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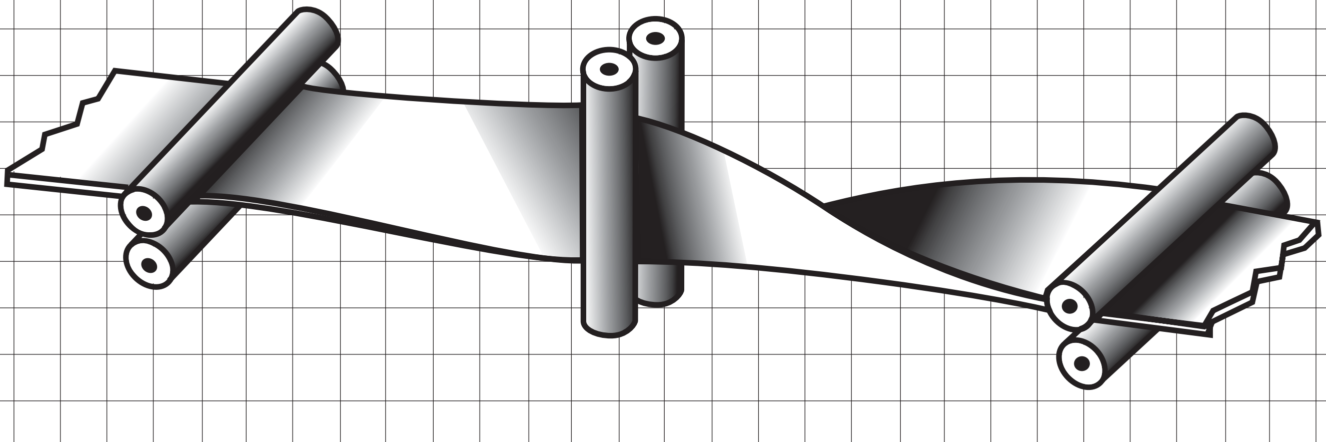
FENNER DUNLOP

CONVEYOR BELTING

Unique Conveyor Problems and Solutions



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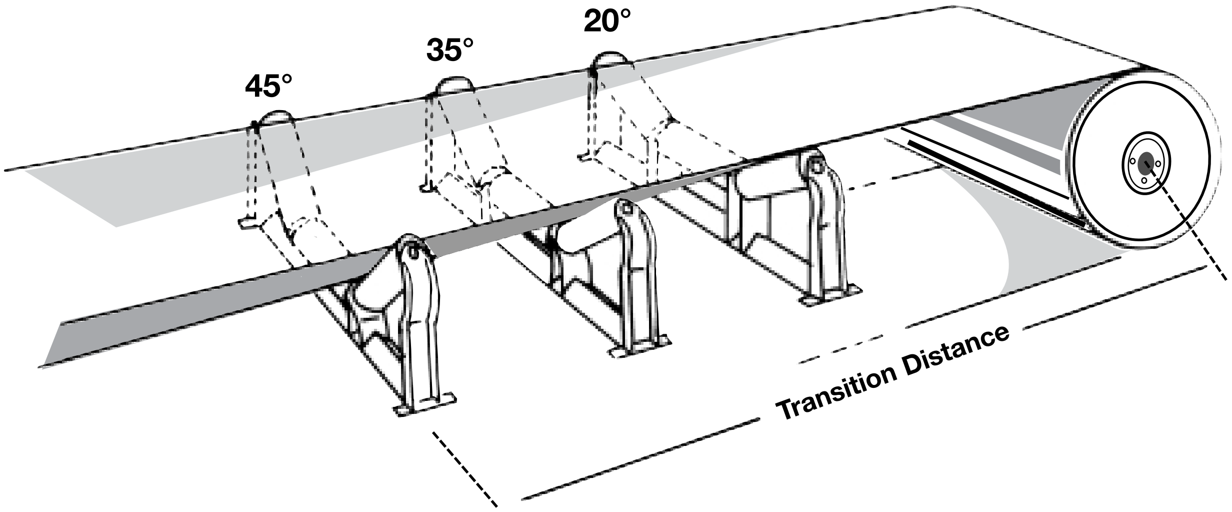
1. TRANSITIONS

The first topic we are going to discuss is transition distances. These distances are measurements from the centerline of a terminal pulley to the first 20° idler, the first 35° idler and the first 45° idler. These distances are applicable from the tail pulley into the loading zone and from the head pulley or tripper discharge pulley back along the line of travel.



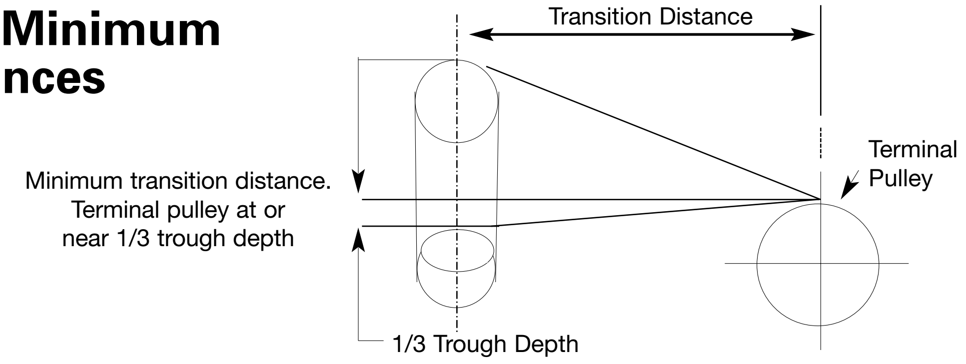
Adequate transition distances are extremely critical with modern technologies use of high modulus, low stretch, conveyor belt carcasses.

As you can see from the picture shown, the belt is flat over a terminal pulley and troughed at various degrees on the conveyor. The distance measured along the edge of the belt in the transition area is greater than the distance measured along the center of the belt to the top of the pulley. With the edge of the belt being stressed in this area, it is critical to maintain proper distances between a terminal pulley and the troughing idlers.

The following tables show the minimum transition distances for fabric belts with loading at 1/3 trough depth, 1/2 trough depth and full trough depth.

Recommended Minimum Transition Distances

1/3 Trough Depth



b=belt width

Idler Angle	% of Rated Tension	Fabric Belts
20°	Over 90	1.2b
	60 to 90	.9b
	Less than 60	.8b
35°	Over 90	2.1b
	60 to 90	1.4b
	Less than 60	1.2b
45°	Over 90	2.6b
	60 to 90	2.0b
	Less than 60	1.6b

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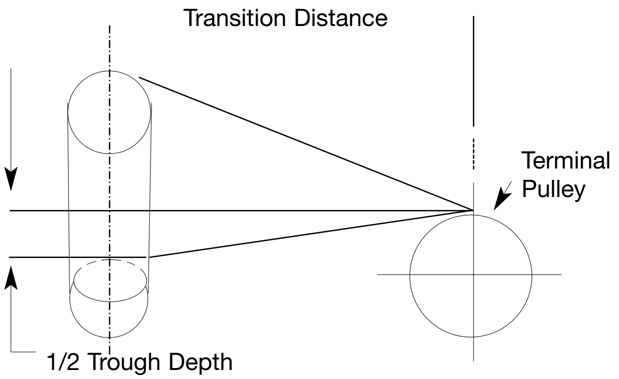
1/2 Trough Depth

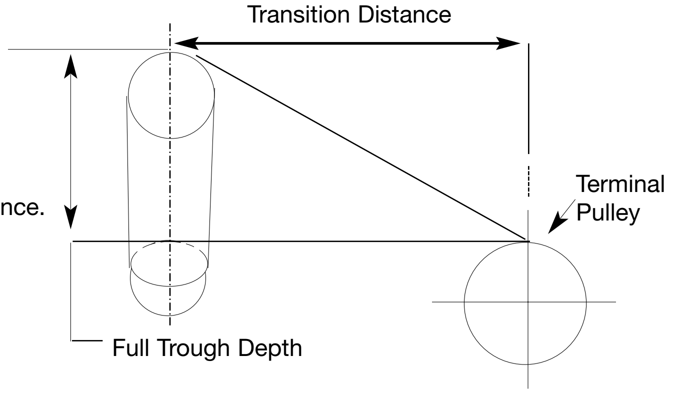


Minimum transition distance.
Terminal pulley at
1/2 trough depth

b=belt width

Idler Angle	% of Rated Tension	Fabric Belts
20°	Over 90	.9b
	60 to 90	.8b
	Less than 60	.6b
35°	Over 90	1.6b
	60 to 90	1.3b
	Less than 60	1.0b
45°	Over 90	2.0b
	60 to 90	1.6b
	Less than 60	1.3b

Full Trough Depth



Minimum transition distance.
Terminal pulley at
full trough depth

b=belt width

Idler Angle	% of Rated Tension	Fabric Belts
20°	Over 90	1.8b
	60 to 90	1.4b
	Less than 60	1.2b
35°	Over 90	3.2b
	60 to 90	2.4b
	Less than 60	2.0b
45°	Over 90	4.0b
	60 to 90	3.0b
	Less than 60	2.4b

Loading depths can be determined by using a straight edge and a tape measure.

When transitions are shorter than suggested minimum, belt damage can occur. At the high tension area, such as the head pulley or tripper discharge, excessive edge tensions will cause adhesions to break along the edge of the belt, and can also rupture the carcass if tensions are severe enough. At a low tension terminal, such as the tail, short transitions will cause a belt to buckle at the bottom of the trough, resulting in splice failure and cover adhesion breakdown in the center of the belt.

CORRECTIVE ACTION

Determine the loading depths by the use of a straight edge and tape measure, mark the frame with the proper transition distances as shown in the preceding charts for the locations of the 1st 20° idler, 1st 35° idler and 1st 45° idler, if 45° troughers are being used.

Removing idlers, drilling all new holes and starting all over again may not be required. Try a simple procedure that in most cases will solve the problem without the drilling of all new holes. Refer to the illustration that we have used in the beginning of this section, as we explain the procedure. Remove the 45° trougher and set it aside. Remove the 35° trougher and place it in the holes where the 45° was located. Remove the 20° trougher and place it in the holes where the 35° was located. Insert an adjustable idler in the holes where the 20° idler was located and set the wing rolls at less than 20°. This procedure works well, providing these idlers in their new locations will give you the required distances.

2. VERTICAL CURVES

Vertical curves come in two different styles, concave and convex. Both types of curves create belt operational problems when the curve is designed with a shorter radius than required.

A. CONCAVE CURVES

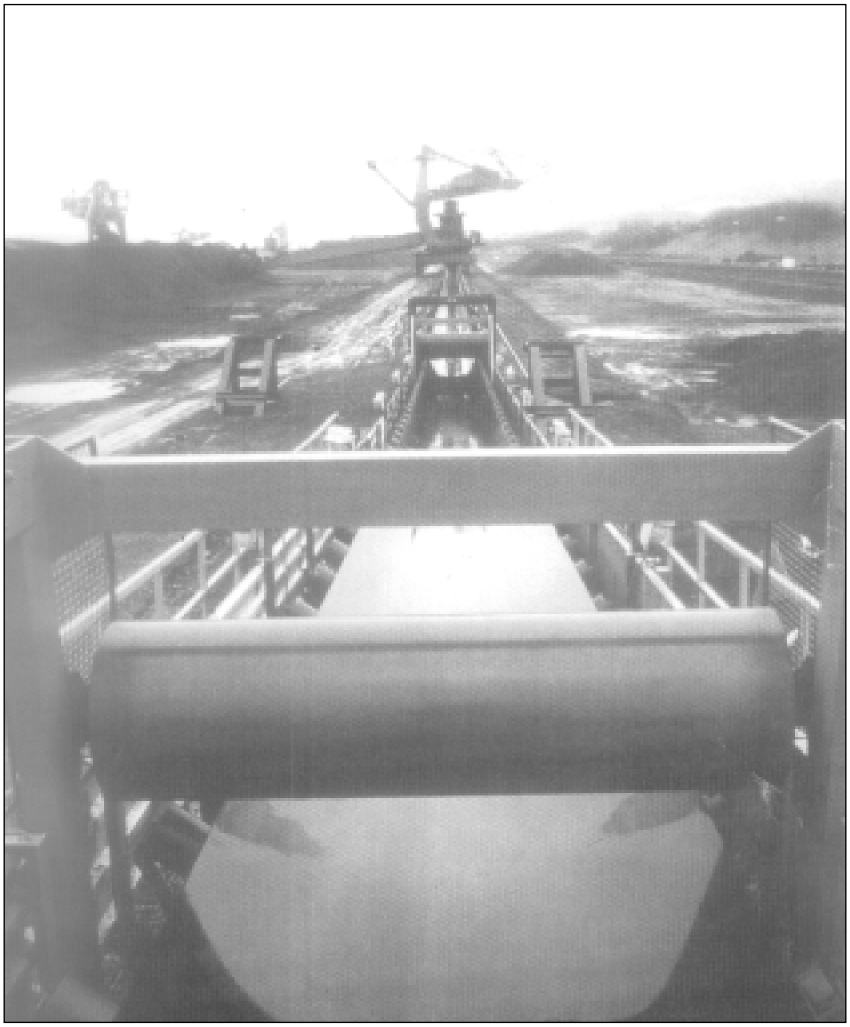
Concave curves, when designed properly, will allow the belt to lay in the idlers when the belt is running empty or full without lifting off of the idlers.

When concave curves are designed with a very short radius, belt lift off is a secondary problem. The stresses that develop in the belt are due to the very short radius and belt failure will happen quickly. The only way to combat the situation is by use of a very low modulus fabric. This fabric is extremely elastic or stretchy. These types of conveyor frames are rare and are usually designed around a low modulus fabric.

The simple concave curve is designed around Radii from around 300 feet to 800 feet. This is the type of curve that may create lift off problems. Lift off results from the curve being designed or built with a radius usually being 100 feet or 200 feet short of ideal. Lift off problems from this type of conveyor can be solved fairly simply.

Concave curve conveyors that have a hard start usually lift off at start up if the belt is empty. A soft start with an extended time delay to full RPM is the first step to keeping the belt in the idlers.

The next step is the installation of a hold down roll in the curve area.



The preceding picture utilizes a hold down roll at the beginning of the curve and another roll in the middle of the curve. The hold down roll is set above the belt, so as not to interfere with product movement. The roll should only touch the belt when lift off occurs with an empty belt.

The last step, if needed, is to select a heavier belt. The increased belt weight will also help keep the belt laying in the idlers. The increase in belt weight will normally not require any change in motor horsepower or take-up weight.

B. CONVEX CURVES

Convex curves create another type of belt problem. A convex curve with a short radius will pull the belt down into the idler gap as the belt goes over the apex of the curve. This will create idler gap failure in the belt resulting in the belt developing a longitudinal slit in the idler junction area. Due to the lines that become visible on the pulley cover of the belt, this type of failure can be spotted early. This line later becomes a crease as the carcass starts to deteriorate. Another way to spot this problem early is to look in the idler junction area. The paint on the idlers in the gap area will wear off due to belt tension in the idler junction area

The apex of the curve, if severe, will cause edge stresses so high that the splice will open up along the belt edge, the belt will suffer from edge delamination and fabric will rupture at the belts edge. A poorly designed curve takes the appearance of a mountain top with one idler placed at the top of the peak. This problem is a result of a short radius curve causing excessive edge tensions.

Further complicating the problem of idler gap failure is that CEMA, the Association that sets idler standards, currently does not have any set standard on the amount of gap in the idler junction area. This gap between the rolls that the belt must span without being forced into the opening can easily approach 1” .

There are numerous ways to overcome problems associated with idler gap failure due to short radius curves and excessive idler gaps. The following is a series of ways to solve this problem.

1. Change the center distance of the idler sets in the curve area. Place the troughing idlers as close as possible to each other. This will help level out the curve.
2. If the conveyor uses 35° idlers, install closely spaced 20° idlers in the curve area, providing spillage does not occur.
3. The most common method is to utilize closely spaced long centered off-set idlers. This type of idler does not have a gap between the rolls.
4. The worst case will require numerous idlers on each side of the apex to be shimmed up to straighten out the curve.
5. The last step is to remove any self training troughed idlers from the curve area.

Self trainers are elevated in order to provide tracking and this elevated feature further accentuates the idler gap problem.

When dealing with excessive edge stress, the same 5 choices listed above should be used. A low modulus or “stretchy type” fabric may be required, in addition to shims and changing idlers.

3. PULLEY CROWNS

LIGHT WEIGHT BELT

Crowned pulleys for lightweight conveyor belts can be trapezoidal or radial shaped. Georgia Duck has products to accommodate both styles, however the amount of crown in either case should not exceed 1/8” total on the pulley diameter. The rate of crown seems to be very important as well as the total amount of crown in the system.

On short center conveyors, we recommend no crown on the drive (avoid putting on drive in every case) and to crown the end pulleys. In a few cases we would also crown additional pulleys, but that depends on the entire design and amount of crown used.

Remember, for crowns to be effective, there must be enough free span/transition for the belt to elongate and conform. Pretension to get pulley crown conformation is important. Too much pretension can cause pulley deflection and bearing problems. Georgia Duck has specific carcass constructions to meet very short center, wide belt applications in the 1:1 ratio of length to width and even less. Please consult factory if you have needs in this area.

A crowned pulley, trapezoidal or radius crown, can be regarded as a special case in our primary rule of tracking..."the belt moves in the direction of first contact with the roller." A simple way of explaining how the crown works is to divide the pulley and the belt into a left half and a right half. As a portion of the belt approaches the crowned pulley, the center edge of each belt half touches the crown (higher area) before the outside edges. The belt wants to move toward the center and will do so until forces in both halves balance one another.

In addition to the **surface effect** described above, there is also a strong internal "balancing of warp yarn tensions". Consider any yarn not directly on the center. If the belt moves off center and this yarn is drawn toward the midpoint of the crown, tension will increase. As the belt revolves and this yarn seeks to move back to its normal position, the tension will diminish. Yarns on both sides of the belt carcass also seek that position, which results in the least amount of stress. This is consistent with the physical structure (pulley) across which they are stressed and consistent with their individual position within the matrix (carcass) of the belt. Accordingly, the belt will move across the crowned pulley until these stresses are balanced and minimized- hopefully centering the belt.

Experience has shown that a crown is most effective when it has a long, unsupported span of the belt approaching the pulley, i.e. the rollers or slider bed must offer a minimum of resistance to belt movement in order for crowns to be most effective.

Georgia Duck goes to great lengths to manufacture balanced carcass belts so that the belt will self center and track on the crown. Our lightweight belts, with or without monofilament, are manufactured with a balanced carcass as this is most critical for short center tracking.

The optimum condition of a long, unsupported span doesn't exist in most lightweight applications. Consequently, crowns are not as effective on the head. Further, it is a detriment as far as lateral distribution of tensions within the belt are concerned. The rule to keep in mind is that snubs, bends, etc. close to a crowned pulley dramatically reduce its effectiveness. Consult the factory about your particular design or need for guidance as to what belt type will better fit.

For reversible conveyors, the location of crowned pulleys and other snub and bend pulleys, is extremely crucial. The shorter the unsupported span, the less effective is the crown.

Georgia Duck recommends a maximum pulley crown of 1/8" per foot of pulley face (not to exceed total of 1/8 in). This 1/8" per foot crown is critical and applies to all types of crowns, trapezoidal, radius or center.

Remember, for the crown to be effective, the belt must conform to the crown. Pulley deflection and bearing loads must be taken into consideration.

HEAVY DUTY BELT

Crowned pulleys are not recommended for high modulus bulk haulage belting. Steel cord belting requires fully machined straight faced pulleys through out the system. If a crowned pulley is used on nylon, polyester or Aramid style belting, the crown should only be placed in a low tension area such as the tail on a conventional head drive conveyor. The tracking forces that the crown exhibits do not affect high modulus bulk haulage belting because the system lacks enough tension to make the crown effective. If you could exert enough tension on the belt to force the belt to conform to the crown, the belt would be subjected to excess stretch and splice failure.

4. RUBBER REVERSION

There is quite a bit of history (25+ years) where certain types of lagging (RT Rubber Belt, SBR rubber, etc.) have reverted (see reversion in Glossary of Industrial Terms) due to heat, friction and stress as the belt contacts the drive pulley. In numerous cases, the initial blame is placed on the belt breaking down. But the mountain of hard evidence has always come back to the lagging that is breaking down into a gummy material that picks up dirt and spreads it around the conveyor. This material builds up on rollers, pulleys and slider beds throughout the system.

The lagging materials proven to work on lightweight belts and unit handling conveyors are chloroprene blends with a minimum of 60% chloroprene and 80-85 shore A hardness, polyurethanes in the 75-80 shore A hardness and PVC lagging materials.

Maintaining properly tensioned belts will improve lagging life by eliminating slippage, friction and heat buildup causing rubber reversion.

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5. LOADING STATIONS

Two of the more common reasons of belt failure are a direct result of the loading stations. These are: Holes being punched in the belt due to impact and carry cover deterioration due to abrasion.

A. IMPACT

Large product lumps such as minus 8" limestone and larger with irregular shapes will punch holes in a conveyor belt if not properly introduced to the belt's surface.

The following are the more common methods of presenting large product lumps to the belt and minimizing the damage that these lumps will cause.

i. Rock Boxes

Rock boxes can absorb the impact from large lumps as the box fills up first, then other lumps bounce off of the pile onto the belt. The purpose of the rock box is to transfer the impact energy to the box rather than the belt. The box also will allow the height of the free fall to the belt to be greatly reduced.

ii. Feeders

Various types of feeders, such as belt feeders, pan feeders, reciprocating feeders and vibratory feeders can be utilized. The particular style of feeder that is used will be determined by product characteristics such as: lump size, product weight, moisture content and abrasion.

iii. Grizzlys

Grizzly bars, if installed properly, can be used rather than a rock box. The bars need to be installed to absorb the impact energy from the lump and re-direct the lump to the belt in the line of travel. Grizzlys must be installed so that the bars do not fill up with product and clog the loading station. Grizzly bars also allow a bed of fines to be introduced to the belt before the larger objects fall to the belt. This cushion effect helps to reduce the damage from larger lumps.

iv. Chutes

Chutes and deflector plates are quite common methods for diverting material onto the belt and reducing the vertical impact onto the belt.

All of these various methods serve the same purpose; The introduction of material to the center of the belt, loading in the direction of travel, loading product with velocity approaching belt speed, and reducing product freefall and impact to the belt.

B. CARRY COVER ABRASION DUE TO LOADING

Rubber cover abrasion due to poor loading is an ongoing problem for the conveyor operator. The first step in correcting this problem is the installation of a rock box, feeder, grizzly bar, chutes or deflector plates into the loading area. As we previously discussed, the product needs to be introduced to the belt with a minimum of impact, loaded in the center of the belt, and product velocity approaching belt speed. As the product free fall decreases, the time required for the product to settle down on the belt also decreases. When products bounce around on the belt before settling down, the rubber cover gets cut, gouged and abraded.

Proper loading is best accomplished when products are loaded on the horizontal. Loading on a decline conveyor increases product movement, increases abrasion and increases the opportunity for spillage. Loading on an incline increases cover cutting, gouging and cover wear.

Introducing large lumps at the feed point can result in impact energy that a given conveyor belt will be unable to absorb without damage to the cover and carcass. Impact energy is calculated in terms of foot-pounds of energy. The product weight times the free fall in feet to the belt will yield the impact energy imparted.

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EXAMPLE:

> A lump of limestone that measures 12” x 10” x 18” equals 2160 cubic inches. 2160 cubic inches divided by 1728 (cubic inches in a cubic foot) equals 1.25 cubic feet. If limestone weighs 100 pounds per cubic foot, this lump would weigh 125 pounds. With a free fall drop of 7 feet, this lump would impart 875 foot pounds of impact energy to the belt. This example does not take into account sharp edges or abrasion. It is suggested that impact energy be calculated by the above method, then refer to the manufacturers catalogue for guidance in belt selection.

In addition to the product side of the belt, you must also consider the support area of the loading station. The belt needs to deflect under impact. Rubber impact idlers should be placed so that the load is introduced to the belt between the impact idlers as much as possible. This will allow the belt to deflect under impact. If the deflection of the belt is excessive, then product will be trapped between the skirt boards and the belt. This causes additional cutting, gouging and abrasion.

When impact idlers are moved closer together, it is impossible to load product between idlers. Loading over the idlers is quite common. With closely spaced impact idlers, it is imperative that these idlers be soft enough to deflect under impact.

Another common method of impact absorption is through the utilization of impact beds. Impact beds have a low durometer soft rubber that allows the impact energy to be transmitted through the belt, into the bed, and then absorbed by the bed. The cushioning of the bed allows the belt to deflect, thereby reducing belt damage due to impact.

6. TAKE-UPS

The automatic gravity take-up is the most common type used on bulk haulage conveyors. A movable pulley with a weight box maintains slack side (T2) tension during starting, stopping and load changes by moving to accommodate elastic stretch in the belt. Underground mine applications commonly use air and hydraulic takeup units to accommodate belt stretch.

Weight take-ups or other automatic types (springs, air or hydraulic) are also used on lightweight belt conveyor applications. Automatic take-ups are generally preferred where space allows them to be used as movable take-ups and they put less stress on the belt.

Gravity take-ups are generally low-maintenance and fool proof. When we try to determine where to position the pulley for its travel during new belt installation, we incur problems. The problems faced by the installer are:

1. Getting slack pulled out during installation
3. Allowing enough spare belt for resplicing
3. Where to set the take-up pulley initially
4. Factors influencing belt elongation such as load, friction, drive type, carcass type, heat etc.
5. Take-up bottoming out due to combination of slack, and elastic stretch
6. How much will this belt stretch?

All these problems must be understood and answered in order to string and splice a new belt, as well as, avoid another shut down to remove a section of belt.

There are several ways to pull slack out when stringing a new belt. Use gravity to help out by stringing the belt from the low end around the conveyor and splicing at the lowest elevation. Use a motor to pull the new belt on; then tie and lock one end off while pulling out the slack. It is easier to pull the slack out of the belt down-hill rather than up-hill.

Determine if a mechanical or vulcanized splice will be used and allow enough extra belt for the type chosen. A good rule-of-thumb for vulcanized splices is 2.5 x belt width for belt consumed in a fabric belt splice.

Positioning the take-up pulley is important. A few feet should be left at top-of-travel due to possible lift up of the take-up pulley during start-up. At least 75% of take-up travel distance should be available for elongation after the initial vulcanized splice. This available travel can be adjusted to reflect splice length, total elongation, changes in loading, environmental changes, type of textile fiber and the carcass design.

One factor influencing elongation is percent of load applied. i.e. a belt with a 440 PIW working tension would stretch

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more at 375 PIW than at 215 PIW. Georgia Duck has stress/strain data for each carcass type so that elastic elongation

can be predicted. Heat can also be a problem as hot belts stretch more due to changes in fiber properties. Higher friction will raise tension resulting in more elongation. Examples of these increases would be frozen idlers, idlers with higher rolling resistance, heavier pulleys, skirtboards, plows, wipers, etc. add friction. Drive location, belt wrap and lagging types influence tension applied to belt. There are various carcass types discussed earlier in the Technical Manual that will influence elongation (plain weave, twill, straight warp). The type of fiber used for the strength member in the carcass will also influence elongation, i.e. there are many types of polyesters, nylons and aramids used in the belt carcass. The fiber chosen can influence elongation several percent (a variation of 0.5% to 7.0 % is not unusual).

The objective for the user and the installer is to involve the belt manufacturer in selecting the best belt for your application. A part of reaching this objective is to understand the difference between elastic and inelastic elongation . Elastic, by definition, means recoverable elongation. Inelastic, by definition, means non-recoverable elongation.

Elastic elongation occurs during acceleration, braking, and load changes during running. This is why automatic take-ups are so common - they compensate for these length changes easily. Typical elastic elongation with polyester fibers is 1% or less, usually more than 1% with nylon fibers and less than 1% with aramids.

Inelastic elongation is permanent and occurs during the break-in period. The fabric weave will influence the amount of stretch to be expected. Highly crimped fabrics have more inelastic elongation than low crimp, straight warp carcass types. The crimped yarns tend to straighten as load is applied and the belt is flexed.

The added benefit of belts made of polyester fibers is that they break-in quickly, usually within the first 3-4 days of running. Nylon warp yarns continue to "creep" and will continue to have permanent elongation (although small in percentage) for the entire life of belt. Aramid yarns also "break-in" quickly (2 to 3 days).

Example: To reach our objective, properly setting the take-up, we must calculate a total belt stretch. A 500 ft. center-to-center conveyor with a taped belt length of 1,080' runs at 280 PIW using a 3 ply 330 PIW rated belt made with plain weave, highly crimped, polyester warp fabric. The belt runs at an ambient temperature of 72° F and is subjected to fully loaded, "hard" starts at a maximum of 392 PIW.

1. We would expect a permanent (inelastic) elongation of 1.2 % x 1,080' or 13 feet.

2. We would expect elastic stretch of 0.8% (@392 PIW) x 1,080' or 9 feet.

3. We want to set the take-up at 5 feet from top of travel (Leaving 10ft. of belt in the take-up).

4. Total take-up travel is then:

   i. 5 feet for position
   ii. 6.5 feet for permanent remember movement of 1'= 2' belt take-up
   iii. 4.5 feet for elastic
   iv. 16 feet total travel (32 feet of belt) or approximately 1% of tape length + 5' for position
   (10.8 + 5 = 15.8')

What if a mechanical take-up or screw take-up is used? How do we tension the belt properly to accommodate elastic and inelastic elongation? First, we must calculate operating tension at the worst possible condition and be able to pre-tension the belt so that the inelastic (permanent) stretch is pulled out. Then we have a belt tightened so that the elastic elongation will be recovered within the length of belt at the slack side. In other words the belt must act like a spring or rubber band. When the belt senses lower tension it must recover and remain tight enough to prevent slippage on the drive.

Usually we cannot achieve the required pretension at initial installation for there is not enough travel and screws cannot exert enough force on the belt face. Therefore we must retension the belt during the break-in period perhaps several times. Mechanical splices may be necessary as the belt will require resplicing to shorten and remain within travel limits of the screw take-up. A high crimp, plain weave carcass will exceed most mechanical take-up movements!

With lightweight belts and mechanical take-ups being more common, we can use the belt as a stress-strain gauge by marking the belt and stretching to some pre-determined length based on load. Contact your Georgia Duck representative for assistance on your conveyor. Unusual conveyor configurations can also result in abnormal belt stresses. Conveyors

such as stackers that have have the drive at or near the tail end will have almost the entire length of belt running at close to full operating tension. This condition can cause abnormal belt stresses.
Reversible conveyors with only one motor and one take-up will exhibit greater than normal belt stretch. This is due to the counterweight being heavier than normal. The increased counterweight is required to maintain tension in the system when the motor is reversed. The increased weight results in greater tension applied to the belt. Georgia Duck has belt calculation methods to determine tensions and stress-strain data related to carcass type, fiber type and resulting modulus of elasticity. We can help you predict take-up requirements on your conveyor.

7. TURNOVERS

A conveyor turnover or twist (see front cover) is usually found on long overland conveyor systems where one turnover is located on the return side behind the drive and the take-up, and the other turnover is located in front of the tail pulley. This configuration is the most common use of the turnover system.

A single turnover can also be found in unusual applications and we will discuss these later.

The dual turnover system means that the carry side of the belt is returned “dirty side up” rather than “dirty side down” and in contact with the return idlers. This means the pulley cover side, or clean side, is in contact with the carry side idlers and the return side idlers also.

The dual system offers a variety of benefits to the conveyor operator such as:
A. With build up on the carry cover you reduce the surface wear on the return idlers.
B. Cover wear on the carry side is reduced because the dirty side is not being pulled across the return idlers.
C. You eliminate the build-up of frozen or sticky material on the return idlers.
D. Carry back or dribble, under the conveyor is concentrated between the terminal pulley and the turnovers, rather than the entire length of the return side.

Design of the turnover system is critical. The length of the twist must be long enough to prevent excess edge stresses and long enough to prevent the center of the belt from buckling. Generally, the turnovers should be on the return side of the conveyor because tensions are lower there.

A general rule of thumb for fabric belts is that the length of a 180° twist should be at a minimum of 1 foot of twist length per 1 inch of belt width. Twist lengths shorter than this can result in tensions at the belt center being too low and can cause instability in the twist area. The belt tensions have to be calculated at the point where the turnovers are located. Tensions such as running tension, acceleration tension, stopping tension and breakaway tension should be calculated in addition to other factors like belt weight, belt rated tension and belt modulus. If the calculated tensions in the twist area are quite low, which will result in an exceptionally long twist length to prevent center buckling, additional counterweight can be added to the system to shorten the twist length. The addition of counterweight or slack side tension must be done with caution so as to not exceed the rated working tension of the belt.

The belt sag between the supporting pulleys could become large enough to affect the tensions in the belt at the turnover area. We suggest that sag be held to a maximum of 2% of the twist length.

Sag can be calculated as follows:

$$\text{SAG (feet)} = \frac{(W_b)\,(L_T)^2}{8\,(T_T)}$$

Where W_b = belt weight in lbs/ft
L_T = horizontal length of twist in feet
T_T = tension at twist in lbs.

The actual amount of belt sag will be approximately 70% of the above calculated value due to stiffness in the vertical position at the center of the turnover. It is also suggested that additional vertical clearance be allowed for the middle of the twist area to allow for changes in tensions that occur during acceleration and deceleration which can cause the belt to jump.

We also recommend the following for all turnover systems:

- All splices should be vulcanized
- When using a dual turnover system the twists should be in the same direction to balance out any edge stresses that occur
- Automatic take-ups should be used to maintain a constant tension in the twist area
- Controlled acceleration and deceleration
- Vertical rolls mid-way in the twist that the belt passes through should be off-set a few inches and adjustable in all directions for containment and tracking purposes
- The horizontal end pulleys should be adjustable for tracking purposes
- The two twist pulleys should be 6” wider than the belt and set for a few degrees of belt wrap.

The single turnover or twist system is constructed in the same manner as the dual system except that there is only one 180° twist in the belt. This means that the conveying run is alternated on the carry cover and then the pulley cover and so on.

The single twist system offers the conveyor operator a different set of benefits than what is found with the double twist system.

A. This system would be used conveying hot materials that will allow the belt to have additional cooling time as the load is alternated from one cover to the other.

B. Some products have an adverse reaction when exposed to rubber covers. Alumina is a typical example when conveyed over Grade II covers. The Alumina attacks the cover by extracting the plasticizers causing the rubber to get hard, crack and cup. The single pass turnover when using RMA Grade II covers will allow the Alumina to attack both sides of the belt at once which will offset the cupping problem. Utilization of the proper rubber cover compound will eliminate the problem completely and then a turnover is not required.

C. Some products are highly abrasive and by using a single pass twist in the belt, both sides of the belt will be worn away rather than the carryside only. This will extend normal belt life.

All rubber covers when exposed to a single pass turnover system should be purchased as balanced covers.
Example: 1/8 x 1/8, 3/16 x 3/16, 1/4 x 1/4.

The design of the single turnover system is concurrent to the double system and the same rules will apply as to turnover lengths, sag amounts and roll layout.

Contact your local Georgia Duck representative for assistance with your questions concerning conveyor belt turnovers.

8. Tension Calculations

The tensions introduced into a belt can be calculated by using several methods. We will mention these calculations in order of simplicity, and offer the drawbacks of each method. This will enable us to proceed to a seldom-used calculation that may supply us the answers that the other methods do not address.

A. The short method is often used to calculate theoretical maximum running tension in a simple conveyor system. The only variables used are motor horsepower, speed, type of take-up, belt wrap on the drive, and whether the drive pulley face is lagged or bare.

The short method uses full motor horsepower rather than the actual power consumed to run the system. Start-up torque and horsepower efficiency are also missing. This could become critical since motors are available with efficiency ratings well over 100% of nameplate rating. The conveyor may only require a small horsepower motor but a much larger one was installed. When this occurs, the tension calculations become distorted.

The simple method of calculating effective belt tension (T_e) is:

$$T_e = \frac{\text{Hp x 33,000}}{\text{Speed (fpm)}}$$

To calculate slack side tension (T_2) use the values for k found in the table below.

$$T_2 = T_e \times k$$

To calculate tight side tension (T_1) at head pulley:

$$T_1 = T_e + T_2$$

Type of Pulley Drive	0 Wrap	Automatic Take-up		Manual Take-up	
		Lagged Pulley	Bare Pulley	Lagged Pulley	Bare Pulley
Single, no snub	180°	0.50	0.84	0.8	1.2
Single, with snub	200°	0.42	0.72	0.7	1.0
	210°	0.38	0.66	0.7	1.0
	220°	0.35	0.62	0.6	0.9
	240°	0.30	0.54	0.6	0.8

For wet belts and smooth lagging, use bare pulley factor. For wet belts and grooved lagging, use lagged pulley factor. If wrap is unknown, assume the following.

Type of Drive	Assumed Wrap
Single-no snub	180°
Single-with snub	210°

If the counterweight is located right behind the drive, then calculate theoretical counterweight:

$$CW = 2 \times T_2$$

To calculate pressure for a hydraulic take-up:

$$\underset{(PSI)}{\text{Pressure}} = \frac{CW\ (lb)}{\underset{(in^2)}{\text{Cylinder area}}}$$

Example: A particular conveyor requires 3000 pounds of take-up tension, and has a hydraulic take-up utilizing a 3 in diameter cylinder. The effective area of the cylinder is 7.068 in^2. The pressure required is 424.41 psi.

B. Another relatively simple method for use on generic conveyors is the use of a hand held calculator programmed for belt calculations. This is reasonably accurate as the program can be written to include numerous variables. The major deficiency is start-up torque and the difference between calculated counterweight and actual counterweight. To better understand how the counterweight difference can play such an important part in trouble free conveying an example best explain this.

Example: A 36 inch wide belt with a conventional snubbed/lagged head drive, and a normal gravity take-up located close behind the head pulley requires 6,080 pounds of take-up tension. This means that slack side tension (T_2) is 3,040 pounds, and the tight side running tension (T_e) is 8,000 pounds. This equates to a total running tension (T_1) of 11,040 pounds or 307 piw. One may select a 3 ply 330 piw belt for this conveyor.

Assume that the belt started slipping and 2800 pounds of counterweight was added. Then the slack side tension (T_2) would increase to 4440 pounds, and the tight side running tension (T_e) would still be 8,000 pounds. Then the total running tension (T_1) would be 12,440 pounds or 346 piw. A 3 ply 330 piw belt is no longer appropriate for this particular system.

With the advent of personal computers, especially laptops, handheld calculators are falling to the wayside. Georgia Duck provides a Windows based program package that can be used on conventional conveyors that do not have undulations, vertical curves, or complex drives. It incorporates horsepower, belt speed, take-up, idler/pulley friction factors and lift to calculate belt tension. This program package is user friendly, and quite accurate for most bulk, package handling, and elevator leg belting applications.

C. The point-to-point program is one of the most detail-oriented programs available. It can calculate tension at any point on the system, which can be helpful when designing complex conveyor systems with inclines, declines, and either convex or concave vertical curves. In order to run this calculation, very detailed descriptions and /or drawings are required. This program can calculate vertical curve radii, and belt turnover distances. The point-to-point also can be set up to simulate conveyor start-up and braking unloaded, partially loaded and fully loaded. If a point-to-point calculation is needed, then please contact either a Georgia Duck Representative or a qualified material handling engineering firm.

9. Bucket Elevators

Bucket elevators transport material vertically by using buckets that are bolted to a belt. Either PVC or rubber belting can be used in this application. Conformation to specific requirements usually determines what kind of belt to use. For instance, OSHA, ASTM, MSHA, and CSA may all have particular belting specifications for grain handling.

There are two types of bucket elevators-Centrifugal and Continuous. Both are loaded at the lower-most pulley (the boot) and discharged at the upper most-pulley (the head). We will discuss each type of elevator.

Centrifugal Discharge

Popular in the grain, coal, and fertilizer industry, centrifugal discharge elevators are high speed when compared to the continuous bucket elevator. This elevator is almost always built completely vertical. The buckets are loaded by digging in the elevator boot. Buckets are spaced in a predetermined uniform cycle in order to allow the material to be thrown out at the head pulley.

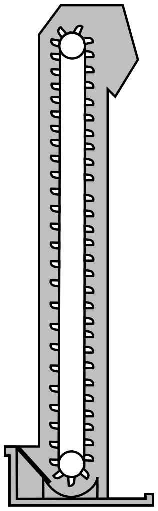

Figure A- Centrifugal

Continuous Bucket

This type of elevator is used to carry either heavy, or large lump material. The continuous elevator is often used to convey materials such as cement or lime. This elevator's speed is much slower than that of the centrifugal discharge elevator.

Most of the time this elevator is at an angle between 15° - 30° from vertical. This allows materials to be dumped directly into the buckets via chute. Buckets are spaced close together so that the bottom of the previous bucket acts as a discharge for the following bucket.

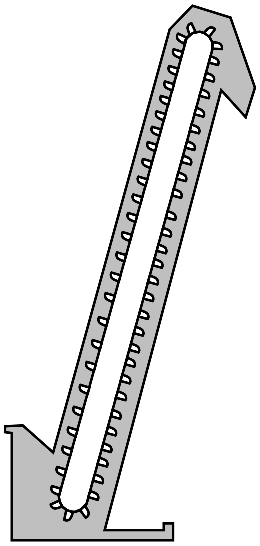

Figure B - Continuous

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