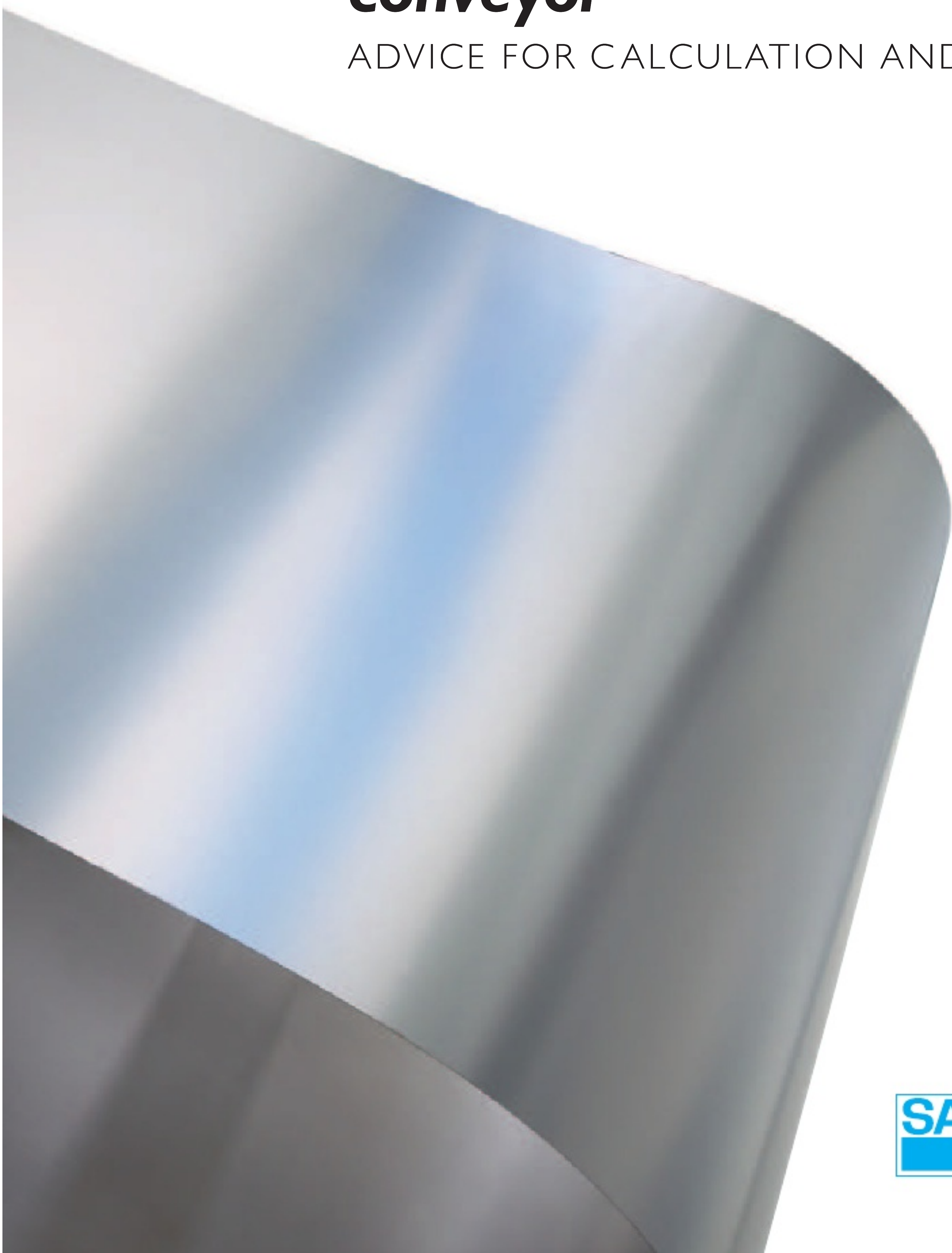
The steel belt conveyor

ADVICE FOR CALCULATION AND DESIGN



SANDVIK

CONTENTS

Introduction 1
Material handling 1
Design criteria and rules 2
Selection of belt grade 2
Selection of belt type 3
Selection of belt size 3
Belt joint 3
Selection of terminal drums for plain
Steel belts 4
Selection of drum profile 5
Selection of terminal sheaves for
True-tracking belts 6
Belt tensioning for steel conveyor belts..... 7
Tensioning mechanisms 8
Selection of belt supports 9
Selection of safety devices and
belt guides 12
Steel belt maintenance 15
Cleaning belts 16
Power calculations 17

The steel belt conveyor

Introduction

"Conveyors" are defined as either fixed or portable devices for moving materials between two fixed points at the same or different elevations, with continuous or intermittent forward movement. Many types and sizes of conveyors have been developed, but our focus is on steel belt conveyors.

During the first four decades of the last century, the steel belt conveyor was used to move a great variety of bulk materials without regard for the belt´s specific properties. Eventually, the specific properties of the steel belt were more fully identified and put to work.

For example:

1. The dense, smooth surface of a steel belt is important for simple discharging of products.
2. The low coefficient of friction is important for sliding and accumulation of products.
3. The elevated temperature properties are important for baking and handling hot products.
4. The hygienic properties, particularly for stainless steel, are important for processing and handling foods.

This publication will deal with the fundamental technical rules for the proper design of steel belt conveyors and the calculations necessary for construction.

In this booklet applications where the material on the steel belt is subjected to a physical or chemical change, referred to as "processing systems" have not been covered. However, the fundamental technical rules for the steel belt conveyor also apply in these applications.

The main types of belts for belt conveyors are:
1. Chain belts
2. Plastic belts
3. Plastic-coated fabric belts
4. Rubber-coated fabric belts
5. Rubber-coated wire reinforced belts
6. Solid steel belts
7. Wire belts

In order to decide which type of conveyor belt to use, some of the following points should be considered:

The first, and probably the most important item, is the product to be moved. For example, if the product is small, a wire belt may not be suitable. If the product is hot, rubber or plastic belts may be eliminated. The quantity and the physical and chemical conditions are considerations necessary for selection of the proper belt.

The second consideration is the distance that the product is going to be moved.

The third is the speed of movement.

The fourth is loading and unloading features, and the integration of the conveyor with other production equipment.

The fifth is maintenance.

And finally, the economic factors.

Material handling

In engineering for material handling, consideration should be given to a number of additional factors. More efficient sorting, loading and unloading of goods increase the reliability, decrease the costs, and reduce the risk of accidents and strain on workers.

Computer-aided material handling systems are a major factor in todays workplace. Each workplace has to be examined to determine which improvements in material handling are necessary.

In analyzing and planning a material handling system, there are a number of factors to be examined.

For example:

1. Product
   a. dimensions
   b. weight
   c. shape
   d. fragility
   e. quantity

2. Workspace
   a. routes
   b. available time
   c. production factors
3. Maintenance and repair
4. Alternative methods

In the final analysis, all these contribute to cost savings – usually the major determining factor for installing a steel belt conveyor.

Design criteria and rules

In the design and selection of the various mechanical components which make up the conveyor, the basic design rules were established many years ago by the first producer of steel conveyor belts, Sandvik. There has been very little published on steel belt and component development, and this is one of the main reasons for this publication.

The present rules for design and calculation provide for optimal performance of the steel belt conveyor.

Selection of belt grade

First, let´s look at belt grades. Each conveyor application has its own unique demands. To meet these demands, it is essential that the proper steel belt grade be selected. Figure 1 can be used as a reference in the selection process. This figure lists critical belt properties and shows the relationship of these properties to the various belt grades.

In general, Sandvik 1200SA and 1300C are most frequently selected for material handling and processing conveyors. As Figure 1 indicates, if an application requires the belt to have high wear resistance and stability against uneven temperature, Sandvik 1300C is very well suited for the purpose.

Hot operating environments.

The maximum practical temperature in which a steel belt should operate is 350 °C. Applications with higher temperatures require special considerations for selecting the belt grade.

It is important that the temperature is uniform across the width of the belt. Local hot spots can cause temporary distortion, erratic belt tracking, and loss of load. Under severe conditions, where uneven heat cannot be avoided, special devices for belt tracking may be required.

Sub-freezing operating environments.

For applications where the steel belt is constantly working in low temperatures below - 20°C or less an austenitic stainless steel is preferred, i. e. Sandvik 1200SA or 1000SA.

GRADE	Static strength	Fatigue strength	Strength at high temperature	Strength at low temperature	Stability against uneven temperature	Corrosion resistance	Wear resistance	Weld factor	Repairability/ weldability
Sandvik									
1000SA	●	●●	●●	●●●●	●	●●●●	●	●●	●●●
1200SA	●●	●●●	●●	●●●●	●	●●●	●●	●●	●●●
1700SA	●●●●	●●●	●●	●●●●	●	●●●	●●	●	●●
1050SM	●●	●●●	●●●	●●●	●●●	●●	●●	●●●●	●●●●
1150SM	●●	●●●	●●	●●●	●●●	●●	●●	●●●●	●●●●
1500SM	●●●●	●●●●	●●●	●●●	●●●	●●	●●●	●●●	●●●
1650SM	●●●●	●●●●	●●●●	●●●	●●●	●●	●●●	●●●	●●●
1850SM	●●●●	●●●●	●●●	●●●	●●●	●●	●●●	●●	●●
1100C	●●	●●●	●●●	●●	●●●●	●	●●●●	●●	●●
1300C	●●●	●●●	●●●	●●	●●●●	●	●●●●	●●●	●●
1320C	●●●	●●●	●●	●●	●●●●	●	●●	●●●	●●●

● = fair ●● = good ●●● = very good ●●●● = excellent

Figure 1
Selection of a belt grade.

Selection of belt type

There are three types of steel belts: **Plain, rubber true-tracked** and **steel spiral true-tracked**. Plain belts are most common, but there are times when the other types are better suited for the conveyor application.

For tracking, the belts can be provided with V-ropes, either rubber or in the form of a specially designed steel spiral. If required, the product side of the belt can be fitted with retaining strips to keep the conveyed material on the belt or with transverse flights to prevent material from sliding backwards when the belt is steeply inclined. Material properties according to PS-SB-5507.

Selection of belt size

For piece goods, the belt width is determined by the size of the individual objects being conveyed, and by the way they are to be placed on the belt. For bulk handling, a calculation must include the following factors:

1. Volume
2. Distance
3. Material Characteristics
4. Incline of the Conveyor

Your local Sandvik office could be contacted for technical assistance. Standard dimension range for Sandvik steel according to PS-SB- 101.

Belt joint

Steel belts for conveyors are joined by either welding or riveting. Special techniques and tools have been developed for both welding and riveting, and unless someone has these skills and tools, it is advisable to have this work done by the belt supplier. Most belts are joined at the work site; however, specialty endless belts may be manufactured at the factory and installed at the work site.

The belt must be carefully prepared and cut so that the two ends fit very closely together before joining.

A special fixture holds the two ends in position as an automatic welding head makes the weld. Depending on the belt grade, the weld is either cold worked or heat treated to restore most of the original mechanical properties, and the surface of the weld is finished to the desired smoothness. Figure 2 shows fixture for automatic welding of belt joint.

Riveted belts have two types of joints: **overlap** and **butt strap**, shown in Figure 3.

The overlapped riveted joint is the easiest to make and can be performed by factory trained maintenance personnel. If a smooth working surface is required, the butt strap joint with flush rivet heads should be used. A butt strap joint is also used on reversible conveyors.

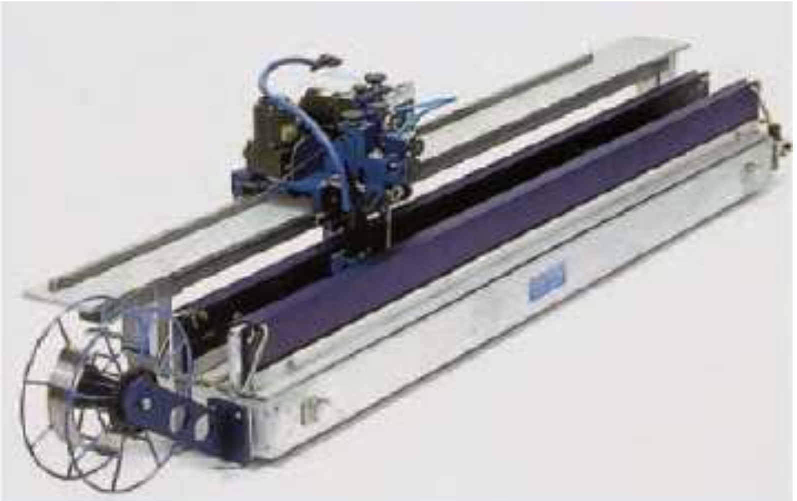


Figure 2
Welding fixture.

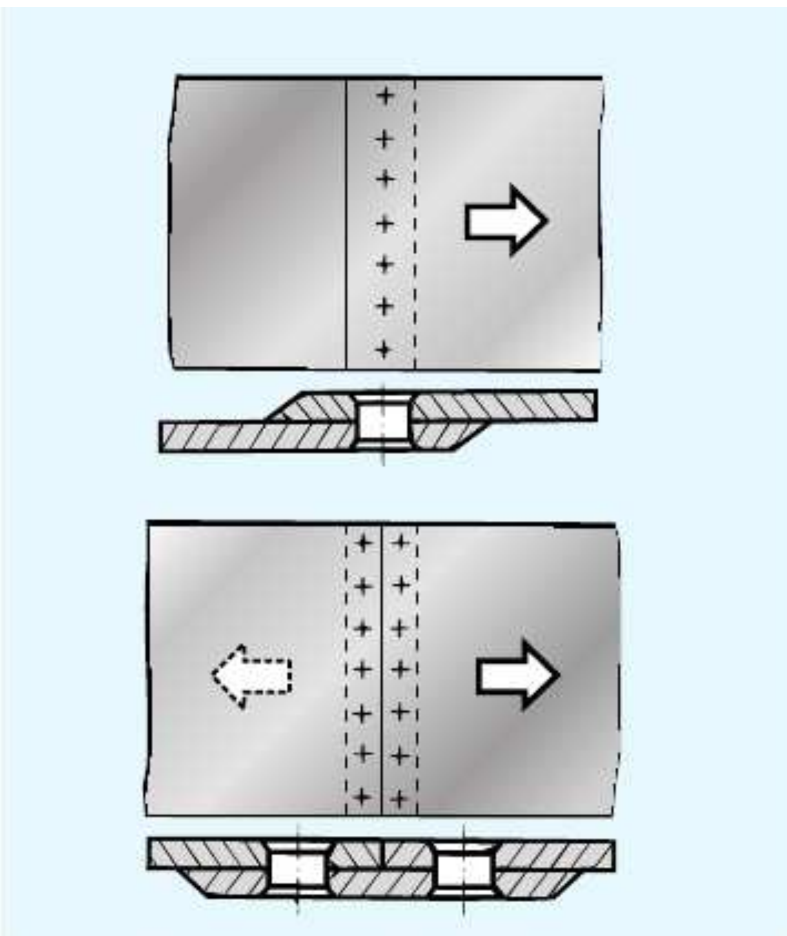


Figure 3
Overlap and butt strap joints.

Selection of terminal drums for plain steel belts

In designing a steel belt conveyor, the anticipated fatigue life of the belt and the belt joint is an important consideration. Preventing premature fatigue failure is the basic objective of having the proper drum diameter.

For a plain belt with riveted joint, a maximum belt tensile stress of 25 N/mm^2 is recommended, and for a plain belt with a welded joint we recommend a maximum of 50 N/mm^2. Tensile stress is defined on page 6. Under such conditions, determine the approximate drum diameter by use of the diagram in Figure 4.

For a welded joint, the diameters given have a safety margin where fatigue should not occur when the belt is operating under normal conditions and where no corrosion, abrasion, or impact are expected. Usually, riveted joints have to be replaced within 10–22 months.

The diagram in Figure 4 is valid for tensile stresses up to 25 N/mm^2. For tensile stresses between 25 and 50 N/mm^2, the drum diameter should be increased by 10 %. This figure is valid for Sandvik 1200SA and 1300C steel belt grades. Diagrams for other steel belt grades are available.

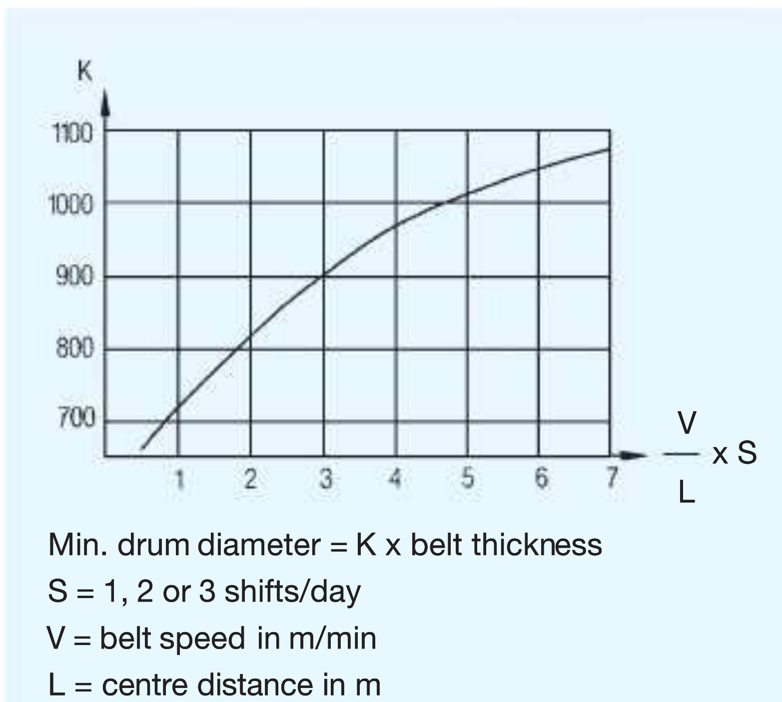


Figure 4
Determination of drum diameter.

Selection of drum profile

The drum profile strongly affects the guidance of the belt. Generally, the overall drum width should be narrower than the belt width. The central portion of the drum face should be cylindrical, and the portions on each side should have a slight taper. See Figure 5.

For special cases, where higher belt stresses are involved, other rules may apply and the belt manufacturer should be consulted.

The tolerance of the drum should be measured as a peak-to-valley measurement while rotating the drum on a shaft. This is usually done with a dial indicator which is moved transversely across the cylindrical face of the rotating drum. The peak-to-valley reading, depending on the width of the cylindrical portion, should not exeed 1.25×10^{-4} x width. This tolerance does not apply to the tapered portions.

The drum should be designed to withstand a maximum deflection of 1:2000 measured between bearing centres.

The conveyor drums are generally made of fabricated steel or cast iron. However, for better friction, the drum face can be lagged with rubber or other high friction materials. Under wet conditions, provision should be made for liquid removal by means of holes, grooves, or a herringbone pattern in order to maintain belt contact with the drum surface.

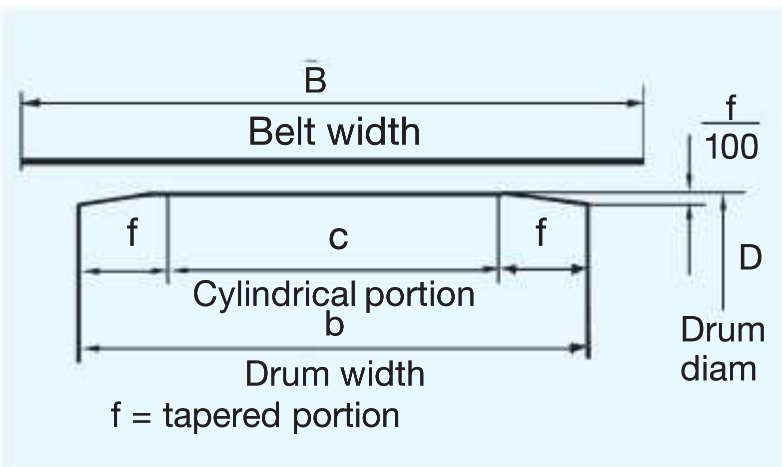


Figure 5
Drum profile.

$$f = \frac{b - c}{2}$$

Dimensions b, c, and f may vary, depending on belt size and belt operating conditions. See Figure 6.

Case	1		2	
B	c	b min	c	b min
≥ 200 ≤ 500	0.5 B	c + 60	B - 60	B - 20
≥ 600 ≤ 800	0.5 B	c + 100	B - 150	B - 50
≥ 1000 ≤ 4500	B - 400	c + 100	B - 200	B - 100

Figure 6
Drum dimensions.

Case 1 is valid for a tensile stress $s_p \le 25$ N/mm²

Case 2 is valid for a tensile stress $\sigma_p > 25$ N/mm² up to ≤ 50 N/mm²

Also valid for belts working above 100 °C.

$$\sigma_p = \frac{\text{Tension force in belt, N}}{\text{Belt width, B x Belt thickness, t, mm}}$$

Selection of terminal sheaves for true-tracking belts

For true-tracking belt conveyors in light duty applications, the arrangement at the terminal ends is quite simple. Narrow belts with a single guiding strip can generally be supported by one true-tracking sheave with a groove. The diameter of the sheave should be determined according to the same rules as described earlier for plain belts. For wider belts and belts with two guiding strips, support sheaves are normally required.

When using more than one sheave on the same shaft, the sheave diameter tolerance is the same as the drum diameter tolerance for plain belts. The number of support sheaves depends on belt width, stress, and the requirements of feeding and discharging. See Figure 7.

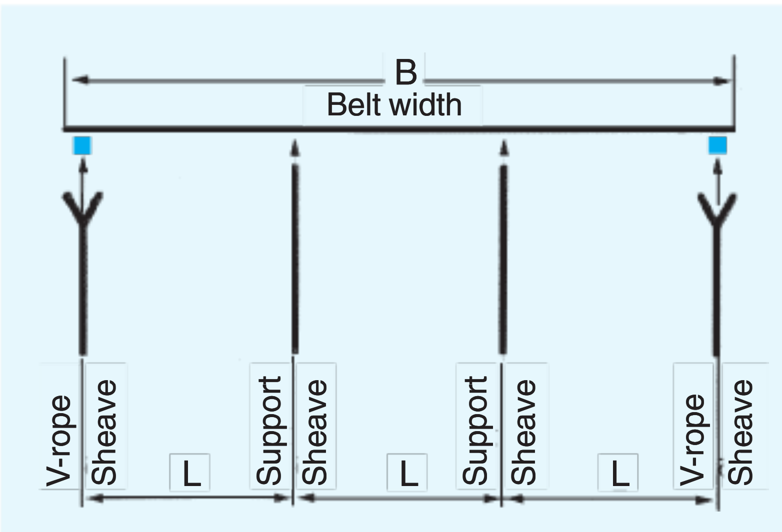


Figure 7
Calculation of the number of required sheaves.

Before calculating the number of sheaves required, the max pull of each sheave has to be checked. See "Minimum belt tension" page 7.

The belt stress at the terminal sheaves is composed of three components:

1. Tensile stress, σ_p, due to the pretension and pulling force, (see page 5).

Normally 15 N/mm² is recommended as the max. tensile stress for a true-tracking belt.

2. Bending stress, σ_b, defined as $\frac{E \times t}{D}$

3. Additional stress, according to Thimoshenko´s "Theory of Plates and Shells."

The maximum permissible belt stress at the terminal sheaves is 400 N/mm².

Referring to Figure 7, the maximum belt section, L, supported by one true-tracking or support sheave can be calculated by using the following formula:

$$L_{max} = \frac{400 - \sigma_b - \sigma_p}{\sigma_p} \times 1.1 \times \sqrt{\frac{E \times t^2}{\sigma b}}$$

D is the diameter of the sheaves (mm); **E** is the modulus of elasticity (N/mm²); **t** is the belt thickness (mm); σ_b is the bending stress (N/mm²); and σ_p is the tensile stress (N/mm²). The resulting value of L_{max} should not be allowed to exceed 400 mm.

As an alternative on belts with more than one guiding strip, sheaves with chamfered sides instead of grooves can be used, see Figure 8. In such a case contact should be made with the steel belt manufacturer, for further advice.

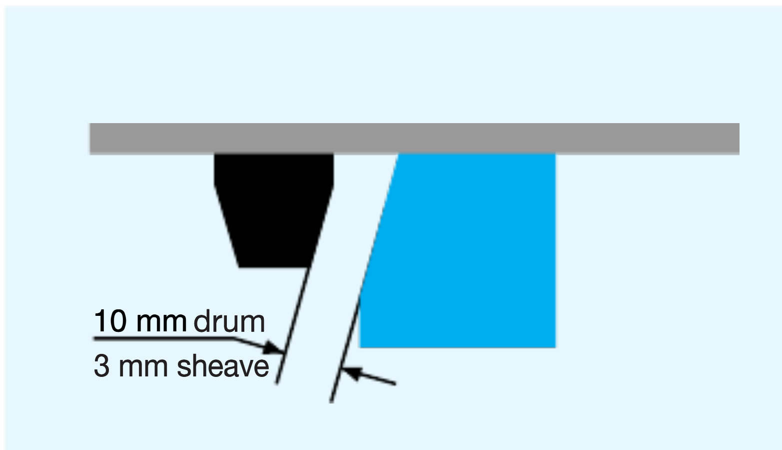


Figure 8
Sheave or drum with chamfered sides.

Sheaves for true-tracking belts with rubber guide strips (EURO standard) should have a groove profile as shown in Figure 9.

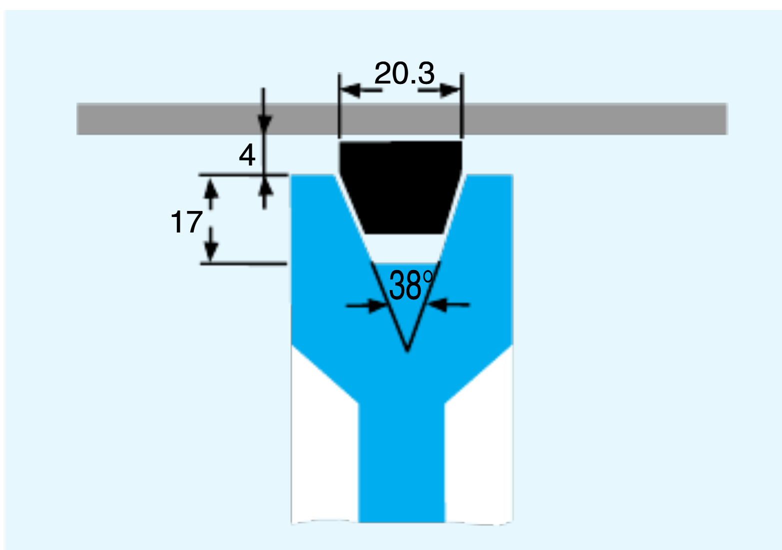


Figure 9
Groove dimensions for rubber true-tracking guide strips.

Sheaves for true-tracking belts with steel-spiral guide strips should have a groove profile as shown in Figure 10.

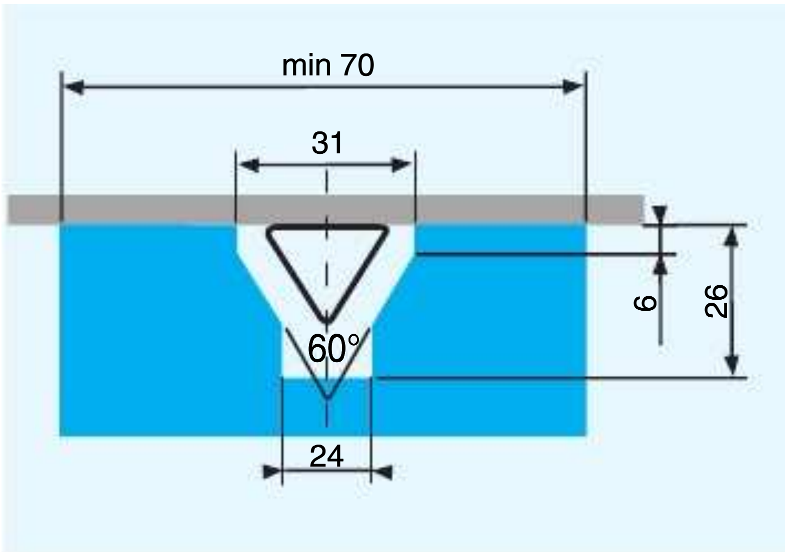


Figure 10
Groove dimensions for steel-spiral true-tracking guide strips.

Belt tensioning for steel conveyor belts

Tension on a steel belt conveyor is generally applied against one of the terminal ends, called the ”tail” drum or sheave.

Because the steel belt will not elongate in normal conveyor usage, the tensioning arrangement is fairly simple. However, at elevated temperatures, consideration has to be given to thermal expansion.

There are four main reasons for tensioning a belt:

1. To obtain enough friction force between the belt and drive drum/sheave for running the belt.
2. To obtain enough pressure between the belt and terminal drums for guidance of plain belts.
3. To obtain optimum belt flatness.
4. To obtain proper belt tension for operational requirements.

Minimum belt tension

When the required pull, P_T (see page 17), to run the belt has been determined, it is necessary to check that the belt tension, Q_P, is sufficient to obtain enough friction between the belt and drum.

For a plain belt and drum arrangement, the following formula applies:

Minimum Qp = 3.3 P_T for a steel drum or 1.67 P_T for a rubber lagged drum.

However, the normal pre-tension stress in plain belts should be no less than 10 N/mm^2. The total min. tension, **Q**, is thus arrived at as follows:

Q = 2 x 10 x B x t

where **B** is the belt width (mm) and **t** is the belt thickness (mm).

The larger, **Q** or **Qp**, should be chosen, as total pre-tension for the plain belt.

For true-tracking belts, other conditions apply with the following assumptions being made: a max pull of 2500 N is assumed for each true-tracking sheave, and a max. pull of 1250 N for each support sheave, to obtain enough friction.

For true-tracking belts, normally a pre-tension stress of 7 N/mm^2 is recommended for true-tracking belts. The normal pre-tension for a true-tracking belt is thus arrived at as follows:

Q = 2 x 7 x B x t

Maximum belt tension

Belt tensions must not be so high, that the maximum tensile stresses – as mentioned on page 5 (plain belts) and page 6 (true-tracking belts) – are exceeded!

Tensioning mechanisms

It is imperative that the components in the tensioning mechanism have sufficient rigidity to provide parallel movement for the two terminal shaft bearings. At least one of the tension shaft bearings should be adjustable for making conveyor and belt tracking corrections. The terminal framework and interconnecting members should also have proper strength and rigidity.

The tensioning arrangement can be designed in many different ways, depending on the operating conditions of the conveyor and the size of the belt. When a new belt is installed, the drum should be positioned near the outermost point in order to provide space for shortening when a future remaking of the joint is required.

An exception to this is where the belt will be exposed to an elevated temperature. In such cases, the thermal expansion has to be considered since the steel belt will expand.

Manual belt tensioning

The simplest device for manually adjusting belt tension consists of two take-up bearings, see Figure 11.

This type is used primarily on conveyor installations with centre-to-centre distances of up to 30 m, operating at room temperature. For adjusting small true-tracking belts, simple pillow blocks with a means of adjustment along the conveyor length axis can be used.

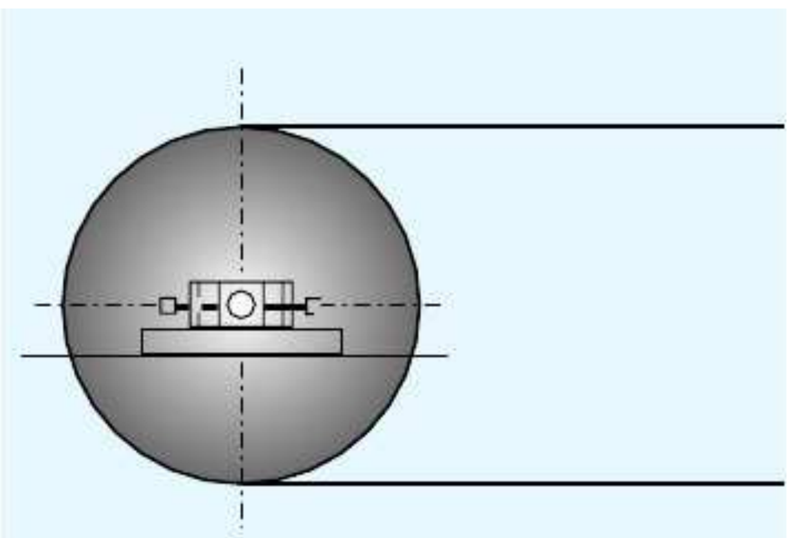


Figure 11
Belt tensioning by means of take-up bearings.

Automatic belt tensioning

An automatic tensioning device should be used on conveyor installations where the centre-to-centre distance is greater than 30 m, or the operating temperature is either lower or higher than the ambient temperature. Automatic tensioning can be achieved in many different ways. Three methods will be illustrated:

1. Tension frame and counterweight.

This method involves the mounting of the tail drum assembly on a moving tension frame which is actuated by a counterweight, see Figure 12.

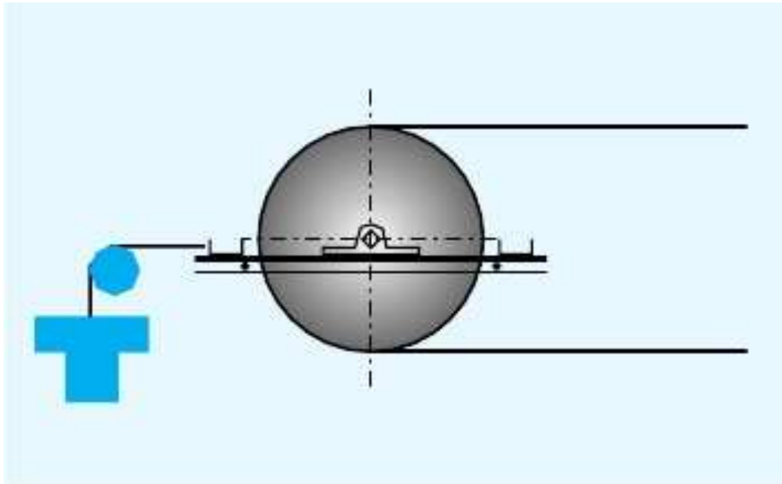


Figure 12
Belt tension by means of a tension frame and a counterweight.

2. Tension frame and coil springs.

This method involves the placement of coil springs between the moving tension frame and the conveyor structure. This design is more compact than the counterweight method. Because the spring pressure varies with the degree of compression, special care must be taken in selecting the spring shape and properties. Hydraulic or pneumatic cylinders with suitable pressure controls can be used instead of the springs. These are particularly advantageous where the tension requires frequent adjustments, see Figure 13.

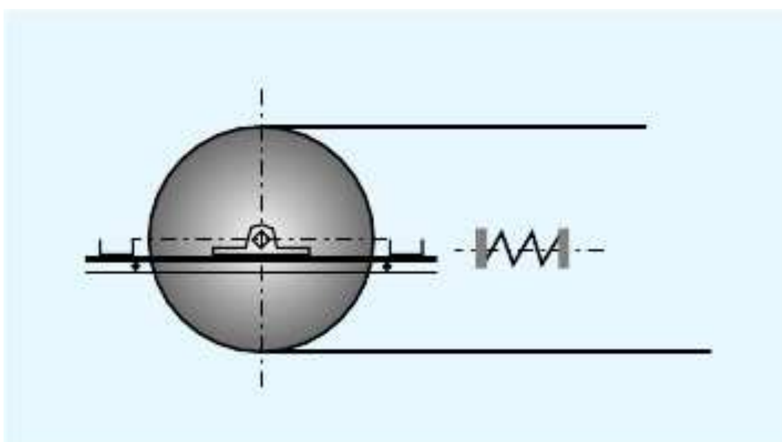


Figure 13
Belt tension by means of a tension frame and compression springs.

3.Torsion shaft.

This method is normally used for wide belts where a tension frame is not practical. When the width of the frame exceeds its length, this can cause structural instability and deflection problems. In such cases, the springs or cylinders can be made to act on a torsion shaft which does not depend on the tension frame for the transmittal of tension forces to the bearings. In the torsion shaft arrangement, the adjustable drum shaft bearings are actuated by threaded rods which provide individual adjustment of the bearings. See Figure 14.

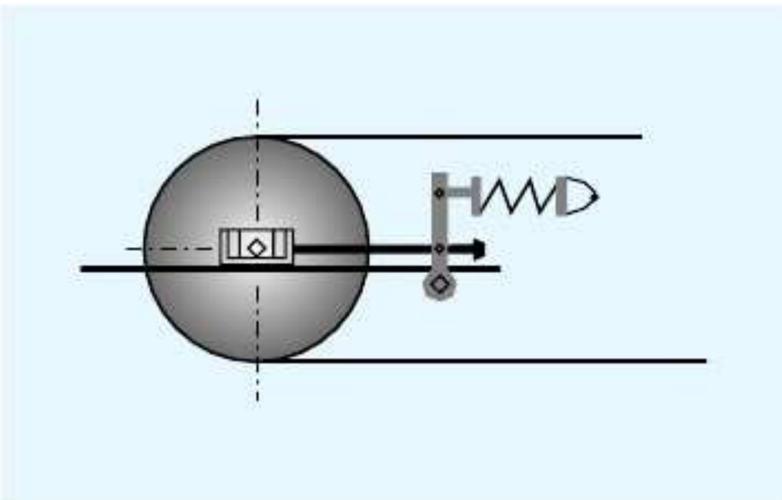


Figure 14
Belt tension by means of a torsion shaft arrangement.

Belt tension formula

Actual belt tension, **S**, in one belt strand can be approximated by checking the catenary, or belt sag, **f**, on the return strand between two belt support rollers. See Figure 15.

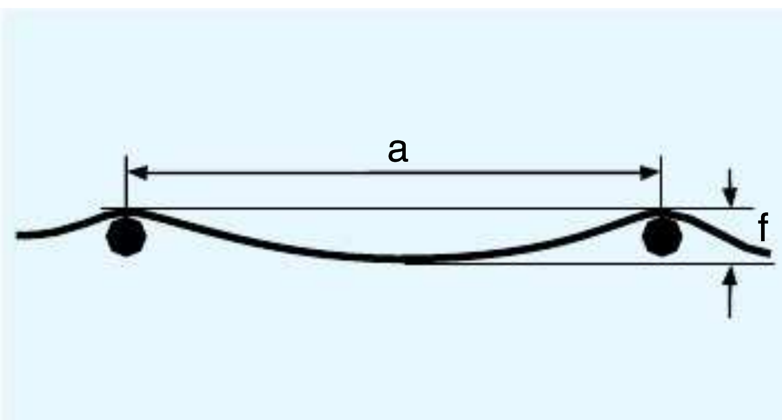


Figure 15
Catenary, or belt sag.

The following formula applies:

$$f = \frac{5}{4} \cdot \frac{q \cdot a^2}{S} \cdot 10^3$$

where **S** is the belt tension (N); **q** is the weight of belt strand (kg/m); **a** is the distance between two rollers (m); and **f** is sag in mm.

NOTE 1: In determining the **weight** of the belt strand, the weight of the true-track strip, if applicable, should be included. The weight of one true-tracking rubber strip is approximately 0.5 kg/m; and, of one true-tracking spiral, 0.25 kg/m.

NOTE 2: The **density** of the steel belt material is approximately 7.9 g/cm^3. See our data sheets for detailed information on density of steel belt grades.

Selection of belt supports

The steel belt can be supported between the terminal ends by either slide supports or idlers. Slide supports are commonly used when the conveyor is short and the load is not extremely heavy or abrasive. Idlers with low friction bearings minimize the pull and wear on the belt.

Slide supports

Carbon steel belts slide easily over metal, wood, or plastic, while stainless steel belts slide easily over wood or plastic. Stainless steel belts should not be used to slide over mild steel, stainless steel or aluminium slide supports.

Idlers

An idler can be either individual wheels on a common shaft or a tubular roller. See Figure 16.

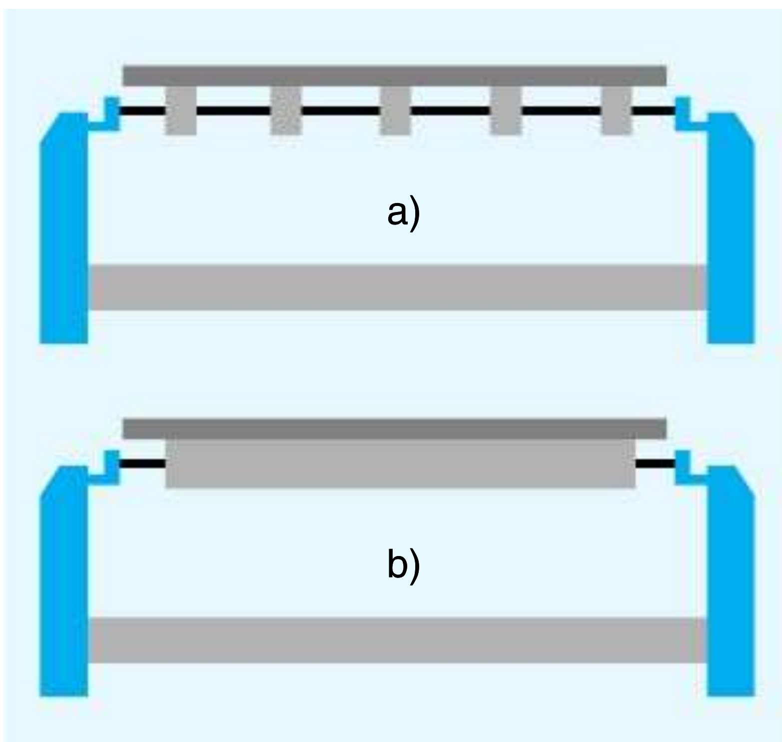


Figure 16
Idlers a) wheels on a common shaft
Idlers b) tubular roller

Generally, the length of the support idlers should be equal to the belt width minus 100 mm. Idlers should be reasonably balanced.

For noise reduction at high belt speeds and for increased friction for improving rotation, rubber rings may be fitted to idler rolls. Rubber rings with elliptical cross-sections to support the return strand are preferred when sticky materials are handled.

Break-point units

Idlers are also used where a belt must change its direction of travel from the horizontal plane. For example, when the return strand must be "caught-up" to clear obstructions. The change of direction should not be more than two degrees per single idler. When greater bends are required, a cluster of closely spaced idlers can be used, called a break-point unit. In such cases, a bend of two degrees maximum per idler applies to the first and last idler in the cluster; a bend of three degrees maximum applies to the intermediate idlers, see Figure 17.

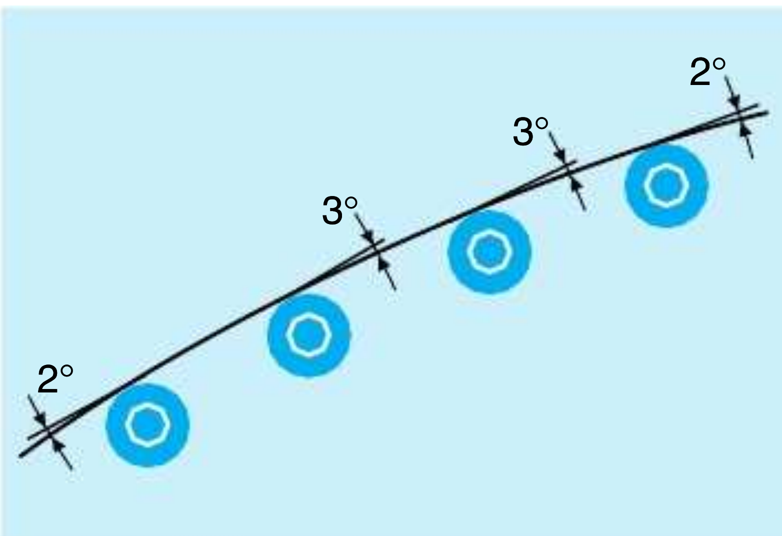


Figure 17
Break-point unit (cluster of idlers with respective bending angles).

The break-point unit should be able to pivot. It has a considerable effect on the guidance of the belt and, therefore, its pivoting bearings should be adjustable both horizontally and vertically. The length of the idlers in the break-point unit should be 50 mm narrower than the belt width.

Idler spacing in the carrying strand

In determining the distance between support idlers in the carrying strand, the objective is to avoid high bending stresses. For a belt operating under normal conditions, the catenary, or sag, between two idlers should be limited to 25 mm. The nomograph in Figure 18 can be used to obtain the idler spacing. The following formula applies:

$$a = \sqrt{\frac{4}{5} \cdot \frac{f}{10^3} \cdot \frac{S}{(q+q_g)}}$$

where **f** is the sag of belt (mm); **q** is the weight of the belt (kg/m); q_g is the weight of the load (kg/m) on the belt; **a** is the distance between idlers (m); and **S** is the tension in one belt strand (**N**).

Depending on the type of process or the product being conveyed the belt sag might have to be less than the above calculated maximum value.

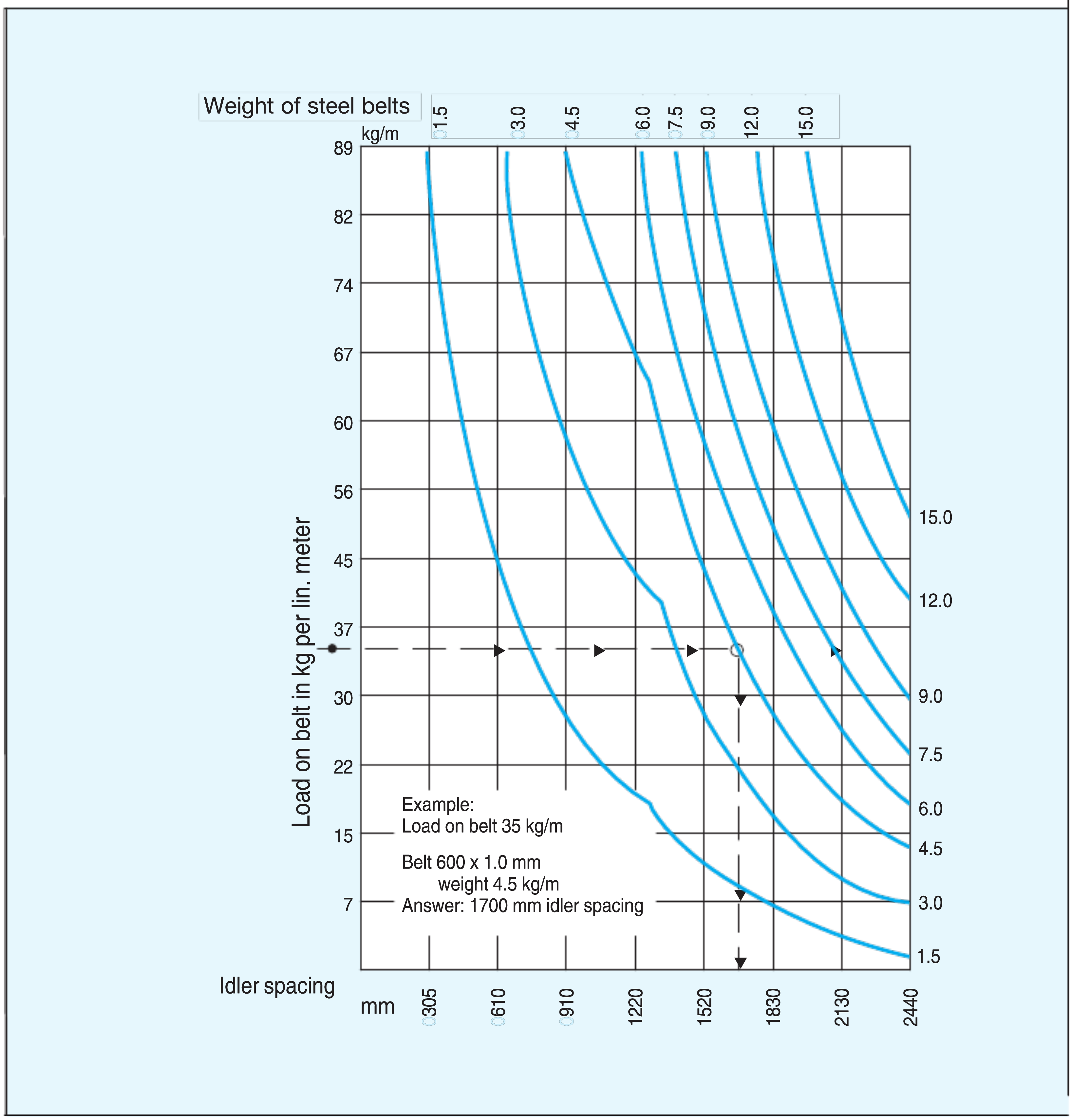


Figure 18

Nomograph for idler spacing.

Idler spacing in the return strand

In the return belt strand, the distance between the tension drum and the first idler should not be more than 3 m. For plain belts, a distance of up to 8 m between idlers can be used if there is sufficient clearance for the sag. For true-tracking belts, the distance can be calculated using the weight of the true-track strips as $\mathbf{q_g}$. See page 10.

Selection of safety devices and belt guides

In the design of a steel belt conveyor, certain safety devices and belt guides have to be included. These can be divided into five groups:

1. Safety scrapers
2. Drum cleaners
3. Belt cleaners
4. Belt guides
5. Limit switches

Safety scrapers

The safety scraper prevents foreign matter from getting between the belt and the tail drum, thereby preventing belt damage which could lead to belt failure. The safety scraper for plain belts is made in the shape of a plough that rides on the return strand just before the drum, see Figure 19.

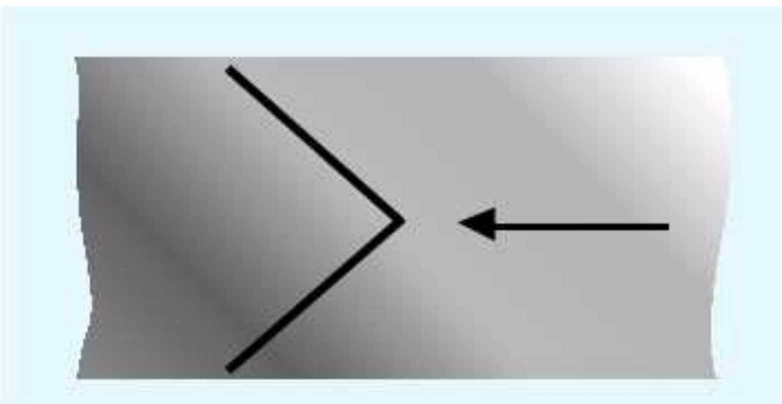


Figure 19
Safety scraper for plain belts.

It is important that the safety scraper floats on the belt and provides full contact. For carbon steel belts, the wear edge should be made of belt material; for stainless steel belts, the wear edge should be made of a softer material, such as plastic, rubber, or wood.

If the safety scraper is wider than the belt, the wear edge extending outside of the belt width must be made of a softer material than the belt. This will help avoid belt edge damage where the belt edges contact the scraper.

The safety scraper for a true-tracking belt has notches in the wear edges to accommodate the guiding strips, see Figure 20. The material adjacent to the true-tracking guide strips should be a soft material, such as rubber.

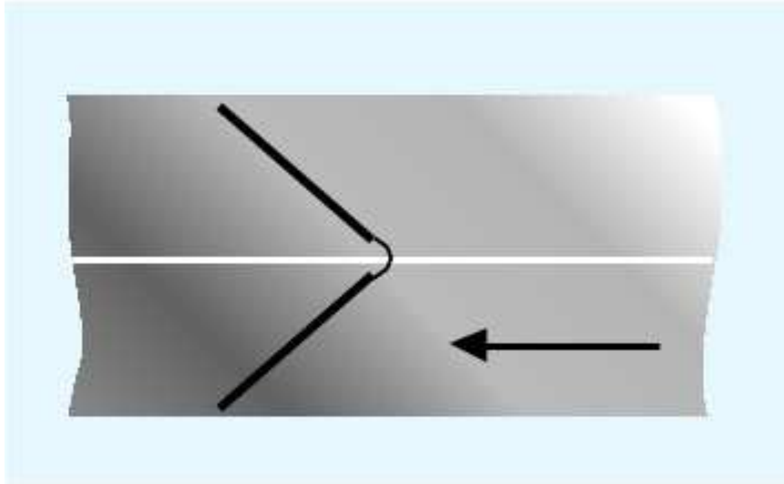


Figure 20
Safety scraper for true-tracking belts.

Drum cleaner

The drum cleaner prevents foreign matter from accumulating on the surface of the tail drum which, in case of build-up, can cause the dimension tolerances of the drum to change. This results in erratic belt tracking. Any material removed by the drum cleaner or groove scrapers should be prevented from falling between the drum and safety scraper, see Figure 21.

For a fabricated steel drum, the drum cleaner blade should be made from plastic or carbon steel belt material. If the drum has tracking grooves, groove scrapers should also be used.

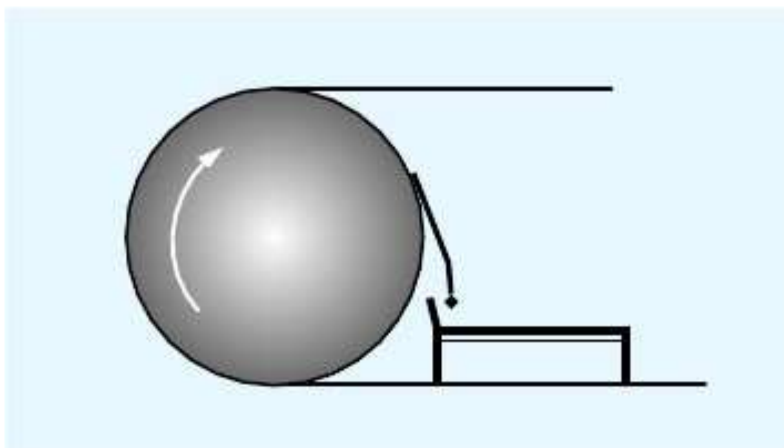


Figure 21
Drum cleaner arrangement.

Belt cleaner

The belt cleaner removes residual material from the product side of the belt, at the discharge location. If not removed, residual material can cause build-up on the return strand idlers, causing erratic belt tracking. The most common belt cleaner is the **tangential** type shown in Figure 22.

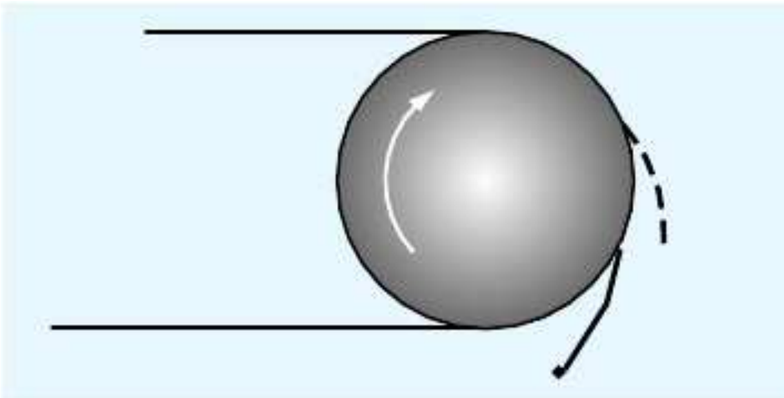


Figure 22
Tangential belt cleaner.

The recommended scraping angle of a tangential belt cleaner is thirty degrees from the tangent as shown in Figure 23.

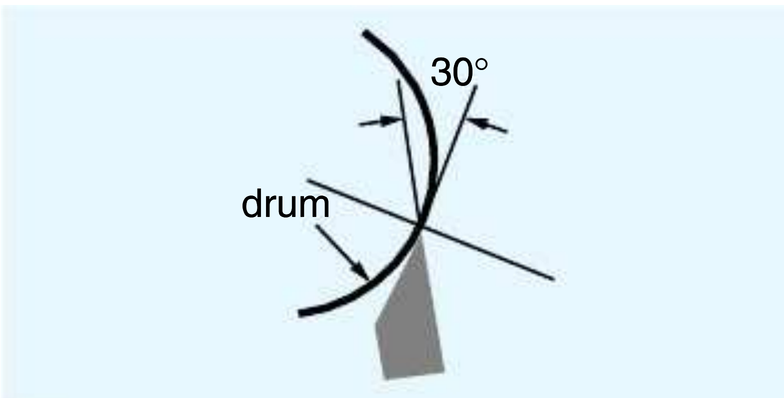


Figure 23
Angle of tangential belt cleaner.

A **radial belt cleaner,** as shown in Figure 24, should be used for a steel belt with a butt strap joint. This type of belt cleaner should also be used when the belt direction is occasionally reversed, or when the scraper blade is made of a soft material, such as rubber or felt.

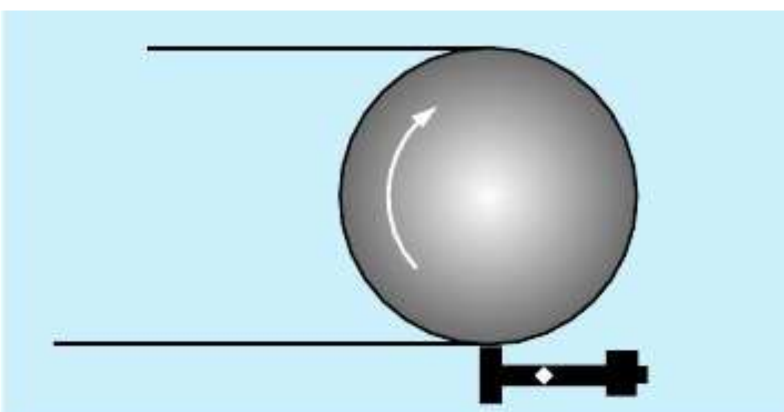


Figure 24
Radial type belt cleaner.

Belt cleaners should not harm the belt in any way. Therefore, in the proper choice of the material for the cleaner blade, the steel belt grade, the product handled, and the temperature of the belt and product should be considered.

A general rule is that the scraper blade for a carbon steel belt should be as hard as the belt. The scraper blade for a stainless belt should be softer than the belt.

For carbon steel belts, scraper blades are normally made of hardened and tempered carbon steel. For stainless steel belts, scraper blades are normally made of brass, polyamide, PVC, or other reinforced plastic material.

Relatively hard scraper blades should be used when handling abrasive products to prevent the hard particles from becoming embedded in the blade.

The scraper blade should not be pressed harder than necessary against the belt. Undue pressure can cause scratches that may deform and harm the belt. An average pressure of 100 N/m of scraper width for tangential cleaners gives sufficient cleaning pressure. For radial type belt cleaners, double the above pressure.

The belt cleaner should be designed so that the scraper blade does not apply pressure on the belt edges. For plain belts, a scraper blade made of metal should be 30–50 mm narrower than the belt width. For true-tracking belts, which have less lateral movement, the blade width can be 6 mm narrower than the belt width. When applying a belt cleaner to a belt with attached retaining strips on the product side, there are other considerations, and these should be discussed with the belt manufacturer.

Belt guiding devices

Theoretically, a steel belt operating under normal conditions, should center itself on the terminal drums or sheaves of a properly designed conveyor. However, it has been found that the terminal guiding effect is limited to a distance of about ten times the belt width. Therefore, a conveyor which falls outside of this ratio, needs some type of guiding devices. These can be either static devices which forcibly limit the lateral movement, or active devices which can increase the force needed to correct the lateral movement. In either case, the devices must not harm the belt edges.

Either static or active belt guides should be located in the top and bottom strands at a distance from both terminals of ten times the belt width. For longer conveyors, additional guides should be located at intervals of 15–20 m.

Keep in mind that all steel belts have a certain deviation from a theoretical straight line. Therefore, it is a good rule to provide sufficient clearance for the belt edges in the conveyor framework, tensioning devices, belt guides, etc. A clearance of 100 mm for each belt edge is suggested for plain belts and 25 mm for true-tracking belts.

When closer belt guiding is required, an automatic belt guiding device should be used.

Static belt guides are often made of wood, brass, or plastic, in the shape of bars with a minimum length of 500 mm. The height of the bars should exceed the anticipated sag of the belt between support idlers, see Figure 25. An alternative to the bars is the use of stationary rollers made of hardened steel.

Figure 25
Static belt guide.

Active belt guides are an arrangement of rollers positioned on a spring-loaded frame, see Figure 26. These rollers should have a minimum diameter of 50 mm for belt speeds up to 90 m/min, and approximately 150 mm for higher speeds.

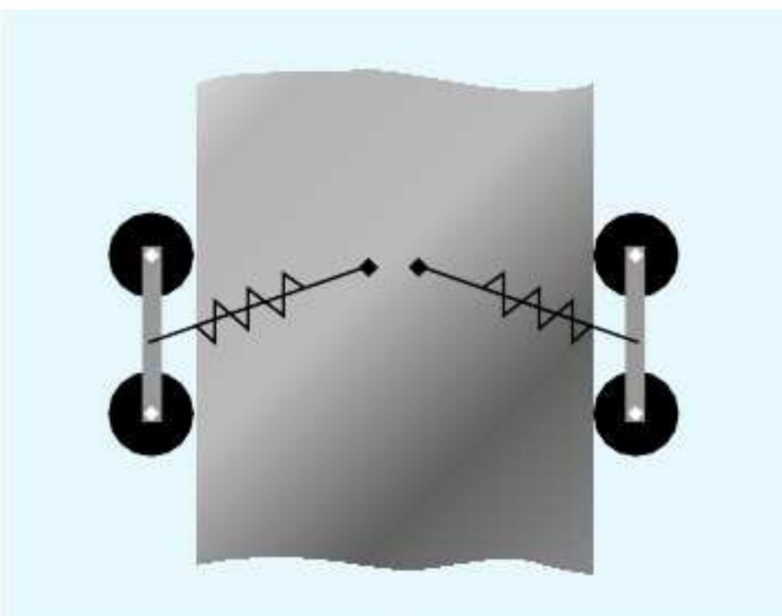


Figure 26
Active belt guide with pairs of spring-loaded rollers.

The springs should have a pre-tension of 100–150 N. Surface hardened rollers are recommended.

Another active guiding device is the adjustment of tension on one side of the belt. If the steel belt moves to one side due to a disturbance from the operation, the tension can be increased on the side towards which the belt moved. The steel belt moves to the side of the lowest tension. By adjusting the terminal bearings, the belt position can be corrected. Usually, only small adjustments are required on the terminal bearings, because the belt is very sensitive to changes in tension.

The movement of the belt can be kept under close surveillance or monitored by the use of a sensor such as a limit switch. The limit switch activates a signal, such as a light or horn, to alert the operator to make manual corrections.

If frequent tracking disturbances are anticipated, or if only minimum lateral belt movement is permitted, an automatic belt tracking device should be used. In automatic systems, the bearing adjustment is accomplished by letting the signal from the limit switch be transmitted to an electric control unit, which in turn activates an electric drive unit which moves one of the terminal bearings, see Figure 27. As an alternative, the sensor can be a pneumatic or hydraulic gauge which in turn acts on a pneumatic or hydraulic cylinder attached to the terminal bearing.

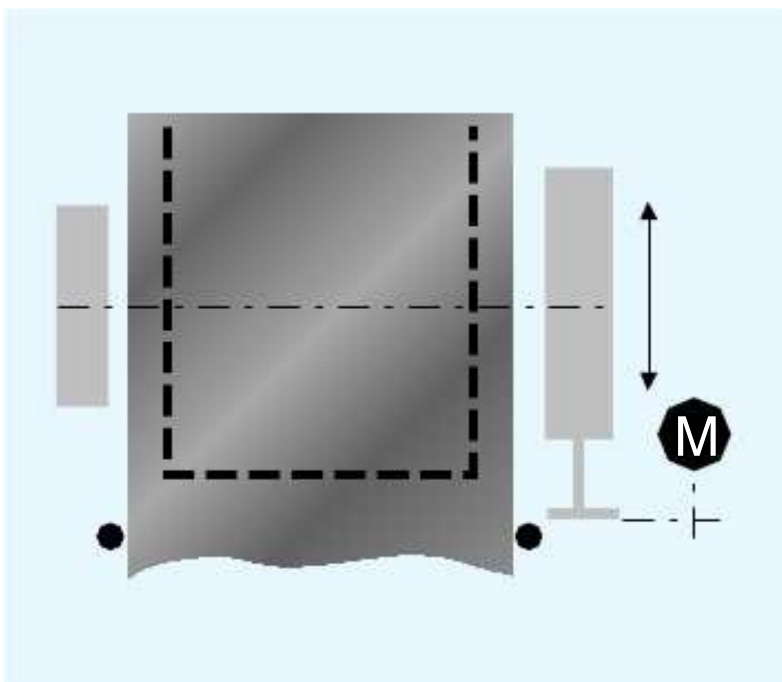


Figure 27
Automatic belt tracking device.

Steel belt maintenance

The service life of a steel belt depends not only on correct design, installation, and operating conditions, but also on proper belt maintenance. Periodic inspections of the belt and the mechanical parts that come in contact with the belt are recommended. Some of the more common items to inspect are as follows:

Deformed or burred edge.

The belt should be closely inspected to make sure it has not come in contact with any conveyor structures, see Figure 28. Such contact can cause belt edge deformation and burrs. The burrs should be removed by filing.

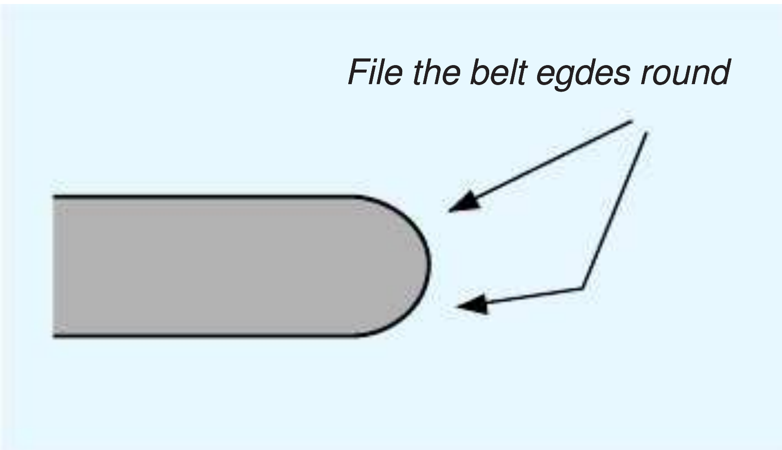


Figure 28
Proper belt edge.

Edge waves and deformations.

In severe cases of improper tracking or uneven pressure from the belt cleaner or scraper, local edge waves may occur in the belt. It is important to remove these deformations. Removing the deformations by cutting is illustrated in Figure 29.

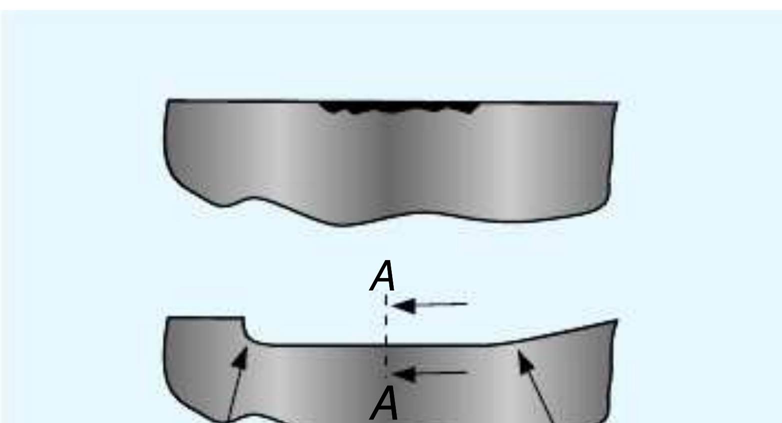


Figure 29
Removing minor deformations by cutting.

Temporary waves and blisters.

Uneven temperatures across the belt can cause waves or blisters to appear in the steel belt. These normally disappear when the belt returns to a uniform temperature condition.

Loss of flatness.

If the entire belt loses its flatness, accumulated foreign matter on drums or belt supports may be the cause. Heavy scratching can also cause belt deformations. Worn cleaner blades can cause surface scratches. Careless loading of an abrasive product which slides when first loaded can cause surface scratches. Where a product impacts the belt on loading, particularly a stainless belt, the belt may deform.

In many cases, the belt can be flattened by trained service people using special service equipment.

Wear on riveted joints.

Normal wear on a riveted joined belt may cause the rivets to loosen. The loose rivets should be removed and replaced with new ones. If necessary, remake the whole joint.

Cracks on welded joints.

Hairline cracks on a welded joint are normally a sign of fatigue. The weld should be repaired or remade.

Cracks in the belt.

Hairline cracks in the belt are also normally a sign of fatigue. If the cracks appear at the belt edge, they may be repaired by welding, or by cutting away a crescent shaped piece and filing the edges round, see Figure 30.

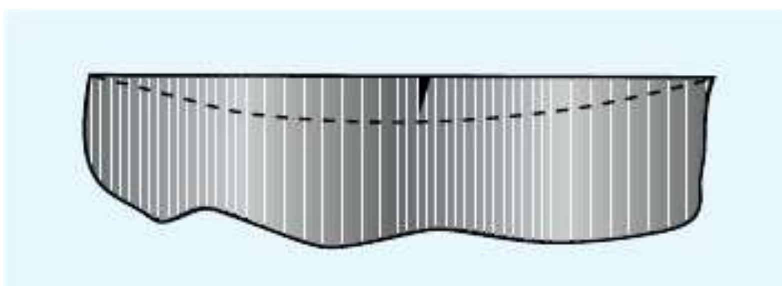


Figure 30
Small edge crack removed by cutting.

If a crack develops from the belt edge and becomes too long for belt edge trimming, and welding is not available for repair, a small hole can be drilled at the end of the crack to prevent further spreading, see Figure 31. Thoroughly remove burrs at the drilled holes.

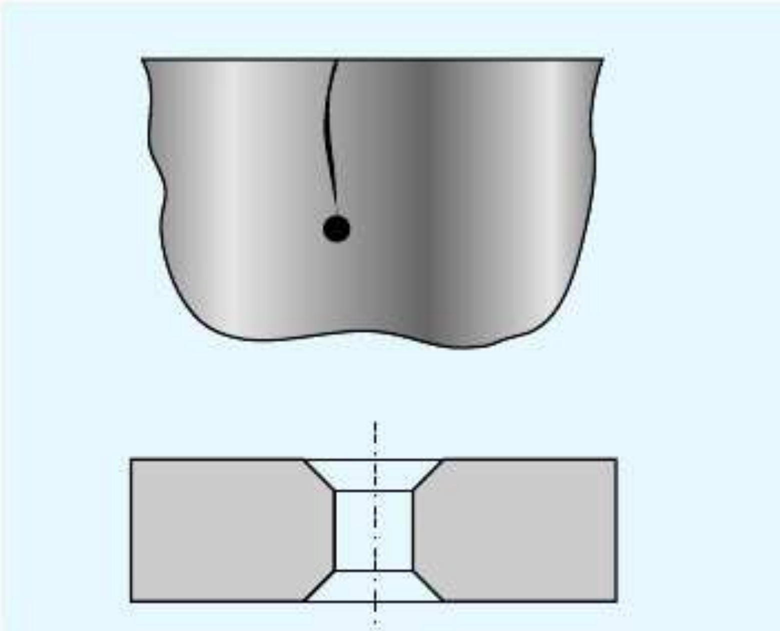


Figure 31

Stopping a crack by "stop drilling".

If a crack appears that is not at the belt edge, small holes may be drilled at each end of the crack to prevent spreading, see Figure 32.

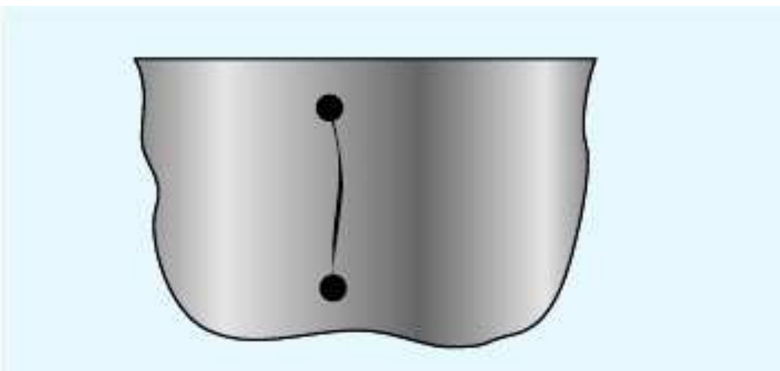


Figure 32

Crack stopped by drilled holes at each end.

The measures described to handle minor fatigue cracks by "stop drilling" have to be considered temporary. If the damage is too large, the best solution may be to splice in a new belt section. The length of such a section should not be shorter than 2/3 of the terminal drum/sheave circumference to prevent two joints from being on the terminal simultaneously.

Rollers.

Check all rollers and other rotating parts to make sure they move freely and that wear-parts are not unduly worn.

Cleaning belts

Periodically, it is advisable to clean a steel belt thoroughly.

The food industry often uses hot water and steam to clean steel belts at routine time intervals. When applying heat, if the belt is running, apply the heat uniformly across the entire width to prevent mistracking. True-tracking belts and belts with bonded retaining strips should not be heated above the max recommended temperature for the bond.

After a carbon steel belt has been cleaned, it should be greased with an acid-free oil to prevent rusting, unless it is immediately put into operation.

Power calculations

Once the steel belt conveyor application is known, the driving power requirements (pull) can be calculated. The pull necessary to overcome friction is generally made up of the following:

P1 = pull required to overcome friction in support rollers.

P2 = pull required to overcome the extra resistance when the belt is supported by rubber-lagged rollers.

P3 = pull required to overcome friction from slide supports.

P4 = pull required to accelerate the load.

P5 = pull required to overcome friction at scrapers when discharging the load.

P6 = pull required to overcome friction in terminal bearings.

P7 = pull required to overcome friction at safety devices.

P8 = pull required to lift the load on inclined conveyors. This can be a negative value.

P9 = pull required to overcome friction of side skirts.

The components listed above can be estimated with sufficient accuracy, using the following formulas which are expressed in Newtons (N).

The variables in the formulas below are expressed in kilograms (kg). The conversion from kg to N have been made in the constants.

$$P1 = 0.1\,[L1 + W1 + (z \times r)]$$

where **L1** is the load on belt (kg/m) times the rolling length with load (m); **W1** is the weight of belt (kg/m) times the total rolling length (m); **z** is the number of rollers; and **r** is the weight (kg) of the rotating part of each roller.

$$P2 = 0.16\,(L2 + W2)$$

where **L2** is the load on belt (kg/m) times the rolling length with load (m) on rubber rollers and **W2** is the weight of belt (kg/m) times the total rolling length (m) on rubber rollers.

$$P3 = k_1\,(L3 + W3)$$

where k_1 is 4 for steel, metal, or plastic supports and 6 for wood supports; **L3** is the load on belt (kg/m) times the sliding length with load (m); and **W3** is the belt weight (kg/m) times the total sliding length (m).

$$P4 = k2 \times q_g$$

where k_2 is 7 for belt speeds under 60 m/min and 9 for belt speeds above 60 m/min. and q_g is the product load (kg/linear m) on the belt.

$$P5 = q_g\,(15Bm + 2)$$

where q_g is the product load (kg/linear m) on the belt and **Bm** is the belt width (m). For sticky products, this value should be doubled.

$$P6 = 0.01 \times Q$$

where **Q** is the tension force (N) related to belt tension, generally 20xBxt (see page 8) where **B** is the belt width (mm) and **t** is the belt thickness (mm). The coefficient 0.01 applies only to roller bearings or ball bearings.

$$P7 = 500 \times Bm$$

where **Bm** is the belt width (m).

$$P8 = 10 \times q_g \times h$$

where q_g is the product load (kg/linear m) on the belt and **h** is the height lifted (m). The value is negative for a declining conveyor.

$$P9 = k_3 \times L4$$

where k_3 is 0.5 when the side skirts have the function of preventing material falling off the belt and normally do not touch the material, is 1.0 when the side skirts touch the material, and is 2.0 if the material is sticky. **L4** is the product load on the belt (kg/linear m) times the length of side skirts (m).

The total pull, P_T, is the sum of **P1** to **P9**. It is suggested that a safety factor of 1.2 be applied to the total pull.

The required effect, **E**, in kilowatts (kW) on the drive terminal shaft can thus be calculated as follows:

$$E = \frac{1.2\,P_T \times V}{60000}$$

where P_T is the total pull (N) and **V** is the belt speed (m/min).

The effect, **E**, arrived at is thus the net effect on the terminal shaft. This value has to be adjusted by the various service factors which apply to the selected drive unit and transmission. Consideration must also be given to the capacity of the drive unit to withstand the initial resistance which occurs during start-up of belts with heavy loads.

Typical steel belt conveyor calculation (example):

Data: Sorting conveyor.

c-c distance:	150 m
Load:	60 kg/m
Speed:	130 m/min
Dimensions:	1000 x 1.2 mm
Grade:	1300C
Shifts:	Two per day

Weight of belt (kg/m):

Density: 7.85 g/cm^3

(1000x1000x1.2x7.85)/(10^{3}x1000) = 9.42 kg/m

Idler spacing: (top strand)

See nomograph (fig 18):

Weight of belt 9.42 kg/m

Load on belt 60 kg/m 1850 mm idler spacing.

Note that maximum parcel weight has not been considered.

Number of idlers: 150000/1850 = 81

Number of rollers in return strand: on page 12 maximum recommended distance between rollers is 8 m. But between tension drum and first roller the distance should be no more than 3 meters. That gives (150-3)/8+1= 20 rollers in return strand.

Power calculation

PT= P1+P2+P3+P4+P5+P6+P7+P8+P9

P1 = Friction in support rollers.

P1= 0.1 [(L_1+W_1+(Z x r)]

Carrying strand:

L_1 = Load on belt for belt carried by rollers = 60 x 150 = 9000 kg

W_1 = Weight of belt times length of belt = 9.42 x 150 = 1413 kg

Z = Number of idlers = 81

r = weight of rotating part of roller = 30 kg

Inserted values give **P1** = 1284 N

Return strand:

L_1 = 0

W_1 = 1413 kg

Z = (150-6)/8+2 = 20

r = 30 kg

Inserted values give P_1 = 201 N

$P1_{tot}$ = 1284 + 201 = 1485 N

P2 = Rubber lagged rollers

P2 = 0.16 (L_2 + W_2)

L_2 = Load on belt times the rolling length = 60x150 = 9000 kg

W_2 = Weight of belt times the rolling length = 9.42x150x2 = 2826 kg

Inserted values gives **P2** = 1890 N

P3 = Friction from slide supports = 0

P4 = Pull to accelerate the load

P4 = k_2 x q_g

k_2 = 9 (belt speed >60m/min)

q_g = 60 kg/m

Inserted values give **P4** = 540 N

P5 = Friction from scrapers during discharge

P5 = q_g (15 Bm + 2)

q_g = 60 kg/m

Bm = Belt width = 1.0 m

Inserted values give **P5** = 1020 N

P6 = Friction from terminal bearings

P6 = 0.01 x 20 x B x t

B = Belt width = 1000 mm

t = Belt thickness = 1.2 mm

Inserted values give **P6** = 240 N

P7 = Friction from safety devices

P7 = 500 x Bm

Inserted values give **P7** = 500 N

P8 = Pull from incline = 0

P9 = Friction from side skirts

P9 = k_3 x L_4

k_3 = 0.5

L4 = 150x60 = 9000

Inserted values give **P9** = 4500 N

Add all contributions and the sum will be:

PT = 10175 N

Drive shaft power

$E = (1.2 \times P \times V)/60000$

Inserted values give $E = 26.5$ kW.

Efficiency for transmission must be added to arrive at required motor power.

Belt tension

According to the formula on page 8 the minimum belt tension should be $Qp = 3.3 \times PT$

This gives $Qp = 3.3 \times 10175 = 33600$ N

Note that Qp is larger than required minimum pre-tention (Q).

Use 33600 N as pre-tension in this case.

Determine tensile stress:

$\sigma = (Q_p/2 + PT)/(1000 \times 1.2) = 33600/2 + 10175/(1000 \times 1.2) = 22.5$ MPa

Determination of drum diameter

See figure 8: $(v \times s)/L = (130 \times 2)/150 = 1.73$

This gives K = 800

Minimum drum diameter 800 x 1.2 = 960, say 1000 mm.

Belt sag in carrying strand

$f = [5 \times 10^3 (q + q_g) \times a^2] / (4 \times S)$ see page 10.

$S = Q_p/2 = 16800$ N, $a = 1.85$ m

$q = 9.42$ kg/m

$q_g = 60$ kg/m gives $f = 18$ mm

Note that the belt sag could be reduced by higher pre-tension or shorter idler spacing.

Standard dimension range for Sandvik steel belts

Sandvik 1000SA

Thickness mm (inches) \ Belt Width mm (inches)	600 (24)	800 (32)	1000 (40)	1200 (48)	1400 (55)	1500 (59)	1560 (61)	3000* (118)	4500** (177)
1.0 (0.040)	●	●	●	●	●	●	●	●	●
1.2 (0.048)				●	●	●	●		

Sandvik 1200SA

Thickness mm (inches) \ Belt Width mm (inches)	200 (8)	300 (12)	400 (16)	500 (20)	600 (24)	800 (32)	1000 (40)	1200 (48)	1400 (55)	1500 (59)	1560 (61)	2080 (82)	3000* (118)	4500** (177)
0.4 (0.016)	●													
0.6 (0.024)	●	●	●	●	●									
0.8 (0.032)		●	●	●	●	●	●							
1.0 (0.040)				●	●	●	●	●	●	●	●		●	●
1.2 (0.048)						●	●	●	●	●	●		●	●
1.6 (0.063)						●	●	●	●	●	●	●	●	●
2.0 (0.079)										●	●	●	●	●

Sandvik 1050SM

Thickness mm (inches) \ Belt Width mm (inches)	600 (24)	800 (32)	1000 (40)	1200 (48)	1400 (55)	1500 (59)	1560 (61)	3000* (118)	4500** (177)
0.8 (0.032)	●	●	●						
1.0 (0.040)	●	●	●	●	●	●	●	●	●
1.2 (0.048)		●	●	●	●	●	●	●	●

Sandvik 1700SA

Thickness mm (inches) \ Belt width mm (inches)	100 (4)	200 (8)	300 (12)	400 (16)	500 (20)	600 (24)	800 (32)	1000 (40)
0.2 (0.008)	●	●	●	●	●	●		
0.3 (0.012)	●	●	●	●	●	●	●	
0.4 (0.016)	●	●	●	●	●	●	●	●
0.6 (0.024)							●	●

Sandvik 1150SM/1650SM/1500SM

Thickness mm (inches) \ Belt Width mm (inches)	800 (32)	1000 (40)	1200 (48)	1400 (55)	1500 (59)	1560 (61)	3000* (118)	4500** (177)
0.8 (0.032)		●						
1.0 (0.040)		●	●	●	●	●	●	●
1.2 (0.048)		●	●	●	●	●	●	●
1.6 (0.063)		●	●	●	●	●	●	●
1.8 (0.071)		●	●	●	●	●	●	●
2.0 (0.079)		●	●	●	●	●	●	●
2.3 (0.091)				●	●	●	●	●
2.7 (0.106)				●	●	●	●	●
3.0 (0.118)				●	●	●	●	●

Sandvik 1300C/1320C (no stock standard)

Thickness mm (inches) \ Belt Width mm (inches)	200 (8)	300 (12)	400 (16)	500 (20)	600 (24)	800 (32)	1000 (40)	1200 (48)	1250 (49)	2380* (94)	3560** (118)	4500*** (177)
0.6 (0.024)	●	●	●									
0.8 (0.032)		●	●	●	●	●						
1.0 (0.040)		●	●	●	●	●	●	●		●	●	●
1.2 (0.048)					●	●	●	●	●	●	●	●
1.4 (0.055)					●	●	●	●	●	●	●	●

Sandvik 1100C: Belt widths 1250-1500 mm (49-59 inches), thickness 1.2 mm (0.048 inch).

Sandvik 1500SM: Belt widths 1200-1560 mm (47-61 inches), thicknesses of 1.2, 1.8, 2.7 and 3.0 mm (0.048, 0.071, 0.106 and 0.118 inch).

Stock standard dimensions indicated within thick lines.

* max. width with one longitudinal weld.

** max. width with two longitudinal welds.

*** max. width with three longitudinal welds.

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