

If you operate a conveyor system that includes turns, this article will reveal the best practices!

Horizontal curve technology or ‘turns’: design, calculation, control.

Terminology/description:

Precise terminology is required to describe the subject.

In everyday language, the term ‘horizontal curve’ is used for conveyors with a ‘winding’ longitudinal profile when viewed from above (**XZ** plane); they are also known as ‘curvilinear conveyors’. By comparison, a conveyor comprising horizontal, ascending and descending straight sections, with convex curves (where the centre of the curve is below the conveyor) and concave curves (where the centre of the curve is above the conveyor), has a perfectly ‘straight’ profile when viewed from above; such a conveyor is typically represented in the **XY** coordinate plane, to ensure it is fully descriptive.

The difficulty here lies in the variety of so-called ‘horizontal’ curves; it would be more accurate to refer to curves in the ‘horizontal plane’ of **XZ** coordinates. But if this horizontal plane is inclined, how are we to understand the definition of a ‘horizontal curve’? To clarify this point, **C3 Expert** uses the term ‘turn’ by analogy with turns in roads, which may have a simple, constant profile or a complex one combining sections with different gradients (inclines) within the same turn.



*Conveyors that are straight or feature one or more convex and/or concave curves have a longitudinal profile (in elevation) that is clearly visible on an **XY** coordinate plane.*

*Where the conveyor comprises simple, constant **Turns**, these are visible in an **XZ** coordinate plane*

*When turns include concave and/or convex curves, a **3D** representation is required to distinguish the subtleties of the alignment.*

Calculation constraints for 'turns'

To calculate a Turn, it is necessary to know the various parameters and their values that characterise it.

In the case of a conveyor project, the calculation has the flexibility to propose different solutions; however, it may have to comply with certain constraints, for example, a layout that imposes a curve radius within a given geographical envelope.

In the case of an existing conveyor, the calculation is constrained by the specified values of numerous parameters; for example, the radius of each turn, the model and pitch of the idler, the location of the drive units, the belt reference and speed, etc.

'Turn' profiles

- 1) A simple turn, with consistent parameters
 - a. In a horizontal plane, ascending, descending
 - b. With a single centre and a single radius.

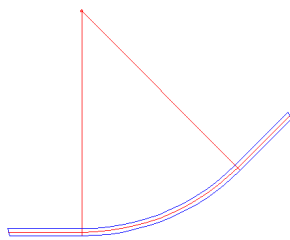


Fig. 1: 1 centre, 1 radius

- 2) Simple turn, with variable parameters
 - a. In a horizontal plane, ascending, descending
 - b. The curved section has several different centres and/or radii.

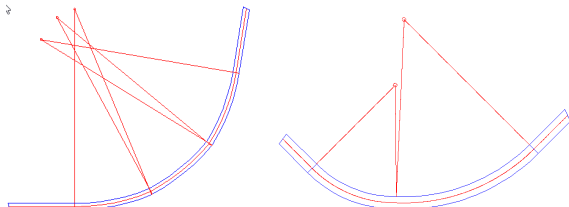


Fig. 2.1: 3 centres, 1 radius ⁽¹⁾ Fig. 2.2: 2 centres, 2 radii

- 3) Complex turn combining one or more curves in the vertical plane (concave, convex) with the same centre and turn radius or with several centres and/or turn radii
 - a. This type of profile is among the most complicated to calculate, as it depends on significant variations in tension across the belt throughout the turn, depending on whether the mass of the material being handled acts as a resistive or driving force. For this type of turn, it is necessary to calculate in advance the minimum and maximum tensions for all subsections of the turn
- 4) A series of turns in alternating directions, either without an intermediate straight section or with a very short intermediate straight section (e.g. a length equal to 1 pitch between two consecutive supports).

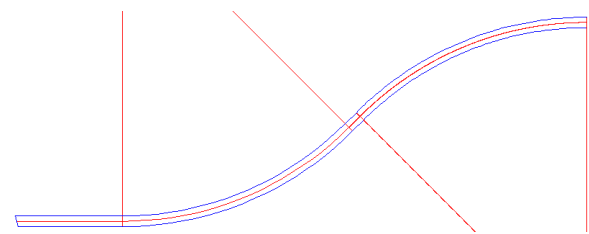


Fig. 4: Two successive turns in opposite directions

Layout of the carrying and return strands

The alignment of the turn, the carrying strand and the return strand are:

- 1) Parallel and superimposed
 - a. The radius of the two curve sections – the carry strand and the return strand – is the same;

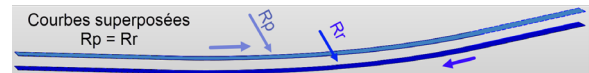


Fig. 5.1: Curves // superimposed main and return runs

- 2) Parallel and side-by-side
 - a. The radius of the return section is increased or decreased by the distance separating it from the longitudinal axes of the two sections, depending on the direction of the turn in question (right-hand or left-hand).

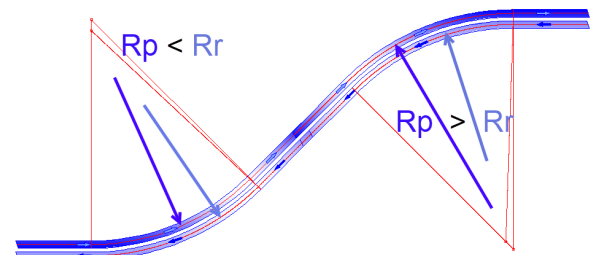


Fig. 5.2: Curves // drive and return sections side by side

- 3) Non-parallel
 - a. Here, there are two separate alignments.
- 4) No 'return strand'.
 - a. This scenario could theoretically exist and would be equivalent to a 'ring' or a loop. The practical benefit of such a layout is unlikely.

Idlers / Design

There is a wide variety of idlers designed to support the belt around turns; they are categorised according to three operating principles:

- 1) The idler is **fixed**, in the sense that, once adjusted, it does not move or shift – either automatically or under control – to vary the angles of inclination and/or orientation. This is the most common type.
- 2) The idler is **'pendular'** (glider), either single or double (carrying rollers, carrier + return). The principle relies on the idler's inclination under the pressure of the belt, whose path tends towards the chord of the curve due to its tensile forces. Thus, the belt presses against the idler's rollers, causing it to tilt whilst pivoting on an axis above the swing arm, up to a limit value defined by a stop.
 - a. This model accommodates the 'variable inclination' of the idler. The angle of orientation of the idler, in this case, is fixed but is affected

- by the 'operational play' of the joint, both when new and when worn;
- This model allows for a shorter turning radius than the fixed model, provided all other turning characteristics (throughput, other factors) remain identical;
 - The limit angle is defined by the angle at which the material being handled begins to slide down the slope during the dynamic phase, its flow rate, etc.;
 - The performance of this model may be advantageous when comparing the manufacturing cost of the swing arms and the conveyor structure with the effectiveness in controlling the belt's trajectory through the turn:
 - The effectiveness of the inclination is determined by calculation and must be combined with the possible but fixed orientation angle of the supports (0° to 3° ⁽²⁾).

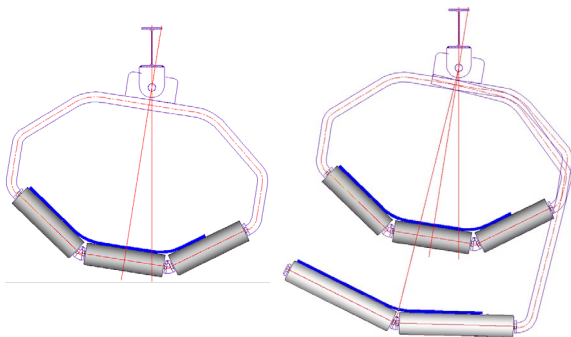


Fig. 6.1: Single glider arm Fig. 6.2: Double glider arm

- The idler is 'variable-orientation' and forms part of n sets of n idlers in the turn. The automatic orientation depends on the trend of the belt's trajectory at time t ,
- t , towards the inside or outside of the curve.

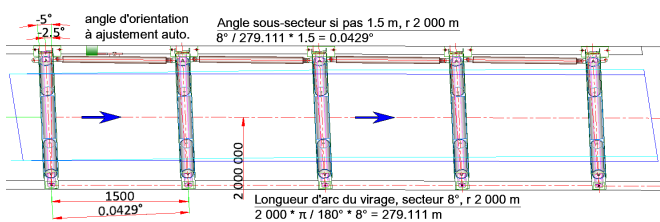


Fig. 7: Controlled variable-orientation system

- This model prioritises the steering force of the rollers, which is considered 1st factor in controlling the belt's trajectory. According to the calculations, its performance is complemented by a fixed angle of inclination;
- This model performs well as the orientation angle of each set of n idlers adjusts automatically to counteract belt drift, depending on the load, belt tension, etc., in the relevant turning sub-section;
- Under these conditions, the system can handle very large variations in load on the belt and/or belt tension. This allows for a shorter turning radius compared with the fixed-support model;

- For this variable-orientation system, it is recommended to use a series of n idlers compatible with load variations, so that the belt does not have time to deviate from its calibrated path when there is a sudden change in load and/or tension (empty, full load, empty);
- The number of idlers in the series is also determined by the force required to manoeuvre their orientation easily;
- This system can also function with a reduced number of supports, which in itself contributes to savings in consumables and energy consumption; this point should be taken into account in the cost-benefit analysis.

Idlers / Orientation angle

In the article above, the swing-arm support model, which prioritises the angle of inclination, is presented before the *variable-orientation* model, as it is more common; however, in terms of efficiency, it is the angle of orientation that takes precedence, supplemented by inclination where necessary (as in the paper industry) ⁽³⁾.

Orientation (direction) angle / variable system:

Throughout the curved section, an orientation angle may be assigned to each idler(s) based on its reference geometric position, which is perfectly perpendicular to the conveyor axis;

- This **initial and reference position** of the idler coincides with the curve radius, passing through the axis of the support's rollers ($= 0^\circ$). Should the steering forces generated by the rollers prove insufficient to control the belt's path, including the maximum permissible offset, this orientation angle must be increased, without, however, exceeding 3° .

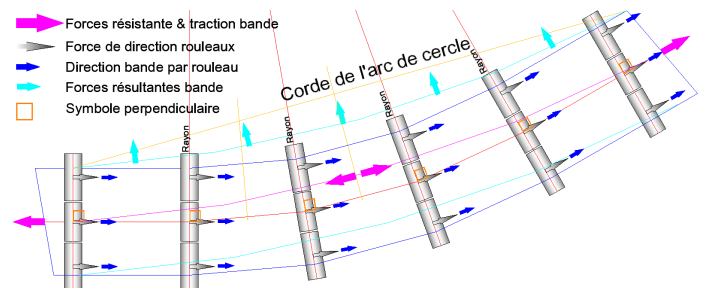


Fig. 8.1: Idlers aligned with each turning radius

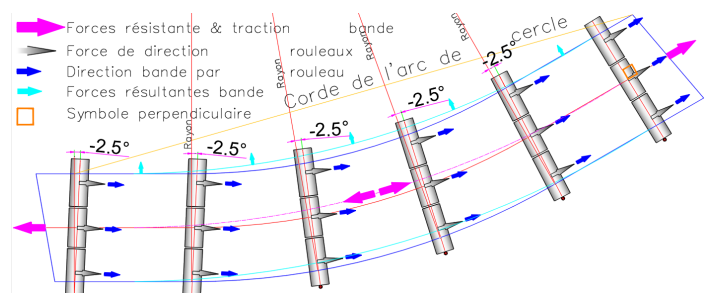


Fig. 8.2: Idlers oriented at -2.5° (ref. roller axis); if the orientation angle of 0.0° is insufficient to limit the extent of the belt's deviation

- $-x^\circ$: orientation opposite to the direction of the turn
- $+x^\circ$: orientation in the direction of the turn

- If the turn comprises several centres of curvature, each sequence of idlers, associated with a sub-section of the turn (same centre and radius), defines an initial reference position for the idlers of 0° at the radius passing through the roller axis, followed by an orientation angle, if necessary $> 0^\circ$.
- If the turn has a complex profile, including concave and/or convex curves, within a generally variable horizontal profile (ascending and descending), the orientation angle of the idlers may differ from one to another, even for the same curve radius within the sub-section. For a given curve radius, it is the variation in tension ⁽⁴⁾ in the belt and the load ⁽⁴⁾ (belt, belt + product), in each sub-section of the turn, that defines the orientation angle of the idlers, step by step, depending on whether the product being handled generates a driving or resisting force applied to the belt.

Principle of the orientation (direction) angle:

In a **straight** section of the conveyor:

- The pulling forces and the resisting forces must be parallel to the conveyor axis and symmetrical to it in magnitude, so as not to create a torque that would cause the belt to shift.
 - In turn sections, this balance of forces in the belt (left and right tension $\neq$) does not exist. This difference in tension increases in proportion to the width of the belt, all other factors being equal.
- The rollers in contact with the belt must generate a guiding force that is perfectly parallel to the conveyor axis, so as not to deflect its path. This requires idlers that are perfectly perpendicular to the conveyor axis.

In a **'turn'** section:

- The tensile and resistive forces acting on the belt tend to pull its path back from its curved trajectory (arc of a circle) towards the **chord** of the circular sector, disregarding the friction forces between the belt and the rollers.
- Consequently, forces must be generated and applied to the belt to counteract the natural tendency of its path to align with the chord of the circular sector's arc (see Fig. 8.2).
 - **Forces 'within' the belt:** this approach involves minimising both the tensile forces and the resistive forces, not only in the turn to be calculated but also across all other sections of the conveyor, including the drive run and the return run. This 1st phase involves a thorough review of all the conveyor's parameters, such as:
 - The belt speed;
 - The **actual** maximum **throughput** to be achieved (to be managed upstream);
 - The **actual nominal throughput**, i.e. the **constant throughput regularly achieved** by the operator, which is often lower than the nominal throughput specified in the original specifications;

- The mass of all components, such as drums and rollers, by reducing their number (those that are unnecessary); with potential savings of 30 per cent, 50 per cent and more.
 - The mass of the belt through a lighter design, such as the thickness of the cover layers and the weight of the carcass, using an aramid or polyester chain instead of a steel one;
 - Optimal distribution of drive units, sometimes involving the addition of a 'booster' (a long conveyor whose length is extended, e.g. behind a tunnel boring machine) and a 'retrobooster' with generators in the case of a long downhill section, in order to regulate the belt tension before the turn.
- **Forces 'on' the belt:** these are the steering forces generated by the rollers of turn, in contact with the belt's running surface; thus, one should say 'under' the belt:
- **Reminder:** if, in the turn, each roller is perpendicular to the conveyor axis because it is aligned with the radius of the curve passing through the axis of the rollers, for the short section between idlers ($\frac{1}{2}$ pitch before, $\frac{1}{2}$ pitch after the idler in question), then the steering forces they generate help to maintain the belt's trajectory.

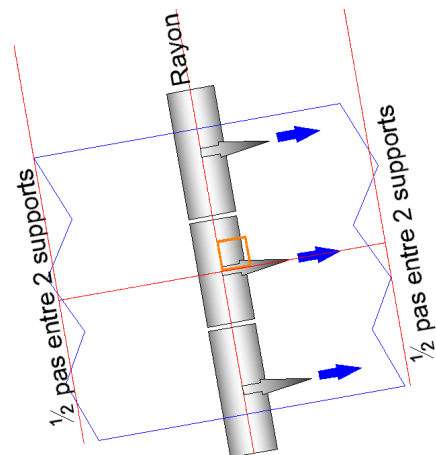


Fig. 9: Curved section between two virtually straight idlers
E.g.: $[48.14^\circ/1680] = 0.028^\circ$ or 1680 = number of idlers in the turn

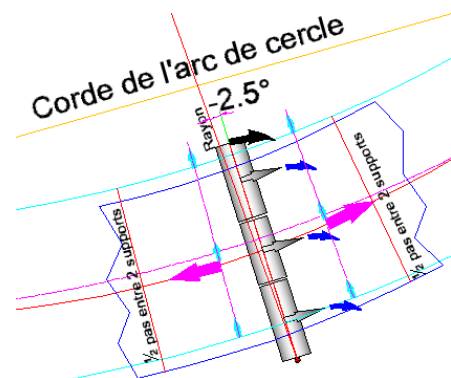


Fig. 10: Example of guidance of the idlers to maintain the belt's path through the turn

- To generate an additional steering force on the belt, **if necessary**, which counteracts the tensions pulling it towards the chord, each idler must be oriented identically so that it generates an opposing steering force that pulls the belt towards the outside of the curve.
- The calculation must determine an orientation angle that is compatible with all phases of the conveyor's operation, taking into account variations in load and tension applied to the belt at the start and end of the calculated turn... Whilst bearing in mind that the tensions in the belt, both upstream and downstream of the turn, also influence the calculation results within the turn.
- Due to the delicate balance of the belt's trajectory, resulting from the various parameters affecting it, it may be advantageous to adopt a design that allows for a greater drift amplitude than the standard value; for example, for an 800 mm wide belt, allowing a drift of 100 mm instead of the standard 40 mm.

This wide range of load and tension applied to the belt favours **variable-orientation idlers**, as the orientation angle adapts to the load and tension (see Fig. 7). Controlling the orientation angle of the idlers makes it possible to reduce the belt drift amplitude compared with that of the **fixed-orientation** model. Consequently, the mass of the side rollers can be reduced, as this system can make do with standard-length rollers (the same length as the centre roller of the idler).

The orientation angle of the supports is always in the opposite direction to that of the turn.

By analogy with the adjustment of idlers in straight sections, it seems that it is more effective to **opt for a low orientation angle**. Here, **the interaction** between the orientation and inclination angles of the idlers must be taken into account, with **sudden changes in equilibrium** depending on the values selected (according to our calculations).

Nevertheless, in practical terms, an orientation angle $> 0.0^\circ$ would be safer (e.g. 0.5° to 1.5°), given the difficulty of ensuring on-site that the orientation angle remains within a realistic adjustment tolerance which is too wide and could result in a negative orientation angle.

Idlers / Pinch angle

With the exception of the single-roller model (flat profile), all roller supports for turns can be manufactured **with** a pinch, as defined in ISO 1537. Whilst the model without a pinch makes good mechanical sense, the calculations must be able to account for the case with a pinch, even if this model is counterproductive in several respects (greater resisting force, etc.). According to the model for idlers with pinch, this can be **neutralised** by appropriate shimming beneath the base of the idler.

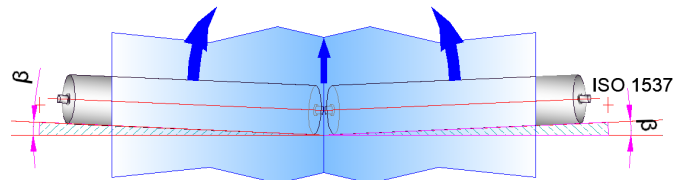


Fig. 11.1: Pinch angle (top view)
V-shaped idler with 2 sectors,

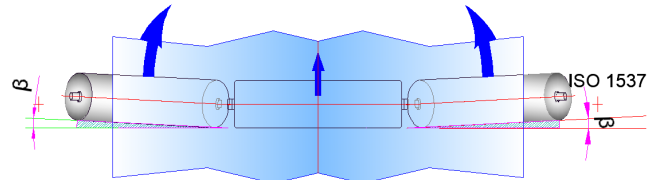


Fig. 11.2: Pinch angle (top view)
3-sector trough-type idler

A plausible explanation for the use of idlers with a pinch angle in turns lies in the pseudo-effectiveness of re-centring the belt with this type of idler when used in straight sections. The principle of the pinch angle lies in the orientation of the side rollers, which are in contact with the belt resulting in a **converging force** towards the conveyor axis, from right to left for the right-hand roller, and vice versa for the opposite roller. When the belt drifts to one side of the conveyor, the surface area of the side roller covered by the belt increases and the steering force generated by this contact increases proportionally, whilst for the opposite roller it decreases in inverse proportion. As for the centre roller, it is neutral and has no effect because it is supposed to be perpendicular to the conveyor axis. For straight sections, the idler's pivot point is its centre, which coincides with the conveyor axis. The same applies to V-shaped supports with two rollers.

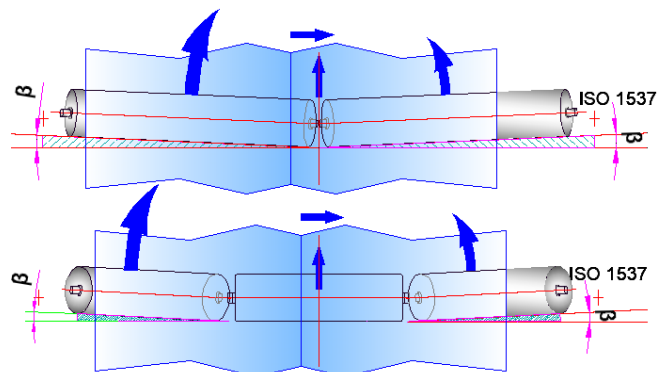


Fig. 12: Belt offset for some reason
The directional force Δ tends to re-centre the belt

In the case of turn sections, for practical (operational) purposes, the reference rotation point or pivot is, by convention, moved to one end of the idler (pivot = fixing bolt on the conveyor frame), on the right or left-hand side of the conveyor (whichever side is most convenient for metrology).

In this context, it is the idler, as a single unit, that is adjusted according to the calculated orientation angle. Another pivot point could be used as a reference, but this would complicate the transfer of the calculation results to the conveyor. In practice, in the **C3 Expert metrology system**, the calculated orientation angle is converted into different vector lengths (calibrated-length tubes) to construct a triangulation, starting from the reference side selected on the conveyor.

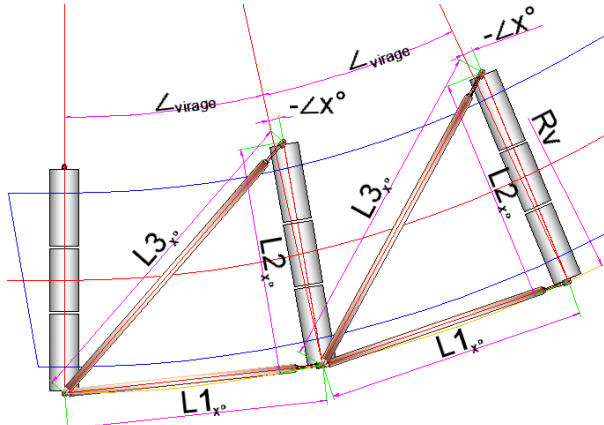


Fig. 13: Orientation angle converted into vectors

Idlers / Angle of inclination

Minimum–maximum inclination / extreme value:

Analysing extreme inclinations helps to understand how the idlers' inclination works. In reality, the solution will be an intermediate value, which depends on a balance between several conflicting parameters.

- **Idler inclination angle = 0°:**
Idler with 1 roller: in this case, only the belt/roller friction forces counteract the tensile forces in the belt to prevent it from following a path that tends towards the chord of the arc, regardless of the orientation forces of the idlers.
Idler with more than one roller: (V-shaped, three-sector trough, etc.) only the roller(s) on the inside of the turn exert a force that counteracts the tension in the belt driving it towards the chord of the arc. The effectiveness of this or these rollers depends on their trough angle.
- **Extreme angle of inclination of the idler = 90°:**
 Like a **wall** or a **stop**, the rollers of the idlers, when in a vertical position, counteract 100 per cent of the tensile forces in the belt to prevent it from following a path that tends towards the chord of the arc. Such a configuration requires the angle of orientation of the supports to be 0°, as any other angle would cause the belt to rise or fall within this vertical plane.
 Obviously, such an inclination reduces the permissible speed of the conveyor belt's carrying strand to 0 t/h. It could be considered for the return strand, provided it is always empty, but in this case the difficulty would be in maintaining the belt at a constant height throughout the turn; this would entail a great deal of complication for little benefit.

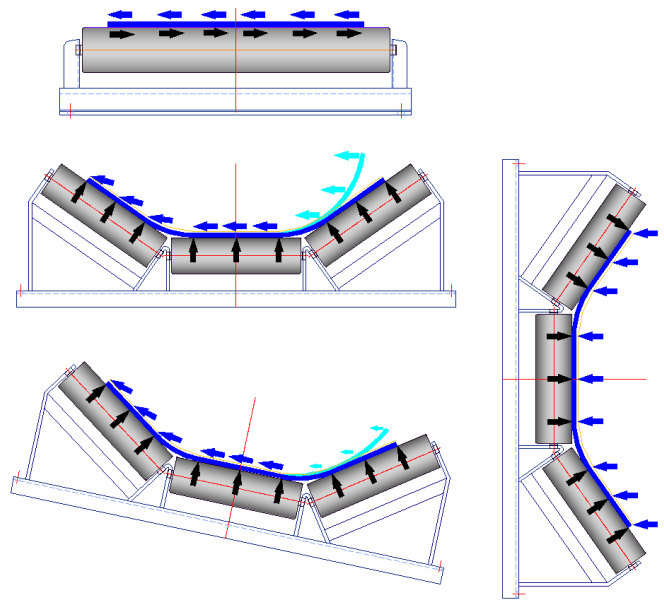


Fig. 14: Belt/roller forces in the Turn
 Fig. 14.1 & 2: Inclination angle 0°, flat, trough-shaped
 Fig. 14.3: Inclination angle 12°
 Fig. 14.4: Inclination angle 90°

Inclination angle of the idlers, 'loaded' strand:

The loaded belt strand is usually the carrying strand; occasionally, the return strand may also be loaded simultaneously and/or according to different operating sequences.

The maximum permissible inclination of the idlers for the loaded belt strand depends on the belt's cross-sectional profile (see the following section) and the product being handled, depending on:

- Throughput,
- Density,
- The particle size distribution
 - including the shape of the largest grains (whether flatter or spherical),
- The internal friction coefficient of the product, characterised by its angle of repose in the dynamic phase,
- The belt filling coefficient, considering a product:
 - spread out to the limit specified by ISO 5048 #7 ($L_{\text{product}} = [(\text{Belt width} \times 0.09) - 50 \text{ mm}]$)
 - concentrated in the centre of the belt's width.

Ultimately, the inclination of the idler must not cause the belt to overflow, resulting in the ejection of the product being handled.

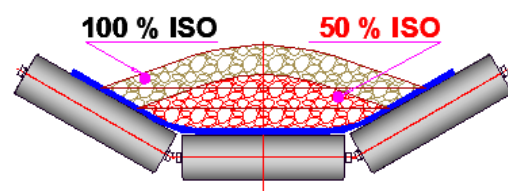


Fig. 15.1: Product spread-to-belt-width ratio

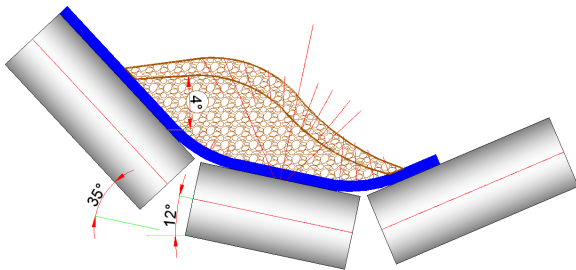


Fig.15.2: Conveyor incline angles, product spill angle

The transverse stiffness of the belt is a characteristic that influences the belt's stability when cornering; it determines the permissible trough angle of the idler, depending on the belt width. This stiffness must be low to ensure the belt can be troughed effectively, particularly in deep troughs. Conversely, this characteristic carries the risk that, across approximately one-third of the belt's width on the outer side of the turn, the radial contact between the belt and the rollers on that same side may be reduced, or the belt may even **lift off the rollers** or **fold back** towards the opposite side (towards the inside of the turn). This phenomenon may only become apparent several years after the belt has been commissioned, arising from errors in: trough transitions, convex curves, belt reversals, and inverted idlers on the return strand. All these causes weaken the stiffness of the belt carcass through fatigue or deformation. This phenomenon of one side of the belt lifting or folding in turns also depends on higher tension on the outer side of the turn for short turn radii, due to a greater difference in tension (Δ) between the inner and outer edges of the belt in the turn, all other parameters being equal.



Fig. 16.1: Carcass too rigid for a deep trough.

Rule: Minimum belt/roller contact = 70% of belt width

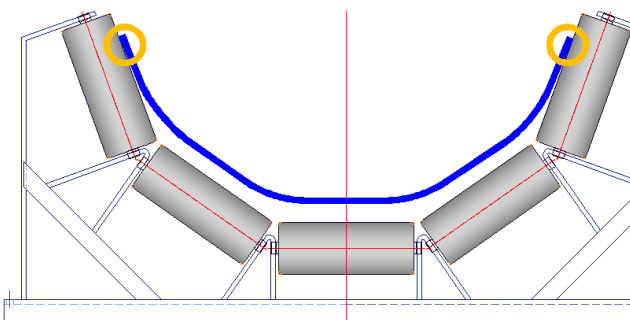


Fig. 16.2: Carcass too rigid. The belt's path will be unstable, but it withstands the stresses of turns well.

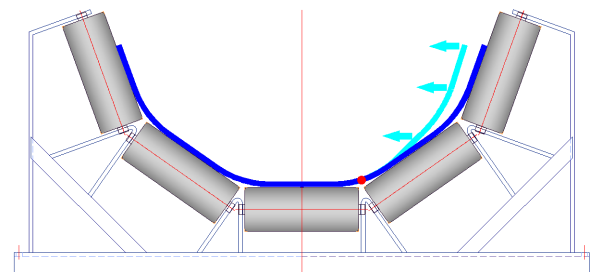


Fig. 16.3: Carcass with good troughing capability, but prone to folding in turns

Idlers / Cross-section

There is a wide variety of cross-sectional profiles for curve idlers, with the additional feature that the side rollers are sometimes longer than the centre roller of the idler, so as to allow a greater range of belt offset than that permitted by ISO 15236-1, clause 7.10; for example: for a strip 800 mm wide, the standard maximum belt drift is 40 mm. The number of sections in the support ranges from 1 to 5, or even more. Idlers with 1 (flat profile) or 2 V-shaped rollers are generally used for the return strand, but in some cases, the return strand may consist of a greater number of rollers (3, 4, 5, ...).

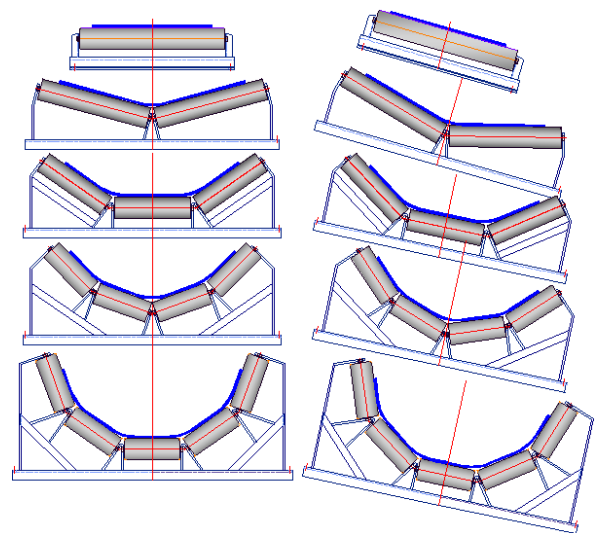


Fig. 17.1: 5 common profiles

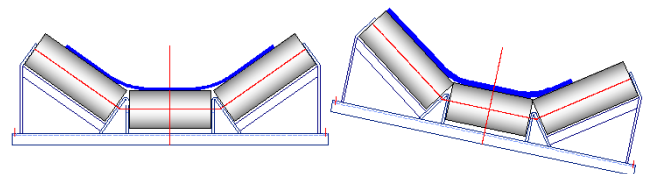


Fig. 17.2: Support with longer side rollers

Whilst, at 1st glance, a **deep** trough idler, for example with 5 sectors (5 rollers), appears more effective at controlling the belt's path through turns, as it combines two types of forces, this advantage remains relative and depends on the belt's ability to form a trough, which is a property that conflicts with its resistance to 'folding' in on itself, a quality that requires a carcass with a sufficiently stiff weft (see Figs. 16.1 & 16.2).

For example, the outer third of the belt width, on the outside of the turn, will experience a reduction in radial support (belt/roller) on its outer side, to the point where it takes off the roller. In this case, the steering force of the outer lateral roller on turn drops to 0 N. This phenomenon can be observed several months after the belt is commissioned, particularly on conveyors with errors in trough transitions, convex curve radii or belt turnaround lengths. Therein lies the problem.

There is another pitfall associated with the use of deep-trough idlers with five or more rollers, when the belt shifts to such an extent that it no longer covers the last roller in the trough; then, when the forces causing this significant drift cease, the belt can no longer re-centre itself because the belt heel abuts against the edge of this final trough roller.

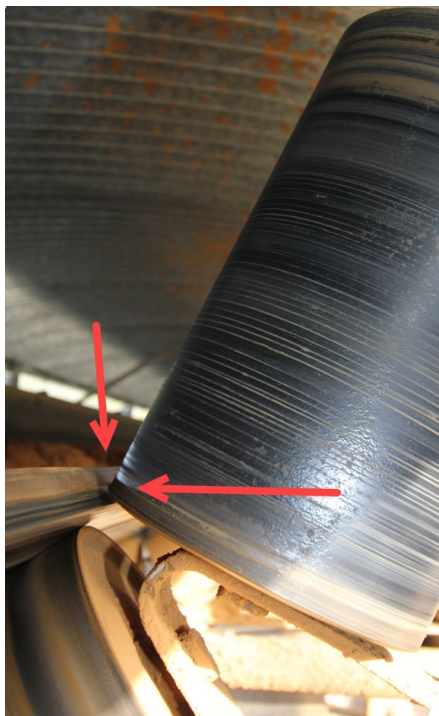


Fig. 18: 5-Idler with 5 rollers, the belt heel abuts against the edge of the outer roller

Idler(s) / Spacing between idlers

Curves, which are trough-belt conveyors, are very long sections that involve a large number of rollers, resulting in significant inertia. A good design for turns should aim to reduce the number of rollers in order to minimise this inertia, which affects the tension in the belt and, consequently, the orientation and inclination angles of the idlers.

The spacing that is, the distance between the idlers in a turn takes into account numerous factors, some of which impose conflicting constraints. A long spacing between the idlers on the carrying section increases the product compression forces, particularly when the product is concentrated towards the centre of the belt's width.

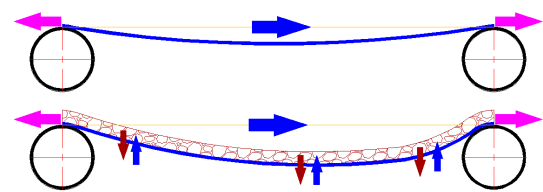


Fig. 19.1: Belt empty: low indentation resistance

Fig. 19.2: Full load, high indentation force

Σ of: load, tension, support pitch

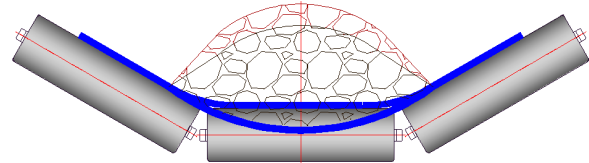


Fig. 19.3: Increased compression with product concentrated in the centre

A long spacing results in a proportionally greater radial force and a larger belt/roller wrap angle than a standard short spacing, for a given weight and tension. A long spacing may prove more advantageous due to the reduction in the belt's longitudinal stiffness, thereby ensuring the long-term effectiveness of the guiding force generated by the rollers.

Calculations show that a short or long spacing has little or no influence on the required curve radius to ensure the belt's trajectory through the turn, for the same angle of rotation and/or inclination of the idler. When the spacing between idlers doubles, the radial pressure and the belt/roller wrap angle double; the same applies for a triple spacing, and so on.

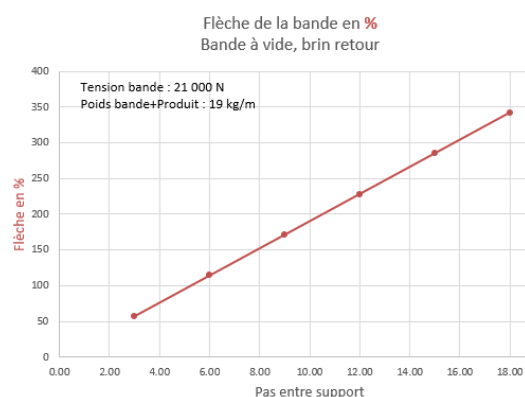
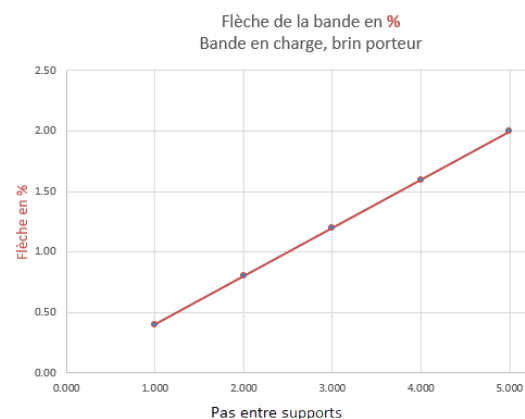


Fig. 20.1, 20.2: The belt deflection, in %, is proportional to the spacing between supports; the wrap angle and the radial belt-to-roller force are proportional, whilst the pressure in mm^2 remains constant. QED

The advantages of a long idlers spacing are as follows:

The '**long spacing**' design:

- Promotes good belt cradling performance; this is particularly true with an empty belt.
 - This allows for a stiffer belt carcass in the weft direction (weft reinforcement, breaker), offering advantages over this design, including:
 - greater resistance to the phenomenon of the belt 'folding back'
- is less susceptible to the risk of complete (delamination) or partial (reduced radial pressure) loss of contact between the belt and the side roller(s) on the outside of the turn.
 - This includes cases of poor belt/roller contact, where there is a speed difference (Δ) between the belt and the roller's peripheral speed.
 - This situation is '*tricky*' because, on site, it is difficult to detect with the naked eye; this leads to misinterpretation in the event of belt misalignment.
- It is less sensitive to variations in the belt/roller interface between a dry and a wet state.
- Reduces **the** belt's '**longitudinal** stiffness', which improves its ability to follow the directional forces exerted by the rollers. This is a crucial factor.

Roll(s) / Uncoated – Coated

Whilst the standard roller design features a steel shell, this can be coated with various materials offering a higher belt/roller coefficient of friction, which enhances the rollers' guiding effect. These coatings are made of rubber, polyurethane or resin containing corundum grains (for example, Belzona product ref. 1821).

However, these high-friction coatings must be used with caution, as their wear and/or that of the belt's running surface coating must be taken into account, given that there is always some belt slippage. The economic aspect must be taken into account and assessed in relation to the benefit gained from the radius of the planned or existing turn (existing conveyor).

The calculation must seek the best compromise between the angle of orientation favouring a small angle for to reduce wear on the roller casings and the stresses exerted on their bearings (axial force), which are not designed for this purpose.

The belt

In the context of a conveyor with turn, the belt is a key factor in calculating the various characteristics of the turn, starting with the curve radius, the spacing, and the orientation and inclination angles of the idlers.

To meet the requirements of a particular material-handling application taking into account the product, its throughput, density and angle of repose, the belt can be designed with different widths, trough profiles and speeds. Because turn conveyors are often very long, the choice between **these three criteria is crucial, and the first two** (width and cross-sectional profile) **are difficult to change at a later stage.**

Belt weight:

This plays a significant role in the calculations in two conflicting ways:

- A high belt weight (kg/m):
This helps to reduce the difference in belt mass between empty and loaded conditions, particularly when a high belt speed is selected in relation to the throughput to be handled, with a low belt fill factor (< 50% of ISO 5048 No. 7).
In this case, the differences between the forces in different directions generated by the turn rollers when the belt is empty and when it is loaded—are relatively small. This small difference is particularly evident on the side rollers when handling a product concentrated towards the centre of the belt's width, resulting in almost constant contact.

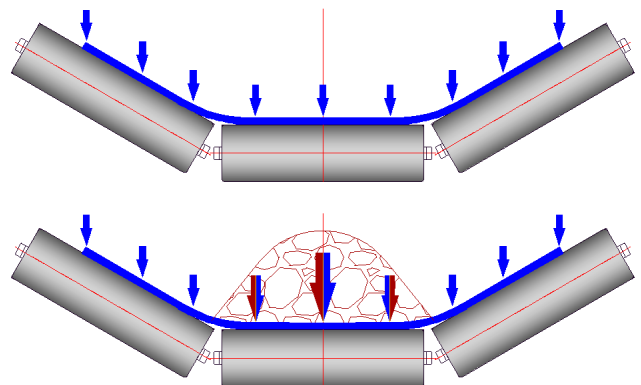


Fig. 21.1: Belt/roller pressure, belt empty conditions

Fig. 21.2: Belt/roller pressure, belt loaded with product concentrated in the centre of the belt

In this same context, where the belt weight is high relative to the product weight (kg/m), there is a reduced difference in 'belt tension' between no-load and full-load conditions.

This high belt weight has the disadvantage of adversely affecting all other sections of the conveyor that are not on a turn, and has a negative impact on the final balance of resistive forces, the total power absorbed by the conveyor, and the energy consumed.

The high belt weight is an aggravating factor when the conveyor includes a **long downward section**. It contributes to lowering the belt tension at the lowest point of the descent and can cause the belt to sag between the idlers; this tension may be close to 0 N, or even negative... **which makes the situation untenable.**

- A relatively low weight (kg/m):
The same arguments as above apply, but in reverse.

Please note:

The weight of the belt is not directly proportional to its breaking strength, as its weight depends on the weight of the carcass material (steel, aramid, polyester), whether or not it contains weft reinforcements, and the thickness and quality of its rubber coatings.

Troughing capability:

As with the weight criterion, the belt's troughing capability (ISO 703-1) must satisfy two conflicting requirements.

- Good troughing capability ensures that the belt will make good contact with all the rollers in the trough (ideally at least 70 per cent of its width). Troughing capability will necessarily be greater for a deep trough. This high troughing capability makes the belt more susceptible to loss of contact with the side rollers on the outside of the curve; a loss of contact renders these rollers ineffective in terms of the steering force applied to the belt (see Fig. 16.3).
- Conversely, low troughing capacity (high transverse stiffness) compromises the belt/roller contact to such an extent that the belt may follow an unstable path, particularly when unloaded, even on straight sections, because its support is limited to its two edges or heels. The effect of such stiffness can be mitigated by increasing the spacing between supports. This transverse stiffness makes the belt less susceptible to detachment from the side rollers on the outer side of the turn, which ensures the directional effect of these same rollers (see Figs. 16.1 & 16.2)... provided, of course, that there is sufficient contact (70 %)!
- A new belt may exhibit good moulding properties for a given width and support profile, but **as it ages**, it may become stiffer and exhibit an unmanageable trajectory. This is particularly true in the case of thick rubber linings made from a rubber compound that hardens over time (aggravating factors: sunlight, heat, ozone), especially when this hardening is more pronounced on the support side than on the running surface. This condition corresponds to a loss of elasticity. This accelerated ageing of the rubber, from 60–63 Shore A to >75 Shore A, may be caused by the covers that enclose the entire conveyor, the design of which creates a confined space under the cover. An outer surface of the cover that is not treated against solar radiation can cause the temperature under the cover to rise to $\approx 85^{\circ}\text{C}$... which is the very process of 'baking' the rubber of the belt's covering!

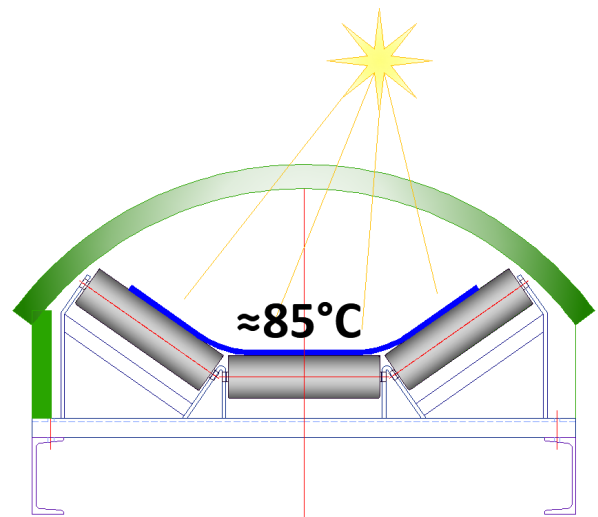


Fig. 22: Confined space / Temperature under the cover

Please note:

For a given belt design, its width influences its ability to form a trough. A belt 1,000 mm wide demonstrates good troughing capability, whilst an 800 mm wide belt is too stiff. This same belt, at 800 mm wide, may have adequate troughing capability for a shallow trough (e.g. 30° instead of 40°).

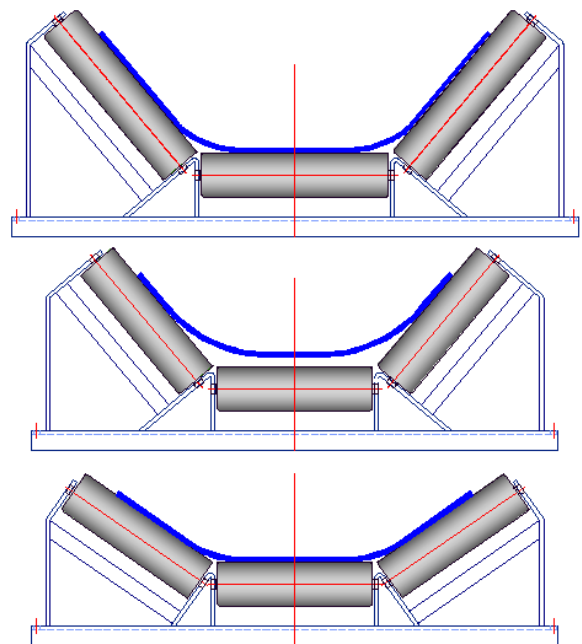


Fig. 23.1: 1000 mm wide belt, 45° trough: OK

Fig. 23.2: 800 mm wide belt, 45° trough: Not OK

Fig. 23.3: 800 mm wide belt, 30° trough: OK

Please note:

In the case of conveyors with long curved sections, **it is costly and sometimes difficult to replace the original supports** with supports whose profile is better suited to the belt.

Thickness of the coverings and cross- belt reinforcement(s):

These two criteria, which define the belt, are described in the preceding paragraphs. They primarily contribute to the belt's weight and its ability to form a trough and/or its transverse stiffness.

The weft reinforcements or breakers add transverse stiffness and weight. These elements also provide greater resistance to stapling, which can be considered a significant advantage in the event of repairs.

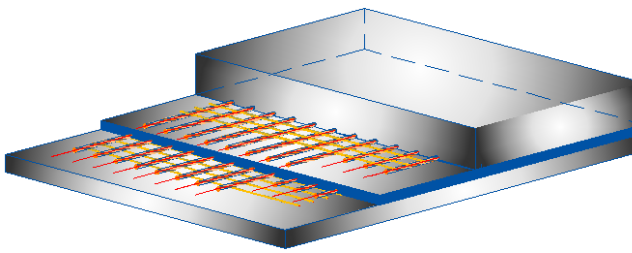


Fig. 24.1: 2 'straight warp' polyester plies + 2 wefts

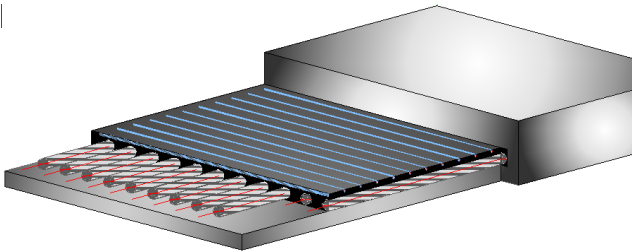


Fig. 24.2: 'Steel cord' chain + 1 steel weft

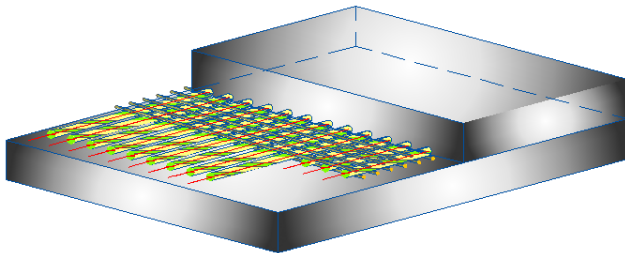


Fig. 24.3: Aramid chain + 1 breaker

The carcass:

Conveyors with wide centre distances are usually fitted with belts featuring a steel-cord carcass. It is primarily within this range of conveyors that configurations with curves are found. Nevertheless, it is worth noting that a steel cord carcass can be advantageously replaced by a **polyester** carcass with **straight cords** for centre-to-centre distances of up to 1,500–2,000 m or more (to be calculated), depending on the overall configuration, and by an aramid carcass for distances beyond that. Although aramid is expensive, in a market that is currently very tight, a technical and economic analysis may reveal significant advantages. The low weight of the aramid carcass makes it worthwhile to review the conveyor design in its entirety and check whether a lower breaking strength for the belt can be considered (e.g. 1st calculation: St 2500 N/mm in steel, 2nd calculation: D 1600 N/mm in aramid) to achieve lower elastic elongation in favour of aramid and a lower purchase cost compared to a 2500 N/mm aramid belt.

Other advantages of the aramid carcass relate to a more comprehensive approach to conveyor design by reducing inertia and tension in the belt. This is particularly relevant in the case of a conveyor with a large downward section.

Width:

Long conveyors are characterised by their narrow belt combined with high speed, so as to require the lowest possible breaking strength (reduced product load in kg/m) and thus keep the conveyor's construction costs under control... but care must be taken with operating costs (energy costs, etc.). When these long conveyors include turns, some designers increase the belt width so that the product being handled is confined to the central third of the belt's width, whilst the left and right side sections remain empty at all times, thereby maintaining a $\approx$ constant radial pressure between the belt and the side rollers (see Figs. 20.1 & 20.2).

The belt width is subject to several constraints, such as those discussed in the preceding paragraphs.

Once the belt width has been defined, it is virtually impossible to change it at a later stage.

The product being handled

In the context of a conveyor with a turn, particular attention must be paid to the product being handled, given its significant influence on the turn parameters.

Product throughput:

This particular attention focuses first and foremost on the **actual throughput** or **throughputs** to be handled.

Experience shows that, very often, specifications state peak throughput or unrealistically high design throughput figures, as these will never be reached not by a long shot.

It should be borne in mind that whoever calculates the conveyor is obliged to do so using the values specified in the specifications. A maximum throughput – whether peak or design throughput, **that is 20 per cent higher than the nominal throughput** should raise concerns and prompt the project partners to do everything possible to limit and regulate this maximum throughput upstream of the conveyor to be designed, and, quite simply, to define realistic values.

Sometimes, right from the start of the project, the operator knows that the throughput will change over time, over a period of 5–10 years or more. In such cases, it is advisable to carry out several simulations incorporating these changes in throughput. The simplest approach is to increase the belt speed over time. **Other solutions involve adding boosters and retro-boosters plus generators** at specific points along the conveyor to maintain constant tension in the belt without increasing its speed (= energy savings).

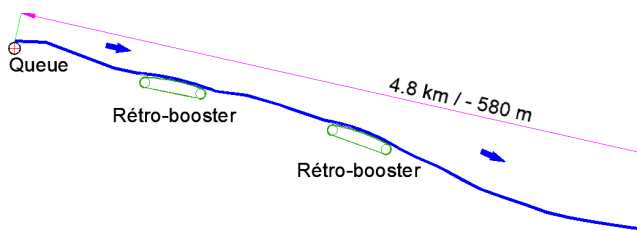


Fig. 25.1: Long descending section with 2 retroboosters and 8 self-piloted generators

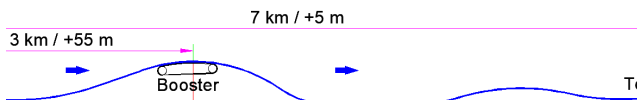


Fig. 25.2: Long, undulating section with 1 booster

Boosters and retroboosters can be used on conveyors that extend over time; for example, a conveyor feeding a tunnel boring machine in a mine where the mining area is expanding.

The effect of the conveyor's throughput is not limited to the turn itself, but also extends upstream and downstream of the turn. For example, in the case of a long downhill section before the turn, or within the turn itself, the material generates a driving force which accelerates the belt's speed through its elastic elongation, so that at the bottom of the descent its tension is close to 0 N, or even negative.

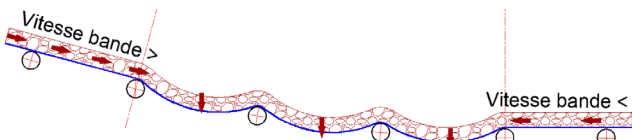


Fig. 26: V_{belt} , T_{belt} = Loops at the bottom of the descent

A reduced throughput, tailored to requirements, will mitigate this phenomenon of belt overspeed, which causes the festoons at the bottom of the descent. Only a calculation can determine whether or not it is necessary to install retroboosters. Furthermore, an uphill section, upstream of the turn, results in a significant resistive force and therefore tension in the belt, which strongly influences the turn parameters, particularly when the upstream section is full and the turn is empty. Only a calculation can determine whether or not boosters need to be installed.

Ultimately, the flow rate influences the curve parameters to a greater or lesser extent depending on the product load on the belt upstream of, within and downstream of the turn.

The product slope angle:

The angle of repose of the product in the dynamic phase determines the trough angle of the side roller on the idler (carrying section) of the turns. This angle of repose limits the angle of inclination of the idlers, to prevent the product from spilling off the belt. (see Fig. 15.2).

Account must also be taken of the presence of blocks in the product which might come to rest on the pile (in the middle or at the edge). This point mainly concerns spherical blocks, as they tend to roll easily. As mentioned above, the angle of inclination of the idlers

must be such that the spherical blocks are not ejected from the conveyor belt.

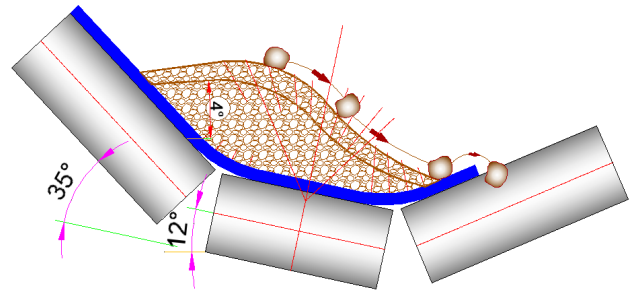


Fig. 27.1: Blocks rolling and being ejected from the conveyor belt

There are thixotropic products, characterised by a very steep static angle of repose and a dynamic angle of repose close to 0°; this zero angle considerably limits the permissible angle of inclination of the idlers.

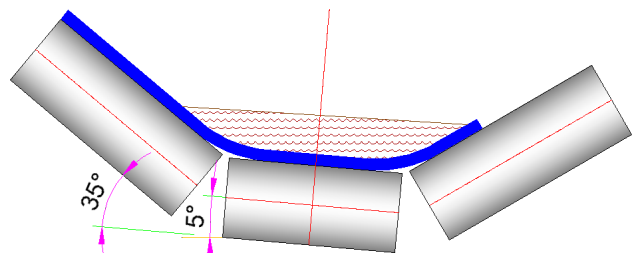


Fig. 27.2: Thixotropic product: dynamic angle of repose $\approx 0^\circ$

Other product characteristics:

The other characteristics of the material being handled have little or no influence on the parameters of the turns, with the exception of hot or very cold materials.

Calculation of turns

From a calculation perspective, a conveyor turn is always treated as two separate turns (one for the carry-strand and one for the return strand), even if they have the same radius (see # *parallel and superimposed turns*); the analysis must be supplemented by the coupled calculation of these two turns to define a common radius.

The 1st essential step in a rational and efficient approach to calculating conveyor turns, following a review of the specifications, particularly regarding throughput rates **is to optimise the overall design of the conveyor as much as possible**, taking into account all its components, including the belt, **in order to minimise resistance forces as much as possible**. This approach is explained in detail in our various previously published articles. Obviously, it is easier to achieve a very high level of optimisation with a conveyor at the design stage than with an existing conveyor, which imposes numerous constraints and entails high modification costs.

In practical terms, to achieve this 1st objective of maximum optimisation, **C3 Expert** offers training tailored to the subject and a licence for its C3v22® software, or the design and calculation services required to carry out this 1st phase.

The 2nd step, again in the context of long conveyors, involves examining several options for the layout of the drive unit(s), so as to **control the tension** in the belt at the **lowest possible levels in the turns**, thereby allowing for small turn radii. Conversely, the higher the potential maximum tension in the belt, the larger the curve radius will be. Here, both the feed section and the return section must be analysed in the same way, and the largest calculated radius is the one that applies.

For example, a long conveyor several kilometres in length, with a long, steeply descending 1st section and fitted with a single drive drum at the tail end (the standard configuration), means that the stresses and tensions in the belt will be more difficult to manage on the return run than on the carry run. In short, such a conveyor may require a larger turning radius for the return run than that corresponding to the drive run.

As these are still very long conveyors, it may be worthwhile to install one or more boosters, of the convex-profile cassette type, positioned at regular intervals along the carrying and/or return sections, so as to reduce the belt tension in the turns for better control of its trajectory, potentially allowing for shorter turning radii (see Fig. 24.2).

Similarly, in the case of a very long downhill section with a steep gradient, it is advisable to install one or more retroboosters with their generators so as to slow the belt down and control its tension at the bottom of the descent, thereby eliminating the risk of belt festoons (see Fig. 24.1).

Once again, **C3 Expert** offers training courses and calculation tools. For example, our calculations and engineering expertise have enabled us to reduce moving masses from **833 t** to **425 t**! Following the same approach and based on Newton's law $E = 1/2 m \cdot V^2$, the calculations have made it possible to reduce the belt's initial speed from 3.6 m/s to 2.0 m/s, ultimately ensuring effective control of the turns and significant energy savings. Can anyone beat that?

Following these two preliminary steps, the minimum and maximum tensions at the start and end of the turn must be extracted from the overall conveyor calculation.

- For a turn with a constant profile, we can assume that the change in tension in the belt is linear throughout the curved section.
- In the case of a turn with a varied and, by definition, complex profile, intermediate calculations must be carried out to determine the minimum and maximum stresses for each homogeneous subsection of the turn.

All these tension values are then entered into our **C3^{HC}** software, which is specifically designed to calculate all the parameters of the turn. Through a series of iterations, the **C3^{HC}** software harmonises the results so that the belt's path is controlled, for both the drive leg and the return leg, whether the routes are

superimposed in parallel (see Fig. 5.1) or side-by-side in parallel (see Fig. 5.2).

Implementation of calculation results on site

Even the best calculation or the best plan will not yield reliable results if implementation on the conveyor is not carried out using the appropriate procedure and resources, bearing in mind that adjustments to the idlers in the turns require tolerances of $\pm 0.3^\circ - 0.2^\circ$, or even less.

If the conveyor frame, with its curve radius ranging from a few hundred metres to several kilometres, is constructed under the supervision of a surveyor, during its construction, adjusting the initial **neutral** position of each idler in the turn (so that it coincides with the radius of the curve passing through the axis of the rollers on each support), followed by their orientation according to the calculated angle, is more difficult to achieve. The difficulties involved in carrying out these adjustments are all the more pronounced in the case of an existing conveyor, due to the covers and other obstacles that have been added over time (conveyor components, various structures, trees and undergrowth, etc.), compounded by difficulties in accessing the conveyor.

To meet this essential need, **C3 Expert** has developed a rigorous procedure combined with metrology tools ⁽⁵⁾, which can be adapted to the conveyor as it stands (with the covers in place) provided that the roller supports are accessible. Our metrology system requires only basic training, as the enclosed instructions are sufficiently clear to allow operators to work independently; nevertheless, this adjustment work demands precision.

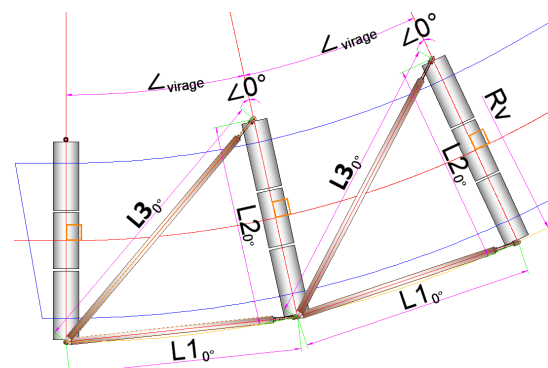


Fig. 28.1: Adjusting supports to the radius of the turn

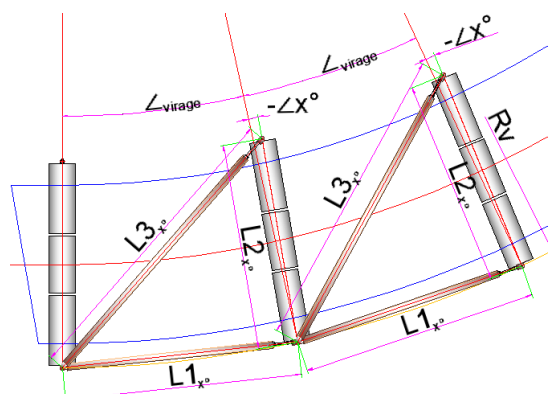


Fig. 28.1: Adding the calculated orientation angle

CONCLUSION

Calculating and controlling the trajectory of conveyor belts with turns is undoubtedly the most complex aspect of the materials handling sector.

The success of a project whether for a new build or the refurbishment of an existing machine requires in-depth knowledge of conveyor and belt technology. This necessitates taking the project's constraints into account and being able to negotiate with the client regarding those that can be revised in the project's best interests, such as throughput rates.

Once the project has been finalised in detail, it will be difficult and costly to modify its specifications at a later stage.

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NB: *We welcome your comments and feedback on this text. Its publication sets out the current state of the art.*

- (1) Fig. 2.1: 3 centres, 1 radius: this situation arises when a Turn features a '**clothoid**' both upstream and downstream, i.e. a curve with a progressive radius so that the change in direction is gradual. This applies to tunnels intended for use by trains and metros.
- (2) Orientation angle up to 3°: this significant angle of 3° is reserved for automatically or controlled-orientation idlers. The greater the orientation angle, the greater the power consumption.
- (3) Variable orientation is more efficient: whilst this is true in absolute terms, in practice it holds true only for systems with variable orientation. Conversely, in the case of a design with 'fixed' support orientation, controlling the belt path becomes 'unmanageable' due to the wide variability of the parameters governing this web path.
- (4) Variations in tension and load determine the angle of the supports: in short, in a 'fixed' orientation design, the greater the amplitude of these variations, the smaller the angle of the supports will be (e.g. 0.5°) in order to limit the amplitude of the belt's alternating deflection towards the chord and then towards the outer tangent of the turn, and vice versa. This scenario requires the use of other arrangements to control the belt's trajectory.
- (5) A rigorous procedure combined with metrology equipment: the alternative to this equipment is the use of a 3-plane (2 perpendicular vertical planes, 1 horizontal plane) green-laser device, to which must be added a laser target and the brackets required to support this equipment, plus various accessories. The disadvantage of a laser system, particularly over long ranges, is that it is difficult to use in sunlight; furthermore, in the case of a conveyor with 'covers', these must all be removed... hence the usefulness of metrology using gauges.