Guidelines for Proper Operation of Wire Mesh Belting

Analysis of service requirements of the particular application is required to determine the best belt design and material selection for optimum performance. Correct belt selection means longer life and lower maintenance. Belt performance is directly related to the condition, setup, and maintenance of the conveyor. For this reason, we send our Service Technicians all over the world to supervise installations. The value of this service is well recognized. This document is to be used as a guide only and is not intended to replace our trained and skilled personnel.

NOTE: Use proper safety equipment, including face and eye protection, as mandated by your company's safety policy.

CAUTION: Due to the large openings in some of the belts there is a risk that operators' fingers or clothing may become caught in belt. Appropriate guards and safeties need to be considered.

STRAIGHT-RUN CONVEYORS

TENSION CALCULATIONS

In making belt selections or determining whether the selected belt is suitable for the application, we must determine the tension. For positive-drive conveyors operating at less than 1000°F, the tension in the belt is typically highest at the drive. Consequently, the tension is zero just after the belt leaves the drive sprockets.

After the belt leaves the drive, the tension increases along the return path and on the load path. The amount of tension that is built-up through the conveyor paths can be estimated from the formulas below. It's recommended that only minimal additional tension be added to the belt in the take-up. Typically, a catenary sag is sufficient to provide enough initial tension in order for the belt to operate. The formulas below are based on the assumption that minimal force has been added in the take-up.

Tension of Simple Flat Conveyors:

For simple conveyors with a discharge end drive operating at room temperature, belt tension can be calculated from the following formula:

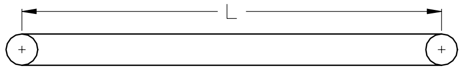
$$\boxed{T = wLf_r + WLf_l}$$

Where: T = Belt Tension, lb/ft of belt width. (Newtons/m of belt width)
w = Belt Weight, lb/ft² (kg/m²)
W = Weight of Product AND Belt, lb/ft² (kg/m²)
L = Conveyor Length (center to center of pulleys), feet (meters)
f_l = Coefficient of Friction between the belt and belt supports on the load path (dimensionless)
f_r = Coefficient of Friction between the belt and belt supports on the return path (dimensionless)

If using Metric units, multiply resultant by 9.8 to convert to Newtons.

Example—Straight-Run Conveyor:

Assume a 100 foot long, level conveyor with drive on the discharge end. Process is cooling pastries at room temperature. Load is 5.0 lb/ft² on UHMWPE-capped belt support rails. Selected belt type is a 48 inch wide true ½ x ½ Flat Wire.



$$\mathbf{T = wLf_r + WLf_l}$$

- **T** = Belt Tension, lb/ft of belt width. (Newtons/m of belt width)
- **w** = belt weight of an A5 = 3.03 lb/ft²
- **L** = conveyor length = 100 feet
- $\mathbf{f_r}$ = friction between belt and supports on return path = 0.35
- **W** = total weight of belt and product = 3.03 + 5.0 = 8.03 lb/ft²
- $\mathbf{f_l}$ = friction between belt and supports on load path = 0.35

T = 3.03 (100) 0.35 + 8.03 (100) 0.35

= **387.1 lb/foot of width**

From the Flat Wire Technical Bulletin, the allowable Tension for an A5 is 500 lb. foot of width; therefore, **Selected belt is strong enough.**

Total Belt Tension:

To determine the total tension at the drive in order to size drive components, multiply the resultant tension by the belt width in feet:

$$\boxed{T_t = T \times BW}$$

Where: BW = Belt Width, ft. (meters)

In the preceding example, for a 48" wide belt,

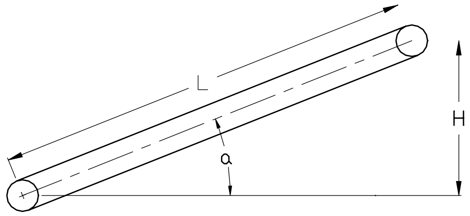
T_t = 387.1 x 4.0 = 1548.4 lb/linear foot

Tension of Simple Incline Conveyors:

For simple incline conveyors with a discharge end drive operating at room temperature, belt tension can be calculated as before but an allowance for pulling the belt up the incline must be added. Note: The reduction of tension due to the weight of the conveyor belt going downhill on the return side usually can be neglected and is omitted from the inclined conveyor formula. The formula changes to:

$$\boxed{T = wLf_r + WLf_l + WH}$$

Where: H = Rise of incline conveyor, feet (meters)



Example—Incline Conveyor:

From the previous example, assume the conveyor had a four foot rise.

T = wLfr + WLfl + WH

H = Incline = 4.0 ft.

T = 3.03 (100) 0.35 + 8.03 (100) 0.35 + (8.03)(4)

= **419.2 lb/foot of width**

Tension of Conveyors with Multiple Segments

To obtain the tension for more complex conveyor layouts, such as shown below, the tension in each segment on both the load and return paths must be summed together:

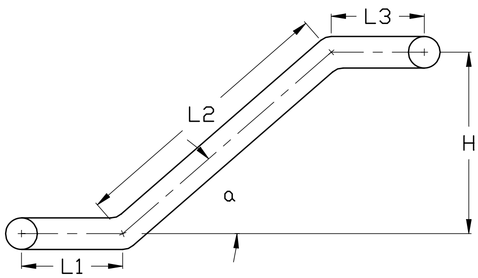
$$\boxed{T = (wL_{r1}f_{r1} + wL_{r2}f_{r2} + wL_{r3}f_{r3}) + (WL_1f_1 + WL_2f_2 + WL_3f_3) + WH}$$

Where: L_{r1} = length of segment 1 on return path… etc.

f_{r1} = friction of segment 1 on return path… etc.

L_1 = length of segment 1 on load path… etc.

f_1 = friction of segment 1 on load path… etc.



If the belt support material is the same for all segments, the formula reduces to:

$$T = wf(L_{r1} + L_{r2} + L_{r2}) + Wf(L_1 + L_2 + L_2) + WH$$

Tension of Accumulating Conveyors

Belt tension in an accumulating conveyor is equal to the tension in the system plus the additional tension created when the belt slips under the product.

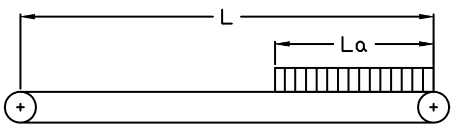
$$\boxed{T_a = T + W_lL_af_p}$$

Where: T_a = Belt tension of an accumulating conveyor, lb/ft of belt width. (Newtons/m of belt width)

W_l = Weight of product, lb/ft² (kg/m²)

L_a = Length of accumulation, ft. (m)

f_p = Coefficient of friction between belt and product (dimensionless)



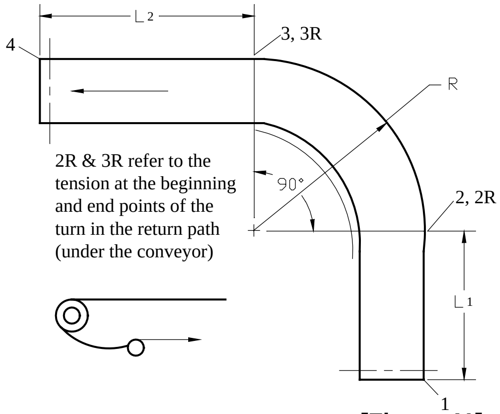
If using Metric units, multiply resultant tension by 9.8 to convert to Newtons.

Turn Curve Calculations

Turn curve belts allow for complex conveyor layouts with a single belt, thereby eliminating the need for transfers while maintaining product orientation. Turn curve belts include the Omni-Pro®, Advantage™, Omni-Grid®, and Omni-Flex® families of belts.

Typically, the inside edge of a belt collapses when entering a turn while the outside edge remains at a constant pitch. As a result, all of the belt tension is concentrated on the outside belt edge. Since the load is not shared across the belt width, the allowable tension is generally quite lower than in a straight-run condition.

However, in a turn curve application, the tension in the belt is concentrated at a single point through the turn.



[Figure 41]

Radial pressure of the belt against a fixed turn rail (see Figure 41) creates considerable belt tension due to sliding friction.

The key to maintaining low tension is making the inside radius as small as allowable for the given belt type, and providing a low friction material on the inside rail.

Inside turn rails provide an acceptable and common means of retaining turn curve belts in a curved path. To reduce wear and friction to acceptable levels, a replaceable wear strip should be used. The material must have resiliency and good abrasion resistance. The lowest possible friction between belt edge and wear strip is desirable and provision should be made for lubrication wherever feasible. A minimal amount of lubricant, compatible with the product and process, is all that is required. A commonly used material is Ultra High Molecular Weight Polyethylene (UHMWPE).

The drag created by sliding friction in a turn can be greatly reduced by the use of a rotating wheel or Ashworth powered turn Edge Drive unit (refer to Technical Product Bulletin “052 Edge Drives” for further information). Increased load carrying capacity and longer belt life are two important benefits.

Turn Ratio

In designing turn curve applications, the Turn Ratio (TR) is critical. The minimum turn ratio of each belt is used to determine the minimum inside turn radius. Consult Product Technical Bulletins to obtain the minimum turn ratio of specific belts.

$$\boxed{IR = TR \times BW}$$

Where: **IR** = Inside Radius of turn

TR = Turn Ratio

BW = Belt Width

Turn Curve Tension Limitations

All of the basic turn curve belts (Omni-Pro®, Omni-Grid®, Omni-Flex®, Small Radius Omni-Grid®, Small Radius Omni-Flex®, Reduced Radius Omni-Grid®, and Advantage™ Series) collapse on the inside edge to negotiate a turn. This places the full belt tension on the outside edge, or on the middle links in the case of the Small Radius belts. The Space Saver Omni-Grid® belts do not collapse on the inside belt edge but expand on the outside belt edge to negotiate a turn. In this case, the tension is concentrated on the second row of links. For all belts, this concentrated stress, if excessive, can cause fatigue failure of the belt components. This is the principal limitation of these belts, as the critical stress is considerably below the actual belt strength in the straight condition.

Tension Calculations—Turns

For a fixed inside rail, tension increases through a turn. This increase is calculated from by the following formula:

$$T_2 = T_1(a) + b(f_s)(R)(W_b + W_1)$$

Where: **T_2** = Tension at the turn exit, lb. (N)

T_1 = Tension at the turn entrance in lb. (N)

a = Turn factor (see below)

b = Turn factor (see below)

f_s = Coefficient of friction between belt and belt supports

R = Radius of turn to the tension link*, feet (M)

W_b = weight of belt, lb/linear ft. (kg/M)

W_1 = weight of product load, lb/linear ft. (kg/M)

If using Metric units (kg, M, etc.) multiply resultant tension x 9.8 to convert to Newtons.

*Refer to Product Technical Bulletin on specific belt; R is usually the radius to the outside edge of the belt.

Turn Factors

Turn factors a and b can be calculated from the following formulas.

$$a = e^{\theta fr}$$

$$b = (a - 1)/f_r$$

Where: **e** = 2.718 (Napierian log base)

θ = Angle of turn in radians (degrees/57.3)

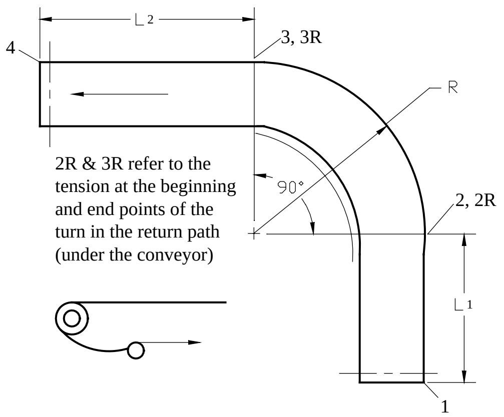
fr = Coefficient of friction for the inside turn rail surface

Turn Factors

Degree of Turn	Inside Turn Rail Coefficient of Friction, fr					
	0.15		0.20		0.25	
	a	b	a	b	a	b
10	1.03	0.20	1.04	0.20	1.04	0.16
15	1.04	0.27	1.05	0.27	1.07	0.27
20	1.05	0.33	1.07	0.35	1.09	0.36
30	1.08	0.53	1.11	0.55	1.14	0.56
40	1.11	0.73	1.15	0.75	1.19	0.77
50	1.13	0.87	1.17	0.85	1.22	0.88
60	1.14	0.93	1.19	0.95	1.24	0.96
70	1.20	1.33	1.28	1.38	1.36	1.43
80	1.23	1.53	1.32	1.61	1.42	1.67
90	1.27	1.80	1.37	1.85	1.48	1.92
100	1.30	2.00	1.42	2.09	1.55	2.19
110	1.33	2.20	1.47	2.34	1.62	2.46
120	1.37	2.46	1.52	2.60	1.69	2.76
130	1.41	2.73	1.57	2.87	1.76	3.05
140	1.44	2.93	1.63	3.15	1.84	3.36
150	1.48	3.21	1.69	3.44	1.92	3.70
160	1.52	3.47	1.75	3.75	2.02	4.08
170	1.56	3.73	1.81	4.06	2.10	4.40
180	1.60	4.00	1.88	4.38	2.19	4.76
190	1.64	4.27	1.94	4.70	2.28	5.12
200	1.69	4.60	2.01	5.05	2.39	5.66

Example: 90-degree Turn Conveyor

Assume belt is driven on the discharge end and employs a catenary sag take-up. The recommended length of the catenary sag is 18 inches (450 mm); therefore the initial tension is equal to:



$$\boxed{T_i = 1.5 \times W_b}$$

The tension at each point along the conveyor is calculated as follows:

$$T_{3R} = T_i + L_2\,(f_r)(W_b)$$

$$T_{2R} = a(T_{3R}) + b(f_r)(R)(W_b)$$

$$T_1 = T_{2R} + L_1(f_r)(W_b)$$

$$T_2 = T_1 + L_1(f_s)(W_b + W_1)$$

$$T_3 = a(T_2) + b(f_s)(R)(W_b + W_1)$$

$$T_4 = T_3 + L_2(f_s)(W_b + W_1)$$

T_4 is the belt tension in lb. (N) at the drive sprockets.

Turns—Other Options

In the first layout calculations, it is clear that fixed rail turns dramatically increase belt tension. Note that in the formula for turn factor formula, “a” is a multiplier of the tension entering the turn (T_1) and that “a” increases with the angle of turn and the coefficient of friction of the inside rail. In planning the layout, try to minimize the number and angle of the turns in order to keep the belt tension down. Keep the layout as simple as possible.

There are other options to fixed rail turns. For instance, if a full diameter, free-rotating turn wheel replaces the fixed inside rail, the tension gain will be considerably reduced. Powering this wheel will offer an even greater improvement. The inside belt support can be incorporated into the wheel to increase its efficiency still further. The following cases illustrate the most practical possibilities. The tension formula is revised accordingly, showing the effect of exit belt tension.

1) **Idler Inside Rail/Stationary Supports (see Figure 42)**

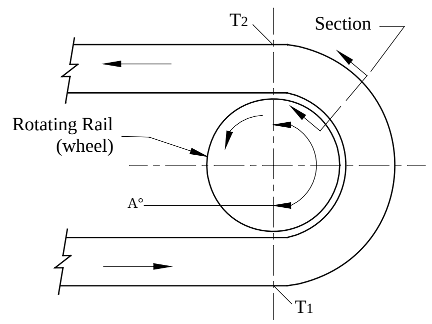
$$T_2 = T_1 + f_s(R)(W_b + W_1)(\theta)$$

Where: θ = A°/ 57.3

(2) **Drive Inside Rail/Stationary Supports (see Figure 42)**

$$T_2 = T_1 + b(f_s)(R)(W_b + W_1)/a$$

If the wheel is driven, a higher f_r is of some benefit. Factors a and b will be increased, producing a lower T_2.



[Figure 42]

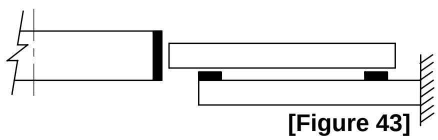
(3) **Idler Inside Rail and Inside Support/Stationary Outside Support (see Figure 43)**

$$T_2 = [T_1 + f_s(R)(W_b + W_1)/2](\theta)$$

Where: θ = A°/ 57.3

(4) **Drive Inside Rail and Inside Support/Stationary Outside Support (see Figure 44)**

$$T_2 = [T_1 + f_s(R)(W_b + W_1)(f_r - f_s)/2]/a$$

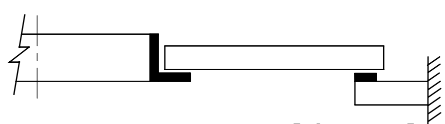


[Figure 43]

(5) **Idler Inside Rail and Supports (see Figure 45)**

$$T_2 = T_1$$

Add bearing friction and initial force to overcome inertia.



[Figure 44]

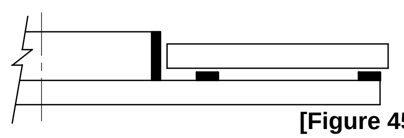
(6) **Drive Inside Rail and Supports (see Figure 45)**

$$T_2 = T_1 - b(f_s)(R)(W_b + W_1)/a$$

(7) **Edge Drives**

(see Product Technical Bulletin “052 Edge Drives”)

$$T_2 = [T_1 + f_s(R)(W_b + W_1)(f_r - f_s)/2]/a$$



[Figure 45]

Comments:

The benefits afforded by formula #4 can also be provided by Edge Drive Units without the space consumption of a full diameter turn wheel on the inside of a turn. A high friction urethane lugged chain, operating in a horizontal plane, powers the inside edge of the belt through the turn.

Turn wheels, which include the belt support bed such as in options 5 and 6 above, can be impractical for wide belts. The transition between straight runs and the turns can be a problem. Option 2 is a more practical choice.

Illustration of Turn Option Benefits:

If we consider a typical case with the following given factors;

$T_1 = 100$

$f_s = 0.2$

$f_r = 0.2$

$R = 6$

$W_b + W_1 = 10$

Angle A = 180°($\theta = 3.14$)

a = 1.88

b = 4.38

For a fixed rail turn, $T_2 = 241$

Applying the same values to the given options, you can see the effect on T_2.

Option (1). $T_2 = 138$

Option (2) $T_2 = 128$

Option (3) $T_2 = 333$

Option (4) $T_2 = 50$

Option (5) $T_2 = 100$

Option (6) $T_2 = 72$

Option (7) $T_2 = 54$

All of the turn wheel options offer some improvement over the fixed rail design. If, in analyzing your layout, the tensions calculated begin to approach or exceed the limitations for the belt to be used, consider the most appropriate option. The alternative is to divide the layout into several independent conveyor sections.

Drive Calculations

Torque at Drive Shaft

The running torque at the drive can be determined for the following formula. This makes no allowance for start-up under load.

$$TQ = T \times PD/2$$

Where:

TQ = Torque, in inch-pounds (N/m)

T = Total Belt Tension, lb. (N)

PD = Pitch Diameter of drive sprockets, in. (m)

Example:

The running torque required at the conveyor drive in the previous example, where the belt tension (T) equals 387.1 lb/foot of width, belt width equals 48 inches and selected sprocket has a pitch diameter of 6.563 inches

TQ = (387.1)(48/12) x 6.563/2 = 5081 inch-pounds of torque

Horsepower Requirements

The suggested horsepower is based on the formula below. A safety factor should be used to allow for transmission losses, start up, loads, etc.

$$HP = T \times S/33{,}000$$

Where: HP = Horsepower

T = Total Belt Tension, lb.

S = Belt speed, ft./min.

Example:

To calculate the horsepower required to drive the conveyor in the previous example, where the belt tension (T) equals 387.1 lb/foot of width, belt width equals 48 inches and belt speed (S) is 50 fpm, would be:

HP = (387.1)(48/12) x 50 / 33,000 = 2.35 horsepower

Drive Shaft Calculations

Shaft Diameter for Combined Torsional & Bending Load

To determine the recommended minimum diameter of the drive shaft, use the following formula. More accurate results are obtained by using the trial and error method.

$$\boxed{D = B \times \{5.1/P \times [(C_b \times M)^2 + (C_t \times Tq)^2]^{1/2}\}^{1/3}}$$

Where:

D = Recommended minimum shaft diameter (inches)

B = Constant; use 1 for solid shafts; or (1 / [1 – K4])1/3 for hollow shafts, where K = (shaft ID/shaft OD)

P = Constant; use 6000 for a shaft with keyway or 8000 for shafts without keyways

C_b = Service Factor in Bending—See table

C_t = Service Factor in Torsion—See table

Tq = Torque (inch-pounds)

M = $(W_r \times L)/8$

W_r = $(R^2 + T^2)^{1/2}$

R = Weight of [Shaft + Sprockets + One Linear Foot of Belt + Load/Linear Foot) (lb.)

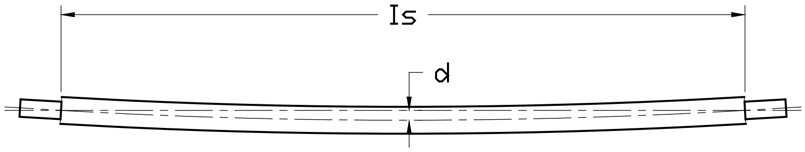
L = Length of Shaft (in.)

Service Factors

C_b	C_t	Type Load
1.5	1.0	gradually applied on steady load
1.5–2.0	1.0–1.5	suddenly applied minor shock load *
2.0–3.0	1.5–3.0	suddenly applied heavy shock load
* most commonly used		

Deflection of Drive Shaft

The maximum recommended deflection of the drive shaft is 0.1 inch (2.5 mm).



When only two outer bearings are used:

$$D = \frac{5 \text{ x } F_1 \text{ x } l_s^3}{384 \text{ x } E \text{ x } I}$$

When a center bearing is used:

$$D = \frac{F_1 \text{ x } l_s^3}{2960 \text{ x } E \text{ x } I}$$

Where:

D = Deflection, inch (mm)

F_1 = Shearing force on the shaft, lbf (N)

$F_1 = \sqrt{T_2 + (w_s \text{ x } l_1 \text{ x } c)^2}$

T = Total Belt Tension at the shaft, lbf (N)

w_s = Weight of shaft, lb/ft (kg/m)

l_1 = Shaft length, ft (m)

l_s = distance between shaft bearings, inch (mm)

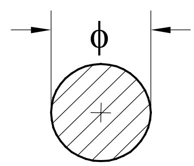
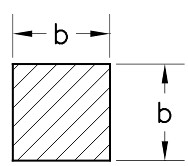
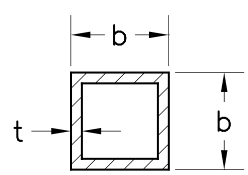
E = Modulus of elasticity of shaft material

Steel: $E = 2.95 \text{x} 10^7$ lbf/inch^2

$= 2.1 \text{x} 10^5$ N/mm^2

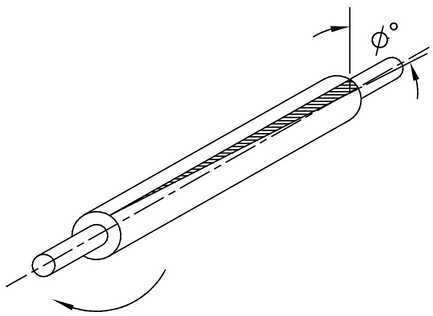
I = Moment of Inertia of drive shaft, inch^4 (mm^4)

C = Force conversion factor
Imperial: 1.0
Metric: 9.8

Shaft Dimension in. (mm)	Moment of Inertia
	$I = \pi \text{x} \Phi^4/64$
	$I = b^4/12$
	$I = [b^4-(b-2t)^4]/12$

Torsion of Drive Shaft

The maximum recommended torsion of the drive shaft is based on the following formulas:
($0.5 \times l_s$ (inch)/39.37 – Imperial) or ($0.5 \times l_s$ (mm)/1000 – Metric).



$$\Phi° = \frac{180 \times T \times d_o \times l_s}{2\pi \times G \times I_t}$$

Where:

$\Phi°$ = Torsion angle

T = Total Belt Tension at the shaft, lbf (N)

d_o = Pitch diameter of drive sprockets, inch (mm)

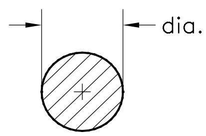
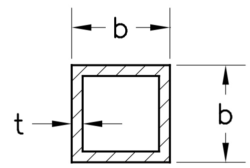
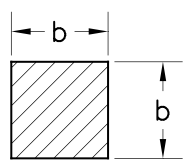
l_s = distance between shaft bearings, inch (mm)

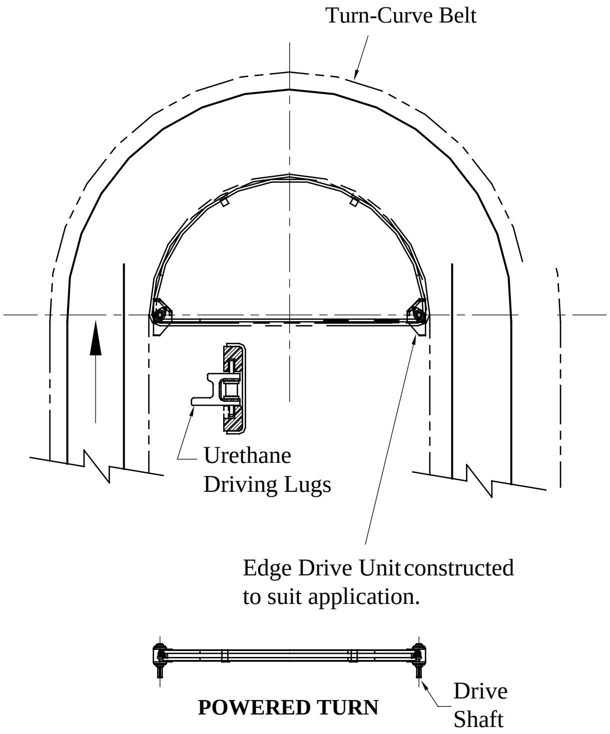
G = Modulus of shear of shaft material

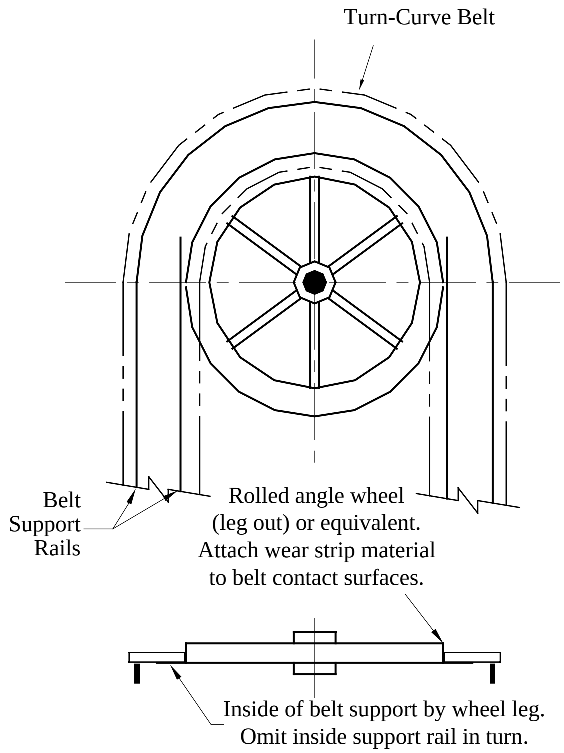
Steel: $G = 11.6 \times 10^6$ lbf/inch^2

$= 81.6 \times 10^3$ N/mm^2

I_t = Inertia force for drive shaft, inch^4 (mm^4)

Shaft Dimension in. (mm)	Inertial Force
	$I_t = 0.1 \times (dia)^4$
	$I_t = t \times b^3$
	$I_t \approx 0.141 \times b^4$





EDGE DRIVE UNIT **IDLER OR DRIVEN TURN WHEEL**

Support Structure

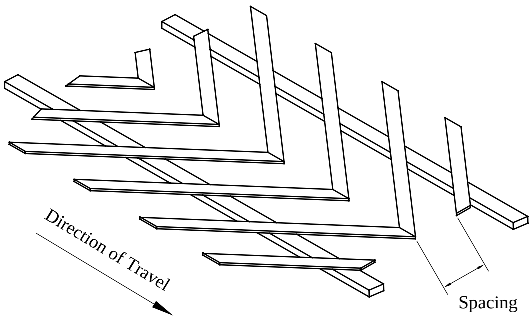
For optimum operation, it is of great importance that the belt has the correct support structure. The conveyor has to be level for even wear on the belt and support structure. An oblique conveyor will reduce the life of the belt and support structure due to expedient wear. The design of the support structure, i.e., the choice and placement of the wear strips, must consider the following factors:

- Belt type
- Load
- Horizontal or inclining conveyor
- Temperature conditions

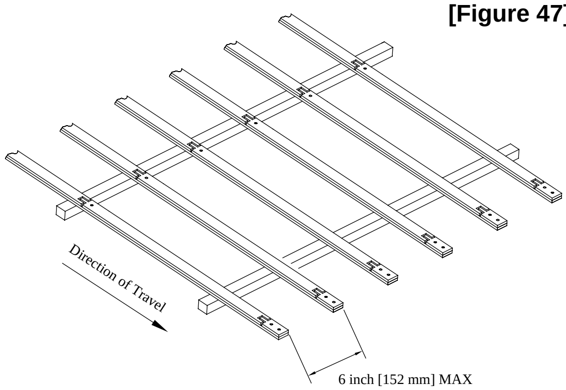
Support structure should extend beyond belt edge for best support.

Typical Support Structures:

[Figure 46]



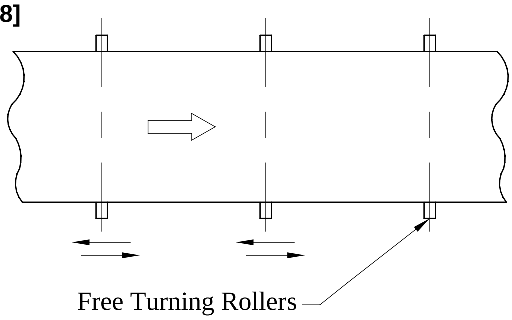
[Figure 47]



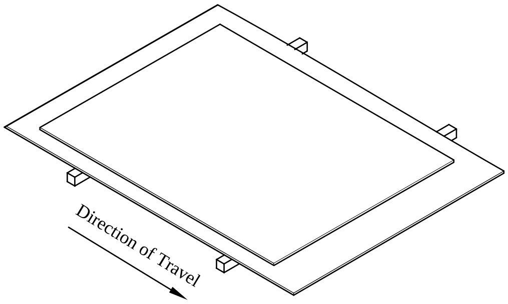
Herringbone rails: Recommended. (Figure 46). Flat wear strips in a "V" configuration with the point of the "V" pointing in the direction of travel. Low friction wear strip material preferred to minimize belt wear. Recommended spacing between rails of 4–12" depending on belt type, load, and other factors. This configuration distributes the wear over the entire belt width.

Longitudinal Rails: (Figure 47) Flat wear strips the full length of the conveyor, parallel to each other and perpendicular to the terminal shafts. Low friction wear strip material preferred to minimize belt wear. Recommended spacing between rails of 4–12" depending on belt type, load, and other factors. This configuration does not distribute wear over the full width of the belt.

[Figure 48]



[Figure 49]

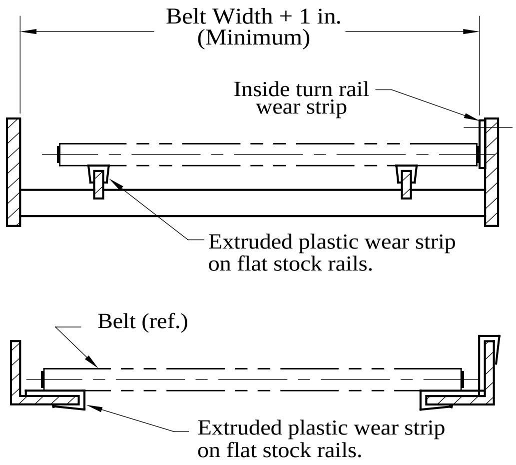


Free Turning Rollers: Recommended. (Figure 48) Roller supports minimize wear on the belt, reduce belt tension, and aid in the tracking of friction-driven belts.

Slider bed: (Figure 49) A slider rail bed of low friction material will, in most cases, afford the best means of providing belt support as it fully supports the belt.

Many excellent **Belt Support Materials** are available. The most commonly used Ultra High Molecular Weight Polyethylene (UHMWPE). UHMWPE is available in numerous shapes and sizes. Special extruded shapes are available in continuous coil lengths for ease of assembly.

For applications that experience **Temperature Fluctuations**, the wear strips should be attached in such a manner as to allow expansion and contraction with temperature. In other words, they should be secured at one end only.



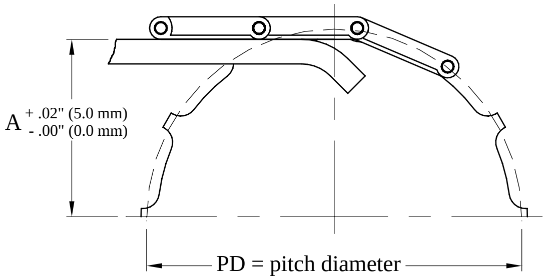
TYPICAL SUPPORT RAIL
AND RETAINING RAIL CONSTRUCTION

As a rule, support rails are required at a maximum 18 inches apart on the load side and 24 inches apart on the return side. Rollers may also be used. For light loads, the support rails may be placed further apart. Contact us for your particular application.

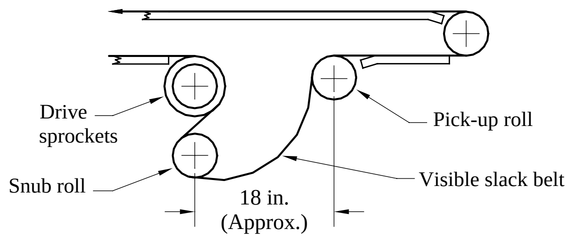
Slider Bed: A slider rail bed of low friction material will, in most cases, afford the best means of providing belt support. Many excellent materials are available, such as Ultra High Molecular Weight Polyethylene (UHM-WPE). Special extruded shapes in continuous coil lengths are available for ease of assembly. In most cases, a roller bed of good quality can be used in straight runs, but is generally impractical for turns. Inclines and declines in a belt path should be located in the straight runs and several feet from any turn wherever possible. On inclines where turns are required, accepted practice is to use hold down rails over the belt edges.

Free Turning Rollers: Recommended. Most oven bands are supported by free turning, horizontally adjustable rollers with externally mounted bearings. Locate bearings outside the oven to provide for adjustments when the band is at baking temperature. Roller supports minimize wear on the band, account for lower tension to overcome friction in the system, and aid in band tracking. For new installations, align all roller supports perpendicular to the oven centerline. If a replacement band is being installed, there is no need to align supports perpendicular to the centerline unless the previous band was severely mistracking. The new band will likely assume the same general path of the old band and tracking adjustments can start from that point.

Wearstrip Placement

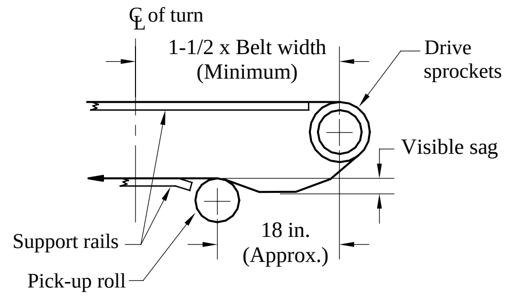


A = ½ x PD – ½ Belt Thickness



RETURN RUN OR CHARGE END DRIVE

- Wear strip placement varies with belts. Reference the Technical Bulletin on the desired belt for more information.
- The values stated are only a guideline; they do not take into account the influence of speed. At speeds above 75 ft/min (23 m/min)it's recommended to increase distance A and shortening the wear strips as much as one-belt pitch in length.



DISCHARGE TERMINAL DRIVE

DRIVE

Belts are designed for either *positive* or *friction* drives.

*Positive drive*s propel the belts by means of sprockets which are designed to engage the fully extended belt. For this reason, the drive sprocket cannot be located immediately adjacent to a turn exit where the links are partially closed. A straight run not less than 1-1/2 times the belt width must be provided to allow the pitch to return to the full extended position. The location of the drive is important and can be critical. Complicated layouts, long conveyors, and heavy loading will certainly require multiple drives. To avoid excessive belt stress in case of a malfunction, a drive safety device such as a torque limiter is recommended.

Friction Drives works by frictional contact between the drive drum and the belt. Typical belts used in this application can be balanced weave, compound balanced weave, and flat wire. Terminal drums must be large enough to ensure good contact and maximum flexibility as the belt travels around the drum. These drums are typically several inches wider than the belt. Each drum must be level, parallel to each other, and perpendicular to the centerline of the conveyor. They must also be clean with no product build-up on the surface. Sometimes drums are lagged with urethane or other high friction material to increase friction between belt and drum. Tracking is important to stabilize belt waver.

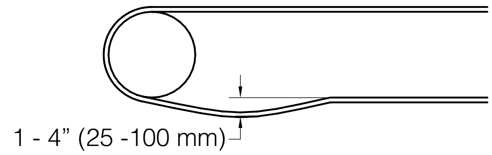
Adjustment Of Belt Length (Take-Up)

Automatic adjustment of the belt length is obtained by providing a catenary sag take-up on the belt return path. Use of a catenary sag (length of unsupported belt) following the sprockets ensures the automatic adjustment of belt length. The catenary sag will serve to absorb the elastic expansion under load, thermal expansion, and the long-term elongation of the belt.

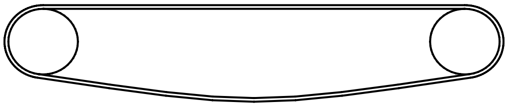
For standard applications, a catenary take-up is placed immediately following the drive sprockets. For applications at elevated temperatures, the expansion of the belt should be calculated first to determine whether sufficient space is available.

Belts running in one direction (uni-directional)

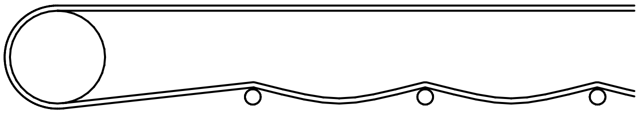
- For conveyors operating within uniform operating conditions catenary sag is only necessary on the first 18 inches (457 mm) after the drive sprockets.



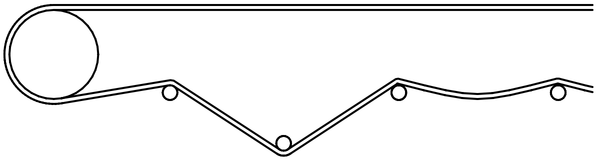
- For short conveyors under 6 feet (1.8 m) the belt can hang freely between the drive and idler terminals.



- For long conveyors over 6 feet (1.8 m) with large temperature changes (50°F [10°C] or greater) the catenary sag should be distributed over longer sections; i.e., support the belt in the return with rollers to distribute the sag over the length of the belt.



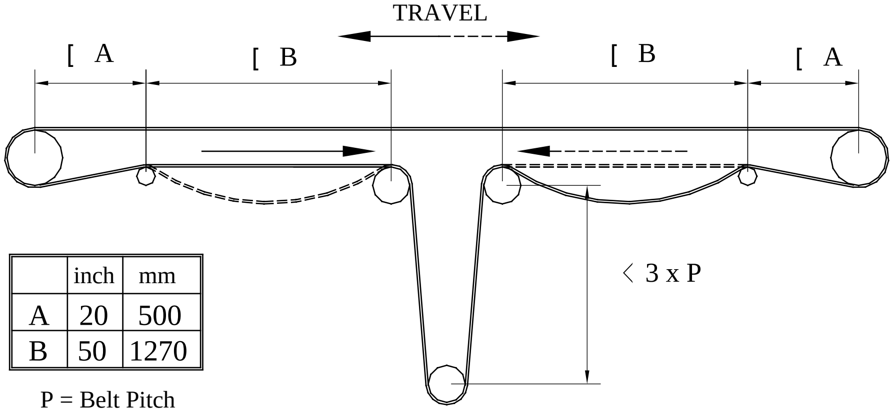
- Alternatively, for long belts and with high temperature variations (50°F [10°C] or greater) a roller can be mounted free hanging in the return path allowing its weight to take-up the slack belt.



Belt Running In Both Directions (Bi-Directional)

Belt take-up can be a source of excessive belt stress. For this reason, where conditions permit, we recommend the use of an unsupported span of belt approximately 18 inches long immediately following the drive, in which the belt loop hangs free with sag of 2 inches or more. If a fixed mechanical take-up is used to facilitate control of excess slack, sag must be visible in the free loop of belt following the drive sprockets. Where other take-up arrangements are necessary, a minimum weight, gravity controlled type is recommended.

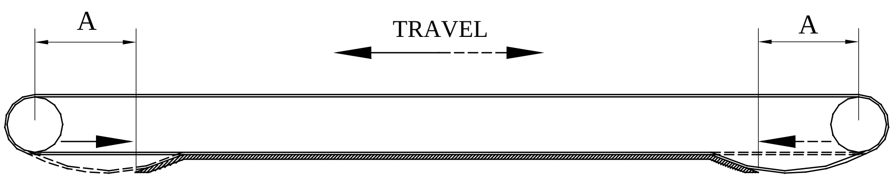
Adjustment of belt running in both directions can be made with either pull/pull or push/pull systems.



	inch	mm
A	20	500
B	50	1270

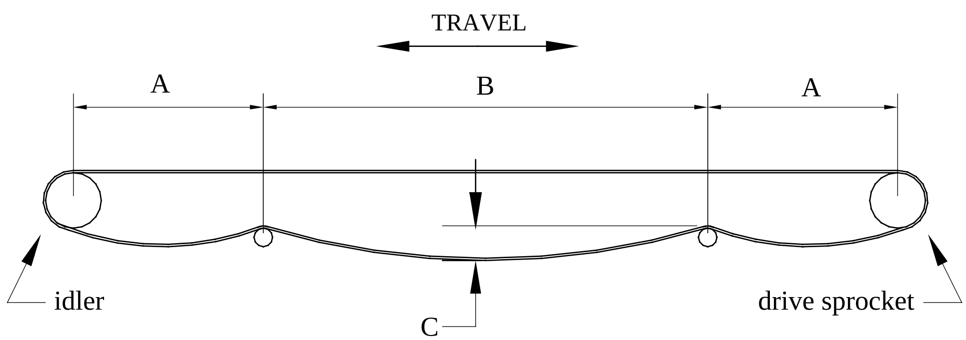
P = Belt Pitch

This pull/pull system requires only one reversible motor.



	inch	mm
A	24	600

This conveyor requires 2 motors (one each end). This construction requires that the motors can be disconnected.



	inch	mm
A	18-24	450-600
B	100-120	2540-3048
C	0.5-1.5	13-38

This pull/pull system requires only one pole-changeable motor. The system is only suitable for short conveyors with small loads. For correct working of the system the load on the belt in the catenary should be approximately 1.3 times higher than the load in the upper belt path. See calculations for catenary sag. The load from the catenary sag must be considered when dimensioning the drive shaft.

Sprockets

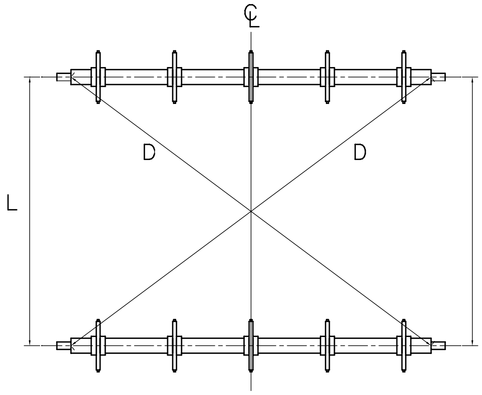
- The basic requirements for belt conveyors are to ensure an evenly distributed load on the belt and support structure, that the belt has the necessary strength, and the placement of the sprockets is correct.
- When constructing a belt conveyor, it is important to allow for adjustment. By using adjustable bearings at the drive end as well as the idler end, it is easy to allow for adjustment.

Note: The use of setscrews in plastic sprockets may cause breaks in the sprockets if the setscrews are over tightened.

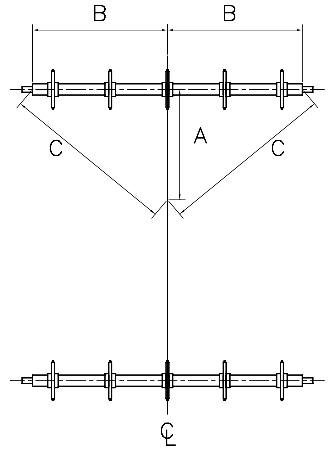
Alignment Of Sprockets

Shafts

Proper alignment of the sprocket shafts is critical for smooth operation. The conveyor must be absolutely level and the shafts must be perpendicular to the centerline of the conveyor. The following methods are acceptable for aligning the drive and idle shafts.



The distance L must be the same at both sides and the length of the two diagonals D must be equal.

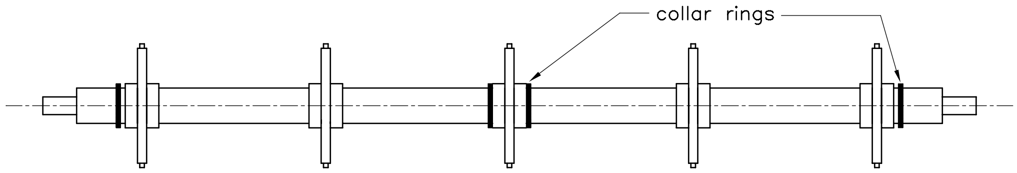


This method is normally used on long conveyors. You choose the measure of A and B as you like. The point is that the size of the two C–meausres are equal.
Note: The centerline must be very accurate. The alignment of sprockets must be performed on both the drive and idle shafts.

Mounting Of Sprockets

Sprockets with a Round Bore

Round bore sprockets are recommended for conveyors with constant ambient temperature and for conveyors with light loading. Sprockets with keyways are used at the drive end and, when sprockets are positioned, the center sprocket is fixed axially with either set screws or collar rings.



Sprockets with a Square Bore

Square bore sprockets are recommended for conveyors with a working temperature different from the surrounding temperature and for conveyors with high tension. Mount the sprockets on the shaft and fix the center sprockets with collar rings to prevent axial movement and ensure even pull.

Determining Number Of Sprockets

The number of sprockets for the belt depends on the load on the belt and the temperature under which the belt is operating.

Factory recommends a minimum of one (1) sprocket per 6 inches (152.4 mm) of belt width for plastic and flatwire type belts. Omni-Grid® and Omni-Pro® Belts typically require only two sprockets per driveshaft.

- For Flat wire belts positively driven with sprockets or a waffle roll (a continuous belt-width toothed member, available via special order), overall diameters will range from 4-1/8 inches (104.8 mm) to 14-11/16 (373.1 mm). The quantity is determined for belt tension, but there is a maximum spacing of 6 inches (152 mm).

  Location—Sprocket should be place in odd numbered openings, ensuring outside sprockets are located in the third openings from each belt edge. This assists the belt in resisting fatigue fractures by providing two load-carrying legs.

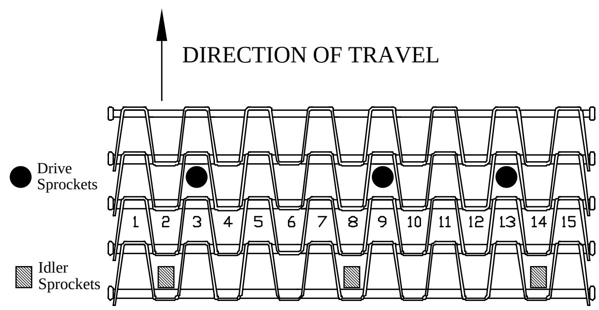
  Hubs—Must be oriented in the same direction to keep teeth perfectly lined up and distribute stress evenly across the belt width. Idlers should be placed in even numbered openings, ensuring that outside sprockets are located in the second openings from each belt edge.

The required number of sprockets for the belt is determined as follows:

Minimum number of sprockets per shaft is calculated by dividing the belt width by sum of maximum sprocket spacing and sprocket width. Round calculated number up to nearest whole number.

Example for a metal belt with a width of 26 inches:
Minimum number of sprockets per shaft = belt width/(max spacing number of sprockets)
26 inch wide belt (660.4 mm)/6 inch = 4.33 sprockets*

- Always round up to the next highest whole number
- A minimum of 5 sprockets is required for both the drive and idler shafts.



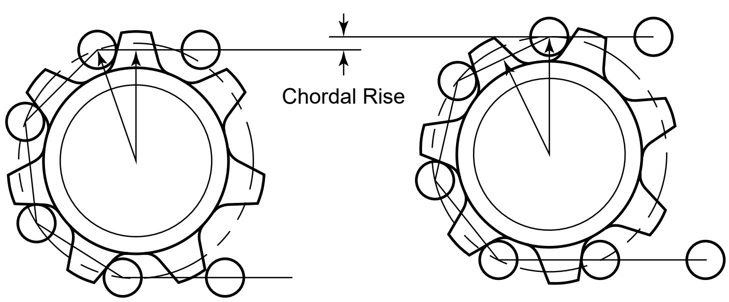
Metric
Number of sprockets = 660.4 mm/152.4 mm = 4.33 sprockets

- Always round up to the next highest whole number
- A minimum of 5 sprockets is required for both the drive and idler shafts.

  Cleatrac® belts are an exception to this rule. Please see Technical Product Bulletin “033 Cleatrac® Belt and Drive System” for Cleatrac® sprocket calculations.

Chordal Action

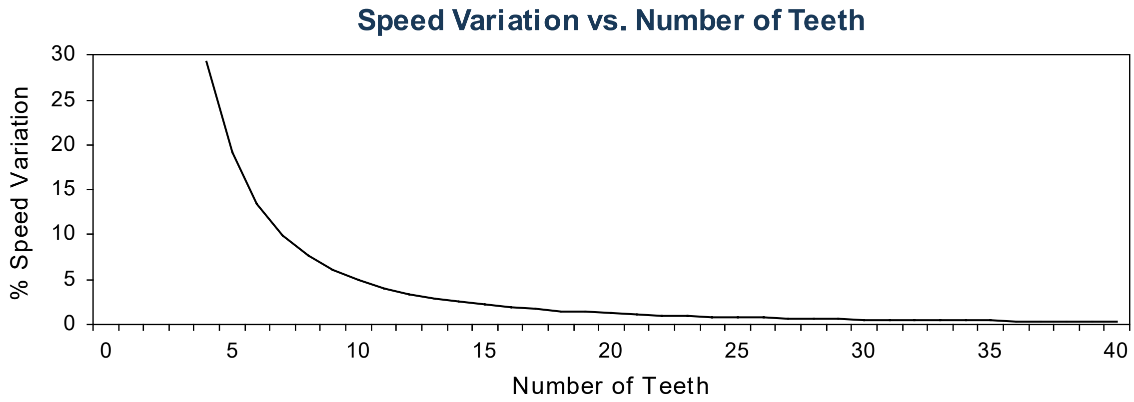
Sprocket driven conveyor belts will experience variation in linear speed as the sprocket drives the belt. Because belts hinge or rotate about the set pitch of the belt they can only bend about the rods or pitch points. This creates a variation in the radius of engagement between the tangent and chord positions. This phenomenon is referred to as chordal action.



When the sprockets are rotating at a constant speed, the belt speed is not steady due to change in engagement radius (chordal rise). Chordal action varies based on the number of teeth on the sprockets.

This variation in speed is calculated as following:

Chordal Action (ratio of speed change) = 1 – cos(180°/N)
Where N equals the number of teeth on the sprocket



When the values for chordal action are graphed it becomes clear that, as the number of teeth are increased on the sprocket, the resulting chordal action or variation in belt speed is reduced dramatically. As noted in the graph above, the speed variation drops to around 4% when an 11-tooth sprocket is used. A speed variation of this amount is seldom noticeable, which is why it's recommended that the selected sprocket for any of these belt types have no less than 11 sprocket teeth.

Sprocket Material Selection

It's recommended to use plastic (UHMWPE) sprockets on the majority of conveyor applications. Plastic sprockets are recommended since they wear extremely well and operate quieter than metal sprockets. However, in applications operating above 150 °F (65°C), metal sprockets are preferred as the plastic will deform and wear faster at elevated temperatures.

Metal sprockets are recommended for applications where elevated temperatures are present, when the product being conveyed is abrasive, or when the belt is being operated at very high tensions. Stainless steel is the material of choice for wet or food applications. For dry applications or the general conveying of non-food or packaged products, cast iron or fully machined plain steel sprockets can be used.

Filler Rolls

Factory recommendation is that the filler rolls be used on shafts where only drive sprockets are used; i.e., Omni-Pro®, Advantage™ , and Omni-Grid®. The filler (or support) rolls are required to keep the belt from deflecting across its width. The maximum diameter for the filler rolls depends on the size of the sprockets being used. The diameter required for the filler rolls can be calculated knowing the pitch diameter of the chosen sprockets.

Max. Dia. = PD x cosine(180/#) – MT

Max. Dia.	= maximum diameter for the filler rolls
PD	= Pitch diameter of sprocket
#	= Number of teeth on the sprocket
MT	= Mesh (Belt) thickness

Mesh thickness for Omni-Pro® and Omni-Grid® belts can be estimated by adding the cross rod diameter plus two times the diameter of the mesh overlay.

Catenary Sag (Vertical Distance from Top to Bottom of Belt Arch)

A free hanging conveyor belt will form a belt arch between two supports. Knowing the amount of belt in the arch and load from the belt is important in determining both load at the sprockets and the required belt length.

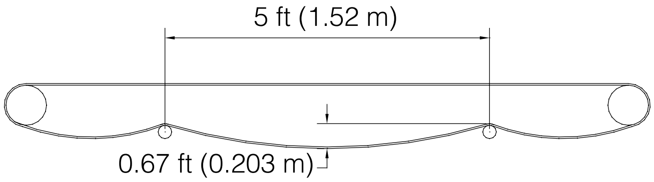
L	= Length of belt in arch–ft (m)
F	= Tension from the belt arch, lbf/ft (N/m) of belt width
D	= Distance between supports–ft (m)
w	= Belt weight–lb/ft^2 (kg/m^2)
S	= Catenary sag–ft (m)
C	= Force conversion factor Imperial: 1.0 Metric: 9.8

Length of belt arch

$$L \approx [(2.66 \times S^2) / D] + D$$

Example: Catenary Sag

W: 1.5 lb/ft^2 (7.5 kg/m^2)



Length of belt arch:

$$L \approx [(2.66 \times S^2) / D] + D$$

Imperial: $L = [(2.66 \times .67^2) / 5] + 5 = 5.24 \text{ ft}$

Metric: $L = [(2.66 \times .203^2) / 1.52] + 1.52 = 1.6 \text{ m}$

Load from belt arch:

$$F = (D^2 \times W \times C) / (8 \times S)$$

Imperial: $F = (5^2 \times 1.5 \times 1) / (8 \times .67) = 7$ **lbf/ft**

Metric: $F = (1.52^2 \times 7.5 \times 9.8) / (8 \times .203) = 104.6$ **N/m**

Expansion/Contraction Of The Belt

Expansion/contraction of the belt may occur at operating conditions where the belt is exposed to changes in temperature. Such changes in the belt width and belt length must be taken into consideration when the conveyor is constructed.

ΔL = length or width expansion–inch (mm)
L = length or width of belt at temperature T_1–ft (m)
T_2 = working temperature °F (°C)
T_1 = surrounding temperature °F (°C)
ρ = expansion coefficient–see table

Expansion coefficients (ρ):

Belt Material	inch /(ft x °F)	mm/(m x °C)
Acetal	6.00E-4	0.12
High Carbon Steel	0.76E-4	0.012
Stainless Steel T304/T316	1.19E-4	0.018

The change in dimensions can be calculated as follows:

$$\boxed{\Delta L = L \times \rho \times (T_2 - T_1)}$$

Example: Expansion/Contraction of the Belt

An example belt application would be: A belt 3 feet (0.91 m) wide, conveyor length 25 ft (7.62 m), ambient temperature +72°F (+22°C), operating temperature +150°F (+65°C)

Expansion in width:

Imperial: $\Delta L = 3 \times 0.001 \times (150–72)$
$\Delta L = 0.23$ inch

Metric: $\Delta L = 0.91 \times 0.15 \times (65–22)$
$\Delta L = 5.8$ mm

Expansion in length:

Imperial: $\Delta L = 25 \times 0.001 \times (150–72)$
$\Delta L = 1.95$ inch

Metric: $\Delta L = 7.62 \times 0.15 \times (65–22)$
$\Delta L = 49.5$ mm

Friction Coefficients

Friction coefficient between Belt and Wear Strips, Metal Belts

Belt Material	Type of Belt Support	f
Stainless Steel or High Carbon	Free Turning Rollers	0.10
	UHMWPE with clean or packaged product	0.20
	UHMWPE with breaded or flour based product	0.27
	UHMWPE with greasy, fried product	0.30
	UHMWPE with sticky, glazed, sugar based product	0.35
Stainless Steel	Mild Steel with temperatures up to 1000°F (538°C)	0.35
	Stainless Steel (not recommended with metal belts)	0.40
High Carbon	Mild Steel with temperatures up to 1000°F (538°C)	0.40
	Stainless Steel (not recommended with metal belts)	0.35

Friction coefficient between Belt and Wear Strips, Plastic Belts

		Wear Strip Material	
Belt Material	Lubrication	Stainless Steel	Polyethylene
Polypropylene	Dry	0.28	0.15
	Water	0.26	0.13
Polyethylene	Dry	0.16	0.32
	Water	0.14	0.30
Delrin—Acetal	Dry	0.30	0.24
	Water	0.23	0.20
Anti-static/Low Friction—Acetal	Dry	0.24	0.19
	Water	0.17	0.15
Super Low Friction—Acetal	Dry	0.20	0.18
	Water	0.15	0.10

Values will be 0.1 to 0.2 higher at the starting moment.

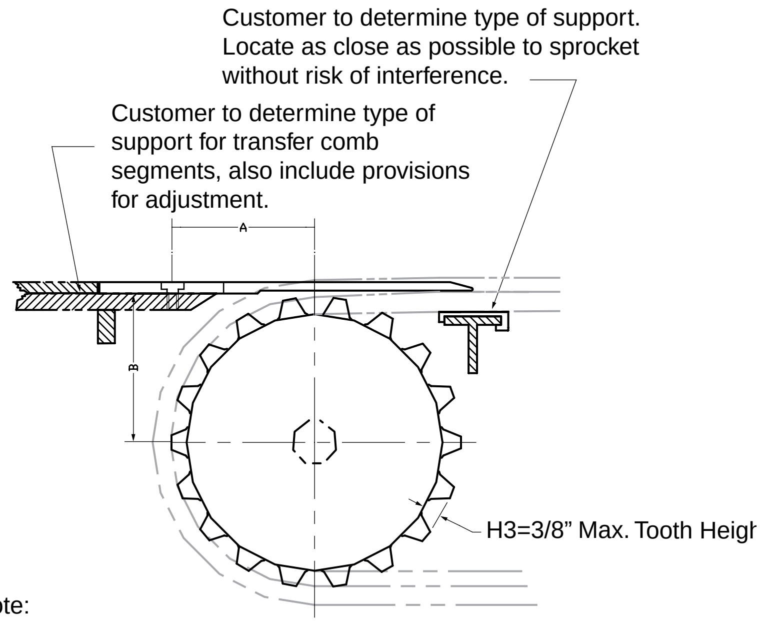
Friction coefficient between Belt and Product

	Product Material				
Belt Material	Lubrication	Glass*	Metal	Plastic	Cardboard
PP	Dry	0.19	0.32	0.17	0.22
	Water	0.17	0.30	0.15	---
PE	Dry	0.10	0.13	0.10	0.15
	Water	0.09	0.11	0.09	---
D-Acetal	Dry	0.18	0.24	0.22	0.27
	Water	0.16	0.21	0.19	---
AS/LF–Acetal	Dry	0.15	0.20	0.18	0.21
	Water	0.12	0.18	0.16	---
SLF-Acetal	Dry	0.12	0.15	0.15	0.19
	Water	0.10	0.14	0.14	---

Values will be 0.1 to 0.2 higher at the starting moment.

* Do not use plastic modules if broken glass comes on the conveyor.

Location of Finger/Transfer Plates



Note:
1. Operate belt and track belt before installing transfer plates.
2. At terminals where the transfer plate is used, sprocket teeth must be reduced to 3/8” overall height for H3 EZ transfer.
3. Return support rails must have a 1” minimum width to insure that pickets do not straddle the rails.
4. Belt has a definite top and bottom and cannot be inverted.

H3 EZ TRANSFER

Sprocket Type	A Dim.		B Dim.	
	in.	mm	in.	mm
#6-18 Tooth	3-1/8 to 3-3/8	79.4 to 85.7	3-5/8	84.1
#8-23 Tooth	3-5/16 to 3-7/16	84.1 to 87.3	4-1/4	108

The diagram shown is to be used as a guide in the placement fingerplates to provide the smoothest possible transfer from the belt to the fingerplate.

The main consideration will be proper clearance for the belt and mounting of the fingerplate because the plates are produced in standard modular sizes. The belt widths must be in intervals coincidental with the finger spacing.

This belt is used with sprockets to insure proper alignment with transfer plate fingers.