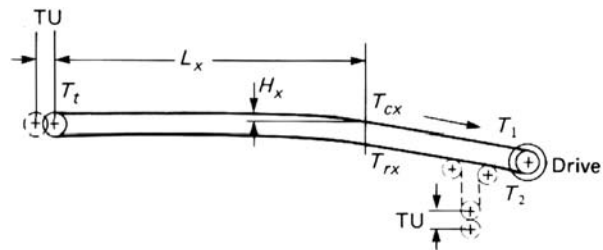
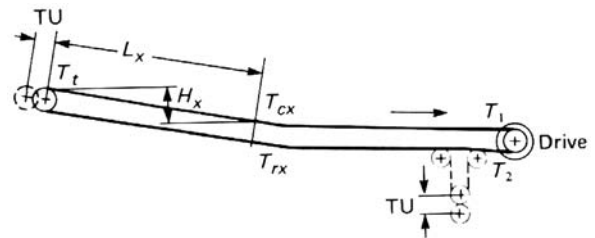


Figure 6.9C Conveyor with concave vertical curve, head pulley drive. Lowering without regenerative load.



**NOTE:** Two takeups are shown only to illustrate alternative. Two automatic takeups cannot function properly on the same conveyor.

Figure 6.10 Head pulley drive — lowering with regenerative load.

$$* T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o$$

Use the larger value of  $T_2$ .

$$T_t = T_{max} = T_1 + T_b + T_{yr} \text{ or } T_t = T_e + T_o + T_b + T_{yr}$$

$$T_{min} = T_2$$

$$T_{cx} = T_t - T_{wcx} + T_{frx}$$

$$T_{rx} = T_t - T_{ywrx} - T_{frx}$$

\*See page 119.

Takeup on return run not recommended to avoid driving through the takeup.

Figure 6.10A Declined belt conveyor with head pulley drive. Lowering with regenerative load.

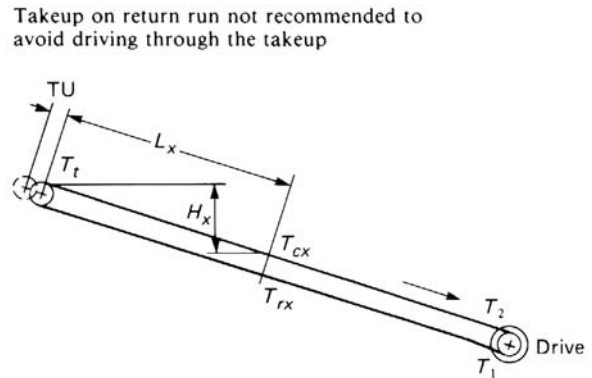


Figure 6.10B Conveyor with convex vertical curve, head pulley drive. Lowering with regenerative load.

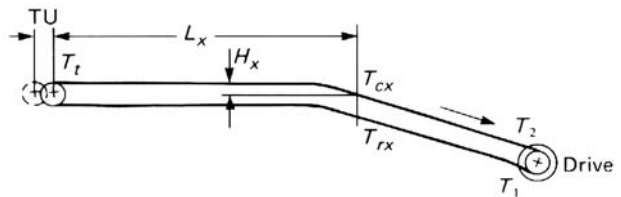


Figure 6.10C Conveyor with concave vertical curve, head pulley drive. Lowering with regenerative load.

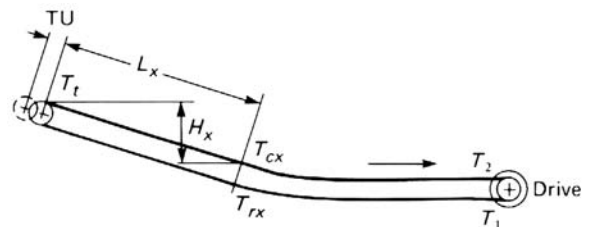


Figure 6.11 Tail pulley drive — horizontal or elevating.

$$T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_0$$

Use the larger value of  $T_2$

$$T_t = T_2$$

$$T_{min} = T_2$$

$$T_{hp} = T_1 - T_{yr} + T_b$$

$$T_{max} = T_1 \text{ or } T_{max} = T_{hp} \text{ Use the larger value of } T_{max}$$

$$T_{cx} = T_2 + T_{wcx} + T_{fcx}$$

$$T_{rx} = T_1 + T_{wrx} - T_{frx}$$

With a tail pulley drive more of the belt is under high tension and it is usually not practical to locate takeup at  $T_2$  just before the loading point. Calculate belt tension at takeup during acceleration and make adequate to avoid belt slip.

Figure 6.11A Inclined conveyor with tail pulley drive.

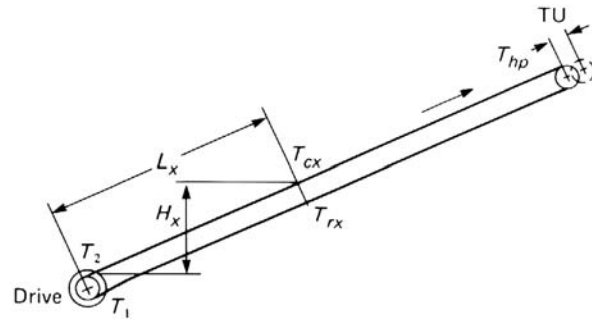


Figure 6.11B Horizontal belt conveyor with concave vertical curve and tail pulley drive.

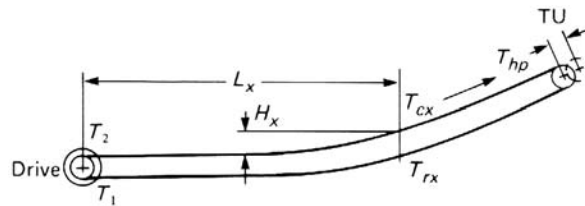


Figure 6.11C Horizontal belt conveyor with convex vertical curve and tail pulley drive.

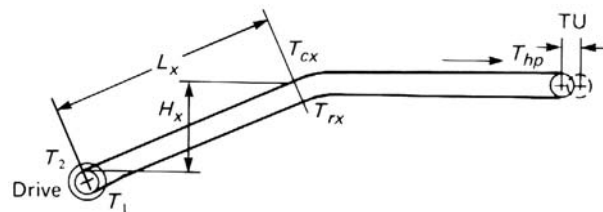


Figure 6.12 Tail pulley drive — lowering without regenerative load.\*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o + T_b + T_{yr} - T_e$$

Use the larger value of  $T_2$

$$T_{hp} = T_o \text{ or } T_{hp} = T_1 - T_b - T_{yr}$$

Use the larger value of  $T_{hp}$

$$T_{hp} = T_{min}$$

$$T_1 = T_e + T_2 = T_{max}$$

$$T_{cx} = T_2 - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_1 - T_{wrx} - T_{frx}$$

\*See page 119.

Takeup on return run not recommended to avoid driving through the takeup.

Figure 6.12A Declined belt conveyor with tail pulley drive. Lowering without regenerative load.

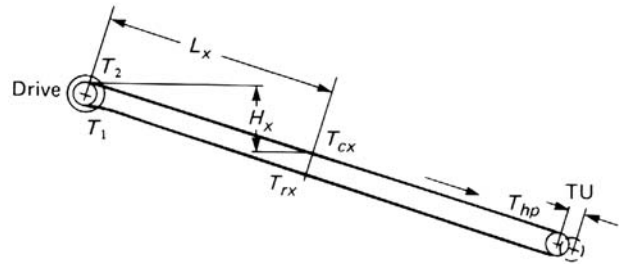


Figure 6.12B Conveyor with convex vertical curve, tail pulley drive. Lowering without regenerative load.

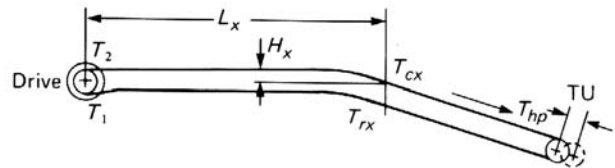


Figure 6.12C Conveyor with concave vertical curve, tail pulley drive. Lowering without regenerative load.

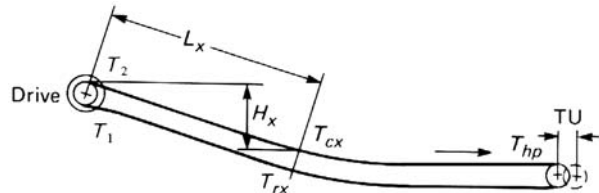


Figure 6.13 Tail pulley drive — lowering with regenerative load.\*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o + T_b + T_{yr}$$

Use the larger value of  $T_2$

$$T_{hp} = T_2 - T_b - T_{yr} \text{ or } T_{hp} = T_o$$

$$T_{hp} = T_{min}$$

$$T_1 = T_{max} = T_e + T_2$$

$$T_{cx} = T_1 - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_2 - T_{wrx} - T_{frx}$$

\*Calculate belt tension required at takeup during acceleration and make takeup adequate to prevent lift-up. See page 119.

Figure 6.13A Declined belt conveyor with tail pulley drive. Lowering with regenerative load.

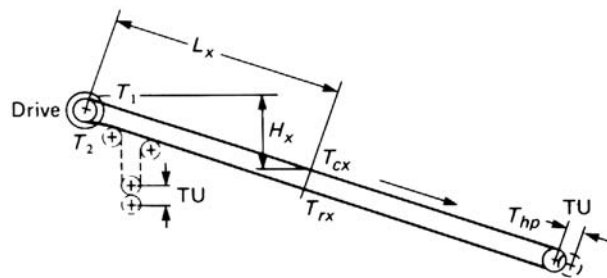


Figure 6.13B Conveyor with concave vertical curve, tail pulley drive. Lowering with regenerative load.

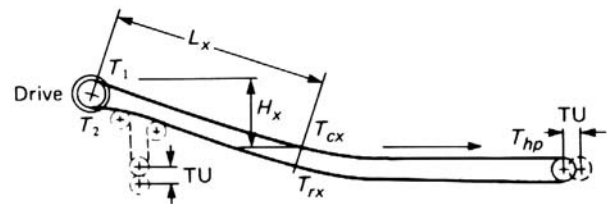
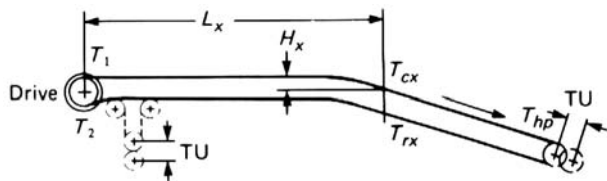


Figure 6.13C Conveyor with convex vertical curve, tail pulley drive. Lowering with regenerative load.



**NOTE:** Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.



Figure 6.14 Drive on return run — horizontal or elevating.\*

$$T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or}$$

$$T_2 = T_o - 0.015W_bL_s + W_bH_d$$

here

$H_d$  = lift to the drive pulley

Use the larger value of  $T_2$

$$T_t = T_{min} \text{ and } T_t = T_o$$

$$T_t = T_2 + 0.015W_bL_s - W_bH_d$$

Use the larger value of  $T_t$

$$T_{hp} = T_e + T_2 + \frac{L - L_s}{L}(T_b - T_{yr}) \text{ or } T_{hp} = T_e + T_o + T_b - T_{yr}$$

$$T_{hp} = T_{max}$$

$$T_{cx} = T_t + T_{fcx} + T_{wcx}$$

$$T_{rx} = T_t - T_{frx} + T_{wrx}$$

Takeups on return run or at tail pulley. \*See page 119

Figure 6.14A Inclined conveyor with drive on return run.

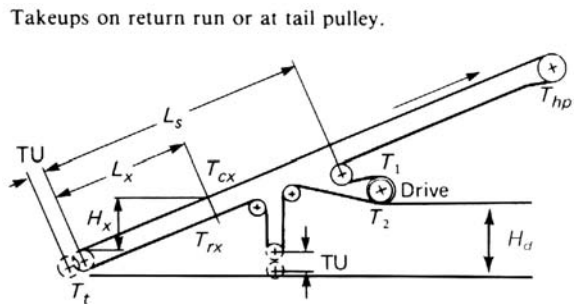


Figure 6.14B Horizontal belt conveyor with concave vertical curve, and drive on return run.

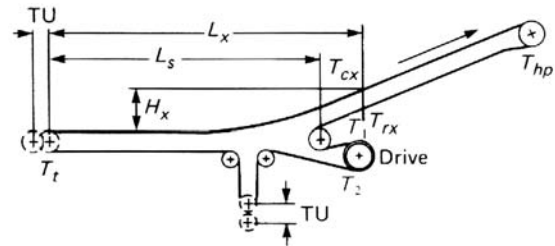


Figure 6.14C Horizontal belt conveyor with convex vertical curve, and drive on return run.

**NOTE:** Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.

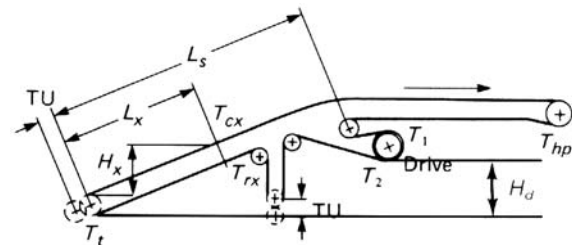


Figure 6.15 Drive on return run — lowering without regenerative load.\*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o - T_e \text{ or } T_2 = T_o - 0.015W_b(t_s) - W_bH_d$$

Use the larger value of  $T_2$

$$T_1 = T_e + T_2$$

$$T_t = T_2 + 0.015W_bL_s + W_bH_d$$

here

$$H_d = \text{lift to the drive pulley, or } T_t = T_o$$

Use the larger value of  $T_t$

$$T_{max} = T_t \text{ or } T_1$$

$$T_{min} = T_t \text{ or } T_1$$

$$T_{hp} = T_t + T_{fcx} - T_{wcx}$$

here

$$L_x = L$$

$$T_{cx} = T_t + T_{fcx} - T_{wcx}$$

$$T_{rx} = T_t - T_{frx} - T_{wrx}$$

Takeup on return run or at tail pulley. \*See page 119.

Figure 6.15A Declined conveyor, with drive on return run. Lowering without regenerative load.

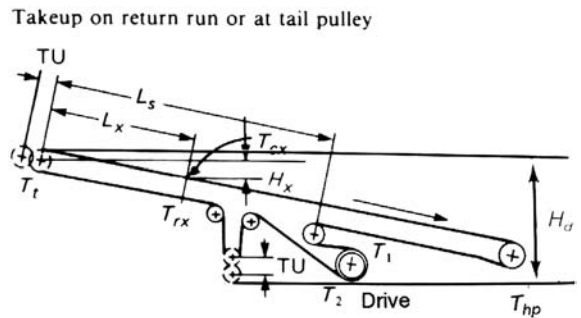


Figure 6.15B Conveyor with convex vertical curve, drive on return run. Lowering without regenerative load.

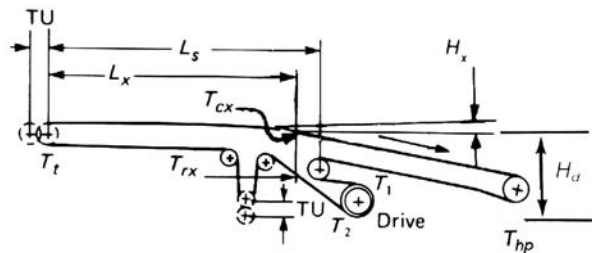
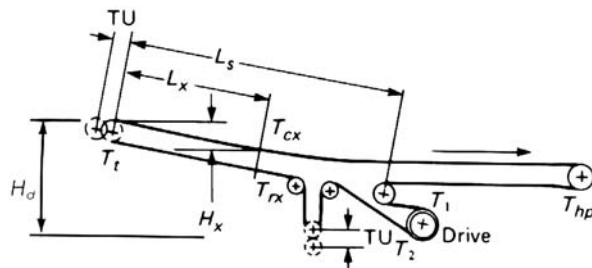


Figure 6.15C Conveyor with concave vertical curve, drive on return run. Lowering without regenerative load.



**NOTE:** Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.

Figure 6.16 Drive on return run — lowering with regenerative load.\*

$$*T_e = T_1 - T_2$$

$$T_2 = C_w \times T_e \text{ or } T_2 = T_o + 0.015 W_b(L - L_s) + W_b(H - H_d)$$

Use the larger value of  $T_2$

$$T_1 = T_e + T_2$$

$$T_t = T_{max} = T_1 + 0.015 W_b L_s + W_b H_d$$

here

$H_d$  = lift to the drive pulley

$$T_{hp} = T_o$$

$$T_{hp} = T_2 - 0.015 W_b(L - L_s) - W_b(H - H_d)$$

Use the larger value of  $T_{hp}$

$$T_{cx} = T_t - T_{wcx} + T_{fcx}$$

$$T_{rx} = T_t - T_{wrx} - T_{frx}$$

\*See page 119

Figure 6.16A Declined conveyor with drive on return run. Lowering with regenerative load.

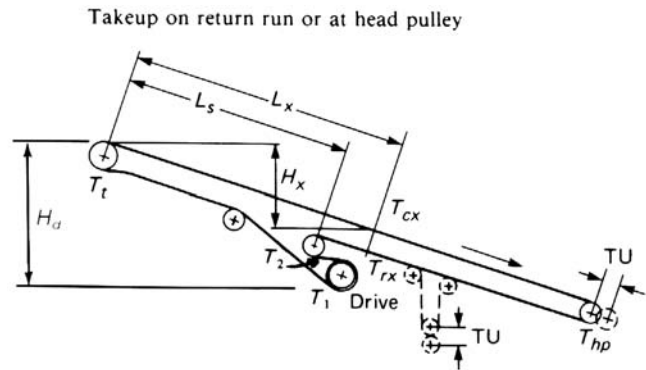


Figure 6.16B Conveyor with concave vertical curve, drive on return run. Lowering with regenerative load.

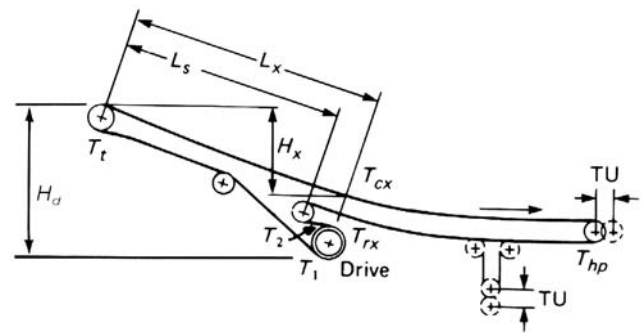
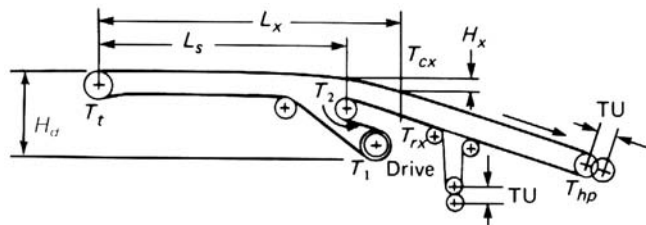


Figure 6.16C Conveyor with convex vertical curve, drive on return run. Lowering with regenerative load.



**NOTE:** Two takeups are shown only to illustrate alternatives. Two automatic takeups cannot function properly on the same conveyor.

**Belt Tension Examples**

A typical calculation of the various tensions in a conveyor belt with single-pulley drive is given below.

*Example 1*

Calculate the various belt tensions for a 30-inch belt conveyor per Figure 6.8A, with 300-ft centers, and a lift of 50-ft. Capacity is 500 tph, of material weighing 100 pounds per cubic ft, at a belt speed of 350 fpm. The belt is carried on 5-inch diam. Class C5 idlers (see Chapter 5) with 3/4-in. shafts, 35 degrees end roll angle. Idlers are spaced every 3 1/2 ft. The material contains 50 percent lumps. Belt weight is 15.0 lbs/ft,  $W_b$ . Material weight is 47.5 lbs/ft,  $W_m$ . Temperature is 60°F.  $T_e$  has been calculated and is equal to 3,030 lbs.

*Step 1:* Determine  $C_w$ . Assume lagged pulley, gravity takeup, and 180 degree wrap. Table 6-8 gives  $C_w = 0.50$ .

*Step 2:* Determine belt tension,  $T_2$ . Minimum  $T_2$  to drive =  $T_e \times C_w = (3,030)(0.50) = 1,515$  lbs.  $T_0$ , the minimum allowable tension for a 2 percent belt sag, per page 120, is as follows:

$$T_0 = 6.25 S_i (W_b + W_m) = (6.25)(3.5)(15.0 + 47.5) = 1,367 \text{ lbs}$$

Using the formula for determining return belt friction tension, pages 87-88,

$$\begin{aligned} L &= 300 \\ W &= 15 \\ K_t &= 1.0, \text{ at } 60^\circ\text{F.} \\ T_{yr} &= 0.015LW_bK_t = (0.015)(300)(15)(1) = 68 \text{ lbs} \\ T_2 \text{ (considering } T_0) &= T_0 + T_b - T_{yr} \\ T_b &= HW_b = (50)(15) = 750 \text{ lbs} \end{aligned}$$

*Therefore:*  $T_2 = 1,367 + 750 - 68 = 2,049$  lbs

Because this is larger than the 1,515 lbs minimum  $T_2$  to drive, use  $T_2 = 2,049$  lbs.

*Step 3:* Calculate the  $T_1$ ,  $T_{max}$ , and takeup tensions.

$$T_{max} = T_1 = T_e + T_2 = 3,030 + 2,049 = 5,079 \text{ lbs}$$

Takeup tension depends upon the location of the gravity takeup. If located near the head end, the belt tension at takeup is  $T_2$  less the weight of nearly 3 ft of belt (45 lbs), or  $2,049 - 45 = 2,004$  lbs. If located near the tail pulley, the takeup tension will be approximately the same as  $T_0$ :

$$T_0 = T_2 + T_{yr} - T_b = 2,049 + 68 - 750 = 1,367 \text{ lbs}$$

(The non-driving pulley frictions have been omitted.)

A typical calculation of the belt tensions of a two-pulley (dual) drive is given below. The conveyor outline is per Figure 6.14A, but with a dual-pulley head drive, per Figure 6.7F.

**Example 2**

Conveyor length = 1,200 ft  
 Belt speed = 400 fpm  
 $T_e$  at the drive pulleys = 20,625 lbs  
 Required hp at the drive pulleys = 250 hp  
 Total motor horsepower = 275 hp  
 Primary drive motor = 200 hp  
 Secondary drive motor = 75 hp  
 Belt weight,  $W_b = 20$  lbs/ft  
 $C_w = 0.11$ , according to Table 6-8 (380° wrap, lagged pulleys)

Step 1: Calculate  $T_{ep}$  and  $T_{es}$  :

$$T_{ep} = \left(\frac{200}{275}\right)(250)\left(\frac{33,000}{400}\right) = 15,000 \text{ lbs}$$

$$T_{es} = \left(\frac{75}{275}\right)(250)\left(\frac{33,000}{400}\right) = 5,625 \text{ lbs}$$

Step 2: Calculate  $T_2$ , which is the minimum value that avoids slippage of the belt on the secondary pulley:

$$T_2 = T_e C_w = (20,625)(0.11) = 2,269 \text{ lbs}$$

Step 3: Calculate  $T_3$ :  $T_3 = T_2 + T_{es} = 2,269 + 5,625 = 7,894$  lbs

Step 4: Calculate  $T_1$ :  $T_1 = T_3 + T_{ep} = 7,894 + 15,000 = 22,894$  lbs

Step 5: Calculate  $C_{wp}$  and  $C_{ws}$  :

$$C_{wp} = \frac{T_3}{T_{ep}} = \frac{7,894}{15,000} = 0.53, \text{ requiring } 180^\circ \text{ wrap angle}$$

$$C_{ws} = \frac{T_3}{T_{es}} - 1 = \frac{7,894}{5,625} - 1 = 0.40, \text{ requiring } 205^\circ \text{ wrap angle}$$

Step 6: Check  $T_2$ , using formula in Figure 6.14A. Assume conveyor is 1,200 ft long, the lift is 60 ft,  $W_b = 20$ ,  $W_m = 80$ , idler spacing 3½ ft, drive at head of the conveyor.

$$T_0 = T_{min} = 6.25(W_b + W_m)S_i = 6.25(20 + 80)3.5 = 2,188 \text{ lbs}$$

(See “Minimum Belt Tension,” page 113.)

$$T_{yr} = 0.015LW_bK_t = (.015)(1,200)(20)(1) = 360 \text{ lbs}$$

(See “Return Belt Friction Tension,” pages 116-117; assume temperature above 32°,  $K_t = 1.0$ )

$$T_b = HW_b = (60)(20) = 1,200 \text{ lbs}$$

(See “Tension,  $T_b$ ” page 116)

## Tension Relationships and Belt Sag Between Idlers

$$\text{Then } T_2 = T_{min} + T_b - T_{yr} = 2,188 + 1,200 - 360 = 3,028 \text{ lbs}$$

Therefore, as  $T_2$  based on  $T_{min}$  is larger than  $T_2$  minimum to prevent slippage (3,028 is greater than 2,269), use  $T_2 = 3,028$  lbs.

Step 7: Calculate corrected values of  $T_3$ ,  $T_1$ ,  $C_{ws}$ , and  $C_{wp}$ :

$$T_3 = T_2 + T_{es} = 3,028 + 5,625 = 8,653 \text{ lbs}$$

$$T_{max} = T_1 = T_3 + T_{ep} = 8,653 + 15,000 = 23,653 \text{ lbs}$$

$$C_{ws} = \frac{T_3}{T_{es}} - 1 = \frac{8,653}{5,625} - 1 = 0.54$$

$$C_{wp} = \frac{T_3}{T_{ep}} = \frac{8,653}{15,000} = 0.58$$

Based on wrap factors from Step 5, which provide the minimum  $T_2$  tension to drive without slippage, the secondary drive pulley would require a 205-degree angle of wrap, and the primary drive pulley would require a 180 degree angle of wrap.

The revised value of  $T_2$  and the correspondingly revised values of  $C_{wp}$  and  $C_{ws}$ , per Step 7, indicate that both drives could have 180 degree angles of wrap. In order to have equal resistance to slip, both drives should have approximately the same wrap angle.

## Belt Tension Calculations

Five illustrative examples are offered to make clear the use of the formulas in determining the belt tensions at point X on the belt conveyor.

### Example 1

The basis for this example is the conveyor profile in Figure 6.8A from page 118, repeated below.

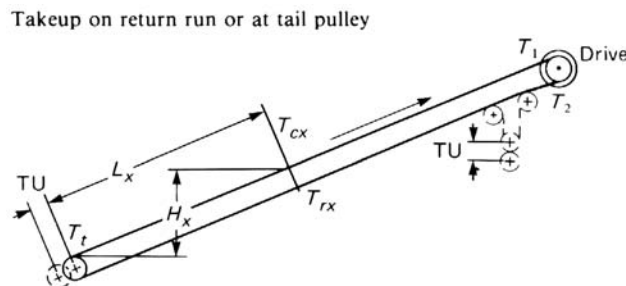


Figure 6.8A.

48-in. belt conveyor

$$W_b = \text{weight of belt} = 15 \text{ lbs/ft}$$

$$W_m = \text{weight of material} = 106.6 \text{ lbs/ft}$$

Troughing idlers, 20-degree angle, Class E6, 6-inch diameter, spaced at 3½ ft, factor  $A_i = 2.8$

Return idlers, Class C6, 6-inch diameter, spaced at 10 ft

$K_t$  = temperature correction factor = 1.0

$T_t$  =  $T_0 = 1,788$  lbs, as  $T_0 = T_{min}$  here

$L_x$  = 1,000 ft

$H_x$  = 31.3 ft

To find the belt tension at point X on the carrying run:

$T_{cx}$  = tension at point X on carrying run

$T_{wcx}$  = tension resulting from weight of belt and material at point X

$T_{fcx}$  = tension resulting from friction on carrying run at point X

$T_{cx} = T_t + T_{wcx} + T_{fcx}$

$T_{wcx} = H_x(W_b + W_m) = (31.3)(121.6) = 3,806$  lbs

$T_{fcx} = L_x K_t (K_x + K_y W_b) + L_x K_y W_m$  when  $K_t = 1.0$ , then

$T_{fcx} = L_x [K_x + K_y (W_b + W_m)]$

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}$$

$$= 0.00068(15 + 106.6) + \frac{2.8}{3.5}$$

$$= 0.883 \text{ (for value of } A_i, \text{ see tabulation on page 91)}$$

$K_y = 0.025$  when conveyor length is 1,000 ft (see Table 6-2). At 3.13% slope,  $W_b + W_m = 121.6$  (use 125 in tables); and 3½ ft standard idler spacing.

$T_{fcx} = 1,000 [0.883 + 0.025(121.6)] = 3,923$  lbs

$T_{cx} = 1,788 + 3,806 + 3,923 = 9,517$  lbs

To find the belt tension at point X on the return run:

$T_{rx}$  = tension at point X on the return run

$T_{wrx}$  = tension at point X on the return run resulting from the weight of the belt

$T_{frx}$  = tension at point X on the return run resulting from return run friction

$T_{rx} = T_t + T_{wrx} - T_{frx}$

$T_{wrx} = H_x W_b = (31.3)(15) = 470$  lbs

$T_{frx} = L_x 0.015 W_b K_t = (1,000)(0.015)(15)(1) = 225$  lbs

$T_{rx} = 1,788 + 470 - 225 = 2,033$  lbs

## Tension Relationships and Belt Sag Between Idlers

### Example 2

The basis for this example is the conveyor profile shown in Figure 6.8B from page 118, repeated below; otherwise, the data is the same as for Example 1, above.

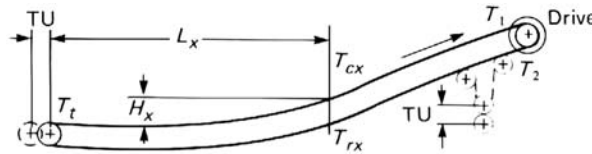


Figure 6.8B.

$$\begin{aligned} L_x &= 1,915 \text{ ft} \\ H_x &= 31.3 \text{ ft} \\ T_t &= 1,788 \text{ lbs} \end{aligned}$$

Horizontal portion 1,565 ft long, inclined portion 835 ft long, on 9 percent slope.

To find the  $T_{cx}$  tension in the carrying run at point X:

$$\begin{aligned} T_{cx} &= T_t + T_{wcx} + T_{fcx} \\ T_{wcx} &= H_x(W_b + W_m) = (31.3)(121.6) = 3,806 \text{ lbs} \\ T_{fcx} &= L_x[K_x + K_y(W_b + W_m)] \text{ Since } K_t = 1.0 \\ T_{fcx} &\text{ is figured in two parts, first for the horizontal portion and then for the} \\ &\text{inclined portion} \end{aligned}$$

For the horizontal portion of the carrying run:

$$\begin{aligned} K_x &= 0.883 \\ K_y &= 0.0277 \text{ when conveyor length is 1,565 ft at 0 degree slope; } W_b + W_m = \\ &121.6, \text{ (use 125 in tables); } 3\frac{1}{2} \text{ ft standard idler spacing} \\ T_{fcx}, \text{ horizontal} &= 1,565 [(0.883) + (0.0277)(121.6)] = 6,653 \text{ lbs} \end{aligned}$$

For the inclined portion of the carrying run:

$$\begin{aligned} K_x &= 0.883 \\ K_y &= 0.0217 \text{ when conveyor length is 1,915 ft at 1.63 percent average slope; } W_b \\ &+ W_m = 121.6, \text{ use 125 in tables; } 3\frac{1}{2} \text{ ft standard idler spacing} \\ T_{fcx}, \text{ incline} &= 352 [(0.883) + (0.0217)(121.6)] = 1,240 \text{ lbs, where 352 ft is the} \\ &\text{distance along the inclined portion to point X. Total } T_{fcx}, \text{ horizontal plus} \\ &\text{incline} = 6,653 + 1,240 = 7,893 \text{ lbs} \\ T_{cx} &= 1,788 + 3,806 + 7,893 = 13,487 \text{ lbs} \end{aligned}$$

To find the  $T_{rx}$  tension in the return belt at point X:

$$\begin{aligned} T_{rx} &= T_t + T_{wrx} - T_{frx} \\ T_{wrx} &= (H_x)(W_b) = (31.3)(15) = 470 \text{ lbs} \\ T_{frx} &= L_x(0.015 W_b) K_t = 1,915(0.015)(15)(1.0) = 431 \text{ lbs} \\ T_{rx} &= 1,788 + 470 - 431 = 1,827 \text{ lbs} \end{aligned}$$



**Example 3**

The basis for this example is the conveyor profile shown in Figure 6.8C from page 118, repeated following. The data is the same as for Examples 1 and 2 above.

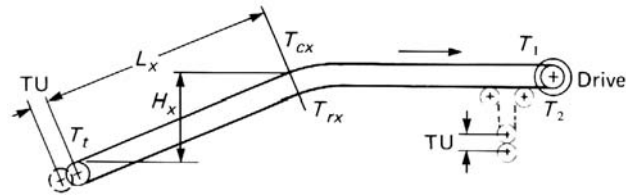


Figure 6.8C.

$$L_x = 350 \text{ ft, on 9\% slope}$$

$$H_x = 31.3 \text{ ft}$$

$$T_t = 1,788$$

To find  $T_{cx}$  at point X on carrying run:

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

$$T_{wcx} = H_x(W_b + W_m) = (31.3)(121.6) = 3,806 \text{ lbs}$$

$$T_{fcx} = L_x[K_x + K_y(W_b + W_m)]$$

$$K_x = 0.883$$

$$K_y = 0.0293 \text{ when conveyor length is 350 ft at 9\% slope; } W_b + W_m = 121.6 \text{ (use 125 in tables); } 3\frac{1}{2} \text{ ft standard idler spacing}$$

$$T_{fcx} = 350 [0.883 + (0.0293)(121.6)] = 1,556 \text{ lbs}$$

$$T_{cx} = 1,788 + 3,806 + 1,556 = 7,150 \text{ lbs}$$

To find  $T_{rx}$  at point X on return run:

$$T_{rx} = T_t + T_{wrx} - T_{frx}$$

$$T_{wrx} = H_x W_b = (31.3)(15) = 470 \text{ lbs}$$

$$T_{frx} = L_x(0.015 W_b) K_t = 350(0.015)(15)(1.0) = 79 \text{ lbs}$$

$$T_{rx} = 1,788 + 470 - 79 = 2,179 \text{ lbs}$$

**Example 4**

This example illustrates the problem of finding belt tensions when the drive is on the return run. The tension calculated at the head and tail pulleys is carried out as follows:

Takeups on return run or at tail pulley.

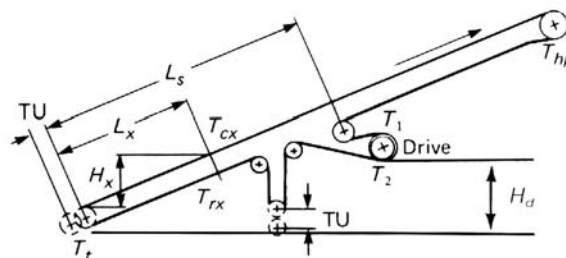


Figure 6.14A.

1. Calculate  $T_e$ ,  $T_1$ ,  $T_2$ , and  $T_t$ , the same as for a conveyor driven at a terminal pulley, using the horsepower formula on page 86, and appropriate tension formulas indicated with the conveyor profiles, Figure 6.14A. The  $T_e$ ,  $T_1$ , and  $T_2$  tensions so calculated apply at the drive pulley regardless of its location along the return run.
2. Calculate the tension,  $T_{hp}$ , at the head pulley using the appropriate formula for  $T_{cx}$  as indicated for conveyor profile with drive on the return run, Figure 6.14. Calculate  $T_{wcx}$  and  $T_{fcx}$  from the formula for determining belt tension at any point. See page 117.

Conveyor data:

$$\begin{aligned}
 W_m &= 120 \text{ lbs/ft} \\
 W_b &= 15 \text{ lbs/ft} \\
 K_t &= 1.0 \\
 K_x &= 0.35 \\
 K_y &= 0.0243 \\
 C_w &= 0.35 \\
 S_i &= 3.5 \text{ ft, idler spacing}
 \end{aligned}$$

36-in. belt conveyor, 600-ft centers, drive located midway of return run, lift 54 ft, slope 9 percent

Calculate the head pulley tension,  $T_{hp}$ , and the tail pulley tension,  $T_t$ .

$$\begin{aligned}
 T_e &= LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) \\
 &= 600 [0.35 + (0.0243)(15) + (0.015)(15)] + 120[(600)(0.0243) + 54] \\
 &= 8,794 \text{ lbs}
 \end{aligned}$$

For 3 percent belt sag,

$$\begin{aligned}
 T_0 &= 4.2 S_i(W_b + W_m) = (4.2)(3.5)(135) = 1,985 \text{ lbs} \\
 T_2 \quad \text{minimum to drive} &= T_e C_w = (8,794)(0.35) = 3,078 \text{ lbs}
 \end{aligned}$$

$$\begin{aligned}
 \text{Corresponding } T_t &= T_2 + (L/2)(0.015 W_b) - (H/2) W_b \\
 &= 3,078 + (300)(0.015)(15) - (27)(15) = 2,741 \text{ lbs}
 \end{aligned}$$

This total tail pulley tension, 2,741 lbs, is greater than 1,985 lbs

Therefore:

$$\begin{aligned}
 T_t &= 2,741 \text{ lbs, and } T_2 = 3,078 \text{ lbs} \\
 T_1 &= T_e + T_2 = 8,794 + 3,078 = 11,872 \text{ lbs}
 \end{aligned}$$

The tension at any point on the carrying run is:

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

Now let

$$L_x = L$$

$$K_t = 1.0$$

Then, tension at the head pulley,  $T_{hp} = T_t + T_{wcx} + T_{fcx}$

$$T_{wcx} = H_x(W_b + W_m) = (54)(135) = 7,290 \text{ lbs}$$

$$\begin{aligned} T_{fcx} &= L_x[K_t K_x + K_y W_b] + L_x K_y W_m, \text{ and since } L = L_x \\ &= 600 [0.35 + (0.0243)(15)] + (600)(0.0243)(120) \\ &= 2,178 \text{ lbs} \end{aligned}$$

Therefore,  $T_{hp} = 2,741 + 7,290 + 2,178 = 12,209 \text{ lbs}$

This is the maximum belt tension. The  $T_1$  tension at the drive pulley may be checked as follows:

$$\begin{aligned} T_1 &= T_{hp} - 27W_b + L/2(0.015W_b) = 12,209 - (27)(15) + (300)(0.015)(15) \\ &= 11,872 \text{ lbs} \end{aligned}$$

This checks with the 11,872 lbs calculated for  $T_1$  from the formula,  $T_1 = T_e + T_2$ .

**Example 5**

This example calculates the belt tension at any point in a declined regenerative conveyor. The calculation is substantially the same as that for a non-regenerative conveyor except that  $\frac{2}{3}K_y$  is used in place of  $K_y$ , and the factor  $A_i$  is eliminated in the formula for  $K_x$ . The value of  $K_y$  is for the length  $L_x$ . Figure 6.10A from page 120 is repeated below.

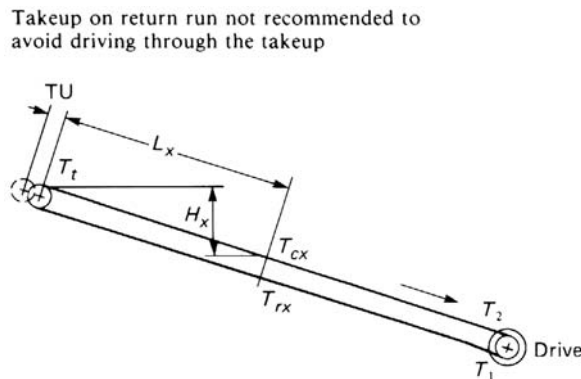


Figure 6.10A.

Conveyor data:

36-in. belt conveyor, 1,000-ft centers, head pulley drive, drop 90 ft, slope 9 per cent

$$W_b = 15 \text{ lbs/ft}$$

## Tension Relationships and Belt Sag Between Idlers

$$\begin{aligned}
 W_m &= 120 \text{ lbs/ft} \\
 S_i &= 3.5 \text{ ft idler spacing} \\
 K_x &= 0.00068 (W_b + W_m) = 0.00068(135) = 0.0918 \\
 K_y &= (0.0169)(0.666) = 0.01126 \text{ for 1,000 ft, 135 for } (W_b + W_m) \text{ and 9 percent} \\
 &\quad \text{slope} \\
 K_t &= 1.0 \\
 C_w &= 0.35 \\
 T_{cx} &= T_t - T_{wcx} + T_{fcx} \\
 T_{rx} &= T_t - T_{wrx} - T_{frx} \\
 T_e &= LK_t(K_x + K_y W_b + 0.015 W_b) + W_m(LK_y \pm H) \\
 T_2 &= T_e C_w \text{ or } T_2 = T_0. \text{ If } T_0 \text{ is the greater} \\
 T_1 &= T_e + T_2 \\
 T_t &= T_1 + 0.015 W_b L + W_b H \\
 T_e &= 1,000 [0.0918 + (0.01126)(15) + (0.015)(15)] + (1,000)(0.01126)(120) - \\
 &\quad (90)(120) = 485.7 + 1,351.2 - 51.2 - 10,800 = -8,963 \text{ lbs}
 \end{aligned}$$

The minus sign merely means that the belt drives the pulley (the conveyor is regenerative).

$$\begin{aligned}
 T_2 &= (8,963)(0.35) = 3,137 \text{ lbs} \\
 T_0 \text{ for 3 percent sag} &= (4.2)(3.5)(135) = 1,985 \text{ lbs}
 \end{aligned}$$

Therefore,  $T_2$  can be taken at 3,137 lbs

$$\begin{aligned}
 T_1 &= T_e + T_2 = 8,963 + 3,137 = 12,100 \text{ lbs} \\
 T_t &= 12,100 + (1,000)(0.015)(15) + (90)(15) = 13,675 \text{ lbs}
 \end{aligned}$$

To calculate  $T_{cx}$  at a point 500 ft from the tail shaft

$$\begin{aligned}
 T_{cx} &= T_t - T_{wcx} + T_{fcx} \\
 T_{wcx} &= H_x(W_b + W_m) = (45)(135) = 6,075 \text{ lbs} \\
 T_{fcx} &= L_x(K_x + K_y W_b) + L_x K_y W_m \text{ when } K_t = 1.0 \\
 &= 500 [0.0918 + (0.01126)(15)] + (500)(0.01126)(120) = 182 + 1,092 = 1,274 \\
 &\quad \text{lbs when } K_y = \frac{2}{3} K_y \text{ for a 500-ft conveyor at 9 percent slope} \\
 \text{Then, } T_{cx} &= 13,675 - 6,075 + 1,274 = 8,874 \text{ lbs}
 \end{aligned}$$

To calculate  $T_{rx}$  at a point 500 ft from the tail shaft

$$\begin{aligned}
 T_{rx} &= T_t - T_{wrx} - T_{frx} \\
 T_{wrx} &= H_x W_b = (45)(15) = 675 \text{ lbs} \\
 T_{frx} &= L_x(0.015) W_b = (500)(0.015)(15) = 113 \text{ lbs when } K_t = 1.0 \\
 \text{Then } T_{rx} &= 13,675 - 675 - 113 = 12,887 \text{ lbs}
 \end{aligned}$$

For conveyor profiles per Figures 6.10B and 6.10C, the portion of the conveyor on a given slope is calculated separately, as in Example 2, page 131, and Problems 5 and 6, pages 172-176.

## Acceleration and Deceleration Forces

Investigation of the acceleration and deceleration forces is necessary for the following reasons.

### Belt Stress

Economy of design dictates the selection of a belt having a carcass strength at or near the normal operating tensions. Consequently, the additional forces resulting from acceleration or deceleration may overstress the belt or its splices, particularly if mechanical splices are used. While this problem is most likely to exist with respect to the belt, there also is the possibility of overstressing the mechanical components such as pulleys, shafts, bearings, takeups, etc.

### Vertical Curves

Two different problems may be encountered with vertical curves.

In the case of concave curves (where the center of curvature lies above the belt) if belt tensions are too high during starting, the belt will lift off the troughing idlers. This is discussed in detail in Chapter 9. It is necessary to analyze this problem in regard to full, partial, and no loads.

In the case of convex vertical curves, where the center of the curvature lies below the belt, there is the possibility of overloading certain idlers.

### Loss of Tension Ratio

During both acceleration and deceleration there exists the distinct possibility of losing the required  $T_1/T_2$  ratio necessary to maintain the desired control of the engagement of belt and drive pulley. This particularly is true if the takeup is located far from the drive.

If a screw takeup is used and improperly adjusted or the travel of a gravity takeup is too limited, the necessary ratio  $T_1/T_2$  may be lost during the attempt to accelerate the belt conveyor.

During deceleration, the effect of the inertia load may cause a loss of the  $T_1/T_2$  ratio necessary to transmit braking forces from the braking pulley to the belt. This would permit the continued motion of the belt and load, after the pulley had been stopped.

### Load Conditions on the Belt

The belt conveyor may operate satisfactorily during stopping or starting if completely loaded or if empty. This, however, may not be so if only portions of the conveyor length are loaded. The conveyor, therefore, has to be analyzed under various conditions of loading.

For example, when a belt conveyor contains a concave curve, a critical condition of starting may be the lifting of the belt at the curve during acceleration because the portion of the belt ahead of the concave vertical curve is loaded, while the remainder of the belt is not. This may not be true if the conveyor is regenerative. Such conditions require careful analysis.

### Coasting

Where there is a system of belt conveyors transferring from one to another, sequence starting or stopping is almost always a prerequisite of design. As an example, a belt with very long centers may transfer to a belt of short centers, in which case the time required to decelerate the two belts must obviously be synchronized, despite the differences in the braking forces required. During the acceleration period, the same synchronization is necessary. In either case, the consequences of not making a

proper analysis and providing the necessary controls will result in a pile-up at the transfer point and possible destruction of the machinery and belt, plus an inoperative system.

### **Takeup Movement**

During both the acceleration and deceleration cycles, where counterweighted takeups are used, the takeup travel may be insufficient unless these forces are considered. The engineer must consider not only the length of travel, but also the rate of travel, particularly where hydraulic, electric, or pneumatic controls are involved.

### **Effect on Material Carried**

In certain instances, the rate of starting and stopping may exert influences on the material which result in intolerable conditions. Obviously, certain materials can be accelerated or decelerated more effectively by the belt than others. For example, if a declined belt conveyor handling pelletized iron ore is stopped too rapidly, the material may start to roll on the belt surface and result in a pile-up at the discharge point. Similarly, starting an inclined belt too rapidly may cause the material to roll backward.

### **Festooning**

Without proper consideration of the starting and stopping forces, it is possible that belt tensions may drop to a point, at some spot in the line, where the belt will festoon (buckle). For example, a belt with a decline from the tail end, and an incline at the head, may be loaded at the tail end only. If braking is applied at the head pulley, the belt may have zero tension or even some slack on the carrying side. The obvious result is load spillage, entanglement, loss of alignment, etc.

### **Power Failure**

In the event of power failure, the belt eventually will stop because of inherent friction forces. Depending upon the profile and conditions of loading, the time required for the friction forces to stop the belt may be intolerably long or short. In the case of a declined regenerative belt conveyor, it may completely unload itself. In a system of belt conveyors, a pile-up of material at transfer points is probable. Therefore, it is obvious that controlled stopping, in the event of a power failure, is very important.

### **Braking Tensions Taken by Return Run and Tail Pulley**

When deceleration is accomplished by means of a brake, the belt tension resulting from the braking force is taken in a direction opposite to that for driving the belt.

For instance, if the drive is at the head end of a horizontal or lift conveyor, power is transmitted from the drive pulley to the carrying side of the belt when the motor is energized. When decelerating with a brake connected to the drive pulley and the motor de-energized, the braking force may be transmitted from the drive pulley to the return belt. The brake application, therefore, may be significant in determining the amount of the counterweight, the design of the takeup, and the shaft sizes.

These are some of the problems that result when acceleration and deceleration forces are ignored or are improperly evaluated. While other difficulties may also exist, those discussed above are sufficient to indicate the importance of proper consideration and analysis.

## Analysis of Acceleration and Deceleration Forces

The accelerating and decelerating forces that act on a belt conveyor during the starting and stopping intervals are the same in either case. However, their magnitude and the algebraic signs governing these forces change, as do the means for dealing with them.

### Acceleration

The acceleration of a belt conveyor is accomplished by some form of prime mover, usually by an electric motor. The resulting forces in a horizontal conveyor are determined by inertia plus friction; in an inclined conveyor, by inertia plus friction plus elevating of the load; in a declined conveyor, by inertia plus friction minus lowering of the load.

### Deceleration

The deceleration of a belt conveyor is accomplished by some form of brake. The resulting forces in a horizontal conveyor are determined by inertia minus friction; in an inclined conveyor, by inertia minus friction minus elevating of the load; in a declined conveyor, by inertia minus friction plus lowering of the load.

If the conveyor contains several portions with different (positive or negative) slopes, a combination of these conditions may result.

### Calculation of Acceleration and Deceleration Forces

The belt conveyor designer is then confronted with the necessity to compute for the conveyor in question the inertia of all its moving parts, the inertia of the load on the belt, total frictional forces, and forces caused by elevating or lowering the load and belt. To be useful, the first two quantities have to be converted to a pound force at the belt line.

Inasmuch as acceleration is defined as the second derivative of displacement with respect to time, and deceleration is simply negative acceleration, time is the basic variable in computing the force. To compute the time, Newton's second law is used. The basic approach is as follows:

$$F_a = Ma$$

where :

$$F_a = \text{accelerating or decelerating force, lbs}$$

$$M = \text{mass, in slugs} = \frac{W_e}{g}$$

$$W_e = \text{equivalent weight of moving parts of the conveyor and load, lbs}$$

$$g = \text{acceleration by gravity} = 32.2 \text{ ft/sec}^2$$

$$a = \text{acceleration, ft per sec per sec (ft/sec}^2\text{)}$$

The force necessary to achieve the acceleration or deceleration is always directly proportional to the mass (or the weight) of the parts and material in motion.

For purposes of calculation, it can be assumed that the belt and the load on it move in a straight line. Other important parts of the system, however, rotate. This is true for all pulleys (including those on takeups and belt trippers), all idlers, and all the rotating parts of the drive.

It appears convenient to use the equation for linear motion as the basis for calculating the acceleration and deceleration forces. This makes it necessary to convert the physical properties of the rotating components of the system to a form in which they can be used in the basic linear relationship:

$$F = \left(\frac{W_e}{g}\right)a$$

In other words, one must find the “Equivalent Weight” of the rotating parts.

For rotating bodies, the mass actually distributed around the center of rotation is equivalent in its effect to the whole mass concentrated at a distance,  $K$ , (the polar radius of gyration, in ft) from that center.

The  $WK^2$  is the weight of the body multiplied by the square of the radius of gyration. If  $WK^2$  is known for the rotating conveyor components, the Equivalent Weight of these components, at the belt line, can be found by solving the equation

$$\text{Equivalent Weight lbs} = WK^2 \left(\frac{2\pi \text{ rpm}}{V}\right)^2$$

where:  $V$  = belt velocity, fpm

Values of  $WK^2$  (expressed in lb-ft<sup>2</sup>), which are difficult to compute, except for very simple shapes, must be obtained for each component from the manufacturers of the conveyor components, motors, transmission elements, etc.

So far considered have been the forces in the system caused by inertia of the moving parts of the conveyor, the moving parts of the drive, and the moving load. Two other forces also are involved, as mentioned on page 138. These are: (1) Forces resulting from friction. (2) Forces resulting from the elevating or lowering of the load and belt. These simply represent the components of the weight of the material and belt, in the direction of motion of the belt, in the various portions of the conveyor.

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## Design Considerations

The belt conveyor designer is confronted with two problems: (1) The necessity to provide a prime mover powerful enough to start the conveyor, sometimes under adverse conditions. (2) To make sure, for the reasons outlined under “Acceleration and Deceleration Forces,” page 138, that the maximum force exerted on the conveyor is within safe limits.

In long, level, high-speed conveyors, a motor large enough for continuous full-load operation may be unable to start the fully loaded conveyor, particularly in cold weather. On the other hand, a motor capable of continuous full-load operation of an inclined conveyor may overstress the belt during starting, unless preventive measures are taken.

The maximum permissible accelerating forces are determined by the factors listed in page 138 of this chapter. Minimum accelerating forces may be dictated by the time during which the prime mover, which usually is an electric motor, can exert its starting torque without being damaged. This limitation is also affected by the frequency of starting the conveyor system.



In the case of deceleration, maximums are governed by the same factors. A minimum deceleration may be dictated by safety or may be necessary because of the material flow at transfer points. In all deceleration calculations involving brakes, the energy-dissipating capacity of the brake will be an important factor to consider.

### Necessary Assumptions

As in all engineering investigations of this type, the first question is, “To what degree of accuracy will the computations have to be carried out?” The answer is not simple. Important factors are the overall size, the importance of the installation, and the type and sensitivity of the equipment adjacent to it.

In any case, numerous simplifying assumptions will have to be made to keep the engineering work within reasonable limits. For examples of simplifying assumptions, refer to the problems connected with belt stretch (elastic elongation from accelerating or decelerating forces) and takeup reactions.

During both the acceleration and deceleration cycles, the transient forces imposed result in extra stretch not encountered during steady state operation. This may result in early splice failure, excessive takeup travel, and other difficulties. Because of the vast differences in carcass construction, from the standpoint of both materials used and methods of manufacture, no single numerical value can express belt stretch as a function of the applied force.

Most manufacturers have listed values of  $B_m$  (elastic constant) for their line of belts. These vary from  $1.3 \times 10^6$  lbs per in. of belt width for steel-cable belt to  $2.3 \times 10^3$  lbs per ply in. of belt width for cotton fabric belts. Other rubber manufacturers may list different values, but they also would vary over the same wide range.

For this reason, as well as many others, the calculations for acceleration and deceleration treat the system as a rigid body. This is a common practice in the solution of problems in dynamics. And while the results usually are quite satisfactory, there is more cause for concern over the accuracy of results in the case of belt conveyors.

No further attempt will be made to justify simplifying the assumption, because this usually is not of major significance. However, the belt conveyor designer should be aware that, for conveyor systems with very long center belts, stretch considerations should not be overlooked.

### Calculations

While the calculations are relatively simple for a conveyor with only one slope, they become increasingly complex for belt conveyors which change slope several times, or which are loaded and unloaded at different points, or which have belt trippers operating on them.

All this results in a great number of possible combinations of load distribution, tripper position, etc. Although theoretically it suffices to investigate only the worst combination of conditions, without analysis, it is usually impossible for even the experienced designer to tell which combination of factors will lead to this extreme case.

In the more complicated cases, it will be necessary to divide the conveyor into portions or sections—within which neither the slope nor the conditions of loading change more than is permitted by the required accuracy of the calculations—and to determine the physical properties for each such portion discussed under “Analysis of Acceleration and Deceleration Forces,” page 138. Any really large rotating member of the belt conveyor, because of its very magnitude, may have to be considered a portion or section by itself.

## Conveyor Horsepower Determination — Graphical Method

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A summation of these weights, forces, and stresses with proper consideration of their algebraic signs will indicate that portion of the system which will impose the most severe limitations to the allowable values for acceleration and deceleration. This, in turn, will permit the selection of the proper prime mover and the necessary control elements.

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## Conveyor Horsepower Determination — Graphical Method

The graphical method shown in Figures 6.17, 6.18, and 6.19 provides the means for estimating horsepower (hp). Belt tensions can be calculated from the resulting horsepower. This method is suitable for conveyors of moderate capacity having relatively straight paths of travel. The results will be sufficiently accurate to establish horsepower requirements when actual weights of belt and revolving parts per foot of conveyor centers are used in Figure 6.17. However, for use in determining tentative or approximate horsepower, a convenient table of typical weights is superimposed on Figure 6.17.

The graphical method is not suitable for final calculations of horsepower for conveyors having decline portions, high capacity, or complex arrangements of terminals, nor for the extended use of rubber skirting and plows that substantially increase the frictional drag on the conveyor belt. On the other hand, it is useful for tentative estimates of horsepower under most of these conditions.

An example of the use of the graphical method follows.

### Determining Required Horsepower — Graphical Method

The following example illustrates a method for determining the required horsepower (hp) for a belt conveyor. The example is the same as Problem 1, page 145, calculated by the analytical method.

In this graphical solution, only the horsepower requirements to move the belt horizontally, elevate the material, and convey the material horizontally are considered. Additional accessory factors such as pulley friction, skirtboard friction, material acceleration, and auxiliary device frictions are included as averages.

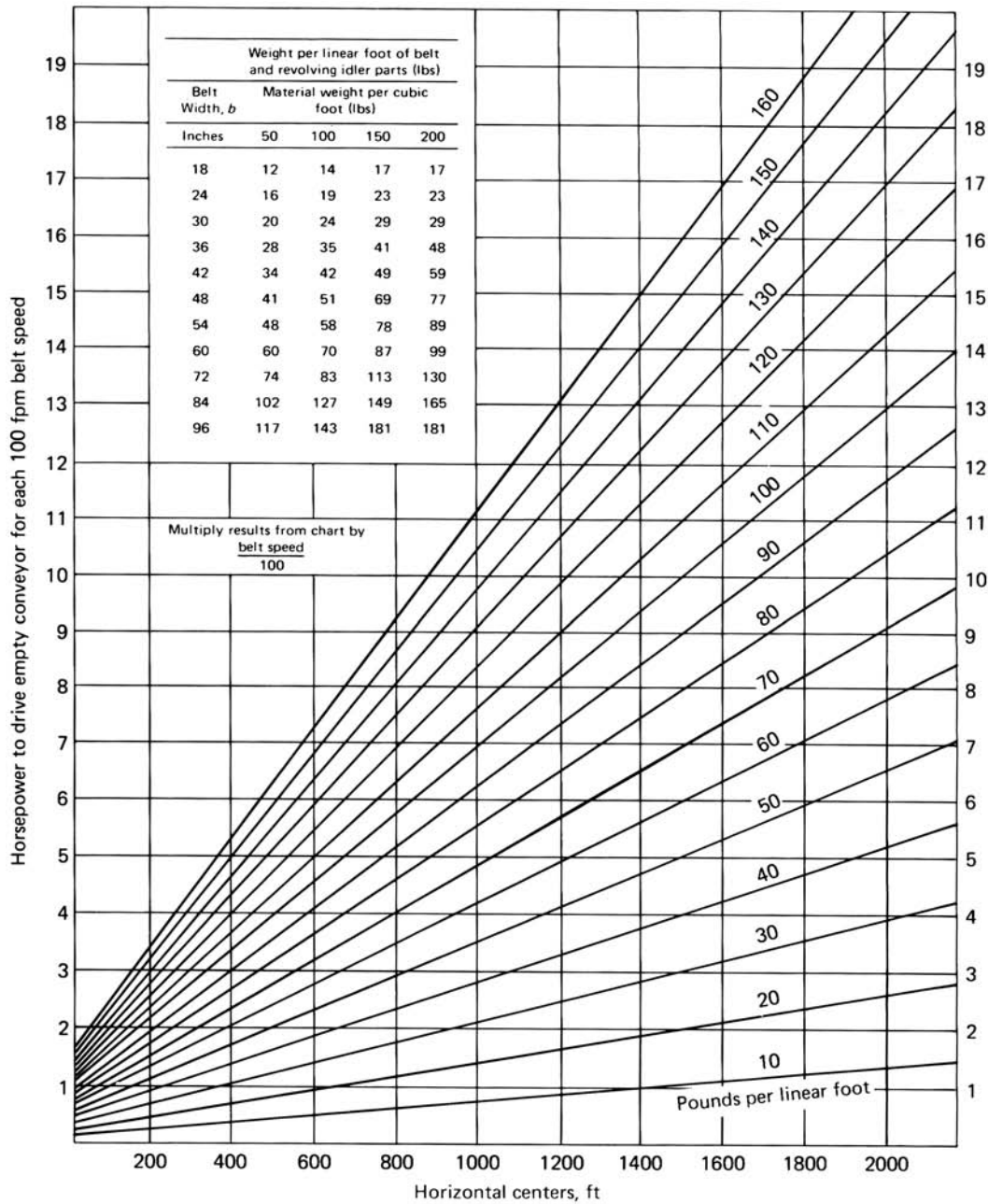


Figure 6.17 Horsepower required to drive empty conveyor.\*

**NOTE:** \*The table of weights is representative of average weights of revolving idler parts, as given in Chapter 5, and estimated belt weights, listed in Table 6-1, page 90. Where actual weights are known, these should be used in the graphical solution.

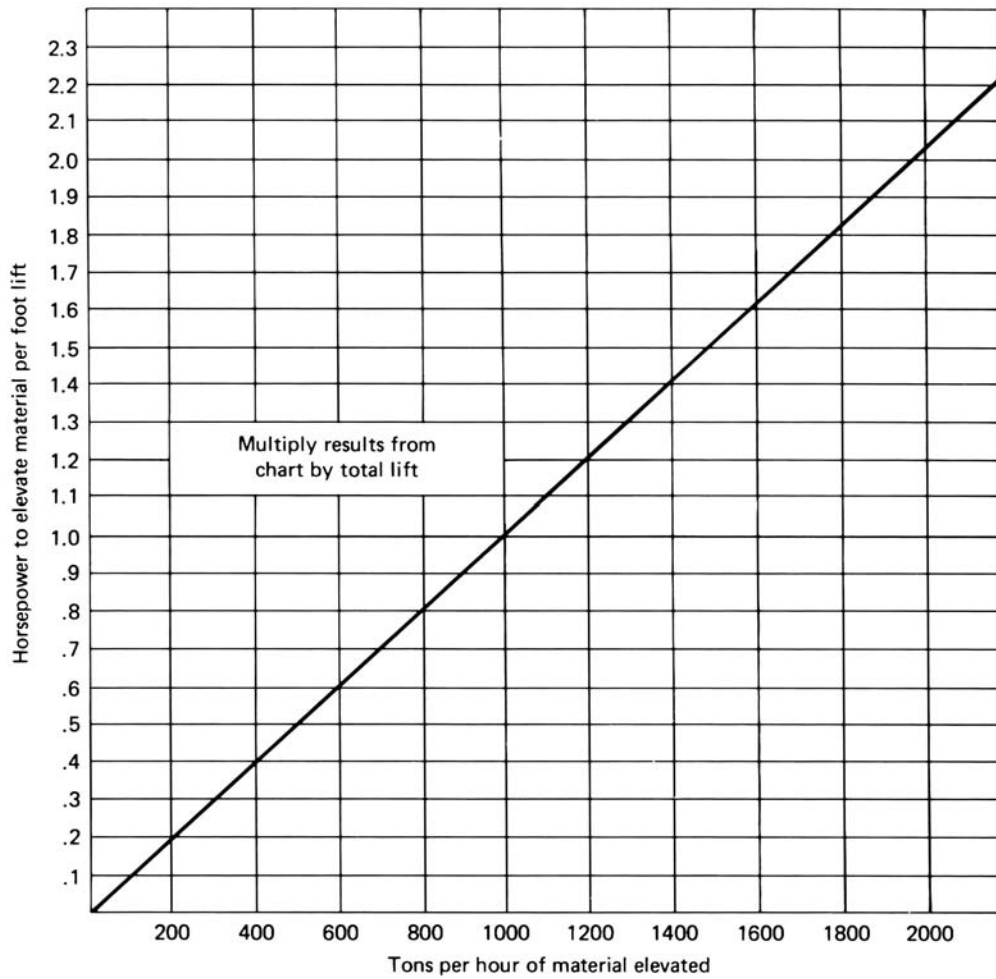


Figure 6.18 Horsepower required to elevate material.

*Conveyor specifications:*

Length,  $L = 2,000$  ft

Lift,  $H = 75$  ft

Capacity,  $Q = 1,600$  tph

Belt speed,  $V = 500$  fpm

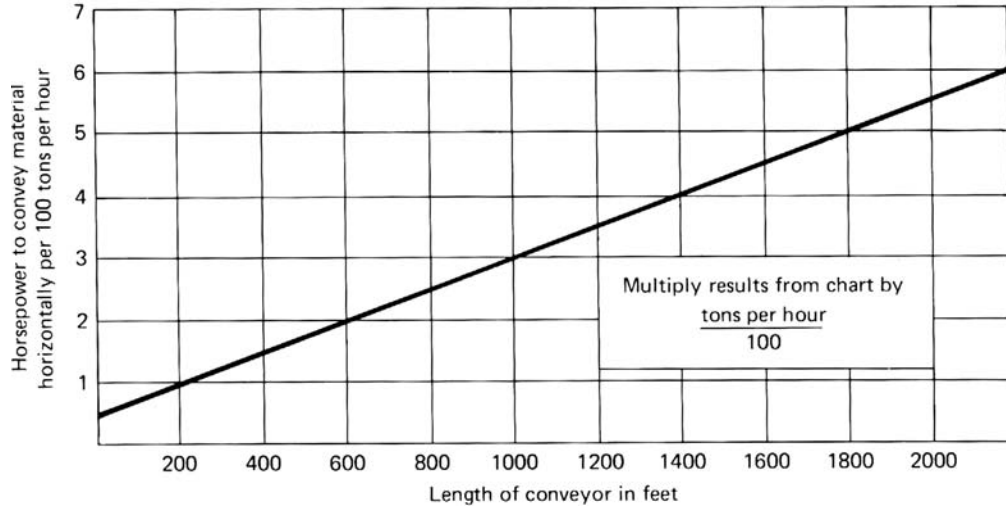
Material density,  $d_m = 100$  lbs/cu ft

Belt width,  $b = 48$  in.

*Graphical analysis.* Referring to Figure 6.17, the weight per foot of belt and revolving idler parts for a 48-in. wide conveyor and 100 pounds per cubic foot of material is given as 51 pounds per foot. Using this value, the horsepower required to drive the empty conveyor at a speed of 100 fpm is 6.5.

Therefore, the horsepower to drive the empty conveyor at 500 fpm is equal to:

$$\frac{6.5 \times 500}{100} = 32.5 \text{ horsepower}$$



**Figure 6.19 Horsepower required to convey material horizontally.**

From using Figure 6.18, the horsepower required to elevate the material can be determined. The horsepower per foot of lift for the 1,600 tph capacity is 1.62. Therefore, the horsepower required to elevate the material 75 ft is given as:

$$1.62 \times 75 = 121.5 \text{ hp}$$

The horsepower needed to convey the material horizontally is determined through the use of Figure 6.19. Using the given conveyor specification of 2,000 ft of conveyor length, the horsepower required to convey 100 tph of material is equal to 5.5 hp. Therefore, the horsepower required for the 1,600 tph capacity is equal to:

$$\frac{5.5 \times 1,600}{100} = 88 \text{ horsepower}$$

The total required horsepower at the belt line is the sum of the above, and equals  $32.5 + 121.5 + 88 = 242 \text{ hp}$ .

Assuming a standard 5 percent loss of power through the drive components due to their inefficiencies, the required motor horsepower is:

$$\frac{242}{.95} = 254.7 \text{ horsepower}$$

A comparison of the horsepower derived by the analytical method, shown on pages 145 to 148 and the above illustrated graphical method shows that the results of the two methods are comparatively very close. This is coincidental inasmuch as the

degree of accuracy of determinations made with the graphical solution is dependable only for estimating purposes. Final design should be made by the analytical method for greatest accuracy.

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### Examples of Belt Tension and Horsepower Calculations — Six Problems

Application of the CEMA horsepower (hp) formula and analysis of belt tensions and power requirements will be illustrated by the following six problems:

- Problem 1 — inclined conveyor;
  - Problem 2 — declined conveyor with regenerative characteristics;
  - Problem 3 — horizontal conveyor;
  - Problem 4 — conveyor with a horizontal section, an inclined section, and vertical curves;
  - Problems 5, 6 — comparison of tension and hp values on two similar conveyors.
- Problems 3 and 4 also include calculation of acceleration and deceleration forces.

#### Problem 1

#### *Inclined Belt Conveyor*

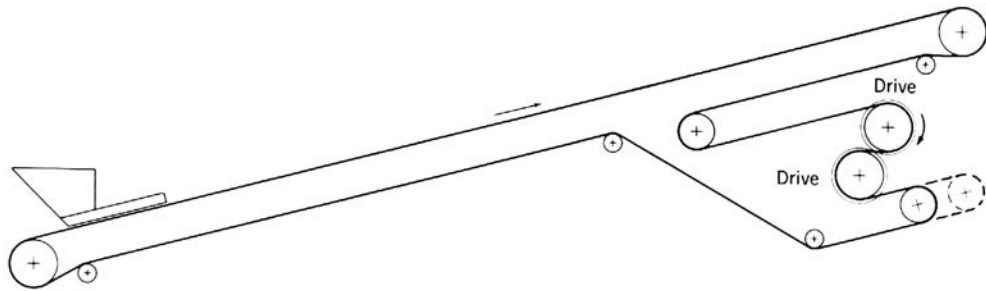


Figure 6.20 Inclined belt conveyor.

#### *Problem:*

Determine effective tension,  $T_e$ ; slack-side tension,  $T_2$ ; maximum tension,  $T_1$ ; tail tension,  $T_t$ ; belt and motor horsepower requirements; and type and location of drive.

In this problem, only two accessories are considered, pulley friction from non-driving pulleys and skirtboard friction. The belt speed is too low to involve any appreciable material acceleration force. The discharge is made freely over the head pulley and no cleaning devices are used.

*Conveyor Specifications:*

$W_b = 15$  lbs/ft from Table 6-1

$L =$  length = 2,000 ft

$V =$  speed = 500 fpm

$H =$  lift = 75 ft

$Q =$  capacity = 1,600 tph

$S_i =$  spacing = 3.5 ft

Ambient temperature = 60°F

Belt width = 48 in

Material = phosphate rock at 80 lb/ft<sup>3</sup>, 15-in. maximum lump from a gyratory crusher

Drive = lagged head pulley or dual drive. Wrap is 240° or 380°, depending on which drive is to be used. See Figures 6.6B and 6.7A; also, Example 2, page 131, and comments.

Troughing idlers = Class E6, 6-in. diameter, 20° angle

Return idlers = Class C6, 6-in. diameter, 10-ft spacing

*Analysis:*

Using Table 6-8, drive factor  $C_w = 0.30$  or 0.11, depending on use of lagged head pulley or dual drive.

$$W_m = \frac{33.3Q}{V} = \frac{33.3 \times 1,600}{500} = 106.6 \text{ lbs per ft}$$

From Figure 6.1, for 60°F,  $K_t = 1.0$

Formula:

$$T_e = LK_t(K_x + K_yW_b + 0.015W_b) + W_m(LK_y + H) + T_{ac}$$

To find  $K_x$  and  $K_y$ , it is necessary to find

$$W_b + W_m = 15 + 106.6 = 121.61 \text{ lbs/ft}$$

Since 3½-ft idler spacing is given and the  $K_x$  value is calculated by using the formula:

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}$$

$$K_x = 0.00068(121.6) + 3.5 = 0.0826 + 0.800 = 0.8826$$

## Examples of Belt Tension and Horsepower Calculations — Six Problems

$K_y$  for  $L = 2,000$  ft, the slope is  $(75/2,000)(100\%) = 3.75\%$ , and  $W_b + W_m = 121.6$  lbs/ft. Table 6-2 gives  $K_y = 0.018$ .

Minimum tension for 3 percent sag,  $T_0 = 4.2 S_i (W_b + W_m)$

$$T_0 = (4.2)(3.5)(121.6) = 1,788 \text{ lbs}$$

*Determine Accessories:*

In this case, the only accessories are nondriving pulley friction and skirtboards. Assume that skirtboards are 15 ft long and spaced apart two-thirds the width of the belt. Then the pull on the belt to overcome skirtboard friction is  $T = C_s L_b h_s^2$ . From the calculation of skirtboard friction,  $h_s = (0.1)(48) = 4.8$  in.  $C_s$  is 0.1086, from Table 6-7, for phosphate rock at 80 pounds per cubic foot. Thus, to solve the equation:

$$T = C_s L_b h_s^2 = (0.1086)(15)(4.8)^2 = 38 \text{ lbs}$$

For the 30 ft of rubber edging on the skirtboards, the additional resistance is  $(3)(30) = 90$  lbs. The total skirtboard resistance is  $38 + 90 = 128$  lbs.

$LK_t K_x$	=	$(2,000)(1)(0.8826)$	=	1,765
$LK_t K_y W_b$	=	$(2,000)(1)(0.018)(15)$	=	540
$LK_t 0.015 W_b$	=	$(2,000)(1)(0.015)(15)$	=	450
$K_y L W_m$	=	$(0.018)(2,000)(106.6)$	=	3,838
$HW_m$	=	$(75)(106.6)$	=	7,995
Nondriving pulley friction	=	$(2)(200) + (2)(150) + (4)(100)$	=	1,100
Skirtboard resistance, $T_{sb}$	=		=	128
Effective tension, $T_e$	=		=	15,816 lbs

*Determine type of drive:*

Analyze head pulley drive @ 240° wrap,  $C_w = 0.30$

$$T_2 = C_w T_e = (0.30)(15,816) = 4,745 \text{ lbs}$$

Analyze dual drive @ 380° wrap,  $C_w = 0.11$

$$T_2 = C_w T_e = (0.11)(15,816) = 1,740 \text{ lbs}$$

However, the minimum tension,  $T_0 = 1,788$  lbs. This minimum should exist close to the loading point on the carrying run of the belt, or at  $T_t$ , to avoid more than 3 percent sag between the troughing idlers spaced at 3.5 ft intervals.

If  $T_t = 1,788$  lbs, the weight of the return belt is  $HW_b = 75 \times 15 = 1,125$  lbs, and the resistance of the return belt is  $0.015LW_b$ , then  $T_2 = 1,788 + 1,125 - 450 = 2,463$  lbs.

Using  $T_2 = 2,463$  lbs, the saving in belt tension with the dual drive over a single-head pulley drive is  $4,745 - 2,463 = 2,282$  lbs, or  $2,282/48 = 48$  lbs/in. width of belt. This saving in belt cost may be enough to offset the cost of a drive.



**Belt Tension, Power, and Drive Engineering**

$$\begin{aligned}
 T_2, \text{ by choice} &= 2,463 \\
 \text{Return run friction } (2,000)(0.015)(15) &= \frac{+450}{2,913} \\
 \text{Less weight of return belt } (75)(15) &= \frac{-1,125}{\text{lbs}} \\
 \text{Tail tension, } T_t &= 1,788 \text{ lbs}
 \end{aligned}$$

*Final Tensions:*

$$\begin{aligned}
 T_e &= 15,816 \text{ lbs} \\
 T_2 &= 2,463 \text{ lbs} \\
 T_t &= T_e + T_2 = 15,816 + 2,463 = 18,279 \text{ lbs}
 \end{aligned}$$

$$T_t = 1,788 \text{ lbs}$$

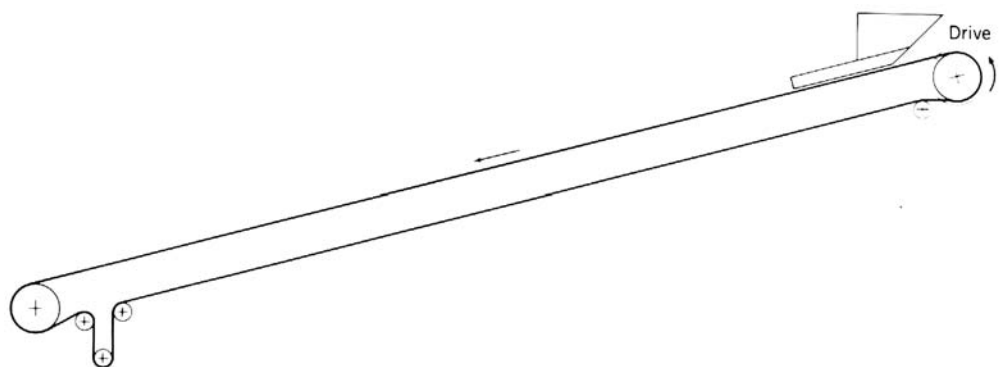
*Horsepower at Motor Shafts:*

$$\begin{aligned}
 \text{Belt hp} &= \frac{T_e V}{33,000} = \frac{(15,816)(500)}{33,000} = 239.64 \\
 \text{Drive pulley friction loss hp} &= \frac{(2)(200)(500)}{33,000} = 6.06 \\
 \text{Add 5 percent for speed reduction loss} &= 0.05 (239.64 + 6.06) = \frac{12.29}{\text{Total hp at motor shafts}} = 257.99 \text{ hp}
 \end{aligned}$$

$$\text{Belt tension} = \frac{18,279}{48} = 381 \text{ lbs per inch of belt width}$$

**Problem 2**

*Declined Belt Conveyor*



**Figure 6.21 Declined belt conveyor.**

Before attempting the solution of a declined conveyor, certain peculiar conditions must be considered.

A declined conveyor, which delivers material below the elevation at which it is received, will generate power if the net change in elevation is more than 2½ percent of the conveyor length. It may generate power at a lower slope, depending on conditions. An electric motor, acting as a generator, is used to retard the conveyor. A brake is used to stop the conveyor.

The motor size is determined by the maximum horsepower, either positive or negative, that it will be called on to produce, and it usually is set by the horsepower generated. The drive is usually located at the tail (feed) end of the conveyor, involving special design problems. One of these is that the motor must start the conveyor by driving through the gravity takeup without lifting the takeup pulley. Care must be taken to check the horsepower and belt tensions for an empty and partially loaded belt.

The brake must be large enough to absorb the torque generated and to decelerate the load. However, the retarding torque must be limited so that it does not overstress the belt. Frequently, on large conveyors, the limiting factor in brake selection will be its holding power within its ability to absorb and dissipate heat. Refer to “Break Heat Absorption Capacity,” pages 191-192.

When a conveyor runs downhill, friction forces increase the belt tension in the direction of motion, while gravity forces decrease the belt tension, by the weight per foot of belt and load, for every foot that the belt and load are lowered.

*Reduced friction.*

The belt, load, and idler friction absorb some of the power that the motor or brake would be compelled to absorb if these quantities did not exist. Therefore, it is important not to overestimate the friction forces or else the selected size of motor or brake might be too small. In order to avoid overestimating the friction forces, the effective tension,  $T_e$  is calculated as follows:

$$T_e = LK_t(K_x + C_1K_yW_b + C_10.015W_b) + C_1W_mLK_y - HW_m + C_1T_{ac}$$

in which factor  $C$  will vary from 0.5 to 0.7 and, for average conditions, will be 0.66.

For *declined conveyors only*, determine  $K_x$  by the formula  $K_x = 0.00068 (W_b + W_m)$ . The additive term  $A_i/S_i$  is omitted because the allowance for grease and seal friction, represented by the factor  $A_i$ , is no longer on the safe side. It may under some conditions approach zero, so the safe course in declined conveyors is to make  $A_i = 0$ .

*Problem:*

Determine effective tension ( $T_e$ ), slack-side tension ( $T_2$ ), maximum tension ( $T_1$ ), tail tension ( $T_t$ ), and belt and motor horsepower requirements.

In this problem, only two accessories are considered, pulley friction of nondriving pulleys and skirtboard friction. The belt speed is too low to involve any appreciable material acceleration and no cleaning devices are employed.

*Conveyor Specifications:*

$W_b = 10$  lbs/ft from Table 6-1

$L =$  length = 1,200 ft

$V =$  speed = 450 fpm

$H =$  drop = 200 ft

$Q =$  capacity = 1,000 tph

$S_i =$  spacing = 4 ft

Ambient temperature = 32°F, minimum

Belt width = 36 in.

Material = limestone at 85 lbs/cu ft, 4-in. maximum lumps

Drive = lagged grooved tail pulley, wrap = 220 degrees

Troughing idlers = Class C6, 6-in. diameter, 20-degree angle,  $A_i = 1.5$

Return idlers = Class C6, 6-in. diameter, 10 ft spacing

*Analysis:*

From Table 6-8, drive factor,  $C_w = 0.35$

From Figure 6.1, for 32°F,  $K_t = 1.0$

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(1,000)}{450} = 74 \text{ lbs per foot}$$

Formula (not considering  $C_l$  factor):

$$T_e = LK_t(K_x + K_yW_b + 0.015W_b) + W_m(LK_y - H) + T_{ac}$$

To find  $K_x$  and  $K_y$ , it is necessary to find

$$W_b + W_m = 10 + 74 = 84 \text{ lbs/ft}$$

$K_x$  must be calculated for two cases. In the first calculation,  $K_x$  is taken at its normal value, so that the tension for the full friction can be determined. In the second calculation,  $K_x$  is taken at its reduced value, so that the tension for the reduced friction can be calculated.

$$\text{Normal } K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i}$$

$$K_x = (0.00068)(84) + \frac{1.5}{4} = 0.05712 + 0.375 = 0.4321$$

$$\text{Reduced } K_x = 0.00068(84) = 0.05712$$

$K_y$  also must be determined for two cases. In the first instance,  $K_y$  will have its normal value as selected from Tables 6-2 and 6-3. This value is then employed for the full friction calculation of the tension. In the second instance,  $K_y$  is modified by the reduced friction factor  $C$ .

The slope is  $(200/1,200)(100\%) = 16.6$  percent. From Table 6-2, it can be seen that 4 ft is not a tabular spacing. For the 16.6 percent slope,  $L = 1,200$  and  $W_b +$

## Examples of Belt Tension and Horsepower Calculations — Six Problems

$W_m = 84$ , the correct value of  $K_y$  is 0.01743, by double interpolation. For simplification in calculations, use  $K_y$  value of .018.

The minimum tension for 3 percent sag is:

$$T_0 = 4.2S_i(W_b + W_m) = (4.2)(4)(84) = 1,411 \text{ lbs}$$

*Determine Accessories:*

The accessories are nondriving pulley friction and skirtboard friction. Assume the skirtboards to be 10 ft long and spaced apart, two-thirds the width of the belt. Then the pull to overcome skirtboard friction is:

$$T = C_s L_b h_s^2$$

$h_s = 10$  percent of belt width or  $(0.1)(36) = 3.6$  in. From Table 6-7,  $C_s$  for limestone is 0.128. Thus, to solve the equation:

$$T = (0.128)(10)(3.6)^2 = 17 \text{ lbs.}$$

For the 20 ft of rubber skirtboard edging, the additional resistance is  $(3)(20) = 60$  lbs. Total skirtboard resistance,  $T_{sb} = 17 + 60 = 77$  lbs

*Full friction,  $T_e$ :*

$LK_t K_x$	=	$(1,200)(1)(0.4321)$	=	518.5
$LK_t K_y W_b$	=	$(1,200)(1)(0.018)(10)$	=	216.0
$LK_t 0.015 W_b$	=	$(1,200)(1)(0.015)(10)$	=	180.0
$K_y L W_m$	=	$0.018(1,200)(74)$	=	1,598.4
$-H W_m$	=	$-(200)(74)$	=	-14,800.0
Nondriving pulley friction = $(2)(150) + (3)(100)$				= 600.0
Skirtboard resistance, $T_{sb}$				= 77.0
Full friction, $T_e$				= <u>-11,610.1</u> lbs

*Reduced friction,  $T_e$ :*

$$T_e = LK_t(K_x + C_1 K_y W_b + C_1 0.015 W_b) + C_1 K_y L W_m - H W_m + C_1 T_{ac}$$

and

$LK_t K_x$	=	$(1,200)(1)(0.05712)$	=	68.5
$LK_t C_1 K_y W_b$	=	$(1,200)(1)(0.66)(0.018)(10)$	=	142.6
$LK_t 0.015 C_1 W_b$	=	$(1,200)(1)(0.015)(0.66)(10)$	=	118.8
$C_1 K_y L W_m$	=	$(0.66)(0.018)(1,200)(74)$	=	1,054.9
$-H W_m = -(200)(74)$	=		=	-14,800.0
Nondriving pulley friction = $[(2)(150) + (3)(100)]0.66$				= 396.0
Skirtboard resistance = $(77)(0.66)$				= 50.8
Reduced friction, $T_e$				= <u>-12,968.4</u> lbs

*Full friction,  $T_2$ :*

$$T_2 = C_w T_e = (0.35)(11,610.1) = 4,064 \text{ lbs}$$

*Reduced friction,  $T_2$ :*

$$T_2 = C_w T_e = (0.35)(12,968.4) = 4,539 \text{ lbs}$$

*Full friction,  $T_t$ :*

$$T_t = T_2 - 0.015LW_b - HW_b = 4,064 - 180 - 2,000 = 1,884 \text{ lbs}$$

*Reduced friction,  $T_t$ :*

$$T_t = T_2 - 0.015C_1LW_b - HW_b = 4,539 - 120 - 2,000 = 2,419 \text{ lbs}$$

*Full friction,  $T_1$ :*

$$T_1 = T_e + T_2 = 11,610 + 4,064 = 15,674 \text{ lbs}$$

*Reduced friction,  $T_1$ :*

$$T_1 = T_e + T_2 = 12,968 + 4,539 = 17,507 \text{ lbs}$$

**Table 6-11. Final tensions, full and reduced friction.**

<b>Final Tensions</b>	<b>Full Friction, lbs</b>	<b>Reduced Friction, lbs</b>
$T_e$	11,610	12,968
$T_2$	4,064	4,539
$T_1$	15,674	17,507
$T_t$	1,884	2,419

*Horsepower at Motor Shaft:*

The horsepower at the motor shaft should be based on the higher of the two values for  $T_e$ .

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(-12,968)(450)}{33,000} = -176.84$$

$$\text{Drive pulley friction hp} = \frac{(200)(450)}{33,000} = +2.73$$

$$\begin{aligned} \text{Additional 5 percent for speed reduction losses} &= +8.71 \\ \text{horsepower at motor shaft} &= -165.4 \text{ hp} \end{aligned}$$

$$\text{Belt tension} = \frac{17,507}{36} = 486 \text{ lbs/in of belt width.}$$

Problem 3

Horizontal Belt Conveyor

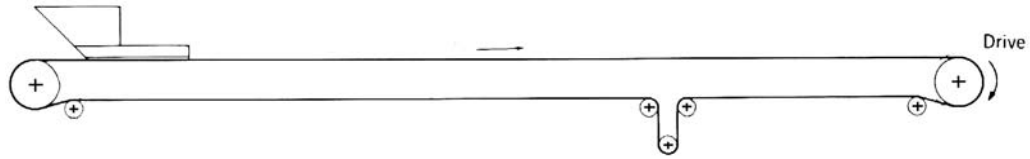


Figure 6.22 Horizontal belt conveyor.

Problem:

Determine effective tension ( $T_e$ ), slack-side tension ( $T_2$ ), maximum tension ( $T_1$ ), tail tension ( $T_t$ ), belt and motor horsepower requirements.

In this problem, only two accessories are considered, pulley friction from non-driving pulleys and skirtboard friction. Material acceleration force has been omitted in this example. The discharge is made freely over the head pulley. No belt-cleaning devices are employed.

Conveyor Specifications:

- $W_b$  = 17 lbs/ft, from Table 6-1
- $L$  = length = 2,400 ft
- $V$  = speed = 500 fpm
- $H$  = lift = 0
- $Q$  = capacity = 3,400 tph
- $S_i$  = spacing = 3 ft

Ambient temperature = 60°F

Belt width = 48 in.

Material = iron ore at 150 lbs/cu ft 10-in. maximum lumps from a gyratory crusher

Drive = lagged and grooved head pulley, 220-degree wrap

Troughing idlers = Class E6, 6-in. diameter, 20-degree angle

Return idlers = rubber-disc type, Class C6, 6-in. diameter, 10 ft spacing

Analysis:

From Table 6-8, drive factor,  $C_w = 0.35$

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(3,400)}{500} = 226.4 \text{ lbs per ft}$$

From Figure 6.1 for 60°F,  $K_t = 1.0$

Formula:

$$T_e = LK_t(K_x + K_yW_b + 0.015W_b) + W_m(LK_y + H) + T_{ac}$$

To find  $K_x$  and  $K_y$ , it is necessary to find

$$W_b + W_m = 17 + 226.4 = 243.4 \text{ lbs/ft}$$

thus:

$$K_x = 1.099, \text{ for } 3.0 \text{ spacing } A_i = 2.8 \text{ and } W_b + W_m = 243.4 \text{ lbs from equation (3)}$$

$$K_y = 0.021, \text{ for } L = 2,400, \text{ slope } 0^\circ \text{ and } W_b + W_m = 243.4 \text{ lbs. Refer to Table 6-2.}$$

$$\begin{aligned} \text{Minimum tension, } T_0, \text{ for 3 percent sag} &= 4.2 S_i (W_b + W_m) = (4.2)(3)(243.4) \\ &= 3,067 \text{ lbs} \end{aligned}$$

*Determine Accessories:*

In this case the only accessories are pulley friction and a loading chute plus skirtboards. Assume that skirtboards are 10 ft long and spaced apart, two-thirds the width of the belt. The pull on the belt to overcome skirtboard friction is  $T = C_s L_b h_s^2$ . From the calculation of skirtboard friction, from page 101, 90,  $h_s = (0.1)(48) = 4.8$  inches.  $C_s$  is (safely).276, from Table 6-7, for iron ore @ 150 pounds per cubic foot. Therefore,  $T = (0.276)(10)(4.8)^2 = 64$  lbs. For the additional 20 ft of rubber edging on the skirtboards, additional resistance is  $(3)(20) = 60$  lbs. Total skirtboard resistance,  $T_{sb} = 64 + 60 = 124$  lbs.

$LK_t K_x$	=	$(2,400)(1)(1.099)$	=	2,638
$LK_t K_y W_b$	=	$(2,400)(1)(0.021)(17)$	=	857
$LK_t 0.015 W_b$	=	$(2,400)(1)(0.015)(17)$	=	612
$K_y L W_m$	=	$(0.021)(2,400)(226.4)$	=	11,411
$H W_m$	=	$(0)(226.4)$	=	0
Nondriving pulley friction = $(4)(100) + (2)(150) =$				700
Skirtboard resistance, $T_{sb} =$				<u>124</u>
Effective tension, $T_e$				= 16,342 lbs
$T_2 = C_w T_e = (0.35)(16,342)$				= <u>+ 5,720</u>
Maximum tension, $T_1 = T_e + T_2$				= 22,062 lbs
Tail tension, $T_t = T_2 + .015 L K_t W_b +$ pulley				
friction = $5,720 + 612 + 700 = 7,032$ lbs				

*Final tensions:*

$$\begin{aligned} T_e &= 16,342 \text{ lbs} \\ T_2 &= 5,720 \text{ lbs} \\ T_1 &= 22,062 \text{ lbs} \\ T_t &= 7,032 \text{ lbs} \end{aligned}$$

*Horsepower at Motor Shaft:*

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(16,342)(500)}{33,000} = 247.61$$

$$\text{Drive pulley hp} = \frac{(200)(500)}{33,000} = 3.03$$

$$\text{Add 5\% for speed reduction loss} = 0.05(247.61 + 3.03) = 12.53$$

$$\text{Horsepower at motor shaft} = 263.17 \text{ hp}$$

$$\text{Belt tension} = \frac{T_l}{\text{width}} = \frac{22,062}{48} = 460 \text{ lbs per inch of belt width}$$

Horsepower at motor shaft = 263 (select 300 hp, 1,750 rpm, motor)

*Acceleration Calculations:*

The following calculations are used to determine the acceleration forces and times:

$WK^2$  of drive (all values are taken at motor speed and should be obtained from the equipment manufacturer):

$WK^2$ of motor	=	101 lb-ft <sup>2</sup>
Equivalent $WK^2$ of reducer	=	20 lb-ft <sup>2</sup> (common practice is to take 1/5 of $WK^2$ of motor)
$WK^2$ of coupling	=	4 lb-ft <sup>2</sup>
Equivalent $WK^2$ of drive pulley	=	5 lb-ft <sup>2</sup>
Total $WK^2$ of drive	=	130 lb-ft <sup>2</sup> , at motor speed

Converting this  $WK^2$  value by using the equation for equivalent weight, page 139, 62,870 lbs is calculated as follows:

$$\text{Drive equivalent weight is } (130 \text{ lb-ft}^2) \left( \frac{1,750 \text{ rpm}}{500 \text{ fpm}} \right)^2 (2\pi)^2 = 62,870 \text{ lbs}$$

For purposes of calculating the equivalent weights, the pulley diameters must first be estimated. The diameters of the head and tail pulleys are assumed to be 42 in.; the rest of the pulleys are assumed to be 30 inches. The actual required pulley diameters are a function of the characteristics of the belt to be used and the belt tension at the pulley. The assumed pulley diameters should be reviewed after this information is known.



Conveyor equivalent weight is as follows:

Pulleys—From Table 8-1, one 42" diameter x 51" pulley with maximum bore of 5" weight = 1,275 lbs

From Table 8-1, five 30" diameter x 51" pulleys with maximum bore of 4", weight = 5 x 780 lbs = 3,900 lbs

6 (total) 1,275 + 3,900 = 5,175 lbs

Thus, 5,175 lbs is the approximate total weight for all of the nondriving pulleys on this conveyor. For more accurate calculations, the equipment manufacturer can supply actual weights. This weight is distributed among all of the elements that make up each pulley (rim, end and center discs, hubs, etc.). The belt must accelerate and decelerate these pulleys.

A generally accepted method for determining the equivalent weight of conveyor pulleys is to use  $\frac{2}{3}$  of the actual total weight. Therefore,

$$\begin{aligned} & \frac{2}{3}(5,175 \text{ lbs}) & = & 3,450 \text{ lbs} \\ \text{Belt, carrying run, from Table 6-1,} & & & \\ & 17 \text{ lbs/ft} \times 2,400 \text{ ft} & = & 40,800 \text{ lbs} \\ \text{Belt, return run, 17 lbs/ft} \times (2,400 \text{ ft} + 30 \text{ ft}) & & = & 41,310 \text{ lbs} \end{aligned}$$

Idlers, troughing, from Table 5-11, for 48" belt width and Class E6, the weight is 81.9 lbs

$$\text{thus, } 81.9 \text{ lbs} \times \frac{2,400 \text{ ft}}{3\text{-ft spacing}} = 65,520 \text{ lbs}$$

Idlers, return, from Table 5-12, for 48" belt width and Class C6, the weight is 48.4 lbs

$$\text{thus, } \frac{48.4 \text{ lbs} \times 2,400 \text{ ft}}{10\text{-ft spacing}} = 11,616 \text{ lbs}$$

Total conveyor equivalent weight = 162,696 lbs

Material load (226.4 lbs/ft)(2,400 ft) = 543,360 lbs

Total equivalent weight for system =  
62,870 lbs + 162,696 lbs + 543,360 lbs = 768,926 lbs

Percent of total within the conveyor:

$$\frac{706,056}{768,926} \times 100\% = 91.8\%$$

Having selected a belt for  $T_1 = 22,062$  lbs, as explained in Chapter 7, at an allowable rating of 90 lbs/in./ply, 6 plies are required and the rated tension is 25,920 lbs.

If the starting tension is limited to 180 percent of the rated tension (see page 113), then the allowable extra belt tension for acceleration is

$$(1.80)(25,920) - (22,062) = 46,656 - 22,062 = 24,594 \text{ lbs}$$

The time for acceleration is found from the equation:

$$F_a t = M \frac{V_1 - V_0}{60}$$

where:

$F_a$  = the allowable extra accelerating tension = 24,594 lbs

$t$  = time, seconds

$V_1$  = final velocity = 500 fpm

$V_0$  = initial velocity = 0 fpm

$M$  = mass of conveyor system =  $\frac{706,056}{32.2} = 21,926$  slugs

Solving for  $t$ :

$$t = \frac{M}{F_a} \times \frac{(V_1 - V_0)}{60} = \frac{21,927}{24,594} \times \frac{(500 - 0)}{60} = 7.43 \text{ seconds}$$

This means that in order not to exceed the maximum permissible belt tension at 46,656 lbs, the time used for acceleration should not be less than 7.43 seconds.

Assume a 300 horsepower motor is used, with a maximum torque of 200 percent of the full-load torque. This corresponds to a force of 39,600 lbs acting at the belt line, if the friction losses of the drive are not considered and the belt speed is 500 fpm. This is not excessive when compared to the 46,656 lbs belt tension allowable at 180 percent of belt rating.

Another limiting factor may be the time that the motor needs to accelerate the system. The average torque – available during acceleration of the chosen motor – taken from its speed torque curve is 180 percent of full load torque. For a drive efficiency of 95 percent, it was found in Problem 3, that the horsepower at the motor shaft to operate the loaded conveyor is 263.17 hp.

$$\text{Horsepower} = \frac{(\text{pull in lbs})(\text{belt speed, fpm})}{33,000}$$

Therefore:

$$(\text{pull in lbs}) = \frac{(\text{hp})(33,000)}{(\text{belt speed, fpm})}$$

Also, the horsepower delivered by the motor is practically proportional to the torque, assuming no appreciable drop in speed from the full-load speed. Therefore, at 180 percent torque, the motor will deliver  $(1.8)(300) = 540$  hp.

The force available for acceleration of the total equivalent mass of the loaded conveyor system, for a belt speed of 500 fpm, is:

$$\begin{aligned}
 F_a &= \left[ \frac{(300)(1.8)(33,000)}{500} - \frac{(263.17)(33,000)}{500} \right] 0.95 \\
 &= 0.95 - [(300)(1.8) - 263.17] \frac{33,000}{500} \\
 &= \frac{(263)(33,000)}{500} = 17,358 \text{ lbs}
 \end{aligned}$$

The total equivalent mass =  $768,926/32.2 = 23,880$  slugs

From the equation,

$$F_a = Ma$$

$$\text{the acceleration, } a = \frac{F_a}{M} = \frac{17,358}{23,880} = 0.727 \text{ ft per sec}^2$$

The time needed is:

$$t = \frac{V_t - V_o}{60a}$$

Therefore:

$$t = \frac{500 - 0}{(60)(0.727)} = 11.46 \text{ seconds.}$$

The time required by the motor to accelerate the loaded conveyor, 11.46 seconds, is greater than the minimum acceleration time to stay within the maximum allowable belt tension, 7.43 seconds. Therefore, the conveyor is safe to start, fully loaded, with the equipment selected.

Had the starting belt stress been limited to 120 percent of the normal belt rating instead of 180 percent, the allowable extra belt tension would have been  $(1.2)(25,920) - (22,062) = 9,042$  lbs and the acceleration time,  $t = (21,927)(500 - 0)/(9,042)(60) = 20.21$  sec minimum. This is more than the time calculated for the motor to accelerate the loaded system, 11.46 seconds. So, if such limitation had been placed on the starting belt stress, the system would not have been safe to start with the equipment selected. In fact, the belt stress during acceleration must be:

$$\begin{aligned}
 \text{extra belt tension} &= \frac{(21,927)(500 - 0)}{(11.46)(60)} = 15,752 \text{ lbs} \\
 \% \text{ of normal belt rating} &= \frac{(15,752) + 22,062}{25,920} (100\%) = 146\%
 \end{aligned}$$

The foregoing assumes that the mass between the slack side of the drive pulley and the takeup is negligible. If the takeup is far removed from the drive, this should be taken into account in the calculations.

In Chapter 13 it is indicated that the acceleration time for NEMA Type C motors, in general, be considered as 10 seconds or less. It, therefore, would be prudent to

check with the motor manufacturer to make sure that the calculated acceleration time of 11.46 seconds would not cause the motor to overheat during starting. In the case of *this particular problem*, the motor manufacturer was asked what maximum safe acceleration time would be for his 300 hp NEMA Type C motor. The manufacturer stated that any time up to 20 seconds would be permissible.

Therefore, the conveyor in this problem could be safely started with the equipment selected, provided the allowable belt tension during starting was 146 percent of normal belt rating, or greater. The use of the 300 hp NEMA Type C motor is justified, provided that operating conditions of this particular conveyor are such that abnormal starting conditions (which would require forces considerably in excess of those calculated) are unlikely to occur. If abnormal starting conditions are likely to occur, even infrequently, consideration should be given to the use of a different means of starting that would satisfy all the requirements described earlier in the section, "Acceleration and deceleration forces."

### *Deceleration Calculations:*

In the preceding example on acceleration, it was found that the total equivalent mass of the conveyor system under normal conditions of operation is equal to 23,880 slugs (see page 158). As these calculations are based on the belt speed of 500 fpm or 8.33 fps, the kinetic energy of the system is:

$$\frac{MV^2}{2} = \frac{(23,880)(8.33)^2}{2} = 828,503 \text{ ft lbs}$$

Earlier in this problem (3), it was found that 263.17 hp is required to operate this conveyor at its rated speed of 500 fpm. Because the conveyor is horizontal, this represents the product of the friction forces and the distance traveled in unit time. This means that the frictional retarding force is:

$$\frac{(263.17)(33,000)}{500} = 17,369 \text{ lbs}$$

The average velocity of the conveyor during the deceleration period would be

$$\frac{500 + 0}{2} = 250 \text{ fpm}$$

Because the total work performed has to be equal to the kinetic energy of the total mass,

$$(t)(250 \text{ fpm})(17,369) = 828,503 \text{ ft lbs}$$

*where:*

$t$  = time in minutes

Therefore:

$$t = \frac{828,503}{(250)(17,369)} = 0.191 \text{ minutes, or } 11.45 \text{ seconds}$$

and the belt will have moved  $(0.191)(250 \text{ fpm}) = 47.75 \text{ ft}$  in this time. As the belt is fully loaded (by assumption), the belt will discharge the following amount of material:

$$\left(\frac{3,400 \text{ tph}}{60}\right)\left(\frac{47.75}{500}\right) = 5.41 \text{ tons}$$

If 5.41 tons of material discharged is objectionable, the use of a brake has to be considered. Such a step, however, can be justified only if the reduced deceleration time is still greater than, or at least equal to, the deceleration cycle of whatever piece of equipment delivers to the conveyor in this example.

Also, another difficulty arises. Suppose it is desirable or necessary to reduce the deceleration time from 11.45 seconds to 7 seconds. Since the total retarding force is inversely proportional to the deceleration time, the additional braking force required must be:

$$17,369 \times \left(\frac{11.45-7}{7}\right) = 11,042 \text{ lbs}$$

If the brake is connected to the drive pulley shaft, the drive pulley is required to transmit to the belt a braking force equal to

$$11,042 \left(\frac{768,926-(62,870)}{768,926}\right) = 10,139 \text{ lbs}$$

The difference between the 11,042 lbs and the 10,139 lbs is the braking force required to decelerate the drive and drive pulley and is not transmitted to the belt.

However, under coasting conditions, the belt tension is principally governed by the gravity takeup which, if located adjacent to the head pulley, would provide a maximum tension equal to  $T_2$ , or 5,720 lbs. Obviously, it is impossible to secure a braking force of 10,139 lbs on the head pulley. Even a much smaller force than this would result in looseness of the belt around the head pulley.

The solution is to provide the braking action on the tail pulley, where it would increase rather than decrease the contact pressure between the belt and pulley. However, a further check on the tail pulley indicates that with 11,042 lbs braking tension, a plain bare tail pulley with 180-degree wrap angle could not produce a sufficient ratio of tight-side to slack-side tension.

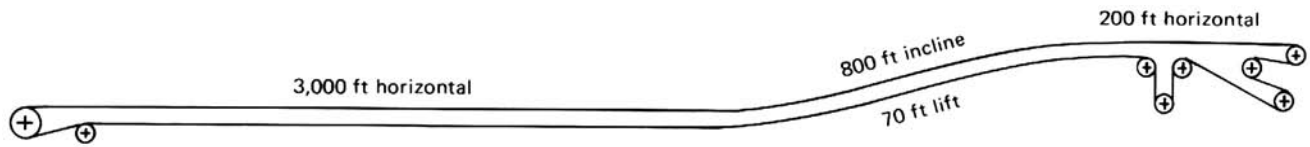
Therefore, it would be necessary to do one or a combination of the following: increase the takeup tension weight, lag the tail pulley, or snub the tail pulley for a greater wrap angle. If the increasing takeup weight should result in a heavier and more costly belt carcass, the second and third remedies are preferable and more economical.

It should be noted that the above calculations are based on maximum friction losses and therefore will give a minimum coasting distance. Since most installations operate under variable conditions, braking and coasting problems should be investigated for a range of friction values. These lower friction values for  $K_x$  and  $K_y$  can be found by the methods outlined in Problem 2, and can result in a lower frictional retarding force approaching 60 percent of the original. This lower retarding force will show greater coasting distances or larger braking forces.

**Problem 4**

*Complex Belt Line*

In the previous examples, the application of the CEMA horsepower formula was limited to belt conveyors with a linear profile and an overall centers length not exceeding 3,000 ft. However, the CEMA horsepower formula can be applied to belt conveyors having more than one change in slope and a total centers length of more than 3,000 ft provided certain procedures are followed. This problem entails a belt conveyor that has two changes in slope and a total centers length of 4,000 ft.



**Figure 6.23 Complex belt line.**

The  $K_y$  factor is dependent upon the average tension in that portion of the belt in which the tension is being analyzed. Tables 6-2 and 6-3 were developed on the basis of the limitations and generalizations stated on page 91, and for normal average tensions in the belts within the limitations specified. For belt conveyors exceeding these limitations, it is necessary first to assume a tentative value for the average belt tension. The graphical method for conveyor horsepower determination, pages 141-145, may be of assistance in estimating this value. After estimating the average belt tension and idler spacing, reference to Table 6-4 will provide values for  $A$  and  $B$  for use in equation (4), page 96. By using this equation, an initial value for  $K_y$  can be calculated. The comparison of this calculated average belt tension with the tentative value will determine the need to select another assumed belt tension. The process should be repeated until there is reasonable agreement between the estimated and calculated average belt tensions.

The following example of a belt tension analysis of a 4000-ft belt conveyor with two changes of slope demonstrates the method of calculation and the use of the table. See Figure 6.23.

Belt conveyors with different profiles can be analyzed in a similar manner, but the various problems, which increase in importance with long and complex belt conveyors, must be carefully analyzed. It is suggested that the designer of such complex conveyors check calculations with a CEMA member company before establishing the final conveyor design.

*Problem:*

Determine the effective tension,  $T_e$ ; slack-side tension,  $T_2$ ; maximum tension,  $T_1$ ; tail tension,  $T_t$ ; concave curve tension at bottom of incline run, convex curve tension at top of incline run; belt horsepower at drive terminal; belt stress; resulting drive factor.

Accessories are omitted in this example in order to clarify the procedure; however, they should be included in actual practice.

*Conveyor Specifications:*

- $Q$  = capacity = 800 tph
- Material = crushed limestone, 85 lbs/cu ft, 8-in maximum lumps
- Ambient temperature, above freezing. Continuous operation
- $L$  = length = 4,000 ft
- $H$  = lift = 70 ft (see Figure 6.23)
- $S_i$  = idler spacing = 4 ft
- $b$  = belt width = 36 in.
- $V$  = speed = 400 fpm
- Drive = dual, 380° wrap, both pulleys lagged
- Troughing idlers = Class C6, 6 in diameter, 20° angle,  $A_i = 1.5$
- Return idlers = rubber-disc type, Class C6, 6 in diameter, 10 ft spacing
- $W_b$  = belt weight = 10 lbs/ft

*Constants:*

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(800)}{400} = 66.6 \text{ lbs per ft}$$

$$K_x = 0.00068(W_b + W_m) + \frac{A_i}{S_i} = 0.00068(10 + 66.6) + \frac{1.5}{4} = 0.427$$

*Analysis:*

Since each portion of this conveyor is analyzed separately, and each is less than or equal to 3,000 ft in length, Table 6-2 can be used to obtain a tentative  $K_y$  factor in order to calculate the average belt tension. This  $K_y$  value is then checked by using Table 6-4, equation (4), page 96, and the average belt tension. The final tensions in each portion of the conveyor are then accurately determined.

The profile is divided into three portions: (1) initial horizontal portion, 3,000 ft long; (2) inclined portion, 800 ft long, 70-foot lift; (3) final horizontal portion, 200 ft long.

Initial horizontal section, 3000 ft long, 3 percent sag in belt,

where:

$$\begin{aligned} K_t &= 1.0 \\ K_x &= 0.427 \\ W_b &= 10 \text{ lbs/ft} \\ W_m &= 66.6 \text{ lbs/ft} \\ (W_b + h_m) &= 76.6 \text{ lbs/ft} \end{aligned}$$

$K_y$  from Table 6-2 would be 0.023.

The *average tension* is:

$$\frac{T_t + K_t[K_x L + K_y L W_b] + K_y L W_m + T_t}{2}$$

Here,  $T_t$  is at least equal to  $T_0$ . And  $T_0$ , for 3 percent sag is

$$4.2 S_i (W_b + W_m) = (4.2)(4)(76.6) = 1,287 \text{ lbs}$$

Thus, the *average tension* is:

$$\frac{1,287 + (0.427)(3,000) + (0.023)(3,000)(76.6) + 1,287}{2}$$

or,

$$\frac{1,287 + 1,281 + 5,860 + 1,287}{2} = \frac{9,140}{2} = 4,570 \text{ lbs}$$

Equation (4) indicates  $K_y = 0.0255$ , for 4,570 lbs average tension, and  $(W_b + W_m) = 76.6$  lbs. Re-estimate using  $K_y = 0.0255$ . *Average tension* is:

$$\frac{1,287 + 1,281 + (0.0255)(3,000)(76.6) + 1,287}{2}$$

or,

$$\frac{1,287 + 1,281 + 5,860 + 1,287}{2} = \frac{9,715}{2} = 4,858 \text{ lbs}$$

Equation (4) checks  $K_y = 0.0255$ , for an average tension of 4,858 lbs, and  $(W_b + W_m) = 76.6$  lbs.

The formula for the actual tension because of friction in the initial horizontal portion (see page 117) is:

$$T_{fcx} = L_x[K_t(K_x + K_y W_b)] + L_x K_y W_m$$



where:

$$\begin{aligned} L_x &= 3,000 \text{ ft} \\ K_x &= 0.427 \\ K_t &= 1.0 \\ K_y &= 0.0255 \\ W_b &= 10 \text{ lbs/ft} \\ W_m &= 66.6 \text{ lbs/ft} \end{aligned}$$

Therefore:

$$\begin{aligned} T_{fcx} &= L_x[K_x + K_y(W_b + W_m)], \text{ since } K_t = 1.0 \\ T_{fcx} &= 3,000[0.427 + 0.0255(76.6)] \\ &= 3,000(2.38) \\ &= 7,141 \text{ lbs} \end{aligned}$$

The tension at the beginning of the vertical concave curve is calculated using the formula for belt tension at any point on the conveyor length, for point X on the carrying run (page 117), at the intersection of the initial horizontal run and the inclined run:

$$\begin{aligned} T_{cx} &= T_t + T_{wcx} + T_{fcx} \\ T_{wcx} &= H_x(W_b + W_m) = (0)(76.6) = 0 \end{aligned}$$

so

$$T_{cx} = 1,287 + 7,141 = 8,428 \text{ lbs}$$

The tension at the bottom of the incline, therefore, is 8,428 lbs. The estimated  $K_y$  is 0.024, for the first approximation of the calculation for the upper end of the incline from Table 6-2 for a value of  $(W_b + W_m) = 76.6$ , and a slope of  $(70/800)(100\%) = 8.8\%$ . Average tension is:

$$\frac{T_t + K_t(K_x L + K_y L W_b) + K_y L W_m + H(W_b + W_m) + T_t}{2}$$

in which  $T_t$  is the tension at the bottom of the incline, or 8,428 lbs, so,

$$\frac{8,428 + (0.427)(800) + (0.024)(800)(76.6) + (70)(76.6) + 8,428}{2}$$

or,

$$\frac{8,428 + 342 + 1,471 + 5,362 + 8,428}{2} = \frac{24,031}{2} = 12,016 \text{ lbs}$$

As stated on page 92, the minimum  $K_y$  value = .016 for 12,016 lbs tension and  $W_b + W_m = 76.6$ . Re-estimate using  $K_y = .016$ . Average tension then is:

## Examples of Belt Tension and Horsepower Calculations — Six Problems

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$$\frac{8,428 + 342 + (0.016)(800)(76.6) + 5,362 + 8,428}{2}$$

or,

$$\frac{8,428 + 342 + 980 + 5,362 + 8,428}{2} = \frac{23,540}{2} = 11,770 \text{ lbs}$$

This checks  $K_y = 0.016$  minimum value, for average tension of 11,770 lbs, and  $W_b + W_m = 76.6$  lbs/ft.

From page 117,  $T_{cx} = T_t + T_{wcx} + T_{fcx}$ . Here,  $T_t$  is 8,428 lbs, the tension at the bottom of an incline.

$$\begin{aligned} T_{wcx} &= H_x(W_b + W_m) = (70)(76.6) = 5,362 \text{ lbs} \\ T_{fcx} &= L_x K_t [K_x + K_y(W_b + W_m)] \text{ where } L_x = 800 \text{ ft and } K_t = 1.0 \\ &= (800)(1.0)[0.427 + 0.016(76.6)] \\ &= 800(1.653) \\ &= 1,322 \text{ lbs} \\ T_{cx} &= 8,428 + 5,362 + 1,322 = 15,112 \text{ lbs} \end{aligned}$$

The tension at the top of the incline, then, is 15,112 lbs.

The final horizontal portion is 200 ft long:

$$\begin{aligned} K_t &= 1.0 \\ K_x &= 0.427 \\ W_b &= 10 \text{ lbs/ft} \\ W_m &= 66.6 \text{ lbs/ft} \\ W_b + W_m &= 76.6 \text{ lbs/ft} \end{aligned}$$

$K_y$  will be at a minimum value because of the high tension that is obvious in this portion of the belt. From page 91, the minimum  $K_y$  of 0.016 is applicable at the indicated *average* tension (obviously more than 15,112 lbs) and with  $W_b + W_m = 76.6$  lbs/ft.

From page 117,  $T_{cx} = T_1 + T_{wcx} + F_{fcx}$ . Here,  $T_t$  is the tension at the beginning of this horizontal section, or 15,112 lbs and  $T_{wcx} = 0$ , since  $H_x = 0$ .

$$\begin{aligned} T_{fcx} &= L_x K_t (K_x + K_y W_b) + L_x K_y W_m \text{ where } L_x = 200 \text{ ft and } K_t = 1.0 \\ &= (200)(1)(0.427 + (0.016)(10)) + (200)(0.016)(66.6) \\ &= 117 + 213 = 330 \text{ lbs} \\ T_{cx} &= 15,112 + 330 = 15,442 \text{ lbs} \end{aligned}$$

In this case,  $T_{cx} = T_1 = 15,442$  lbs

The final tension at the head pulley is 15,442 lbs.  $T_e = T_1 - T_2$ . To find  $T_2$ , refer to Figure 6.7A, where  $T_t = T_2 - T_b + T_{yr}$ . The tail tension,  $T_t$ , was taken at  $T_0 = 1,287$  lbs to avoid more than 3% belt sag between idlers.

Thus,

$$1,287 = T_2 - T_b + T_{yr}$$

$$T_b = HW_b = (70)(10) = 700 \text{ lbs}$$

$$T_{yr} = 0.015LW_bK_t = (0.015)(3,000 + 800 + 200)(10)(1) = 600 \text{ lbs}$$

Thus,

$$1,287 = T_2 - 700 + 600, \text{ or } T_2 = 1,387 \text{ lbs}$$

$$T_e = T_1 - T_2 = 15,442 - 1,387 = 14,055 \text{ lbs}$$

With the  $T_e$  and  $T_2$  tensions now known, it is necessary to check the wrap factor,  $C_w$ . A 380-degree wrap, dual-pulley drive with lagged pulleys, requires a  $C_w = 0.11$ . (See Table 6-8.) From the known tensions:

$$C_w = \frac{T_2}{T_e} = \frac{1,387}{14,055} = 0.099$$

Since this is less than the required 0.11, the belt may slip on the drive pulleys. This situation can be corrected in one of two ways: (1) The wrap on the drive pulleys can be increased from 380 to 405 degrees, or (2) the takeup weight can be increased until  $T_2/T_e = 0.11$ . This requires an increase in all tensions of  $(0.11 \times 14,055) - 1,387 = 160$  lbs.

Assume all tensions are increased by 160 lbs:

$$T_1 = 15,442 + 160 = 15,602$$

$$T_2 = 1,387 + 160 = 1,547$$

$$T_t = 1,287 + 160 = 1,447$$

$$\text{Belt stress} = \frac{T_1}{\text{Belt width}} = \frac{15,602}{36} = 433 \text{ lbs per inch of width (PIW)}$$

Horsepower at belt line, excluding all accessories, is as follows:

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(14,055)(400)}{33,000} = 170.36$$

If drive efficiency = .94, horsepower at motor shaft =  $170.36/.94 = 181.23$  hp. Acceleration and deceleration calculations for this example follow. For radii of concave and convex curves for this example, refer to Chapter 9.

**Examples of Belt Tension and Horsepower Calculations — Six Problems**

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*Acceleration Calculations:*

$$T_e = 14,055 \text{ lbs}$$

$$T_2 = 1,547 \text{ lbs}$$

$$T_1 = 15,602 \text{ lbs}$$

$$T_t = 1,447 \text{ lbs}$$

$$b = \text{Belt width} = 36 \text{ in.}$$

$$L = \text{Length} = 4,000 \text{ ft}$$

$$H = \text{Lift} = 70 \text{ ft}$$

$$Q = \text{Capacity} = 800 \text{ tph}$$

$$V = \text{speed} = 400 \text{ fpm}$$

Material = crushed limestone at 85 lbs/cu ft

$$W_m = \text{Material weight} = 66.6 \text{ lbs/ft}$$

$$W_b = \text{Belt weight} = 10 \text{ lbs/ft}$$

Troughing idlers Class C6, 6-in. diameter 20° angle, at 4 foot spacing

Rubber-disc return idlers, Class C6 6-in. diameter at 10 foot spacing

Horsepower at motor shaft = 181.23 (select one 75 hp and one 125 hp motor, each to be 1,750 rpm)

$WK^2$  of drive (all values are taken at motor speed and should be obtained from the equipment manufacturer)

$$WK^2 \text{ of motor} = 58.0 \text{ lbs-ft}^2$$

$$\text{Equivalent } WK^2 \text{ of reducer} = 11.6 \text{ lbs-ft}^2$$

(It is common practice to estimate the  $WK^2$  of the reducer to be 20 percent of the  $WK^2$  of the motor)

$$WK^2 \text{ of coupling} = 2.0 \text{ lbs-ft}^2$$

$$\text{Equivalent } WK^2 \text{ of drive pulley} = 2.0 \text{ lbs-ft}^2$$

$$\text{Total } WK^2 = 73.6 \text{ lbs-ft}^2, \text{ at motor speed}$$

Converting this  $WK^2$  value to the equivalent weight at the belt line,

$$\text{Drive equivalent weight (lbs)} = (WK^2) \left( \frac{2\pi \text{ rpm}}{V} \right)^2$$

$$\text{Drive equivalent weight is } (73.6 \text{ lb-ft}^2) \left( \frac{1,750 \text{ rpm}}{400 \text{ fpm}} \right)^2 (2\pi)^2 = 55,615 \text{ lbs}$$

Conveyor equivalent weight is as follows:

Pulleys — for reasons given on page 155, first assume the non-drive pulley diameter: two at 48 in. and four at 36 in.

From Table 8-1, two (48 in. diameter x 38 in.) pulleys with max. bore of 5 in., weight = 2 x 1,270 = 2,540 lbs

From Table 8-1, four (36 in. diameter x 38 in.) pulleys with max. bore of 4 in., weight = 4 x 715 = 2,860 lbs

## Belt Tension, Power, and Drive Engineering

$$\begin{aligned} \text{From page 156, } \frac{2}{3}(2,540 \text{ lbs} + 2,860 \text{ lbs}) &= 3,600 \text{ lbs} \\ \text{Belt, carrying run, from Table 6-1, } 10 \text{ lbs/ft} \times 4,000 \text{ ft} &= \underline{40,000} \text{ lbs} \\ \text{Subtotal} &= 43,600 \text{ lbs} \end{aligned}$$

$$\text{Belt, return run, } 10 \text{ lbs/ft} \times (4,000 \text{ ft} + 50 \text{ ft}) = 40,500 \text{ lbs}$$

Idlers, troughing, from Table 5-11, for 36-in. belt width and class C6, the weight is 43.6 lbs.

$$43.6 \left( \frac{4,000 \text{ ft}}{4\text{-ft idler spacing}} \right) = 43,600 \text{ lbs}$$

Idlers, return, from Table 5-12, for 36-in. belt width and Class C6, the weight is 37.6 lbs.

$$37.6 \left( \frac{4,000 \text{ ft}}{10\text{-ft idler spacing}} \right) = \underline{15,040} \text{ lbs}$$

$$\begin{aligned} \text{Total conveyor equivalent weight} &= 142,740 \text{ lbs} \\ \text{Material load } (66.6 \text{ lbs/ft})(4,000 \text{ ft}) &= 266,400 \text{ lbs} \\ \text{Total equivalent weight of system} &= \\ 55,615 \text{ lbs} + 142,740 \text{ lbs} + 266,400 \text{ lbs} &= 464,755 \text{ lbs} \end{aligned}$$

Percent of total within the conveyor:

$$(142,740 + 266,400)/464,755 \times 100\% = 88\%$$

Having selected a belt for  $T_1 = 15,602$  lbs, as explained in Chapter 7, at an allowable rating of 70 lbs/in./ply, 7 plies are required and the rated tension is 17,640 lbs.

If the starting tension is limited to 180 percent of the rated tension (see page 113), then the allowable extra tension is:

$$(1.80)(17,640) - (15,602) = 31,752 - 15,602 = 16,150 \text{ lbs}$$

The time for acceleration is found from the equation:

$$F_a t = M \frac{V_1 - V_0}{60}$$

Where:

$F_a$  = the allowable extra accelerating tension = 16,150 lbs

$t$  = time, seconds

$V_1$  = final velocity = 400 fpm

$V_0$  = initial velocity = 0 fpm

$M$  = mass of conveyor system =  $\frac{409,140}{32.2} = 12,706$  slugs

Solving for  $t$ :

$$t = \frac{M(V_1 - V_0)}{F_a \cdot 60} = \frac{12,706}{16,150} = \frac{(400 - 0)}{60} = 5.24 \text{ seconds}$$

## Examples of Belt Tension and Horsepower Calculations — Six Problems

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This means that in order not to exceed the maximum permissible belt tension at 31,752 lbs, the time used for acceleration should not be less than 5.24 seconds.

In determining the starting tension in the belt, the first step is to find the total horsepower available, in the form of tension. From this value, subtract the total tension to operate the loaded conveyor. The result will be the force available to accelerate the total system. The total horsepower, in the form of tension, available to accelerate the entire system comes from the 75 hp and 125 hp motors. The starting torque available from these NEMA Type C motors is a variable that should be confirmed by the motor manufacturer. For this example, the value is assumed to be 200 percent of the motor rating.

Then, the total tension available is:

$$\frac{2(75 + 125)(33,000)}{400} = 33,000 \text{ lbs}$$

From this value we subtract the tension required to operate the loaded conveyor:

$$33,000 - \frac{14,055}{0.94(\text{drive efficiency})} = 18,048 \text{ lbs}$$

The resulting tension available to accelerate the loaded conveyor is 18,048 lbs. The acceleration of the total system consists of the acceleration of the drive (12 percent of the total system) and of the conveyor (88 percent of the total system). However, in the process of acceleration, some amount of the available force (tension) is absorbed by the frictional losses (heat) in the drive machinery. Since this is a small amount compared to the total, a conservative approach is to ignore these losses because it is our aim to determine the effect of acceleration on the belt and its capacity to withstand tensile forces. Therefore,  $0.88 \times 18,048 = 15,882$  lbs, which is the acceleration force (expressed in lbs of belt tension). The operating  $T_1$  is then added to this value in order to obtain the actual starting tension in the belt, which is  $15,602 + 15,882 = 31,484$  lbs. This is not excessive when compared to the 31,752 lbs belt tension allowable at 180 percent of belt rating.

Another limiting factor may be the time that the motor needs to accelerate the system. The average torque available during acceleration of the chosen motor taken from its speed torque curve is 180 percent of full-load torque. For a drive efficiency of 94 percent, it was found in Problem 4, page 166, that the horsepower at the motor shaft to operate the loaded conveyor is 181.23 hp.

$$\text{Horsepower} = \frac{(\text{pull in lbs})(\text{belt speed, fpm})}{33,000}$$

*Therefore:*

$$(\text{pull in lbs}) = \frac{(\text{hp})(33,000)}{(\text{belt speed, fpm})}$$

The force available for acceleration of the total equivalent mass of the loaded conveyor system, for a belt speed of 400 fpm, is

$$\begin{aligned} F_a &= \left[ \frac{(200)(1.80)(33,000)}{400} - \frac{(181.23)(33,000)}{400} \right] 0.94 \\ &= \frac{0.94 \times 33,000}{400} (200 \times 1.8 - 181.23) \\ &= 13,864 \text{ lbs} \end{aligned}$$

The total equivalent mass =  $464,755/32.2 = 14,433$  slugs. From the equation,

$$F_a = Ma$$

$$\text{the acceleration, } a = \frac{F_a}{M} = \frac{13,864}{14,433} = 0.96 \text{ ft per sec}^2$$

The time needed is:

$$t = \frac{V_1 - V_0}{60a}$$

Therefore:

$$t = \frac{400 - 0}{(60)(0.96)} = 6.94 \text{ seconds}$$

The time required by the motor to accelerate the loaded conveyor, 6.94 seconds, is greater than the minimum acceleration time to stay within the maximum allowable belt tension, 5.24 seconds. Therefore, the conveyor is safe to start, fully loaded, with the equipment selected.

Had the starting belt stress been limited to 140 percent of the normal belt rating instead of 180 percent, the allowable extra belt tension would have been  $(1.4)(17,640) - (15,602) = 9,094$  lbs and the acceleration time  $t = (12,706)(400 - 0)/(9,094)(60) = 9.31$  sec minimum. This is more than the time calculated for the motor to accelerate the loaded system, 6.94 seconds. If such a limitation had been placed on the starting belt stress, the system would not have been safe to start with the equipment selected. In fact, the belt stress during acceleration must be:

$$\begin{aligned} \text{extra belt tension} &= \frac{(12,706)(400 - 0)}{(6.94)(60)} = 12,206 \text{ lbs} \\ \% \text{ of normal belt rating} &= \frac{12,206 + 15,602}{(17,640)} (100\%) = 158\% \end{aligned}$$

The foregoing assumes that the mass between the slack side of the drive pulley and the takeup is negligible. If the takeup is far removed from the drive, this should be taken into account in the calculations.

In Chapter 13, "Accelerating time," it is indicated, in general, that the acceleration time for NEMA Type C motors be considered as 10 seconds or less. It is always prudent to check with the motor manufacturer to make sure that the calculated acceleration time will not cause the motor to overheat during starting.

### *Deceleration Calculations:*

In the foregoing example of acceleration calculations, it was found that the total equivalent mass of the conveyor system under normal conditions of operation is equal to 14,433 slugs. As these calculations are based on the belt speed of 400 fpm or 6.67 fps, the kinetic energy of the system is:

$$\frac{MV^2}{2} = \frac{14,433 \times 6.67^2}{2} = 320,733 \text{ ft-lbs}$$

On page 167 it was calculated that 181.23 hp is required at the motor shaft to operate this conveyor at its rated speed of 400 fpm. This represents the product of the friction plus gravity forces and the distance traveled in unit time. This means that the frictional plus gravitational retarding force is:

$$\frac{(181.23)(33,000)}{400} = 14,951 \text{ lbs}$$

The average velocity of the conveyor during the deceleration period would be:

$$\frac{400 + 0}{2} = 200 \text{ fpm}$$

Because the total work performed has to be equal to the kinetic energy of the total mass:

$$(t)(200 \text{ fpm})(14,951) = 320,733 \text{ ft lbs}$$

where:

$t$  = time in minutes

Therefore:

$$t = \frac{320,733}{(200)(14,951)} = 0.1073 \text{ minutes, or 6.44 seconds}$$

and the belt will have moved  $(0.1073)(200 \text{ fpm}) = 21.46 \text{ ft}$  in this time. As the belt is fully loaded (by assumption), it will discharge the following amount of material:

$$\left(\frac{800 \text{ tph}}{60}\right)\left(\frac{21.46}{400}\right) = 0.72 \text{ tons}$$

If 0.72 tons of material discharge is objectionable, the use of a brake has to be considered. Such a step, however, can be justified only if the reduced deceleration time is still greater than, or at least equal to, the deceleration cycle of whatever piece of equipment delivers to the conveyor in this example.

Also, another difficulty arises. Suppose it is desirable or necessary to reduce the deceleration time from 6.44 seconds to 5 seconds. Since the total retarding force is inversely proportional to the deceleration time, the additional braking force required must be:



$$14,951 \times \left( \frac{6.44 - 5}{5} \right) = 4,306 \text{ lbs}$$

If the brake is connected to the drive pulley shaft, the drive pulley is required to transmit to the belt a braking force equal to

$$4,306 \text{ lbs} \times .88 = 3,789 \text{ lbs}$$

The difference between the 4,306 lbs and the 3,789 lbs is the braking force required to decelerate the drive and drive pulley and is not transmitted to the belt.

However, under coasting conditions, the belt tension is principally governed by the gravity takeup which, if located adjacent to the head pulley, would provide a maximum tension equal to  $T_2$ , or 1,547 lbs. Obviously, it is impossible to develop a braking force of 3,789 lbs on the head pulley. Even a much smaller force than this would result in looseness of the belt around the head pulley.

The solution is to provide the braking action on the tail pulley where it would increase rather than decrease the contact pressure between the belt and pulley. However, a further check on the tail pulley indicates that with 3,789 lbs braking tension, a plain, bare tail pulley with a 180-degree wrap angle could not produce a sufficient ratio of tight-side to slack-side tension.

Therefore, it would be necessary to do one or a combination of the following: increase the takeup tension weight, lag the tail pulley, or snub the tail pulley for a greater wrap angle. If the increased takeup weight should result in a heavier and more costly belt carcass, the second and third remedies are preferable and more economical.

It should be noted that the above calculations are based on maximum friction losses and therefore will give a minimum coasting distance. Since most installations operate under variable conditions, braking and coasting problems should be investigated for a range of friction values. These lower friction values for  $K_x$  and  $K_y$  can be found by the methods outlined in Problem 2, and can result in a lower frictional retarding force approaching 60 percent of the original. This lower retarding force will show greater coasting distances or larger braking forces.

### *Problems 5 and 6*

#### *Comparison of Tension and Horsepower Values on Two Similar Conveyors*

The two belt conveyors compared here in Problems 5 and 6 have the same load capacity, carry the same bulk material, have the same length, the same operating speed, and the same lift. The only difference is that one conveyor has a concave vertical curve while the other has a convex vertical curve.

The CEMA formula for power to operate belt conveyors determines the effective tension,  $T_e$ . The previous examples show how to obtain  $T_2$ ,  $T_1$ , and  $T_t$ . *Frequently, the belt conveyor designer will require belt tensions elsewhere*; for instance, in the determination of the radius of a concave vertical curve. For a discussion of the belt tensions at any point on a belt conveyor, see "Belt Tension at Any Point, X, on Conveyor Length," page 117. Formulas for the belt tensions in belt conveyors having concave and convex vertical curves are given in Figures 6.8 through 6.16, inclusive.

The comparison of the two belt conveyors in Problems 5 and 6 shows how the factor  $K_y$  changes with increasing belt tension.

## Examples of Belt Tension and Horsepower Calculations — Six Problems

In Problem 5, Figure 6.24, the  $K_y$  factor for the tail half,  $L_1$ , of the conveyor is selected for a 300-foot horizontal conveyor. The  $K_y$  factor for the inclined drive half,  $L_2$ , is selected for the total conveyor length of 600 ft with an average slope of lift/total length =  $36/600 = 6$  percent, because the belt tension is higher than it would be for a 300-foot inclined conveyor, due to the belt pull at the end of the horizontal portion.

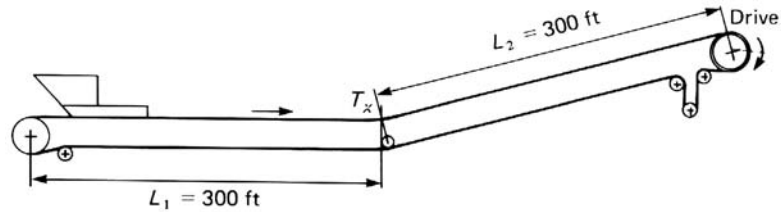


Figure 6.24 Belt conveyor with concave vertical curve.

In Problem 6, Figure 6.25, the  $K_y$  factor for the tail half,  $L_1$ , of the conveyor is selected for a 300-foot conveyor inclined at a slope of lift/300 ft =  $36/300 = 12$  percent. The  $K_y$  factor for the drive half,  $L_2$ , of the conveyor is less than it would be for a 300-ft horizontal conveyor, because of the high belt tension existing at the top of the inclined portion. The criterion for determining the  $K_y$  value to use for the horizontal drive half,  $L_2$ , of this conveyor is the  $K_y$  value of a 600-foot inclined conveyor at a 6 percent slope. The  $K_y$  value for the horizontal portion cannot be more, and probably is a little less than this  $K_y$  value.

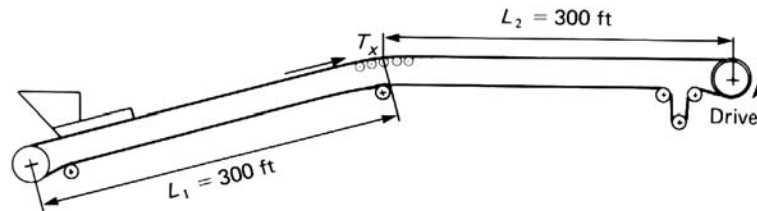


Figure 6.25 Belt conveyor with convex vertical curve.

The difference in the calculated effective tensions in Problems 5 and 6 is small. But larger and longer conveyors would entail more significant differences.

$$W_b = 10 \text{ lbs/ft from Table 6-1}$$

$$H = 36 \text{ ft}$$

$$L = 600 \text{ ft}$$

$$L_1 = 300 \text{ ft}$$

$$L_2 = 300 \text{ ft}$$

$$V = 500 \text{ fpm}$$

$$Q = 1,000 \text{ tph}$$

$$S_i = 4.5 \text{ ft}$$

$$\text{Ambient temperature} = 60^\circ\text{F}$$

$$\text{Belt width} = 36 \text{ in.}$$

$$\text{Material} = 100 \text{ lbs/cu ft}$$

$$\text{Drive} = \text{lagged head pulley, wrap} = 220^\circ$$

Troughing idlers = Class E6, 6-in. diameter 20° angle,  $A_i = 2.8$

Return idlers = Class C6, 6-in. diameter, 10-foot spacing

To simplify the calculations, all accessories are omitted.

*Analysis (Problem 5, Figure 6.24):*

From Table 6-8, wrap factor,  $C_w = 0.35$ . From Figure 6.1, for 60°F,

$K_t = 1.0$

$$W_m = \frac{33.3Q}{V} = \frac{(33.3)(1,000)}{500} = 66.6 \text{ lbs per ft}$$

$$W_b + W_m = 10 + 66.6 = 76.6 \text{ lbs/ft}$$

$$T_0, \text{ minimum tension for 3 percent sag} = 4.2 S_i (W_b + W_m) = (4.2)(4.5)(76.6) \\ = 1,448 \text{ lbs}$$

$T_1$  is taken as  $T_0 = 1,448$  lbs

$$T_2 = T_t - 0.015LW_b + HW_b = 1,448 - (0.015)(600)(10) + (36)(10) \\ = 1,448 - 90 + 360 = 1,718 \text{ lbs}$$

$$K_x = (0.000568)(W_b + W_m) + \frac{A_i}{S_i} = (0.00068)(76.6) + \frac{2.8}{4.5} = 0.6743$$

The effective belt tension,  $T_e$ , is figured individually for each half of the conveyor. Horizontal portion, 300 ft long,  $K_y$  from Table 6-2, for 0° slope, 300 ft and  $(W_b + W_m) = 76.6$ , is 0.0347. Corrected for 4.5-foot idler spacing, Table 6-3, gives  $K_y = 0.0349$ .

From "Belt Tension at Any Point, X, on Conveyor Length," page 117, tension is:

$$T_{fcx} = T_t + T_{wxc} + T_{fcx}$$

but,  $T_{wxc} = H_x(W_b + W_m) = 0$ , for a horizontal belt

and  $T_{fcx} = L_x[K_t(K_x + K_yW_b)] + L_xK_yW_m$

and  $K_t = 1.0$  for 60°F

*Therefore:*

$$T_{fcx} = L_xK_x + L_xK_yW_b + L_xK_yW_m = L_xK_x + L_xK_y(W_b + W_m)$$

Thus,

$$T_{cx} = T_t + 0 + L_xK_x + L_xK_y(W_b + W_m)$$

## Examples of Belt Tension and Horsepower Calculations — Six Problems

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Calling  $L_x$  equal to  $L_1$  for the first (horizontal) half of the conveyor:

$$\begin{aligned} T_{cx} &= T_t + L_1 K_x + L_1 K_y (W_b + W_m) = 1,448 + (300)(0.6743) \\ &\quad + (300)(0.0349)(76.6) = 1,448 + 202 + 802 \\ &= 2,452 \text{ lbs} \end{aligned}$$

For the inclined portion, drive half  $L_2$ .

$$K_x = 0.6743$$

$K_y$  is 0.028, for a slope of  $(36/600)(100\%) = 6\%$ , and  $W_b + W_m = 76.6$ , and a length of 600 ft, from Table 6-2 for the tabular idler spacing. The corrected value is 0.0298, from Table 6-3, for a 4½-foot spacing.

$T_{cx} = T_t + T_{wcx} + T_{fcx}$ . However,  $T_t$  is the tension existing at the bottom of the incline, so:

$$\begin{aligned} T_{cx} &= 2,452 + T_{wcx} + T_{fcx} = 2,452 + H_x(W_b + W_m) \\ &\quad + L_x K_x + L_x K_y (W_b + W_m) \end{aligned}$$

substituting  $L_2$  for  $L_x$ , and 36 for  $H_x$

$$\begin{aligned} T_{cx} &= 2,452 + 36(W_b + W_m) + L_2 K_x + L_2 K_y (W_b + W_m) \\ &= 2,452 + (36)(76.6) + (300)(.6743) + (300)(.0298)(76.6) \\ &= 2,452 + 2,757.6 + 202.3 + 684.8 \\ &= 6,097 \text{ lbs} \end{aligned}$$

Adding to  $T_{cx}$  the nondriving pulley friction,  $(2)(150) + (4)(100) = 700$  lbs, the tension in the belt at the head pulley =  $T_{cx} + 700 = T_1 = 6,097 + 700 = 6,797$  lbs

$$T_e = T_1 - T_2 = 6,797 - 1,718 = 5,079 \text{ lbs}$$

$$\text{Belt horsepower} = \frac{T_e V}{33,000} = \frac{(5,079)(500)}{33,000} = 77 \text{ hp}$$

*Analysis (Problem 6, Figure 6.25):*

$T_0$  has been calculated in Problem 5 as 1,448 lbs.

$$\begin{aligned} \text{Take } T_t &= T_0 = 1,448 \text{ lbs} \\ T_t &= T_2 + L(0.015W_b) - H_{wb} \\ T_2 &= T_t - L(0.015W_b) + HW_b = 1,448 - (600)(0.015)(10) + (36)(10) \\ &\quad + (36)(10) = 1,448 - 90 + 360 = 1,718 \text{ lbs} \end{aligned}$$

Inclined portion, 300 ft long. The slope of the incline is  $(36/300)(100\%) = 12\%$ . For this slope, a length of 300 ft and  $W_b + W_m = 76.6$ , the value of  $K_y$ , from Table 6-2,

is 0.0293. Corrected for 4½-foot idler spacing, from Table 6-3,  $K_y = 0.0312$ .  $K_x$  already has been calculated in Problem 5 as 0.6743.

From “Belt Tension at Any Point, X, on Conveyor Length,” page 117, the tension at point X in the carrying run is:

$$T_{cx} = T_t + T_{wcx} + T_{fcx}$$

Since  $K_t = 1.0$  for 60°F, the equation can be written:

$$T_{cx} = T_t + H_x(W_b + W_m) + L_x K_x + L_x K_y (W_b + W_m)$$

So, for the inclined tail half ( $L_1 = L_x$ ) of the conveyor,

$$\begin{aligned} T_{cx} &= 1,448 + (36)(76.6) + (300)(0.6743) + (300)(0.0312)(76.6) \\ &= 1,448 + 2,757.6 + 202.3 + 716.9 = 5,125 \text{ lbs} \end{aligned}$$

$T_{cx} = 5,125$  lbs. This, then, is the tension in the belt at the top of the incline and at the beginning of the horizontal portion.

Horizontal portion = 300 ft long

$$K_x = 0.6743$$

$K_y$  is dependent on the average belt tension, which, as is seen from the preceding calculation, will be very high. Used as a criterion of  $K_y$ , the value of  $K_y$  is 0.028, calculated from a 600-foot long inclined conveyor, at an average slope of 6%, from Table 6-2. Corrected for 4½-foot idler spacing, Table 6-3 gives  $K_y = 0.0298$ .

From “Belt Tension at Any Point, X, on Conveyor Length,” page 117, the tension at point X in the carrying run (at the head pulley, in this case) is  $T_{cx} = T_t + T_{wcx} + T_{fcx}$ .

However,  $T_t$  is the tension at the start of the horizontal run = 5,125 lbs.  $T_{wcx} = H_x(W_b + W_m)$ . And since  $H_x = 0$ , then  $T_{wcx} = 0$ . Also,  $L_x = L_2 = 300$  ft.

$$\begin{aligned} T_{cx} &= 5,125 + T_{fcx} = 5,125 + L_2 K_x + L_2 K_y (W_b + W_m) \\ &= 5,125 + (300)(0.6743) + (300)(0.0298)(76.6) \\ &= 5,125 + 202 + 685 = 6,012 \text{ lbs} \end{aligned}$$

Add to  $T_{cx}$  the nondriving pulley friction  $(2)(150) + (4)(100) = 700$  lbs

$$T_{cx} + 700 = T_1 = 6,012 + 700 = 6,712 \text{ lbs}$$

$$T_e = T_1 - T_2 = 6,712 - 1,718 = 4,994 \text{ lbs}$$

$$\text{Belt hp} = \frac{T_e V}{33,000} = \frac{(4,994)(500)}{33,000} = 75.7 \text{ hp}$$

## Belt Conveyor Drive Equipment

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The comparison of these two conveyors, each having the same length, lift, size, speed, and capacity, and handling the same material, shows that the conveyor with the concave curve will have a higher belt tension at the head pulley, will require a higher effective tension, and will require more horsepower than the conveyor with the convex curve.

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## Belt Conveyor Drive Equipment

The engineering of practically all belt conveyor installations involves a comprehensive knowledge of the proper application of conveyor drive equipment, including speed reduction mechanisms, electric motors and controls, and safety devices.

### Belt Conveyor Drive Location

The best place for the drive of a belt conveyor is at the location that results in the lowest maximum belt tension. For horizontal or inclined conveyors, the drive usually is at the discharge end, while for declined conveyors the drive is usually at the loading end. For special conditions and requirements, it may be advisable to locate the drive elsewhere. See “Drive Arrangements” and “Analysis of Belt Tensions.”

Economics, accessibility, or maintenance requirements may make it preferable to locate the drive internally on the conveyor. Frequently, for the larger conveyors, a saving in supporting structures can be realized by doing so. Inclined boom conveyors sometimes are driven at the loading end for this reason.

### Belt Conveyor Drive Arrangement

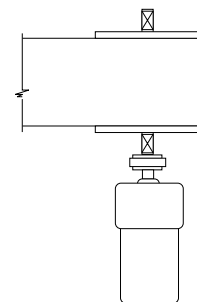
Belt conveyor drive equipment normally consists of a motor, speed reduction equipment, and drive shaft, together with the necessary machinery to transmit power from one unit to the next. The simplest drive, using the minimum number of units, usually is the best. However, economic reasons may dictate the inclusion of special-purpose units in the drive. These special-purpose units may be required to modify starting or stopping characteristics, to provide hold-back devices, or perhaps to vary the belt speed.

### Speed-Reduction Mechanisms

The illustrations in Figures 6.26 through 6.33 show most of the belt conveyor drive equipment assemblies currently in common use.

The following comments apply to these figures:

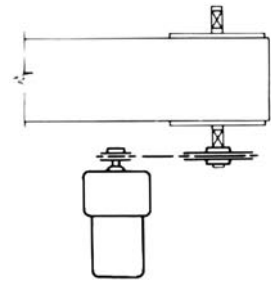
- Figure 6.26, Gear motor directly connected by flexible coupling to drive shaft, is a simple, reliable and economical drive.



**Figure 6.26 Gear motor is directly connected, by a flexible coupling, to the motor's drive shaft.**

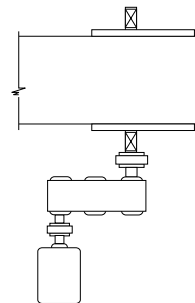
- Figure 6.27 — Gear motor combined with chain drive to drive shaft — is one of the lowest cost flexible arrangements and is substantially reliable.

**Figure 6.27 Gearmotor combined with chain drive or synchronous belt drive to drive shaft — is one of the lowest cost flexible arrangements, provides additional reduction, and is substantially reliable.**



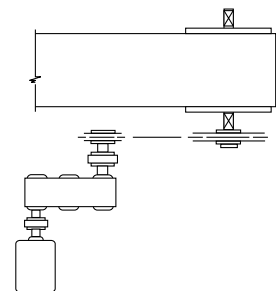
- Figure 6.28 — Parallel-shaft speed reducer directly coupled to the motor and to drive shaft — is versatile, reliable, and generally heavier in construction and easy to maintain.

**Figure 6.28 Parallel-shaft speed reducer directly coupled to the motor and to drive shaft — is particularly well suited to large conveyors, is versatile, reliable, and generally heavier in construction and easy to maintain.**



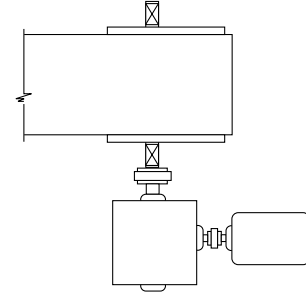
- Figure 6.29 — Parallel-shaft speed reducer coupled to motor, and with chain drive, to drive shaft — provides flexibility of location and also is suitable for the higher horsepower requirements.

**Figure 6.29 Parallel-shaft speed reducer coupled to motor, and with chain drive, to drive shaft — provides flexibility of location and also is suitable for low speed high torque requirements.**



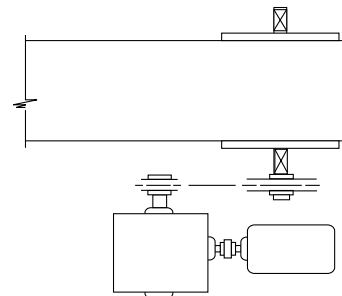
• Figure 6.30 — Spiral-bevel helical speed reducer, or worm-gear speed reducer, directly coupled to motor and to drive shaft — is often desirable for space-saving reasons and simplicity of supports. The spiral-bevel speed reducer costs substantially more than the worm-gear speed reducer but is considerably more efficient.

**Figure 6.30** Spiral-bevel helical speed reducer, helical-worm speed reducer, or worm-gear speed reducer, directly coupled to motor and to drive shaft — is often desirable for space-saving reasons and simplicity of supports. The spiral-bevel speed reducer costs somewhat more than the helical-worm speed reducer and considerably more than the worm-gear speed reducer but is more efficient than the helical-worm and considerably more efficient than the worm gear.



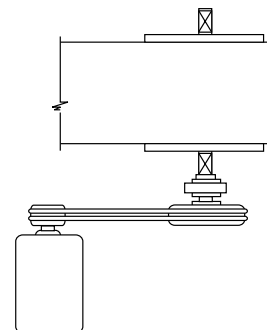
• Figure 6.31 — Spiral-bevel helical speed reducer, or worm-gear speed reducer, coupled to motor and, with chain drive, to drive shaft — is a desirable selection for high reduction ratios in the lower horsepower requirements. This drive is slightly less efficient, but has lower initial costs and is most flexible in terms of location.

**Figure 6.31** Spiral-bevel helical speed reducer, helical-worm speed reducer, or worm-gear speed reducer, coupled to motor and, with chain drive to drive shaft — is a desirable selection for high reduction ratios in the lower horsepower requirements. This drive is slightly less efficient, but has lower initial costs and is most flexible in terms of location.



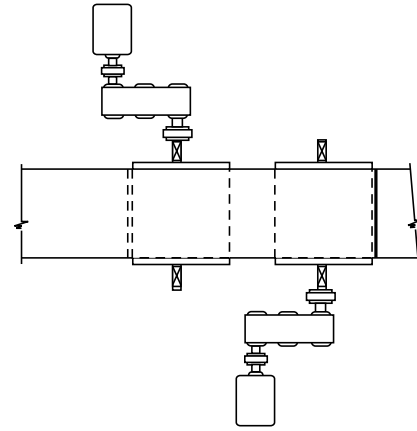
• Figure 6.32 — Drive-shaft-mounted speed reducer with V-belt reduction from motor — provides low initial cost, flexibility of location, and the possibility of some speed variation and space savings where large speed reduction ratios are not required and where horsepower requirements are not too large.

**Figure 6.32** Drive-shaft-mounted speed reducer with direct drive of V-belt reduction from reducer mounted motor — provides low initial cost, flexibility of location, and the possibility of some speed variation and space savings where large speed reduction ratios are not required and where horsepower requirements are not too large.





- Figure 6.33 — Dual-pulley drive, shown in Figure 6.33, is used where power requirements are very large, and use of heavy drive equipment may be economical by reducing belt tensions.



**Figure 6.33 Two motors (dual-pulley drive) coupled to helical or herringbone gear speed reducers, directly coupled to drive shafts.**

Selection of the type of speed-reduction mechanism can be determined by preference, cost, power limitations, limitations of the speed-reduction mechanism, limitations of available space, or desirability of drive location. The use of speed reducers in the drives for belt conveyors is almost universal today. However, space-saving considerations and low initial cost sometimes may dictate the use of countershaft drives with guarded gear or chain speed reductions.

All of the drives shown can be assembled in either left- or right-hand arrangement.

### Drive Efficiencies

To determine the minimum horsepower at the motor, it is necessary to divide the horsepower at the drive shaft by the overall efficiency of the speed reduction machinery.

To determine the overall efficiency, the efficiencies of each unit of the drive train are multiplied together. The final product is the overall efficiency.

The efficiencies of various speed-reduction mechanisms are listed in Table 6-11. These efficiencies represent conservative figures for the various types of drive equipment as they apply to belt conveyor usage. They do not necessarily represent the specific efficiencies of the drive units by themselves. Rather, they take into account the possible unforeseen adverse field conditions involving misalignment, uncertain maintenance, and the effects of temperature changes. While there are some variations in efficiency among different manufacturers' products, the data in Table 6-11 generally cover the efficiencies of the various speed-reduction mechanisms.

As an example of the application of the overall drive efficiency — the result of combining equipment unit efficiencies — consider a belt conveyor drive consisting of a double-helical-gear speed reducer and an open-guarded roller chain on cut sprockets. The approximate overall efficiency, according to Table 6-11, is  $(0.94)(0.93) = 0.874$ . If the calculated minimum horsepower at the drive shaft is 13.92 hp, then the required motor horsepower is  $13.92/0.874 = 15.9$  hp. Therefore, it is necessary to use at least a 20-hp motor.

## Belt Conveyor Drive Equipment

### Mechanical Variable Speed Devices

The most common mechanical methods of obtaining variable speeds of belt conveyors are: V-belt drives on variable-pitch diameter sheaves or pulleys, variable-speed transmissions, and variable-speed hydraulic couplings.

The choice of these devices depends upon the power and torque to be transmitted, the speed range and accuracy of control required, how well the chosen control works into the system, and the relative initial and maintenance costs.

### Creep Drive

In a climate with low temperatures that cause ice to form on the conveyor belt, with resulting loss in conveyor effectiveness, it is good practice to consider the installation of a creep drive in connection with the drive equipment. The creep drive can also be used to provide an effective means for inspecting a conveyor. The creep drive consists of an auxiliary small motor and drive machinery, which, through a clutch arrangement, takes over the driving of the empty conveyor at a very slow speed. This creep drive is arranged to be operative at all times when the conveyor is not handling any load, thus preventing the formation of harmful ice deposits on the conveyor belt. Creep drives are normally run at about 10 percent of normal belt speed.

**Table 6-12. Mechanical efficiencies of speed reduction mechanisms.**

Type of Speed Reduction Mechanism	Approximate Mechanical Efficiency
V-belts and sheaves	0.94
Roller chain and cut sprockets, open guard	0.93
Roller chain and cut sprockets, oil-tight enclosure	0.95
Single reduction helical or herringbone gear speed reducer or gearmotor	0.98
Double reduction helical or herringbone gear speed reducer or gearmotor	0.97
Triple reduction helical or herringbone gear speed reducer or gearmotor	0.95
Double reduction helical gear, shaft-mounted speed reducers	0.97
Spiral bevel-helical speed reducer single, double, or triple reduction	*See note below
Low-ratio (up to 20:1 range) helical-worm speed reducers	0.90
Medium-ratio(20:1 to 60:1 range) helical-worm speed reducers	0.85
High-ratio (60:1 to 100:1 range) helical-worm speed reducers	0.78
Low-ratio (up to 20:1 range) worm-gear speed reducers	0.90
Medium-ratio(20:1 to 60:1 range) worm-gear speed reducers	0.70
High-ratio (60:1 to 100:1 range) worm-gear speed reducers	0.50
Cut gear spurs	0.90
Cast gear spurs	0.85

\*Efficiencies for bevel-helical speed reducers are slightly less than the efficiencies shown for helical-gear speed reducers.

## Backstops

A loaded inclined belt conveyor of sufficient slope tends to move backwards, when forward motion is stopped by a cessation or interruption of power or a mechanical failure in the driving machinery. Should the loaded belt move backward, the material would pile up at the tail end of the conveyor. This could seriously damage the belt, impose a safety hazard, and result in the need to clean up and dispose of the spilled material. To prevent this reversal of motion, a backstop is used.

A backstop is a mechanical device that allows the conveyor to operate only in the desired direction. It permits free rotation of the drive pulley in the forward direction but automatically prevents rotation of the drive pulley in the opposite direction.

There are three general backstop designs: ratchet and pawl, differential band brake, and over-running clutch. See Figures 6.34, 6.35, and 6.36.

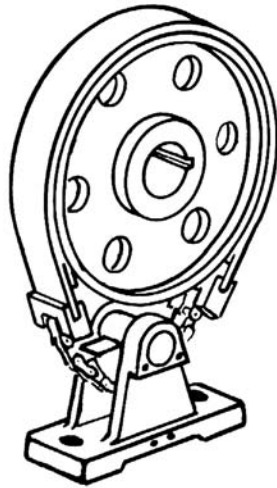


Figure 6.34 Typical differential band brake backstop.

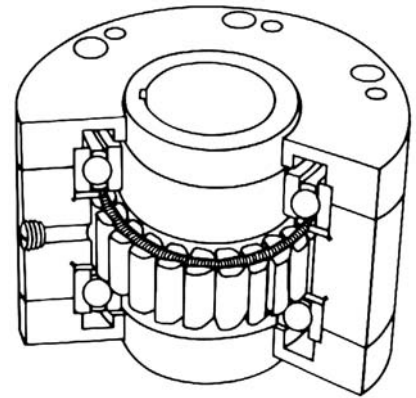


Figure 6.35 Sprag type holdback.

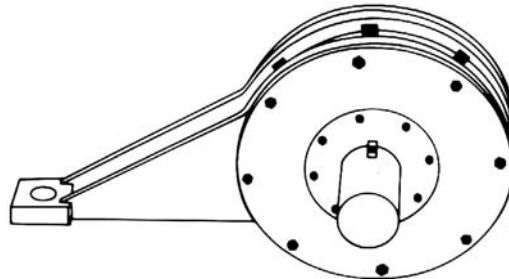


Figure 6.36 Over-running clutch backstop.

**Determine Need and Capacity of Backstop, Inclined Conveyors**

When the force required to lift the load vertically is greater than one-half the force required to move the belt and load horizontally, a backstop is required. That is, when:

$$HW_m > \frac{L(K_x + K_y W_b + 0.015 W_b) + W_m LK_y}{2}$$

See the formula for  $T_e$ , page 87, omitting  $T_p$ ,  $T_{am}$ , and  $T_{ac}$ .

Because a backstop is a safety device, it is important that the friction forces that retard the reverse motion of the conveyor are not overestimated. The above formula reduces these friction forces by 50 percent and *eliminates both the temperature correction factor and the friction introduced by the conveyor accessories.*

Backstops are rated on the basis of the pound-ft (lb-ft) of torque they can safely develop. To determine the approximate amount of torque required of a backstop mounted on the drive pulley shaft, the following analysis applies:

$r$  = radius of drive pulley, ft

rpm = revolutions per minute of drive pulley shaft

*Torque required of backstop:*

$$\text{Torque} = r \left[ HW_m - \frac{L(K_x + K_y W_b + 0.015 W_b) + W_m LK_y}{2} \right]$$

$$\text{Torque} = \frac{\text{hp} \times 5,250}{\text{rpm}} \quad \text{and} \quad \text{hp} = \frac{\text{rpm} \times \text{Torque}}{5,250}$$

*Horsepower of brake:*

$$\text{hp}_b = \left( \frac{\text{rpm} \times r}{5,250} \right) \left[ HW_m - \frac{L(K_x + K_y W_b + 0.015 W_b) + W_m LK_y}{2} \right]$$

This analysis applies to straight inclined conveyors. For conveyors with irregular profiles, a special analysis must be made.

Most manufacturers of backstops recommend size selection based on maximum breakdown or stalled torque of motor. Refer to specific manufacturer for his selection procedure.

**Brakes**

A loaded declined regenerative conveyor, when operating, is restrained from running away by the power source. Any interruption of power or mechanical failure of the drive will permit the belt and load to run out of control. To prevent this, a properly located brake is required.

A horizontal conveyor, or a declined conveyor that is not regenerative, may coast to a degree that is not tolerable. In such cases, a brake is used to regulate the stopping time and distance.

A brake is a friction device for bringing a conveyor belt to a controlled stop. While brakes are used to bring a conveyor to rest in the event of power failure or mechanical drive failure, they also are used to control the coasting distance of a conveyor as it is being decelerated, in order to limit the amount of material that will discharge over the

head pulley during the stopping interval. Brakes are used instead of backstops on inclined reversible conveyors, because backstops are unidirectional.

The brakes used in belt conveyor control operate on the principle that the braking surfaces are engaged by springs and disengaged either by a magnet or by hydraulic pressure induced by an electric motor-hydraulic pump combination. These two types generally are classified by the method of disengaging the braking surfaces. Eddy-current brakes are also used for deceleration.

Practically all conveyors involving lift or lowering need, in addition to the braking force, a holding action after the conveyor has come to a standstill, if for no other reason than safety. In the case of an inclined conveyor, this holding action could be provided by a backstop. However, for any declined conveyor, there is an obvious need for some device that permits application of a controlled torque to decelerate the load at a reasonable reduction in the rate of speed, yet allows sufficient holding power to keep the conveyor belt securely at a standstill when fully loaded but not in operation.

Any conveyor which, under some condition of loading, is regenerative must, for purposes of deceleration analysis and holding power of the brake, be considered as a declined conveyor.

### **Mechanical Friction Brakes**

The mechanical friction brakes are commonly operated electrically. For safety reasons (power failure) such brakes should be spring-set, and power-released.

These mechanical friction brakes provide both the necessary decelerating torque and final holding action. They are interconnected electrically with the motor such that when the power to the motor is off, the holding coil, on the brake also is de-energized, thus allowing a spring to set the brake. For this reason, these brakes are “fail-safe.”

The designer should bear in mind, though, that a friction brake is not a precision device, because of the inherently disadvantageous properties of brake linings. The coefficient of friction of brake lining, and with it the actual braking torque, is affected by temperature, humidity, and the degree to which the lining has become worn.

### **Eddy-Current Brakes**

Eddy-current brakes produce a dynamic braking torque by means of a smooth drum that rotates in a magnetic field produced by a stationary field coil. Eddy currents are generated in the surface of the drum as it rotates. A magnetic attraction between these eddy currents and the poles of the field assembly produces a braking torque on the drum. This torque varies directly with the field current and the speed of the drum. It can be adjusted in a stepless manner by a control system.

For holding action, because the eddy-current brake is not effective in case of power failure, it should be combined with an auxiliary mechanical friction brake. As an eddy-current brake drum slows down, the torque that it is capable of exerting diminishes and is zero when the drum ceases to rotate. Thus, an eddy-current brake cannot be expected to hold a conveyor belt in a standstill position. The auxiliary friction brake also serves to decelerate the conveyor in case of power failure.

Deceleration can also be achieved within the drive motor and its control. There are three basic ways of achieving this braking action, none of which provides holding power after the conveyor belt has come to rest. For this reason, some type of auxiliary external brake is always needed to hold the conveyor belt at a standstill.

## Brakes and Backstops in Combination

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### Plugging the Motor

Here, the current is reversed and counter torque is developed. This force attempts to rotate the motor in a direction opposite to the existing motion. The energy is dissipated as heat. The motor must be de-energized when zero speed is reached, otherwise the motor will attempt to accelerate in the reverse direction. Among others, squirrel-cage motors are most suitable for this application. There is no holding effect at zero speed and the electrical power losses during plugging are high.

### Dynamic Braking

Dynamic braking is a system of electric braking in which motors are used as generators and the kinetic energy of the load is employed as the actuating means of exerting a retarding force. To dynamic brake a.c. motors, it is necessary to provide a source of d.c. excitation during the braking period. The control is so arranged that when the stop button is depressed and the a.c. line contactor is opened, another contactor closes to connect the d.c. excitation to one phase of the motor primary. The motor now acts as a generator and is loaded by the induced current flowing through its squirrel-cage winding. The braking torque, which varies in proportion to the exciting current, rapidly increases as the motor slows down but then decreases at near zero speed. The braking torque disappears near zero speed and there is no holding effect at zero speed.

### Regenerative Braking

Squirrel-cage motors operating above synchronous speed have inherent retarding torque characteristics. This retarding condition, known as regenerative braking, is applicable above the synchronous speed of the motor (or for multi-speed motors above their synchronous speeds). The energy generated by the motor flows back into the electric power line. Care must be taken to insure that the electric power system is capable of absorbing the power generated by the motor.

This fundamental type of braking is found to be especially useful for declined conveyors operating at a speed that drives the motor at its synchronous speed, plus slip.

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## Brakes and Backstops in Combination

Often a brake is used to control the stopping interval on an inclined conveyor. If the conveyor is a large and important one, which may reverse and run backward in the event of a mechanical failure, prudence dictates the use of a mechanical backstop as a safety precaution, in addition to the electrically operated brake.

Friction surfaces on brakes, and brakes used as backstops, do not develop the design friction factors until the braking surfaces have worn in to effect full contact. Therefore, friction brakes used as such or as backstops must be adjusted to compensate for this "wearing in" process.

### Restraint of Declined Conveyors

Declined conveyors of the regenerative type, are restrained in normal operation by the drive motor which acts as a generator, when the belt and its load force the motor to rotate faster than its synchronous speed. The motor may fail to restrain the belt and load when it is forced to a speed where its current output is excessive and the overload protective device breaks the circuit. Proper selection of the motor and controls will avoid this contingency. Nevertheless, a brake must be supplied, one which will set when the power circuit is broken.

A centrifugal switch is often used on declined belt conveyors to open the electrical control circuit at a predetermined overspeed, and thus to set the brake. This will act as a safety against mechanical failure in the drive machinery.

A brake is usually located at the tail end of a declined conveyor.

**Table 6-13. Backstop and brake recommendations.**

<b>Type of Conveyor</b>	<b>Backstop</b>	<b>Brake</b>	<b>Forces to be Controlled</b>
Level or horizontal conveyor	Not required	Required when coasting of belt and load is not allowable or needs to be controlled	Decelerating force minus resisting friction forces
Inclined conveyor	Required if hp of lift equals or exceeds hp of friction	Not usually required unless preferred over backstop	Incline load tension minus resisting friction forces
Declined conveyor	Not required	Required	Decelerating force plus incline load tension minus resisting friction

**Backstop and Brake Recommendations**

Table 6-13 lists recommendations for the use of backstops and brakes on horizontal, inclined, and declined conveyors.

**Deceleration by Brakes**

Brakes are a necessity on declined conveyors so that the loaded belt may be stopped without excessive or runaway coasting. Brakes are also applied to horizontal and inclined belt conveyors for the same reason. Excessive coasting may discharge far more material than the succeeding conveyor or other units can handle. Mathematical calculation and the careful selection of a properly sized brake will eliminate such difficulties.

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**Devices for Acceleration, Deceleration, and Torque Control**

**Starting the Conveyor**

Smooth starting of a conveyor belt is important. It can be accomplished by the use of torque-control equipment, either mechanical or electrical, or a combination of the two. The belt conveyor designer should investigate acceleration stresses of conveyor components to insure that the overall stresses remain within safe limits.

Smooth starting can be an important consideration, where excess horsepower may have been installed to provide for future increased capacity or for future extensions of the conveyor. In cases of conveyors having vertical curves or trippers, too rapid a start may cause excessive lifting of the belt from the idlers. This would necessitate a provision for gradual acceleration of the conveyor belt.

### **Controlled Acceleration**

Acceleration can be controlled by several types of electrical devices.

#### *Wound-Rotor Motors with Step Starting*

By the addition of external resistance in the secondary winding, electrically accessible through slip rings, starting torque can be controlled by planned steps. This allows a program designed to suit the particular conveyor, and overcome the problems of excessive belt tension, shape of the vertical curves, and other problems that are solved by starting time control.

This type of electrical control device has been widely used for many years on large belt conveyor systems.

#### *Squirrel-Cage Induction Motor with Autotransformer*

Another method of controlling the torque, and with it the acceleration time, is the use of an induction motor (normal or high-torque) with autotransformer starting. Its use must be checked because the low-starting torque caused by the reduced voltage may not be enough to overcome the breakaway static friction in level or inclined conveyors.

#### *Eddy-Current Couplings*

These are electromagnetic devices composed of three basic parts: a rotor made up of multiple pole pieces (and secured to one shaft), a hollow iron cylinder or drum that surrounds the rotor (and is secured to the other shaft), and a stationary electromagnetic coil that surrounds both rotor and drum and provides the magnetic field in which they operate.

The electromagnetic coil is energized by a low-power, direct current supply. When either the rotor or the drum is rotated, eddy currents are induced. These eddy currents set up a secondary field and thus create a torque between the rotor and drum. The driven or output member never attains the same speed as the driving or input member. This inherent difference in speed is called "slip." The slip loss appears as heat, which must be dissipated by air or water cooling.

In a conveyor drive, the eddy-current coupling is placed between the squirrel cage motor and speed reducer, on the motor shaft, and on the speed reducer input shaft. Because the degree of excitation of the coil determines the slip between the driving and driven members, it is obvious that eddy current couplings provide an ideal means of controlled acceleration. Excitation of the coil can be increased over a definite time period, or it can be made responsive to tachometer feedback speed-regulating control. Sophisticated electronic control can be employed to regulate the coil excitation to produce virtually any desired result.

There are several advantages of eddy-current couplings. (1) They require low-power coil excitation. (2) They permit smooth, controlled starting. (3) The motor can be started and accelerated without connecting the load. On frequent start and stop applications the motor can run continuously. (4) Variable speed can be obtained. However, in variable speed applications the additional slip creates more heat that must, of course, be dissipated. (5) They make possible the use of squirrel-cage motors and across-the-line starters. (6) A modified eddy-current coupling can be used as a decelerating brake (not as a "holding brake," however).

The disadvantages of eddy-current couplings are: (1) They require additional drive space. (2) Water cooling must be provided for the larger sizes. (3) Generally, they are more expensive than a wound-rotor motor and reduced-voltage starting.



### *Fluid Couplings*

These are mechanical two-piece mechanisms consisting of an impeller and a runner, both within a housing filled with oil. The impeller is connected to the driving shaft; the runner is connected to the driven shaft. In conveyor drives, the fluid coupling usually is placed between the motor and speed reducer. When the impeller is revolved, the oil is accelerated to the periphery and passes to the blades of the runner, producing a torque on the runner proportional to the weight and rate of fluid flow. The fluid coupling basically is a slip clutch. And, as with an eddy current coupling, the slip loss appears as heat. Unlike its electrical counterpart, a fluid coupling is not used as a variable-speed device.

When properly applied, a fluid coupling can produce reasonably smooth acceleration of high-inertia loads. The motor speed rises rapidly to a point near the maximum torque condition before the load is engaged. This makes the standard squirrel-cage motor an ideal driver, as its peak torque is about 200 percent of full-load torque. The fluid coupling permits the use of squirrel-cage motors with across-the-line starters.

Fluid couplings also provide an excellent solution to voltage drop problems by allowing the motor to achieve full speed before it experiences load. In addition, fluid couplings limit belt forces imparted due to higher voltage at start-up, thus preventing belt slippage.

### *Variable-Speed Hydraulic Couplings*

These devices have been used very successfully worldwide. The variable-speed hydraulic coupling consists of a fluid coupling with input and output shafts, heat exchanger, charging oil pump, and associated control. The amount of oil in the coupling is variable by the position of an adjustable scoop tube. The control can be manual or completely automatic. Speed variations over a 4:1 range are possible. The device allows the AC motor to start under no load. The torque buildup starts at zero and the buildup can be controlled over an extended time. Smooth acceleration times up to 200 seconds are achievable. The device can also act as a clutch without AC motor shutdown.

### *Dry Fluid Couplings*

These are similar to oil-filled fluid couplings, except that they consist of a housing that is keyed to the motor shaft and a rotor that is connected to the load. The housing contains a charge of steel shot instead of a fluid. When the motor is started, centrifugal force throws the charge of steel shot to the inner periphery of the housing, where it packs around the rotor. Some slippage takes place before the housing and rotor are finally locked together. Power thus is transmitted from the motor to the load.

The amount of the charge of steel shot determines the torque during acceleration. It can also determine the torque-limiting feature of this coupling.

### *Miscellaneous Fluid Couplings*

These are similar to dry fluid couplings except, in lieu of a charge of steel shot, these fluid couplings employ silicone fluid or other environmentally suitable liquids. Characteristics of these couplings are such that manufacturers should be consulted for specific performance details before an application is made.

### *Centrifugal Clutch Couplings*

These consist of a driving hub, a driven sleeve or drum, and a series of shoes connected to or driven by the hub. The periphery of each shoe is provided with a brake-lining material. The hub is carried by the driving shaft, the drum by the driven shaft.

## Brake Requirement Determination (Deceleration Calculations)

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When the hub rotates, centrifugal force impels the shoes outward against the inside of the drum, to transmit power to the load. Slippage occurs, which produces the effect of smooth starting.

### *Flywheel*

Mechanical control of starting and stopping can be accomplished by means of a flywheel, which adds to the WK2 of the prime mover, thus increasing the starting time and limiting the torque input to the belt conveyor system, as well as increasing stopping time and distance.

### *Mechanical Clutches*

This device can effectively control starting torque; it allows adjustment of the amount of torque, as well as the rate at which it is applied. The mechanical clutch can be preset for both rate of application and maximum limit.

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## Brake Requirement Determination (Deceleration Calculations)

In order to determine whether any braking action, other than the friction forces inherent in the system, is required, several different circumstances under which the conveyor might be stopped have to be considered. For instance, is the stopping intentional, or is it the result of power failure? Also, if another conveyor feeds onto it, or if the conveyor in question delivers its load to an additional belt, it is necessary to consider their respective motions and deceleration cycles.

It is obvious that the drives of a conveyor system that consists of more than one belt, and in which at least one belt feeds into another, have to be interconnected electrically in such a way that if one conveyor is stopped for any reason, the one feeding onto it is also stopped. This precaution alone does not suffice, however, if the physical properties of the first conveyor are such that it would coast longer than the second one. If this were to occur, it would result in a pile-up of material on the second belt and could cause a hazardous situation.

Generally speaking, in any system with more than one conveyor, the length of the deceleration cycle of any successive conveyor should be equal to or more than that of the preceding one.

If the inherent properties of the various units do not result in deceleration cycles that agree with this basic rule, two remedies are possible. (1) A brake can be applied to those conveyors that coast too long. This is a straightforward solution, and relatively easy to accomplish. (2) The stored energy of those conveyors that come to a stop too quickly can be augmented, for instance, by a flywheel. Although a flywheel will lengthen the stopping distance of a conveyor, it will also increase its acceleration time. This must be taken into consideration by the belt conveyor designer.

However, in most cases, the application of a brake will be found more convenient, unless its use overstresses any member of the unit to which it is applied.

### **Material Discharged During Braking Interval**

To determine the amount of material discharged during the braking interval, it must be assumed that the conveyor decelerates at a constant rate. Therefore, the distance travelled, while stopping from full speed, is the average velocity multiplied by the time of braking interval.

$$\text{Distance, ft, conveyor travels} = \left(\frac{V+0}{2}\right)\left(\frac{t_d}{60}\right) = \frac{Vt_d}{120}$$

where :

$V$  = belt speed, fpm  
 $t_d$  = actual stopping time, seconds

If the amount of material that can be safely discharged to the succeeding conveyor (or other) unit is known, the maximum length of time of the braking or decelerating interval can be determined as follows:

$$W_d = \frac{V t_m}{120} (W_m)$$

Therefore:

$$t_m = \frac{120 W_d}{W_m V}$$

where :

$t_m$  = maximum permissible stopping time, seconds (braking or deceleration interval)  
 $W_d$  = weight, lbs, which can be discharged  
 $W_m$  = weight of material, lbs/ft of belt

### **Forces Acting During Braking or Deceleration**

The forces that act on the conveyor during a braked stop (deceleration) include inertia; frictional resistance; gravity material load force; inclines or declines; and braking force.

The frictional resistance forces and the gravity material load forces, if any, are equal to  $T_e$ . The braking force is equal to the algebraic sum of the other forces.

Therefore, for horizontal, inclined, and non-regenerative declined belt conveyors, the braking force = inertial forces +  $T_e$  or:

$$F_d = \frac{M_e V}{60 t_m} - T_e = \frac{W_e V}{60 g t_m} - T_e$$

For regenerative declined belt conveyors, braking force = inertial forces +  $T_e$ , or:

$$F_d = \frac{M_e V}{60 t_m} + T_e = \frac{W_e V}{60 g t_m} + T_e$$

where

$F_d$  = braking force, lbs, at belt line  
 $M_e$  = equivalent moving mass, slugs  
 $g$  = acceleration of gravity, 32.2 ft/sec<sup>2</sup>  
 $W_e$  = equivalent weight of moving parts of the conveyor and its load, lbs. See Problems 3 and 4, pages 153-161 and 161-172, respectively.  
 $V$  = speed of belt, fpm  
 $t_m$  = maximum permissible stopping time, seconds (braking or deceleration interval)  
 $T_e$  = effective or driving horsepower tension, lbs

### Brake Location

An analysis of the belt tension diagram during deceleration should be made to determine the appropriate pulley on which to apply the brake. The braking force will be additional to the friction and positive lift forces.

If the brake is installed on a head-end drive pulley, the automatic takeup force must be sufficient to transmit the braking force through the takeup. The wrap factor at the braking pulley must be checked for adequacy during braking. Also, the minimum belt tension in the carrying run of the conveyor must be maintained during braking. The maximum permissible belt tension must not be exceeded during deceleration.

For inclined or short horizontal conveyors, it may be possible to brake through the head or drive pulley, providing the takeup has sufficient force to absorb the braking force and still maintain a slack-side tension to meet the wrap factor requirements. If this is not practical, as in the case of a long horizontal or declined conveyor, then the braking force must be applied to the tail pulley.

The maximum belt tension during deceleration should be calculated to insure that it does not exceed the recommended allowable starting (or braking) tension. (See page 113, and Chapter 13, "Controlled Deceleration.") If it is found that the belt tension does exceed the allowable amount, a heavier belt may be required. Or the belt conveyor can be reanalyzed to provide for a smaller braking force acting over a longer time period. If the conveyor is subjected to frequent stops, the pulleys and shafts must be selected for the higher tensions introduced during deceleration.

### Braking Torque

The braking force (lbs) determined above and acting at the belt, multiplied by the radius (ft) of the braked conveyor pulley, gives the required torque rating of the brake (lb-ft), provided the brake is installed on the same shaft that carries the braking pulley.

$$\text{Torque} = F_d r$$

where:

$F_d$  = braking force at the belt

$r$  = radius of conveyor pulley, ft, on the same shaft as the brake

If the brake is to be installed on some shaft other than the pulley shaft, the torque requirement is converted by multiplying the above torque by the revolutions/minute of the shaft for which the torque was determined. This product is then divided by the revolutions/minute of the shaft on which the brake will be mounted. Select the brake with the next higher torque rating.

### Brake Heat Absorption Capacity

The discussions above relate to the selection of a brake on the basis of torque only. Stopping a moving mass involves the absorption of the kinetic energy of the belt, the load, and the moving machinery. This energy only can be dissipated in the form of heat at the brake. The resulting temperature rise of the brake elements must not damage the brake. For this reason, a discussion of brake design and brake heat absorption follows.

Industrial brake linings usually are made of either woven or molded material, plus various fillers and adhesives. The coefficient of friction of these linings against a brake wheel varies considerably with different ambient conditions. Because of the

nature of these variations, definite values of the friction coefficient cannot be given. Nevertheless, some variations that can be expected are approximated below.

Coefficients, and consequently torque values, may vary widely for new linings and/or new wheels, until both lining and wheel surfaces are worn in. This requires approximately 4,000 to 6,000 full-torque brake-setting operations. During this period static torque may drop as much as 30% below the initial setting, and dynamic torque as much as 50%. For this reason, the discussion will relate only to well-worn-in linings and wheels.

Both static and dynamic torque vary with wheel surface temperatures. At 50°C to 75°C, the static torque may be as much as 30 percent to 35 percent high. But it then drops off rather rapidly with increase in wheel temperature. At 115°C to 135°C, the static torque is about normal. At 150°C, it may be 5% to 7% below normal. The dynamic torque may be 10 percent to 15 percent high at 40°C to 60°C, and then rise rapidly, until at 115°C to 150°C it may be as high as 140 percent. It then drops off rather rapidly with further temperature rise.

Because of these variations, brake wheels are rated at 120°C rise for normal energy dissipations. The ratings, which are expressed in “hp seconds,” are based on a maximum temperature rise of 120°C at the brake wheel, when the brake is applied at the listed time intervals. The brakes have lower ratings for more frequent stops because they will not cool sufficiently between stops to absorb the heat of rapidly repeated stopping.

Humidity will also have an adverse effect on braking torque because industrial brake linings absorb moisture. If a brake is allowed to stand inoperative for some time in high ambient humidity, the braking torque may be reduced as much as 30% when the brake is first set. This condition is self-correcting, because the heat generated in braking rapidly drives off the moisture. Usually the torque will be restored to almost normal at the end of the first braking cycle. In this case, the only effect is a longer time than usual to make the first stop.

Variations in any given lining material, and in surface conditions of the lining and wheel, may result in a 10 percent plus or minus variation in torque during successive stops.

From the above factors, it is evident that industrial brakes are not precision devices. The normal method of setting brake torque by measuring either spring length or adjusting-bolt length is at best an approximation. Where braking effects are important to a conveyor operation, the brake should be readjusted for optimum braking by actually stopping and holding the load after the brake first is installed. For critical conveyor applications, it may be necessary to readjust the brake more than one time during the break-in period for new linings.

### Brake Calculations

To check the brake wheel heat absorption for a single stop of a loaded conveyor, first determine the actual stopping time for the brake selected.

$$t_d = \frac{W_e V}{(32.2)(60)} \left( \frac{Z_b}{r} \right) \left( \frac{\text{rpm}_b}{\text{rpm}_p} \right) + T_e$$

where:

- $t_d$  = actual stopping time, seconds
- $W_e$  = equivalent weight of the moving mass, lbs
- $V$  = velocity or speed of belt, fpm
- $Z_b$  = torque rating or setting of brake, lb-ft
- $\text{rpm}_b$  = revolutions/minute of brake shaft
- $\text{rpm}_p$  = revolutions/minute of drive pulley shaft

**NOTE:** For regenerative belt conveyors,  $T_e$  may be negative.

The energy that must be absorbed by the brake when making a single stop of a loaded conveyor is expressed as follows:

$$\text{Energy, in horsepower seconds} = \frac{(Z_b)(\text{rpm}_b)(t_d)}{10,500}$$

The symbols are the same as above, but with the brake on the drive pulley shaft,  $\text{rpm}_p = \text{rpm}_b$ .

The heat absorption should be approved by the brake manufacturer for the anticipated duty cycle.

If the brake selected does not have the heat-absorption capacity required, either a modified or larger brake with the necessary heat-absorption capacity should be used. The spring should be adjusted to the torque desired.

**Example**

As a numerical example of brake selection, the belt conveyor specifications for Problem 3, page 153, will be used. Since the  $WK^2$  and total equivalent weight for the conveyor have been calculated, only the essential portions of these specifications are repeated here.

*Conveyor Specifications:*

$V$ = belt speed	=	500 fpm
$W_m$ = weight of material/ft of belt	=	226 lbs
$T_e$ = effective tension	=	16,405 lbs
$T_2$ = slack-side tension	=	5,720 lbs
$T_0$ = minimum tension	=	3,067 lbs
$T_t$ = tail tension	=	7,054 lbs
Equivalent weight of conveyor moving parts	=	162,696 lbs
Weight of material load	=	543,360 lbs
Total equivalent weight for belt tension determination	=	706,056 lbs
Drive equivalent at belt	=	<u>62,870</u> lbs
$W_e$ = total equivalent weight for brake determination	=	768,926 lbs

Assuming that the conveyor discharges into a hopper that holds 9,000 lbs of material, the maximum permissible stopping time is as follows:

$$t_m = \frac{120 W_d}{W_m V} = \frac{(120)(9,000)}{(226)(500)} = 9.54 \text{ seconds}$$

Brake force at belt line is:

$$F_d = \frac{W_e V}{60 g t_m} - T_e = \frac{(768,926)(500)}{(60)(32.2)(9.54)} - 16,342 = 4,517 \text{ lbs}$$

*Analysis:*

If the total equivalent retarding force of 4,517 lbs is applied to the head-end drive shaft, a proportion equal to:

$$\frac{(62,870)(500)}{(60)(32.2)(9.54)} = 1,706 \text{ lbs}$$

of the equivalent force would be absorbed in retarding the drive components. The remainder,  $4,517 - 1,706 = 2,811$  lbs of the equivalent force, would be transmitted to the belt by the pulley to retard the conveyor moving parts and the load. This force is  $T_{eb}$ .

During braking, the highest tension in the belt will be  $T_{1b}$ , on the return run just past the drive pulley. If the automatic takeup is to be on the verge of yielding to the braking force,  $T_{1b}$  can be assumed to be equal to  $T_2$ , the slack-side tension during normal operation of the belt. Because:

$$T_{1b} - T_{2b} = T_{eb}$$

and substituting  $T_2$  for  $T_{1b}$ :

$$T_2 - T_{2b} = T_{eb} = 5,720 - T_{2b} = 2,811$$

*Therefore:*

$$T_{2b} = 5,720 - 2,811 = 2,909 \text{ lbs}$$

This is the tension in the carrying run at the head-end drive pulley during braking. It is insufficient, for the minimum tension,  $T_0 = 3,067$  lbs. Also,  $T_{2b} = (C_{wb})(T_{eb})$ , or:

$$C_{wb} = \frac{T_{2b}}{T_{eb}} = \frac{2,909}{2,811} = 1.03 \text{ wrap factor during braking}$$

This is sufficient, as the wrap factor for the drive is 0.35 to prevent slip between the pulley and belt.

When it is shown that braking at the head-end drive produces too low a tension in the carrying run, or too small a wrap factor, it is necessary to increase the belt tensions by increasing the automatic takeup force.

## Brake Requirement Determination (Deceleration Calculations)

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The alternative to braking at the head-end drive pulley is to apply the braking force to the tail pulley. In this case, the entire braking force of 4,517 lbs must be transmitted to the belt.

When the power of the drive is cut off, and just as the brake takes effect, the tension in the return run at the tail pulley will be:

$$T_{2b} = T_2 + \text{pulley friction} + \text{return-belt idler friction} \\ - \text{inertia of moving parts of the return run}$$

$$\text{Pulley friction} = (4)(100) + (1)(150) = 550 \text{ lbs}$$

$$\text{Return idler friction} = L(0.015 W_b) \\ = (2,400)(0.015)(17) = 612 \text{ lbs}$$

Equivalent weights of the moving parts of the return run are:

$$\text{Return belt, } LW_b = (2,400)(17) = 40,800 \text{ lbs}$$

Return idler weight of rotating parts is 48.4 lbs, from Table 5-14, for a 48-in.-wide belt and Class C6 idlers.

$$\text{Total return idler rotating weight} = \left(\frac{2,400}{10}\right)(48.4) = 11,616 \text{ lbs}$$

Rotating weight of pulleys, from Problem 3, page 156, is 3,450 lbs

$$\text{Total equivalent moving parts of return run} = 40,800 + 11,616 + 3,450 = 55,866 \\ \text{lbs}$$

Equivalent force at the belt line

$$= \frac{W_e V}{60gt_m} \\ = \frac{(55,866)(500)}{(60)(32.2)(9.54)} \\ = 1,516 \text{ lbs}$$

Therefore:

$$T_{2b} = 5,720 + 550 + 612 - 1,516 = 5,366 \text{ lbs}$$

And because  $T_{cb} = 4,517$ , and  $C_{wb} = T_{2b}/T_{cb}$ , then

$$C_{wb} = \frac{5,366}{4,517} = 1.19$$

This is very satisfactory, since a 180-degree wrap, bare pulley requires only that the wrap factor,  $C_{wb}$ , be 0.84 or larger. See Table 6-8. The maximum belt tension when braking =  $4,517 + 5,366 = 9,883$  lbs. This is well within the maximum of  $1.80(25,920) - 9,883 = 36,773$  lbs (see page 157). It therefore is appropriate to place the brake on the tail pulley shaft of this conveyor.



Assuming the tail pulley radius is 1.5 ft and the pulley is revolving at 53 rpm, the torque at the tail pulley shaft is:

$$F_d r = (4,517)(1.5) = 6,776 \text{ lb-ft}$$

As the brake will be mounted directly on the tail pulley shaft, the required brake torque will be 6,776 lb-ft.

For this problem, assuming alternating current electric power is available, the brake selected from the brake manufacturers' catalog is an a.c. magnetic brake with a 10,000 lb-ft rating. This is the next larger size than that calculated at 6,776 lb-ft.

Actual stopping time, using this 10,000 lb-ft brake is:

$$\begin{aligned} t_d &= \frac{\frac{W_e V}{(32.2)(60)}}{\left(\frac{Z_b}{r}\right)\left(\frac{rpm_b}{rpm_p}\right) + T_e} \\ &= \frac{(768,926)(500)}{(32.2)(60)} \\ &\quad \left(\frac{10,000}{1.5}\right)\left(\frac{53}{53}\right) + 16,342 \\ &= 8.65 \text{ seconds} \end{aligned}$$

This is less than the maximum of 9.54 seconds that is permissible.

Energy absorbed is:

$$p = \frac{(Z_b)(rpm_p)(T_d)}{10,500} = \frac{(10,500)(53)(8.65)}{10,500} = 432.62 \text{ hp seconds}$$

The brake selected is capable of heat absorption of 3,400 hp-seconds every 15 minutes. This indicates that a loaded stop can be made safely without overheating the brake.