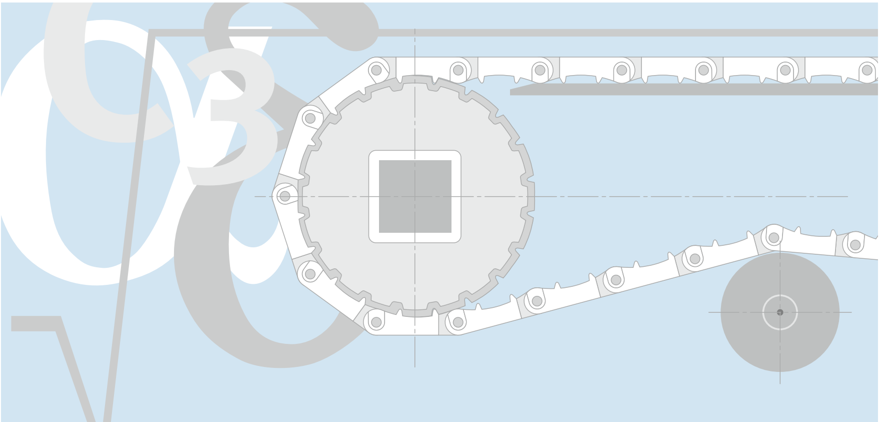
siegling prolink
modular belts

Recommendations for constructing and calculating conveyors



You can obtain detailed information on Siegling Prolink plastic modular belts in the overview of the range (ref. no. 800) and the data sheets on the individual series.

Please note:
When using Prolink Series 11 and the Combo belts (a combination of Prolink Series 5 ST and Prolink Series 11) please refer to:
Series 11/combo belts · Design guidelines and recommendations for use (ref. no. 201).

Content

Siegling – total belting solutions

MOVEMENT SYSTEMS

Belt support

Skid plate

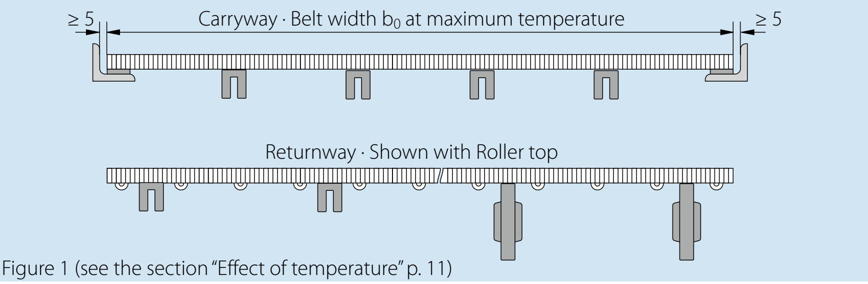


Figure 1 (see the section "Effect of temperature" p. 11)

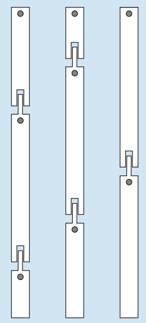


Figure 2

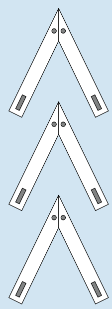


Figure 3

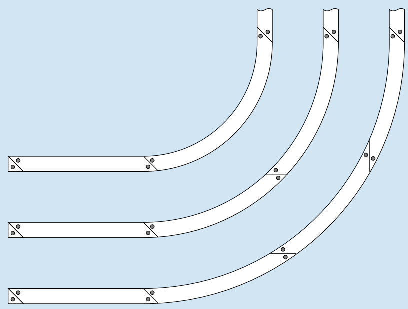


Figure 4

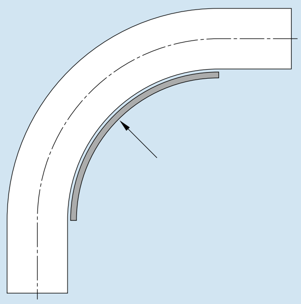


Figure 5

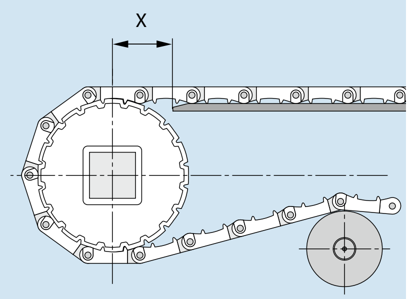


Figure 6

The belt can be supported in the following ways:

- Continuous plate support made of steel or plastics such as PE 1000. We recommend this for conveyors with heavy loads.
- Straight parallel wearstrips (figs. 1 + 2) made of steel or plastics. This is an inexpensive solution for applications with minimal loads. The belt wear is limited to the areas where the wearstrips support the belt. We recommend a distance of approx. 120 – 150 mm between the wearstrips for the upper side and approx. 200 mm for the return side. Alternatively, snub rollers can be used. Support is always provided in areas which do not have profiles, rollers etc. fitted.
- The belt is supported over the entire width by a V-shaped arrangement of the wearstrips (figs. 3 + 4). This spreads the wear and tear evenly and means heavy loads can be applied.
- Around the curves the belt is supported by plastic guides at the sides, for example PE 1000 or a plastic with lubricating properties, on the inner radius (see fig. 5).

Suitable plastic wearstrips are available from specialized dealers. The width should be approx. 30 – 40 mm, whereby the thickness depends on the height of the screw heads.
The permissible temperature ranges, as given by the manufacturer, must also correspond to the expected operating conditions.

Thermal expansion and contraction must also be taken into consideration when mounting the support. These effects can be eliminated by slots and appropriate distancing between the wearstrips (see the section "Effect of temperature").

- Distance X ≤ 1.5 x module pitch
- Place the snub roller on the return side so that the arc of contact on the drive and idle shafts ≥ 180°. (This does not apply to conveyors with e ≤ 2 m. Rollers on the return side are not necessary here.)

Roller support

Rollers are not generally used to support the belt on the upper face. Unavoidable belt sag between the rollers as well as the chordal action of the drive unit (see page 11) mean the goods are tipped which can cause problems. Sometimes rollers are used for conveying bulk goods.

Shafts

Drive Shaft

In general, we recommend the selection of a square shaft. The main advantage of this design is that positive drive and tracking are possible without keys and keyways. This saves on additional manufacturing costs. In addition, this form facilitates the lateral movement of the sprockets in the case of temperature variations.

Occasionally round shafts with feather keys are also used for low-loaded, narrow belts. Specially designed sprockets with bore and keyway are available.

Fastening the sprockets

Usually only 1 sprocket (as near as possible to the centre) must be fastened axially on each idle or drive shaft. The design of this sprocket enables positive tracking of the belt.

Examples of possible methods for fastening a sprocket are shown below:

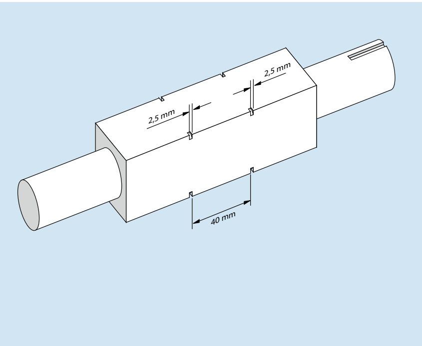


Figure 1

Shaft 40 x 40 mm
Fastening the sprocket with a retainer ring in accordance with DIN 471 (Seeger circlip ring)
d = 56 mm.

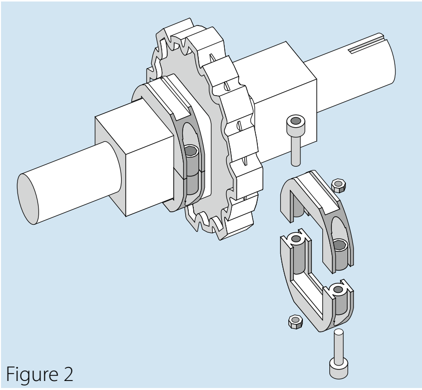


Figure 2

Siegling Prolink Retainer Rings provides a quick, easy and reliable solution for fixing the sprocket (see ref. No 412 for details).

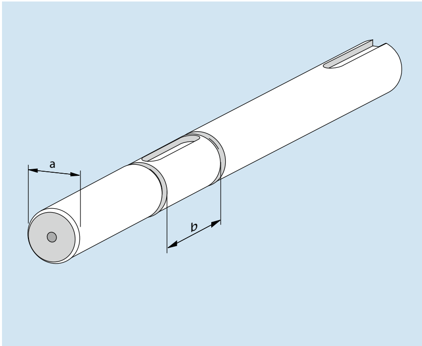


Figure 3

Fixation of the sprocket with retainer rings in accordance with DIN 471 (Seeger circlip ring).

MOVEMENT SYSTEMS

Deflection

Large belt widths and/or high tensile loads can lead to excessive deflection, preventing perfect belt-tooth engagement in the drive area. This results in uneven stress on the teeth of the sprocket, and it is possible that the sprockets do not engage properly, leading to “jumping” of the teeth when the belt is loaded. The borderline value permitted is the tooth engagement angle α_z and depends on the shape of the gear ring and module. For the Siegling Prolink linear belts this is 1.2°.
If the borderline values are exceeded, additional intermediate bearings must be applied or a larger shaft selected.
The tooth engagement angle α_z is calculated using this formula:

$$\alpha_Z = \arctan\left(\frac{y_W}{l} \cdot 2\right)$$

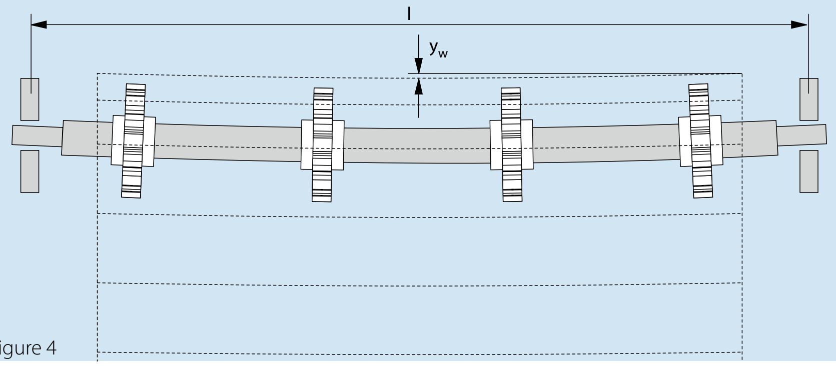
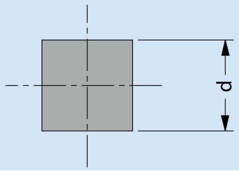


Figure 4

The shaft deflection y_W is calculated using the following formula

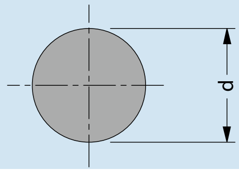
Figure 5



Solid shaft

$$y_W = 0.156 \, \frac{F_W \cdot l^3}{E \cdot d^4} \quad \text{[mm]}$$

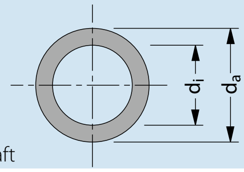
Figure 6



Solid shaft

$$y_W = \frac{80 \cdot F_W \cdot l^3}{E \cdot d^4 \cdot \pi \cdot 96} \quad \text{[mm]}$$

Figure 7



Hollow shaft

$$y_W = \frac{80 \cdot F_W \cdot l^3}{96 \cdot E\,(d_a^4 - d_i^4) \cdot \pi} \quad \text{[mm]}$$

F_W = shaft load [N]
l = bearing centre distance [mm]
E = shaft’s modulus of elasticity [N/mm²] (e.g. for steel = $2.1 \cdot 10^5$ N/mm²)
d = length of side of square shaft [mm]
d, d_i, d_a = diameter of shaft [mm]
y_W = shaft deflection

Conventional conveyors

Belt sag/control of belt length

There are various causes for changes in the belt length, e.g.

- Elongation or contraction of the belt due to temperature variation
- Wear of the connecting rods as well as enlargement of the connecting rod holes in the modules after a certain "break-in time" (enlargement of holes, 0.5-mm larger holes in a 50 mm module result in an elongation of 1 %).

Therefore we recommend not supporting one (or several) sections on the return side and using the resulting belt sag to compensate for the increase in length. It is important that perfect engagement between belt and sprocket is ensured. Following are several examples:

a) Short conveyor (fig. 1)

b) Medium length conveyors, up to a centre distance of approx. 4,000 mm (fig. 2)

c) Long conveyors: centre distance > 20,000 mm and low speeds centre distance < 15,000 mm and high speeds (fig. 3)

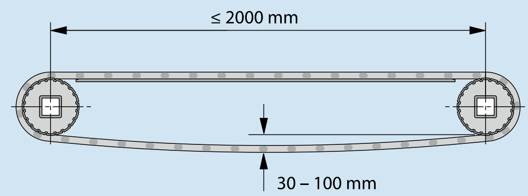


Figure 1

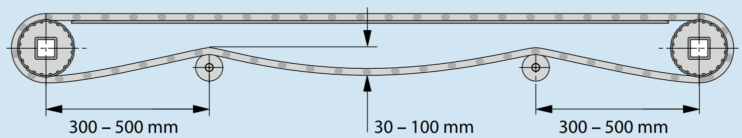


Figure 2

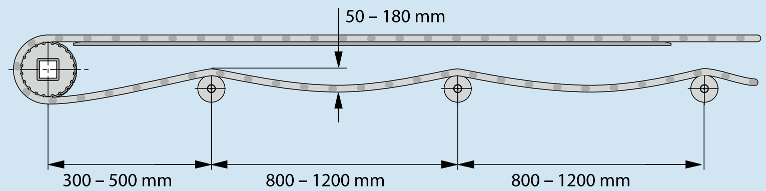


Figure 3

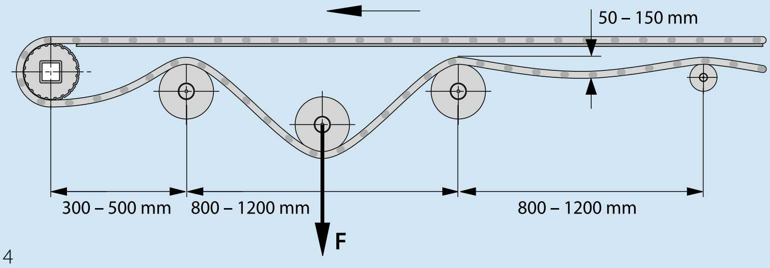


Figure 4

Another effective method for compensating for belt elongation is a load-dependent take-up system (e.g. weighted roller). This should be located as closely to the drive shaft as possible since the take-up system will ensure even tension on the return side and therefore perfect engagement between sprocket and belt (fig. 4).

For series 1, 3 and 7 we recommend a weighted roller, 150 mm in diameter and a weight of approx. 30 kg/m belt width.

For series 2 and 4.1 we recommend a weighted roller, 100 mm in diameter and a weight of approx. 15 kg/m belt width.

For series 6.1 we recommend a weighted roller, 100 mm in diameter and a weight of approx. 60 kg/m belt width.

For series 8 and 10 we recommend a weighted roller, 100 mm in diameter and a weight of approx. 30 kg/m belt width.

For series 13 we recommend a weighted roller, 50 mm in diameter and a weight of approx. 10 kg/m belt width.

Reversible conveyors

Two-motor design

Advantages: Low tension on the return side, making smaller shaft loads possible

Disadvantage: Increased costs due to additional motor and electronic control. For larger conveyors with relatively heavy loads, however, this system may still be the most reasonably priced.

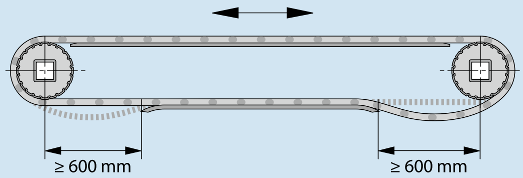


Figure 1

Centre drive

For reversing operation the drive shaft must be located as close to the middle as possible. To the right and the left of the drive unit, areas with belt sag are to be provided, since these are necessary for the required belt tension. The 180° arc of contact on the drive shaft means belt and sprocket engage perfectly making reliable power transmission in both operational directions possible.

The location of the drive unit causes more stress on the shafts at the ends of the conveyor as there is effective pull on both the upper and return sides in the form of belt tension.

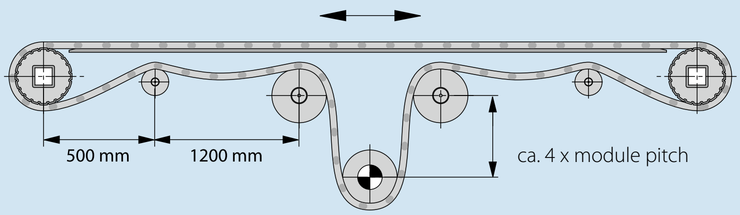


Figure 2

Alternating tail-head drive configuration

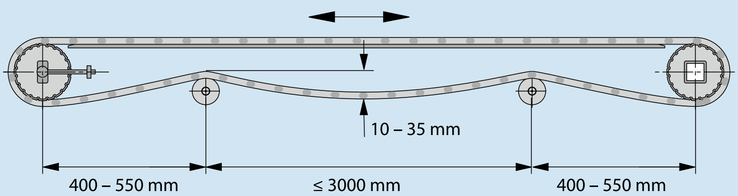


Figure 3

In the case of head drives the conveyor is like a conventional conveyor. It is only when conveying direction is reversed that the conveyor become tail-driven and the drive unit has to push the belt and its load. If the tension on the return side is not greater than that on the upper side it will jump sprockets.

An approximate value for the tension on the return side is 1.2 – 1.3 x F_U. This automatically leads to a greater shaft load.
$F_W \approx 2.2 - 2.3 \times F_U$

Inclined conveyors

Inclined conveying

We always recommend the following:

- Only operate with a head drive, i.e. use the upper shaft as the drive shaft.
- There is always a screw-operated take-up system or a load-dependent tension take-up on the return side since tension decreases with increasing inclination (caused by the belt sag).
- If sprockets are used at upper intermediate points, the centre sprockets may not be fastened axially.
- If rollers are used at upper intermediate points, a minimum radius of approx. 80 mm is required.
- When shoe or wearstrips are used, the radius should be as large as possible in order to keep wear to a minimum. See page 11 for recommended minimum radius. The width of the shoe should not be smaller than 30 mm.
- If the belt is more than 600 mm wide, we recommend providing further supports on the belt surface or on the profiles on the return side.

Rough guideline on achievable inclines:

- Flat top surface (FLT) 3 – 5°
- Friction top surface (FRT) 20 – 40°
- Straight profiles < 60°
- Bent profiles < 90°

Testing is always recommended to determine the actual possible incline angle for a particular product/application.

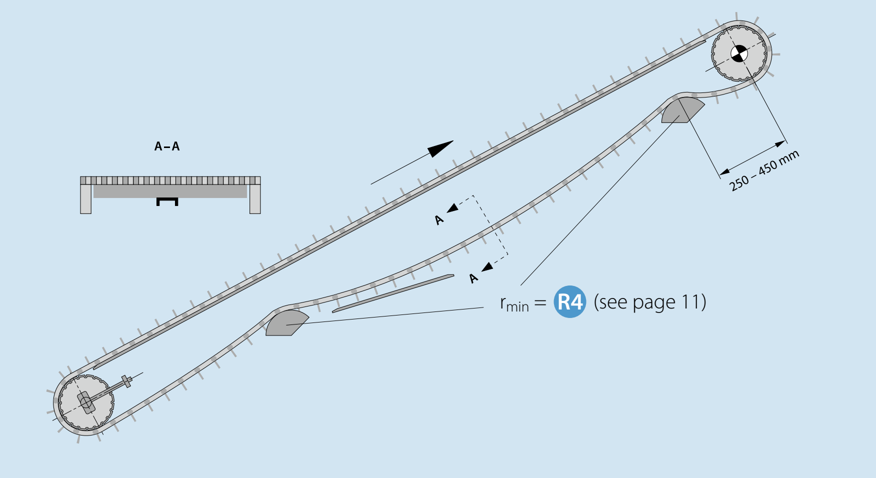


Figure 1 – Belt with lateral profiles

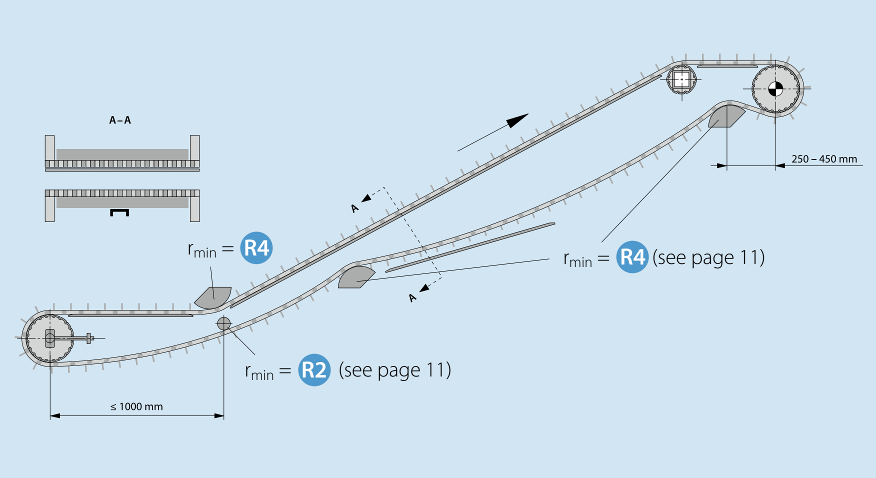


Figure 2 – Belt with lateral profiles

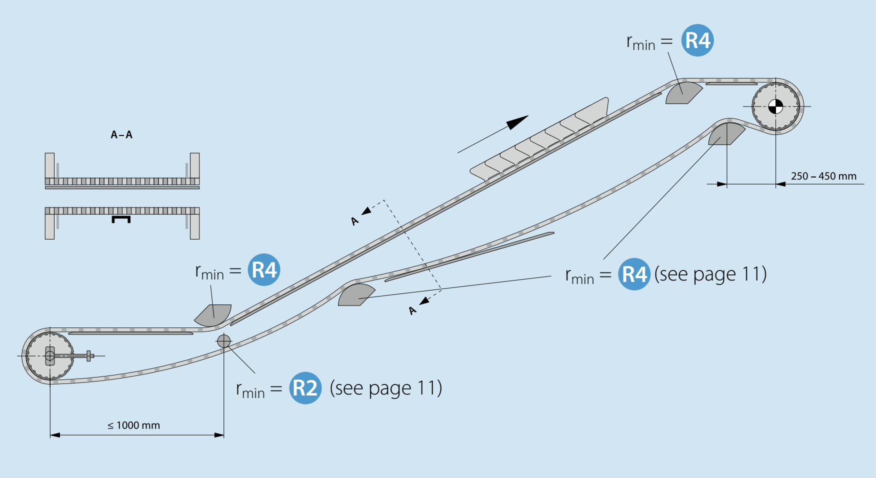


Figure 3 – Belt with side guards

Declined conveying

For this conveyor design, a tail drive unit is possible if there is an active load-dependent tension take-up at the lower idle shaft (e.g. gravity, spring or pneumatic). Otherwise the general recommendations given above apply here.

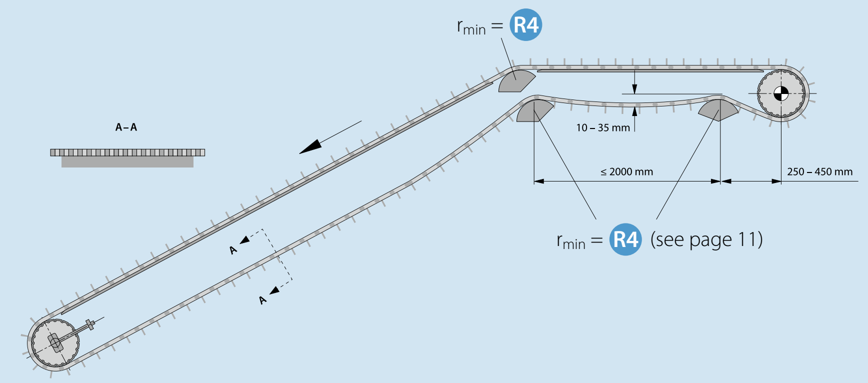


Figure 4 – Belt with lateral profiles

Nose bar conveyors

Tight transfer

Narrow transfer gap ensures smooth transfer of small and delicate products. The radius of the nose bar must be suitable for the selected belt type. For the recommend radius see below.

For optimum product stability (smooth belt run) we recommend:

- a belt wrap angle of 65° to 70°
- a straight support section of minimum 2 x belt pitch before and after the nose bar.

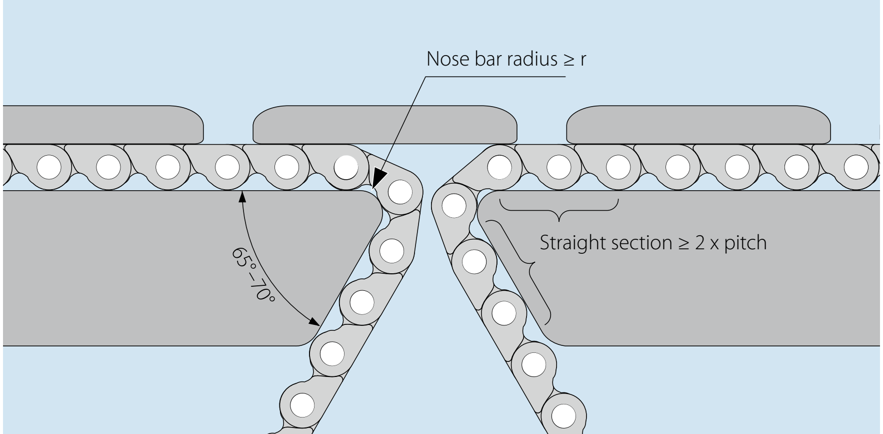
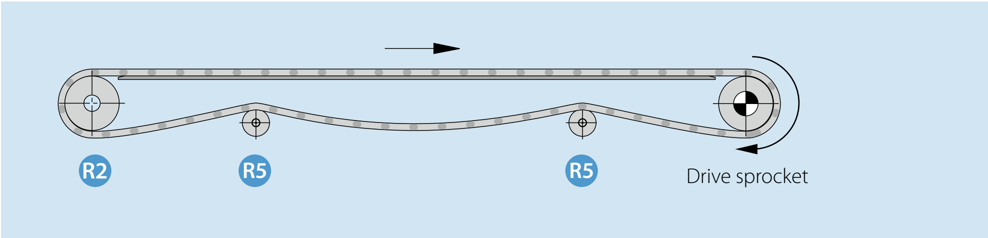


Figure 1

Belt series	Nose bar, r_{min} [mm]
S13	3
S4.1	11
S2, S5, S8, S10 & S11	25

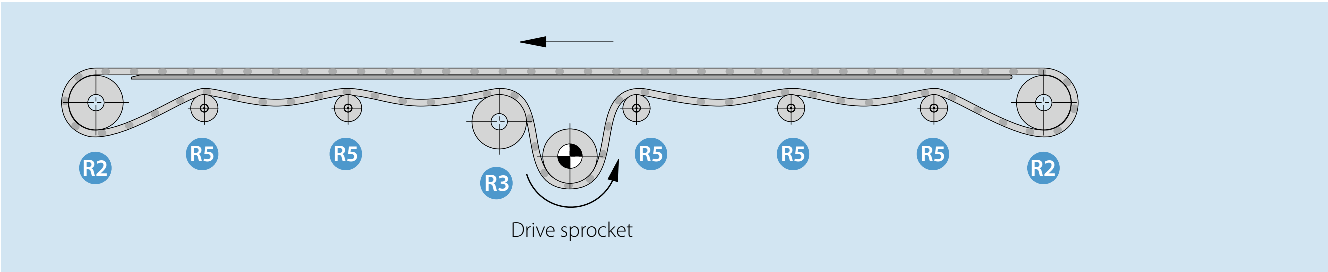
Recommended minimum radii

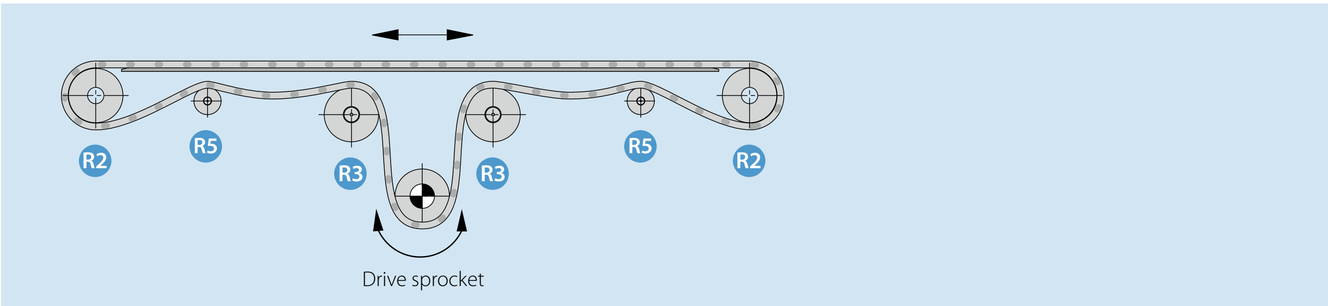
Standard conveyors



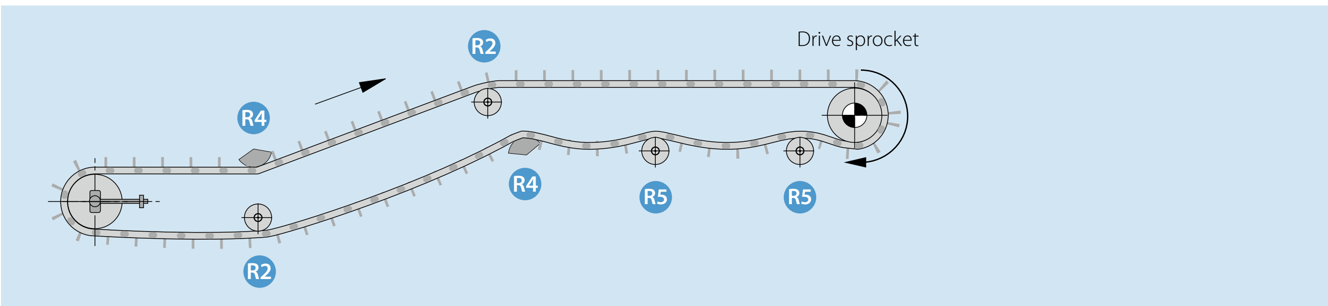
- R1 Side flex radius
- R2 Front flex radius
- R3 Load bearing roller
- R4 Hold down shoe
- R5 Back flex roller

Centre drive conveyors

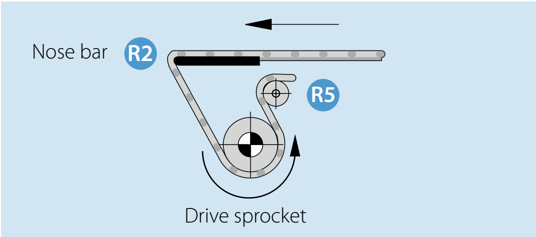




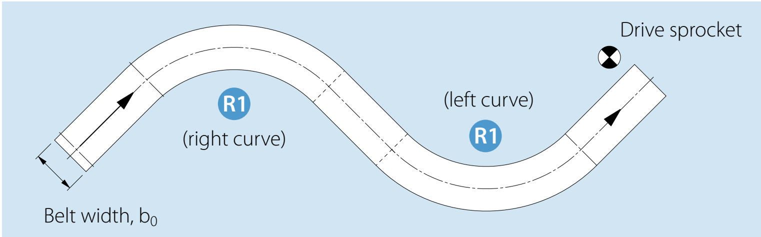
Inclined conveyors



Nose bar conveyors



Curved conveyors



Belt types	Pitch [mm]	Side flex	Front flex	Back flex		
		Min. inner radius [mm] R1	Min. radius on rollers* [mm] R2	Min. radius on load bearing rollers [mm] R3	Min. radius on hold-down shoes [mm] R4	Min. radius on rollers [mm] R5
S1-x FLT / NSK / FRT / SRS S1-PMU with SG **	50		50	100	150	50 150
S2-x FLT / GRT S2-57 RRB S2-x PMU with SG **	25		25	50	75	25 50 50
S3-x FLT / LRB S3-x with SG **	50		50	100	150	50 150
S4.1-x FLT / NPY / NTP S4.1-0 FRT1	14		11	25	38	12.5 16.5
S5-45 GRT / NTP / FRT R S5-45 PMU with SG ** S5-45 G / RG	25	2 x b_0	25 50	50	75	25 75 25
S6.1-x FLT / CTP / NPT / PRR S6.1-x PMU with SG **	50		50	100	150	50 150
S7-x FLT / NSK / FRT / SRS / PRR	40		40	80	120	40
S8-x FLT / NSK / RAT / FRT / SRS / PRR S8-0 RTP A90 S8-0 PMU with SG **	25		25	50	75	25 30 75
S9-57 GRT / NTP S9-57 PMU with SG **	50	1.8 x b_0	50	100	150	50 150
S10-x FLT / NTP / LRB S10-0 PMU with SG **	25		25	50	75	25 75
S11-45 GRT / NTP / FRT S11/S5 combo	25	1.4 x b_0 1.45 x b_0	25	50	150 75	25
S13-0 FLT / NPY	8		3***	16	24	8

Using larger radii than listed will reduce wear on belt, rollers and/or hold-down shoes. Larger radii will also usually reduce noise levels and make the belt run more smoothly.

* Depending on the application smaller radii are possible (e.g. knife edges).
Speed, possible band noise/jiggle and type of conveyed goods must be considered in these cases.
** Back flex radius depends on profile height and distance
*** Knife edge/nose bar

Curve conveyors

Meshing

The teeth must mesh into the modular belting in the areas marked by the arrows. (fig. 1)

Attention!
In the case of Prolink series 11 and Combo belts (a combination of Prolink series 5 ST and Prolink series 11) different dimensions and characteristics must be taken into account.

Please refer to: **Series 11/Combo belts · Design guidelines and recommendations for use** (ref. no. 201).

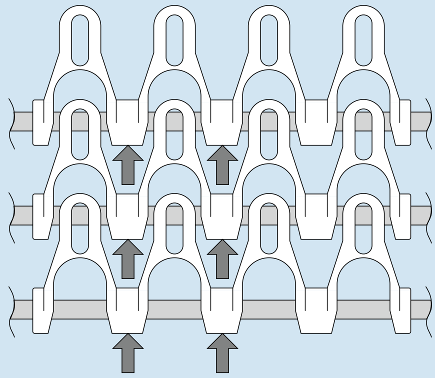


Figure 1

Inner radius

Siegling Prolink inner radius r_{min}
for curved belts

Series 5:	$r_{min} = 2 \times b_0$
Series 9:	$r_{min} = 1.8 \times b_0$
Series 11:	$r_{min} = 1.4 \times b_0$
Combo belts	
Series 5 ST/Series 11:	$r_{min} = 1.45 \times b_0$

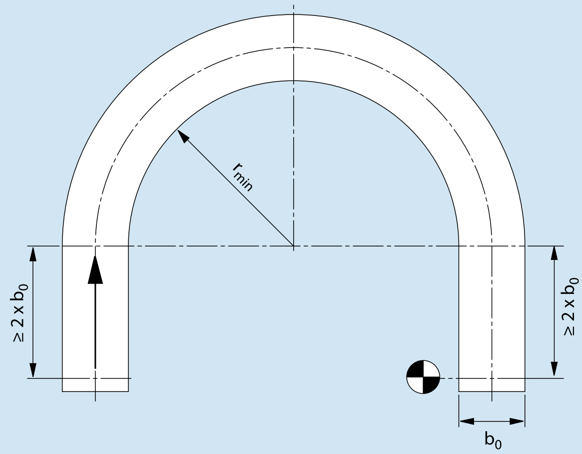


Figure 2 – 180° Curve

Belt tension

The three usual tensioning methods are possible to create the correct belt tension:

- Screw-operated take-up system
- Gravity take-up system
- Catenary sag on the return side near the drive drum

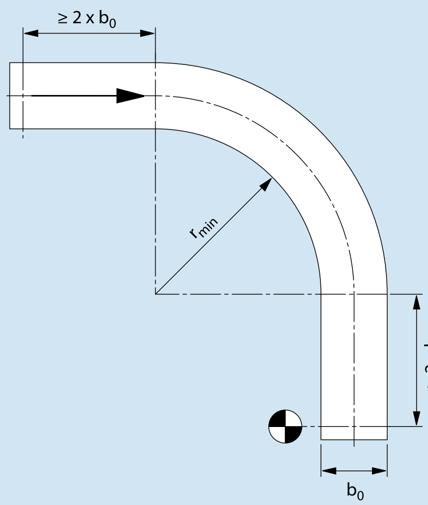


Figure 3 – 90° Curve

Geometries of curves

Please consult us if you cannot construct the conveyor according to the drawings because space is restricted.

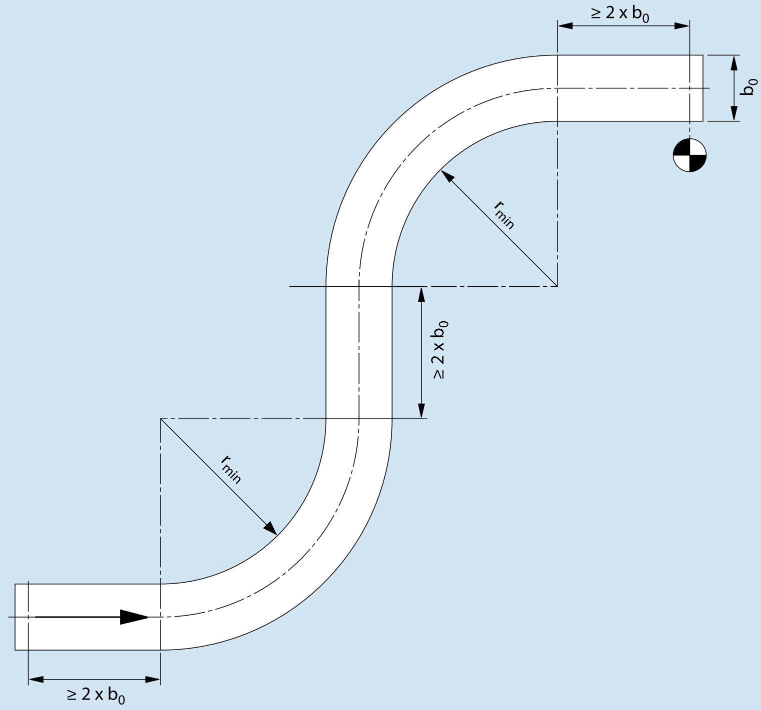


Figure 4 – S-shaped curve

Spiral conveyors

Possible conveyor designs

Fig. 1:
Example of declined conveying to join two production units with different heights.

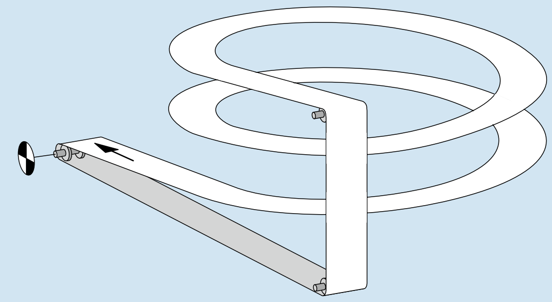


Figure 1

Fig. 2:
For inclined conveying, the drive unit must be located at the end of the curve at the top. Make sure that the arc of contact on the drive shaft is approx. 180°. This type of design (without driven inner cage) should not have more than 2 – 3 tiers.

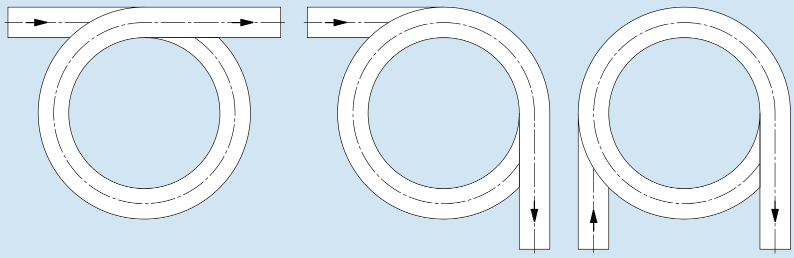


Figure 2

Fig. 3:
The main drive system is the driven inner cage, which as a rule consists of vertical rods. The curved belt is supported on the inner radius by the cage and is moved by traction between the belt and the cage. The direction of rotation of the cage determines whether the conveying is inclined or declined.
The drive and tensioning unit depicted in the sketch provides the necessary belt tension. The speed of the motor must be coordinated with the speed of the cage drive.
It should be possible to move the tensioning unit a distance corresponding to approx. 1 % of the belt length.
The belt can be supported by wearstrips as described on page 2.

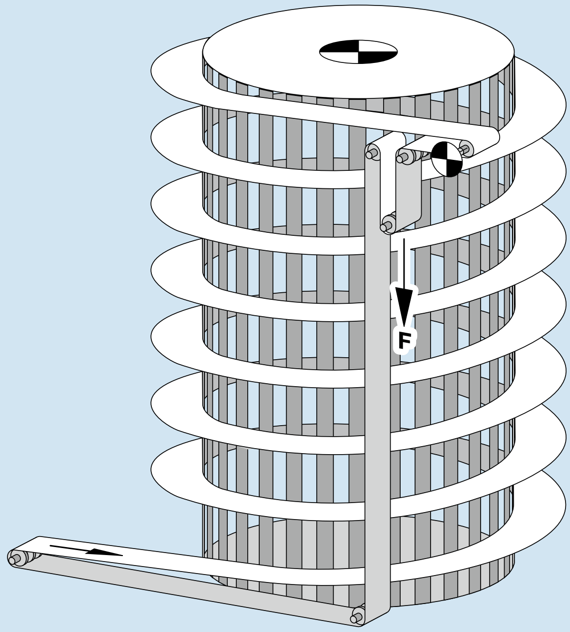


Figure 3

Further information

Effect of temperature

Plastics can expand or contract significantly when temperatures fluctuate. The construction engineer must make allowances for changes in belt lengths and widths if the operating temperature is not the same as the ambient temperature. Essentially, this affects the belt sag on the return side and the lateral clearance on the conveyor frame.

Material	**Coefficient of thermal expansion a [mm/m/°C] ***
PA	0.12
PA-HT	0.10
PBT	0.16
PE	0.21
POM	0.12
POM-CR	0.12
POM-HC	0.12
POM-MD	0.12
PP	0.15
PXX	0.15
PXX-HC	0.15

* Average values for the permissible temperature range

Calculation of changes in length and width:

$$\Delta l = l_0 \cdot (t_2 - t_1) \cdot a$$

$$\Delta b = b_0 \cdot (t_2 - t_1) \cdot a$$

Calculation example:
Ambient temperature 20 °C, the belt is used for the conveying of hot goods, resulting in an operating temperature of 90 °C. Belt length 30 m, belt width 1 m, belt material polypropylene.

$$\Delta l = 30 \cdot (90 - 20) \cdot 0.15$$

$$\Delta l = 315 \text{ mm}$$

$$\Delta b = 1 \cdot (90 - 20) \cdot 0.15$$

$$\Delta b = 10.5 \text{ mm}$$

The increase in belt length of 315 mm is not insignificant which means that the return side must be designed in such a way that the additional belt sag is absorbed. In order to accommodate the increase in width, the conveyor frame must have a wider design.
When operating at temperatures below 0°-C, the length and width contract. This must also be accomodated in the conveyor design.

Δl = change in length in mm
+ = elongation
– = contraction

l_0 = belt length at initial temperature in m

b_0 = belt width at initial temperature in m

t_2 = operating temperature °C

t_1 = initial temperature °C

a = coefficient of thermal expansion mm/m/°C

Chordal action

What is known as chordal action is typical for all sprocket-driven belts, chains etc. The rise and fall of a module during the slewing motion cause changes in the linear speed of the belt. The number of teeth on sprocket is the decisive factor for these periodic fluctuations in speed.

As the number of teeth increases, the percentual change in speed decreases. In practice this means that the largest number of teeth possible must be used if the goods are not to tip or for other reasons an even belt speed is required.

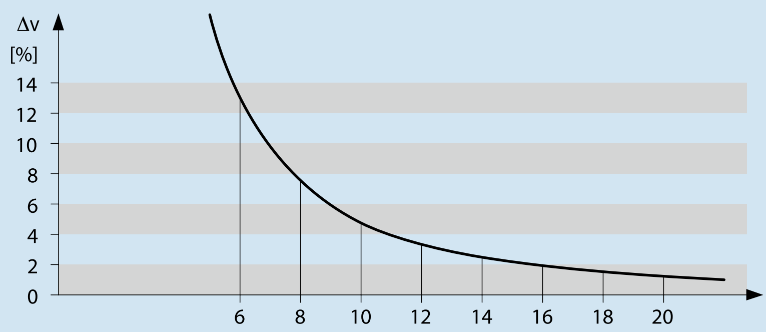


Figure 1 – Number of teeth on sprocket

Calculation

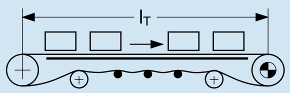
Key to the symbols

Designation	Symbols	Unit
Effective belt pull	F_U	N
Force determining belt selection	F_B	N
Shaft load	F_W	N
Calculated power at drive drum	P_A	kW
Coefficient of friction with accumulated goods	μ_{ST}	–
Coefficient of friction with skid plate	μ_T	–
Operational factor	C_1	–
Temperature factor	C_2	–
Adjusted belt pull	C_3	N/mm (lb/ft)
Allowable belt pull	C_3 max	N/mm (lb/ft)
Acceleration due to gravity	g	9.81 m/s^2
Conveyor length	l_T	m
Height of lift	h_T	m
Mass of entire belt (see data sheet)	m_B	kg
Total load	m	kg
Mass of drive drum	m_W	kg
Angle of conveyor	α	°
Belt width	b_0	mm
Belt speed	v	m/min

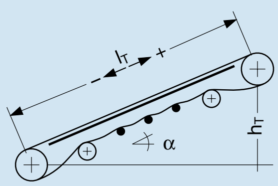
Loading examples to determine the effective pull F_U

A

One of the three following formulae is used to calculate F_U, depending on the design of the conveyor.



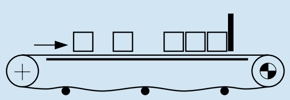
$F_U = \mu_T \cdot g \cdot (m + m_B)$ [N]



$F_U = \mu_T \cdot g \cdot (m + m_B) \overset{(-)}{+} g \cdot m \cdot \sin \alpha$ [N]

(+) inclined

(–) declined



$F_U = \mu_T \cdot g \cdot (m + m_B) + \mu_{ST} \cdot g \cdot m$ [N]

Mass of rotating parts on the return side was ignored.

Coefficients of friction μ_T between belt and wearstrip

The figures stated have been established under ideal conditions. When operating under other conditions we recommend using higher friction coefficients. ("–" = not recommended combinations)

	Belt material	PE & PE-MD			PP, PXX PXX-HC			POM incl. CR, HC & MD			PA-HT		
Wearstrip material	Operating conditions	clean	regular	soiled	clean	regular	soiled	clean	regular	soiled	clean	regular	soiled
Hardwood	dry	0.16	0.16	0.24	0.22	0.39	0.59	0.16	0.22	0.32	0.18	0.19	0.29
	wet	–	–	–	–	–	–	–	–	–	–	–	–
HDPE	dry	–	–	–	0.14	0.19	0.29	0.08	0.19	0.29	0.15	0.23	0.34
	wet	–	–	–	0.12	0.17	0.26	0.08	0.12	0.25	–	–	–
Lubric. PA	dry	0.18	0.28	0.45	0.13	0.24	0.35	0.12	0.20	0.30	0.16	0.24	0.36
	wet	–	–	–	–	–	–	–	–	–	–	–	–
Steel	dry	0.14	0.23	0.38	0.25	0.31	0.47	0.18	0.23	0.35	0.20	0.31	0.45
	wet	0.13	0.21	0.33	0.24	0.29	0.44	0.14	0.17	0.26	–	–	–
UHMW PE	dry	0.30	0.31	0.47	0.13	0.22	0.35	0.13	0.17	0.32	0.18	0.24	0.38
	wet	0.27	0.28	0.45	0.11	0.20	0.32	0.11	0.15	0.28	–	–	–

Coefficients of friction μ_{ST} between belt and conveyed product ("–" = not recommended combinations)

	Belt material	PE & PE-MD			PP, PXX & PXX-HC			POM incl. CR, HC & MD			PA-HT		
Conveyed product	Operating conditions	clean	regular	soiled	clean	regular	soiled	clean	regular	soiled	clean	regular	soiled
Cardboard	dry	0.15	0.19	0.34	0.22	0.31	0.55	0.20	0.30	0.50	0.20	0.30	0.50
	wet	–	–	–	–	–	–	–	–	–	–	–	–
Glass	dry	0.10	0.15	0.25	0.16	0.24	0.41	0.13	0.20	0.35	0.13	0.20	0.33
	wet	0.09	0.13	0.22	0.17	0.21	0.37	0.13	0.18	0.33	–	–	–
Metal	dry	0.13	0.20	0.33	0.32	0.48	0.60	0.17	0.27	0.45	0.20	0.30	0.50
	wet	0.11	0.17	0.28	0.29	0.45	0.58	0.16	0.25	0.42	–	–	–
Plastic	dry	0.10	0.13	0.25	0.15	0.21	0.37	0.15	0.25	0.41	0.13	0.20	0.33
	wet	0.08	0.11	0.22	0.14	0.19	0.34	0.14	0.21	0.36	–	–	–

Force determining belt selection F_B

B

$$F_B = F_U \cdot \frac{C_1}{C_2} \quad [N]$$

Operational factor C_1

	C_1
Smooth operating conditions (smooth start)	+ 1.0
Start-Stop-operation (start when loaded)	+ 0.2
Tail drive (push configuration)	+ 0.2
Belt speed greater than 30 m/min	+ 0.2
Inclined or swan-neck conveyor	+ 0.4
Total C_1	_ _ _ _

Temperature factor C_2

The tensile strength of the different materials increases at temperatures below 20 °C but at the same time other mechanical properties are reduced at low temperatures. Therefore the C_2 factor is set to 1.0 at temperatures below 20 °C.

The temperatures relate to the actual belt temperature. Depending on the application and conveyor layout the temperature of the conveyed product may be different.

	Belt material				
Temperature [°C]	**PE**	**PP**	**POM**	**PA**	**PA HT**
– 60	1.0	–	–	–	–
– 40	1.0	–	1.0	–	–
– 20	1.0	–	1.0	1.0	1.0
0	1.0	–*	1.0	1.0	1.0
+ 20	1.0	1.0	1.0	1.0	1.0
+ 40	0.90	1.0	1.0	1.0	1.0
+ 60	0.62	0.85	0.96	0.95	1.0
+ 80	–	0.65	0.75	0.72	1.0
+ 100	–	0.45	–	0.50	1.0
+ 120	–	–	–	0.40	1.0
+ 140	–	–	–	–	1.0
+ 155	–	–	–	–	1.0

* below + 7 °C avoid impact and ensure smooth start

Counter-checking the Siegling Prolink type selection

C

$$\frac{F_B}{b_0} = C_3 \leq C_3\,max$$

Allowable belt pull C_3 max

Material / Type	PE [N/mm (lb/ft)]	PP [N/mm (lb/ft)]	POM [N/mm (lb/ft)]	PA [N/mm (lb/ft)]
S1	18 (1233)	30 (2055)	40 (2740)	–
S2	3 (206)	5 (343)	7 (480)	–
S3	6 (411)	12 (822)	16 (1096)	–
S4.1	3 (206)	5 (343)	10 (685)	–
S5 straight	10 (685)	18 (1233)	25 (1713)	–
S5 curved	–	1000 N/225 lb	1800 N/405 lb	
S5 ST straight	10 (685)	18 (1233)	25 (1713)	–
S5 ST curved	–	1200 N/270 lb	2100 N/473 lb	–
S6.1	13 (891)	18 (1233)	30 (2055)	30 (2055)
S7	18 (1233)	30 (2055)	50 (3425)/60 (4110)*	–
S8	15 (1028)	20 (1370)	40 (2740)	–
S9 straight	12 (822)	22 (1507)	30 (2055)	24 (1644)
S9 curved	–	1600 N/360 lb	2800 N/630 lb	2240 N/504 lb
S10	6 (411)/3 (206)*	8 (548)/5 (343)*	20 (1370)/11 (754)*	20 (1370)/11 (754)*
S11 straight	–	9 (887)	15 (1028)	15 (1028)
S11 curved	–	600/135 lb	1000/225 lb	1000/225 lb
S13	–	–	4 (274)	–

* depending on belt configuration

Number of sprockets on the drive drum (guidelines)

Where centre distances are substantial, the number of drive sprockets still depends on the engagement ratio between teeth/module (i.e. on the belt length).

$C_3 \leq 20\,\%$	from C_3 max, the distance between the sprockets should then be approx. 160 mm (6.3 in).
$C_3 \leq 40\,\%$	from C_3 max, the distance between the sprockets should then be approx. 100 mm (3.9 in).
$C_3 \leq 60\,\%$	from C_3 max, the distance between the sprockets should then be approx. 80 mm (3.1 in).
$C_3 \leq 80\,\%$	from C_3 max, the distance between the sprockets should then be approx. 60 mm (2.4 in).
$C_3 > 80\,\%$	from C_3 max, please inquire.

Shaft load F_W

D

$$F_W \approx F_U \cdot C_1 + m_W \cdot g \qquad [N]$$

Power requirement at the drive drum P_A

E

$$P_A = \frac{F_U \cdot v}{1000 \cdot 60} \qquad v \text{ in m/min} \qquad [kW]$$

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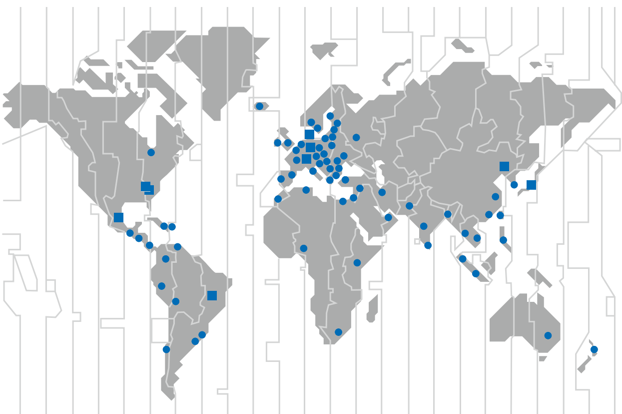

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