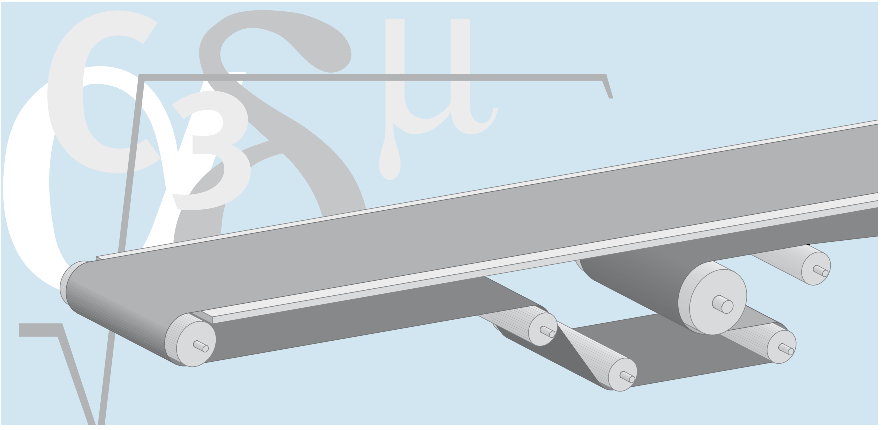
siegling transilon
conveyor and processing belts

Calculation methods – conveyor belts



This brochure contains advanced equations, figures and recommendations, based on our longstanding experience. Results calculated can however differ from our calculation program B_Rex (free to download from the Internet at www.forbo-siegling.com).

These variations are due to the very different approaches taken: while B_Rex is based on empirical measurements and requires a detailed description of the machinery, the calculation methods shown here are based on general, simple physical equations, supplemented by certain factors that include a safety margin.

In the majority of cases, the safety margin in calculations in this brochure will be greater than in the corresponding B_Rex calculation.

Further information on machine design can be found in our brochure, ref. no. 305 "Recommendations for machine design."

Contents

Siegling – total belting solutions

MOVEMENT SYSTEMS

Terminology

Key to the abbreviations

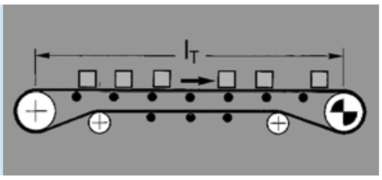
Designation	Abbreviation	Unit
Drum and roller width	b	mm
Belt width	b_0	mm
Calculation factors	C..	–
Drum and roller diameter	d	mm
Drive drum diameter	d_A	mm
Rolling resistance of support rollers	f	–
Tensile force	F	N
Maximum belt pull (on the drive drum)	F_1	N
Minimum belt pull (on the drive drum)	F_2	N
Force of the tensioning weight	F_R	N
Effective pull	F_U	N
Tensioning drum weight	F_{TR}	N
Steady state shaft load on the drive drum	F_{WA}	N
Initial value of the shaft load	$F_{Winitial}$	N
Relaxed shaft load on the return drum	F_{WU}	N
Acceleration due to gravity (9.81 m/s^2)	g	m/s^2
Difference in the drum radii (crowning)	h	mm
Conveying height	h_T	m
Relaxed belt pull at 1% elongation per unit of width	$k_{1\%}$	N/mm
Support roller pitch on upper side	l_0	mm
Transition length	l_S	mm
Support roller pitch on return side	l_u	mm
Geometrical belt length	L_g	mm
Length of conveyor	l_T	m
Mass of the goods conveyed over the entire length conveyed (total load)	m	kg
Mass of the goods conveyed on the top side (total load)	m_1	kg
Mass of the goods conveyed on the return side (total load)	m_2	kg
Mass of the belt	m_B	kg
Mass of the goods conveyed per m length conveyed on the upper face (line load)	m'_0	kg/m
Mass of all rotating drums, except for drive drum	m_R	kg
Mass of the goods conveyed per m length conveyed on the return side (line load)	m'_u	kg/m
Mechanical motor power	P_M	kW
Mechanical power calculated on the drive drum	P_A	kW
Production tolerance	Tol	%
Friction coefficient when running over roller	μ_R	–
Friction coefficient for accumulated conveying	μ_{ST}	–
Friction coefficient when running over table support	μ_T	–
Belt velocity	v	m/s
Volume flow for bulk goods conveying	$\dot{V}$	m^3/h
Total take-up range	X	mm
Belt sag	y_B	mm
Drum deflection	y_{Tr}	mm
Margin for take-up range	Z	mm
Machine’s angle of inclination	α	°
Arc of contact on the drive drum (or snub roller)	β	°
Opening angle on the tensioning drum	γ	°
Belt elongation (pre-tensioning with weight)	ΔL	mm
Permitted angle of inclination for unit goods	δ	°
Elongation at fitting	ε	%
Maximum belt elongation	ε_{max}	%
Drive efficiency	η	–
Bulk density of goods conveyed	ρ_S	kg/m^3

Unit goods conveying systems

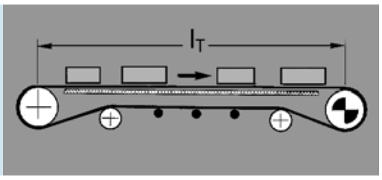
Load examples to establish the maximum effective pull F_u [N]

$m = l_T \cdot$ Weight of conveyed goods per metre

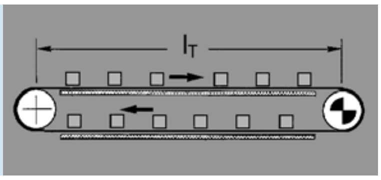
$F_U = \mu_R \cdot g \cdot (m + m_B + m_R)$ [N]



$F_U = \mu_T \cdot g \cdot (m + \frac{m_B}{2}) + \mu_R \cdot g \, (\frac{m_B}{2} + m_R)$ [N]



$F_U = \mu_T \cdot g \cdot (m_1 + m_2 + m_B)$ [N]

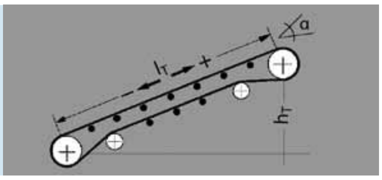


Direction conveyed upwards:

$F_U = \mu_R \cdot g \, (m + m_B + m_R) + g \cdot m \cdot \sin \alpha$ [N]

Direction conveyed downwards:

$F_U = \mu_R \cdot g \, (m + m_B + m_R) - g \cdot m \cdot \sin \alpha$ [N]

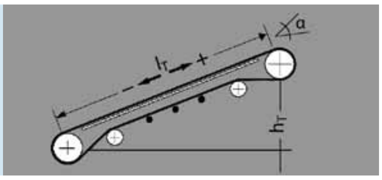


Direction conveyed upwards:

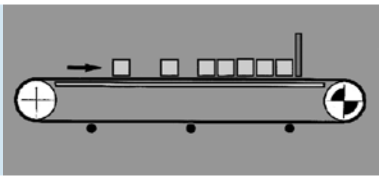
$F_U = \mu_T \cdot g \, (m + \frac{m_B}{2}) + \mu_R \cdot g \, (\frac{m_B}{2} + m_R) + g \cdot m \cdot \sin \alpha$ [N]

Direction conveyed downwards:

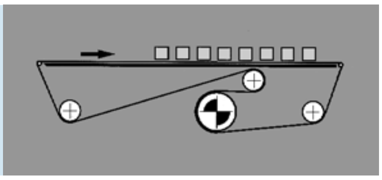
$F_U = \mu_T \cdot g \, (m + \frac{m_B}{2}) + \mu_R \cdot g \, (\frac{m_B}{2} + m_R) - g \cdot m \cdot \sin \alpha$ [N]



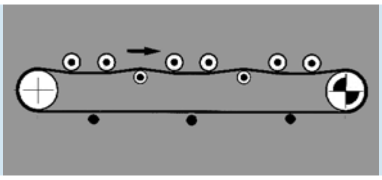
$F_U = \mu_T \cdot g \, (m + \frac{m_B}{2}) + \mu_R \cdot g \, (\frac{m_B}{2} + m_R) + \mu_{ST} \cdot g \cdot m$ [N]

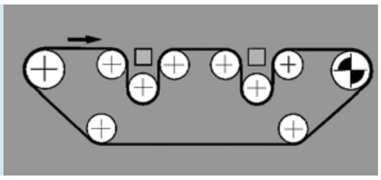


F_U = please enquire [N]



F_U = please enquire [N]





Friction coefficients μ_S for various coatings (guidelines)

	0, A0, E0, T, U0, P	NOVO	U1, V1, VH	UH, V2H, U2H, V5H, V10H	TXO (Amp Miser)
μ_T (table)	0.33	0.33	0.5	0.5	0.18
μ_T (galvanised slider beds)	–	–	–	–	0.24
μ_R (roller)	0.033	0.033	0.033	0.033	–
μ_{ST} (accumulated)	0.33	0.33	0.5	0.5	–

Please note:
The friction coefficients stated are based on experience with older friction surfaces that have been subjected to standard wear and tear and soiling. These friction coefficients are about 1.5 times higher than those for new surfaces.

Maximum belt pull F_1

$$F_1 = F_U \cdot C_1 \quad [N]$$

If effective pull F_U can be calculated

$$F_1 = \frac{P_M \cdot \eta \cdot C_1 \cdot 1000}{v} \quad [N]$$

If the effective pull F_U cannot be calculated, F_1 can be established from the motor power installed P_M.

Factor C_1
(applies to the drive drums)

Siegling Transilon Underside coating	V3, V5, U2, A5, E3			V1, U1, UH, U2H, V2H, V5H		
Arc of contact β	**180°**	**210°**	**240°**	**180°**	**210°**	**240°**
Smooth steel drum						
dry	1.5	1.4	1.3	1.8	1.6	1.5
wet	3.7	3.2	2.9	5.0	4.0	3.0
Lagged drum						
dry	1.4	1.3	1.2	1.6	1.5	1.4
wet	1.8	1.6	1.5	3.7	3.2	2.9

Siegling Transilon Underside coating	0, U0, NOVO, E0, A0, T, P			TX0 (AmpMiser)		
Arc of contact β	**180°**	**210°**	**240°**	**180°**	**210°**	**240°**
Smooth steel drum						
dry	2.1	1.9	1.7	3.3	2.9	2.6
wet	not recommended			not recommended		
Lagged drum						
dry	1.5	1.4	1.3	2.0	1.8	1.7
wet	2.1	1.9	1.7	not recommended		

Factor C_2
Checking the Transilon type selected

$$\frac{F_1}{b_0} \leq C_2 \quad \left[\frac{N}{mm}\right]$$

if the value $\frac{F_1}{b_0}$ is larger than C_2,

a stronger belt type (with a higher $k_{1\%}$ value) must be used.

C_2 indicates the max. permitted belt pull per unit width for the belt type:

$$C_2 = \varepsilon_{max} \cdot k_{1\%}$$

You can find details on the maximum elongations in the product data sheets.
If these are not available, the following can be assumed (but not guaranteed):

Tension member Type	Polyester Polyester (key letter "E")	Aramide (key letter "AE")
Examples of type classes	E 2/1, E 3/1, E 4/2, E 6/1, NOVO, E 8/2, E 10/M, E 12/2, E 15/2, E 15/M, E 18/3, E 20/M, E 30/3, E 44/3	AE 48/H, AE 80/3, AE 100/3, AE 140/H, AE 140/3
ε_{max} in %	2.0	0.8

Note: If belts have been perforated, b_0 must be reduced by the total width of the holes at a typical cross section. In the case of extreme temperatures, the C_2 factors change. Please enquire.

Minimum diameter of the drive drums d_A

$$d_A = \frac{F_U \cdot C_3 \cdot 180}{b_0 \cdot \beta} \quad [mm]$$

Factor C_3
(applies to the drive drums)

Siegling Transilon Underside coating	V3, V5, U2, A5, E3	V1, U1, UH	0, U0, NOVO, T, P
Smooth steel drum			
dry	25	30	40
wet	50	Not recommended	Not recommended
Lagged drum			
dry	25	25	30
wet	30	40	40

Mechanical capacity calculated on the drive drum P_A

$$P_A = \frac{F_U \cdot v}{1000} \quad [kW]$$

Mechanical capacity required P_M

$$P_M = \frac{P_A}{\eta} \; [kW] = \text{ the next largest, standard motor is selected}$$

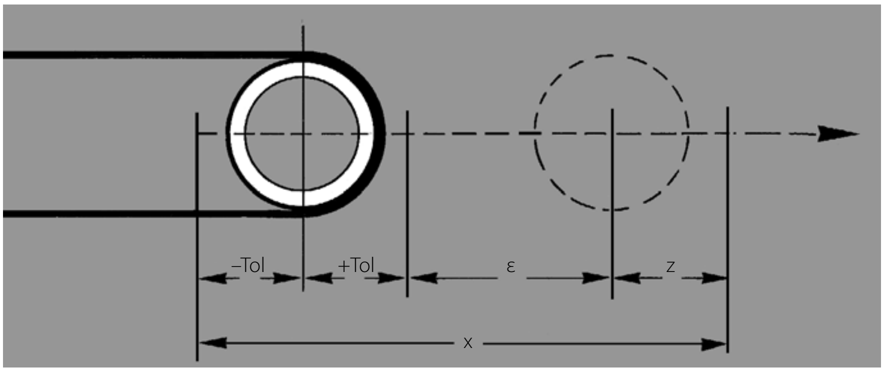
Take-up range for screw-operated take-up systems

The following factors must be taken into account when establishing the take-up range:

1. The approximate magnitude of elongation at fitting ε of the belt, resulting from the belt load. To establish ε, see pages 7 and 8.

2. The production tolerances (Tol) of the belt as regards the length.

3. Any external influences that might necessitate greater elongation (tensioning) than usual, or might require a safety margin, such as for example the impact of temperature, stop-and-go operation.

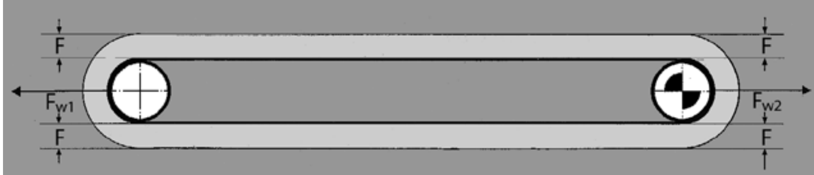


Generally, depending on the load, elongation at fitting, ranging from approx. 0.2 % to 1 %, is sufficient, so that normally a take-up range x of approx. 1 % of the belt length is adequate.

Guidelines for shaft load at rest with tensile force F

When you are estimating the shaft loads, please assess the different levels of belt pull when the conveyor is at rest and in a steady state.

At rest



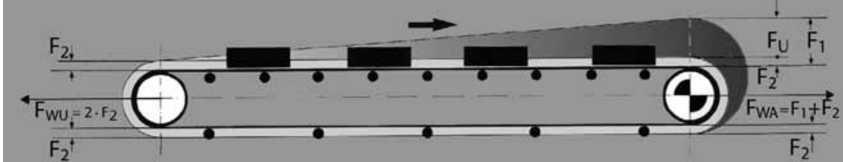
$F_{W1} = F_{W2} = 2 \cdot F$ $\quad F \approx ε\% \cdot k_{1\%} \cdot b_0$ $\quad$ [N]

Guidelines for elongation at fitting ε for head drives

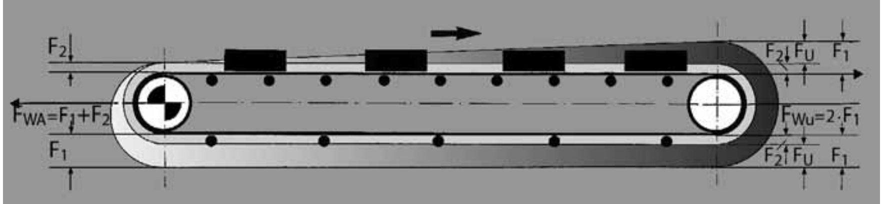
The minimum elongation at fitting for head drives is:

$$ε \approx \frac{F_U/2 + 2 \cdot F_2}{2 \cdot k_{1\%} \cdot b_0} \quad [\%]$$

Head drive in steady state forces



$F_2 = F_1 - F_U$ $\quad F_{WA} = F_1 + F_2$



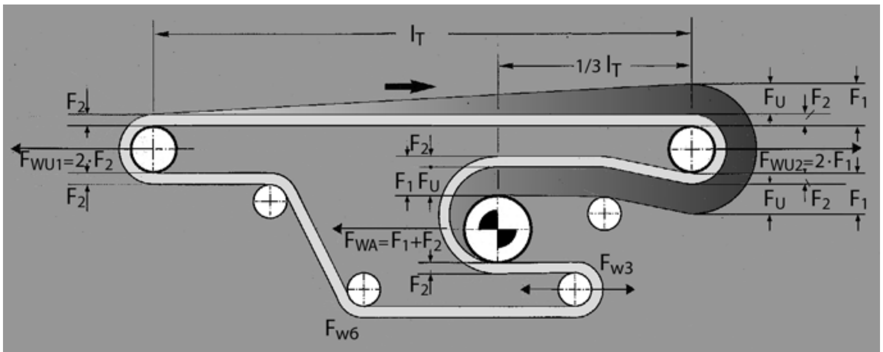
Tail drive in steady state forces

$F_2 = F_1 - F_U$

Guidelines for elongation at fitting ε for tail drives

The minimum elongation at fitting for return side drives is:

$$\varepsilon = \frac{F_U/2 + 2 \cdot F_2 + F_U}{2 \cdot k_{1\%} \cdot b_0} \quad [\%]$$



Return side drive in steady state

Guidelines for elongation at fitting ε for return-side drives

The minimum elongation at fitting for operating head drives is:

$$\varepsilon = \frac{F_U\,(C_1 - K)}{k_{1\%} \cdot b_0} \quad [\%]$$

Guidelines for steady state shaft load

Typical drive drum β = 180°

$$F_{WA} = F_1 + F_2 \quad [N]$$

Typical end drum β = 180°

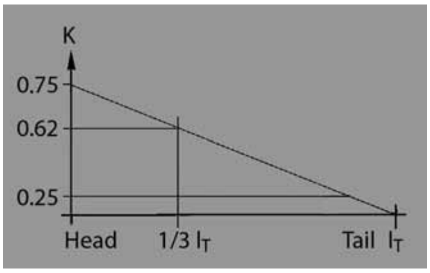
$$F_{W3} = 2 \cdot F_2 \quad [N]$$

Typical snub roller β = 60°

$$F_{W6} = \sqrt{2 \cdot F_2 \cdot \sin(\beta/2)} \quad [N]$$

Typical drive druml β ≠ 180°

$$F_{WA} = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos\beta} \quad [N]$$



K for head drives = 0.75
K for return-side drives = 0.62
K for tail drives = 0.25

Shaft load when tensioning belts

Tension members made of synthetic materials display significant relaxation behaviour. As a result, the relaxed $k_{1\%}$ value is taken as a basis for calculating belts in line with ISO 21181. It describes the probable long-term force-elongation properties of the belt material that has been subjected to stress due to deflection and load change. This produces the calculation force F_W.

This implies that higher belt forces $F_{Winitial}$ will occur when tensioning the belt. They will have to be taken into account when dimensioning the drum and its components (bearings). The following value can be assumed as a reference:

$$F_{Winitial} = F_W \cdot 1.5$$

In critical cases, we recommend you contact application engineers at Forbo Siegling.

Dimensioning force-dependent take-up systems

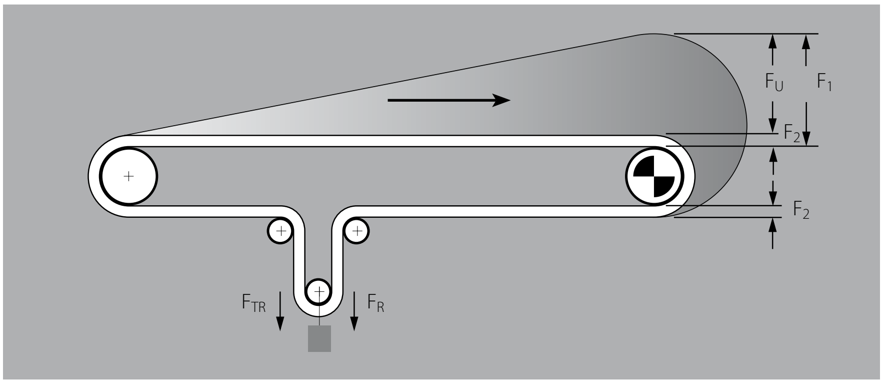
Establishing F_R

In weight-loaded take-up systems, the tension weight must generate the minimum belt pull F_2 to achieve perfect grip of the belt on the drive drum (spring, pneumatic and hydraulic take-up systems work on a similar principle).

The tension weight must be able to move freely. The take-up system must be installed behind the drive section. Reverse operation is not possible. The take-up range depends on the effective pull, the tensile force F_2 required, elongation of the belt ΔL, the production tolerance Tol, the safety margin for tensioning Z and the belt selected.

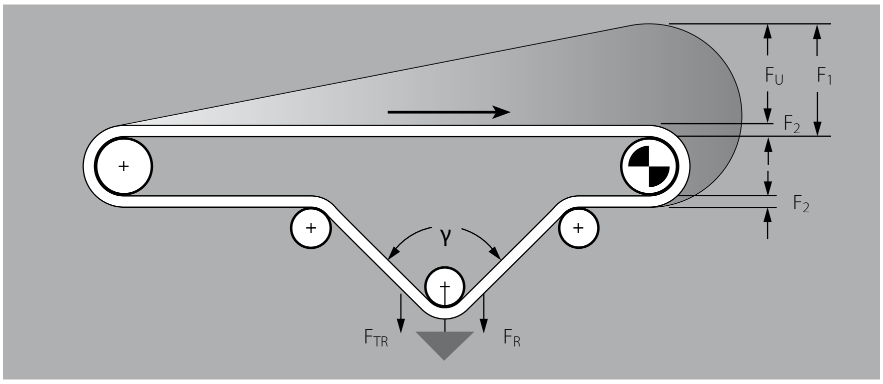
$$F_R = 2 \cdot F_2 - F_{TR} \qquad [N]$$

Example for establishing the tension weight FR [N] at 180° arc of contract (F_{TR} = tensioning drum weight [N]).



$$F_R = 2 \cdot F_2 \cdot \cos \frac{\gamma}{2} - F_{TR} \qquad [N]$$

Example for establishing the tension weight F_R [N] at an angle γ according to the drawing (F_{TR} = tensioning drum weight [N]).



Establishing belt elongation ΔL

In force-driven take-up systems, the overall elongation of the belt changes, according to the level of the effective pull. The change in belt elongation ΔL has to be absorbed by the take-up system. For head drives ΔL is calculated as

$$\Delta L = \frac{F_U/4 + F_{TR} + F_R}{k_{1\%} \cdot b_0} \cdot L_g \qquad [mm]$$

Bulk goods conveying systems

Longitudinal angle of inclination δ

Guidelines for the longitudinal angle of inclination δ permissible in various bulk goods. The machinery's actual angle of inclination α must be less than δ.

These values depend on the particle shape, size and mechanical properties of the goods conveyed, regardless of any conveyor belt coating.

Bulk goods	δ (approx.°)
Ash, dry	16
Ash, wet	18
Soil, moist	18 – 20
Grain, except oats	14
Lime, lumps	15
Potatoes	12
Gypsum, pulverised	23
Gypsum, broken	18
Wood, chips	22 – 24
Artificial fertilizer	12 – 15
Flour	15 – 18

Bulk goods	δ (approx.°)
Salt, fine	15 – 18
Salt, rock	18 – 20
Loam, wet	18 – 20
Sand, dry, wet	16 – 22
Peat	16
Sugar, refined	20
Sugar, raw	15
Cement	15 – 20

Bulk density of some bulk goods ρ_S

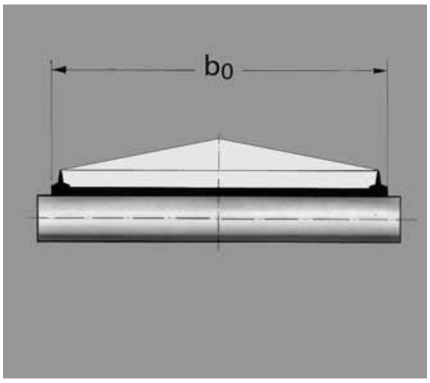
Goods conveyed	Bulk density ρ_S [10^3 kg/m^3]
Ash, cold, dry	0.7
Soil, moist	1.5 – 1.9
Grain (except oats)	0.7 – 0.85
Wood, hard	0.6 – 1.2
Wood, soft	0.4 – 0.6
Wood, chips	0.35
Charcoal	0.2
Pulses	0.85
Lime, lumps	1.0 – 1.4
Artificial fertilizer	0.9 – 1.2
Potatoes	0.75
Salt, fine	1.2 – 1.3
Salt, rock	2.1
Gypsum, pulverised	0.95 – 1.0

Goods conveyed	Bulk density ρ_S [10^3 kg/m^3]
Gypsum, broken	1.35
Flour	0.5 – 0.6
Clinker	1.2 – 1.5
Loam, dry	1.5 – 1.6
Loam, wet	1.8 – 2.0
Sand, dry	1.3 –1.4
Sand, wet	1.4 – 1.9
Soap, flakes	0.15 – 0.35
Slurry	1.0
Peat	0.4 – 0.6
Sugar, refined	0.8 – 0.9
Sugar, raw	0.9 – 1.1
Sugarcane	0.2 – 0.3

Volume flow $\dot{V}$ for belts lying flat

The table shows the hourly volume flow (m^3/h) at a belt velocity of v = 1 m/s. Conveyor belt lying flat and horizontal. The belt is equipped with 20 mm high longitudinal profiles T20 on the belt edges of the top face.

b_0 mm	400	500	650	800	1000	1200	1400
Angle of surcharge 0°	25	32	42	52	66	80	94
Angle of surcharge 10°	40	57	88	123	181	248	326



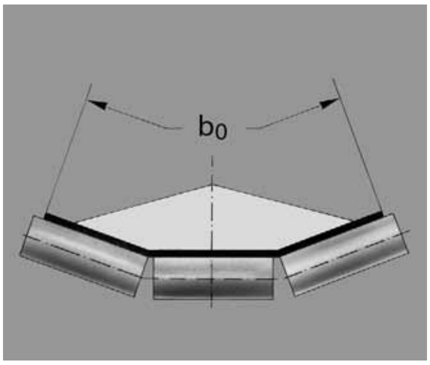
MOVEMENT SYSTEMS

Volume flow $\dot{V}$ for troughed conveyor belts

in m^3/h at a belt velocity of 1 m/s.

Note:
Under real world conditions, the theoretical values for volume flow are hardly ever reached as they only apply to horizontal belts with perfectly even loads. Uneven loads and the properties of the goods conveyed can decrease the amount by approx. 30 %.

b_0 mm	400	500	650	800	1000	1200	1400
Troughed angle 20°							
Angle of surcharge 0°	21	36	67	105	173	253	355
Angle of surcharge 10°	36	60	110	172	281	412	572
Troughed angle 30°							
Angle of surcharge 0°	30	51	95	149	246	360	504
Angle of surcharge 10°	44	74	135	211	345	505	703



Factor C_6

In inclined conveying, the theoretical quantity of goods conveyed is slightly less. It is calculated by applying the factor C_6 which depends on the conveying angle α.

Conveying angle α [°]	2	4	6	8	10	12
Factor C_6	1.0	0.99	0.98	0.97	0.95	0.93
Conveying angle α [°]	**14**	**16**	**18**	**20**	**22**	
Factor C_6	0.91	0.89	0.85	0.81	0.76	

Factor C_4

Additional effective pull, for example from scrapers and cleaning devices, is taken into account by including the factor C_4.

l_T [m]	25	50	75	100	150	200
C_4	2	1.9	1.8	1.7	1.5	1.3

Rolling resistance for support rollers f

f = 0.025 for roller bearings
f = 0.050 for slide bearings

Establishing the mass of goods conveyed m

$$m = \frac{\dot{V} \cdot \delta_S \cdot l_T \cdot 3.6}{v} \quad [kg]$$

$$F_U = g \cdot C_4 \cdot f\,(m + m_B + m_R) \pm g \cdot m \cdot \sin\alpha \quad [N]$$

Calculation as for unit goods

Establishing the effective pull F_U

(–) downwards
(+) upwards

Support roller pitches

The support roller pitch depends on the belt pull and the masses. The following equation is used to calculate it:

If maximum sag of 1 % is permitted, (i.e. $y_B = 0.01\ l_0$)

Recommendation $l_0\ \text{max} \leq 2b_0$
$l_u \approx 2 - 3\ l_0\ \text{max}$

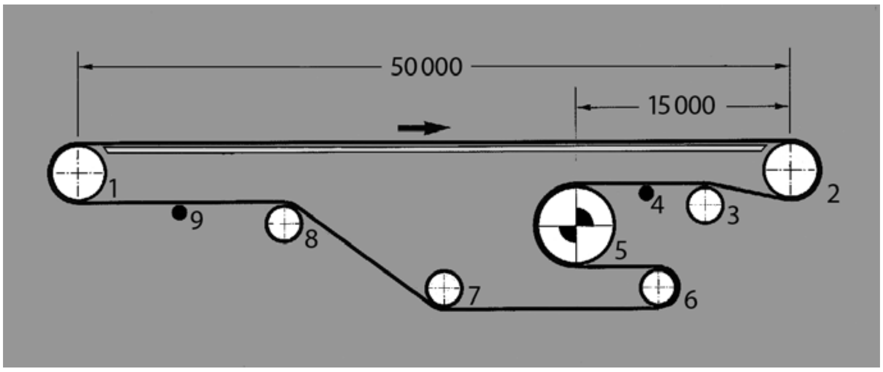
$$l_0 = \sqrt{\frac{y_B \cdot 800 \cdot F}{m'_0 + m'_B}} \quad [mm]$$

$$l_0 = \frac{8 \cdot F}{m'_0 + m'_B} \quad [mm]$$

l_0 = Support roller pitch on upper side in mm
y_B = Maximum conveyor belt sag in mm
F = Belt pull in the place concerned in N
$m'_0 + m'_B$ = Weight of goods conveyed and belt in kg/m

Calculation example for unit goods conveying

In a goods sorting system, conveyor belts are loaded with goods and sent to the distribution centre. Horizontal conveying, skid plate support, return drive systems as shown on the sketch, drive via the top face of the belt, drive drum with lagging, screw-operated tensioning system, 14 support rollers. Proposed belt type: Siegling Transilon E8/2 U0/V5H MT black (900026) with $k_{1\%} = 8$ N/mm.



End drums 1, 2, 6
Snub rollers 3, 7, 8
Drive drum 5
Support rollers 4, 9, and various tension drums 6.

Length of conveyor	l_T	= 50 m
Geometrical belt length	L_g	= 105000 mm
Belt width	b_0	= 600 mm
Total load	m	= 1200 kg
Arc of contact	β	= 180°
v = ca. 0.8 m/s	g	= 9.81 m/s^2
Mass rollers	m_R	= 570 kg (all drums except for 5)

Effective pull F_U [N]

$$F_U = \mu_T \cdot g\,(m + \frac{m_B}{2}) + \mu_R \cdot g\,(\frac{m_B}{2} + m_R)$$

$$F_U = 0.33 \cdot 9.81\,(1200 + \frac{157.5}{2}) + 0.033 \cdot 9.81\,(\frac{157.5}{2} + 570)$$

$$F_U \approx 4340 \text{ N}$$

m = 1200 kg
μ_R = 0.033
μ_T = 0.33
m_B = 157.5 kg (from 2.5 kg/m^2 · 105 m · 0.6 m)

Maximum belt pull F_1 [N]

F_U = 4350 N
C_1 = 1.6

$$F_1 = F_U \cdot C_1$$

$$F_1 = 4350 \cdot 1.6$$

$$F_1 \approx 6960 \text{ N}$$

Checking the belt type selected

F_1 = 6960 N
b_0 = 600 mm
$k_{1\%}$ = 8 N/mm

$$\frac{F_1}{b_0} \leq C_2$$

$$\frac{6960}{600} \leq 2 \cdot 8 \text{ N/mm}$$

11,6 N/mm ≤ 16 N/mm

The belt type has been chosen correctly.

F_U = 4340 N
C_3 = 25
β = 180°
b_0 = 600 mm

$$d_A = \frac{F_U \cdot C_3 \cdot 180°}{b_0 \cdot \beta} \quad [mm]$$

$$d_A = \frac{4340 \cdot 25 \cdot 180°}{600 \cdot 180°} \quad [mm]$$

d_A = 181 mm

d_A dimensioned at 200 mm

Minimum drive drum diameter

F_U = 4350 N
v = 0.8 m/s

$$P_A = \frac{F_U \cdot v}{1000} \quad [kW]$$

$$P_A = \frac{4350 \cdot 0.8}{1000}$$

$P_A \approx 3.5$ kW

Power P_A on the drive drum

P_A = 3.5 kW
η = 0.8 (assumed)

$$P_M = \frac{P_A}{\eta} \quad [kW]$$

$$P_M = \frac{3.5}{0.8} \quad [kW]$$

$P_M \approx 4.4$ kW

P_M at 5.5 kW or higher

Motor power required P_M

F_U = 4350 N
C_1 = 1.6
K = 0.62
$k_{1\%}$ = 8 N/mm for E8/2 U0/V5H black
b_0 = 600 mm

$$\varepsilon = \frac{F_U\,(C_1 - K)}{k_{1\%} \cdot b_0} \quad [\%]$$

$$\varepsilon = \frac{4350\,(1.6 - 0.62)}{8 \cdot 600} \quad [\%]$$

$\varepsilon \approx 0.9$ %

Minimum elongation at fitting for return drive

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Shaft load in steady state drum drum 2 (return drum)

Simplified calculation assuming $\beta = 180°$

$F_1 = 6960$ N

$F_{W2} = 2 \cdot F_1$

$F_{W2} = 2 \cdot 6960$ N

$F_{W2} \approx 13920$ N

Shaft load in steady state drum drum 1 (return drum)

$F_2 = F_1 - F_U$
$F_2 = 6960 - 4350$
$F_2 = 2610$ N

$F_{W1} = 2 \cdot F_2$

$F_{W1} = 2 \cdot 2610$ N

$F_{W1} \approx 5220$ N

Shaft load in steady state drum drum 5 (Drive drum)

$F_1 = 6960$ N
$F_2 = F_1 - F_U$
$F_2 = 6960 - 4350$
$F_2 = 2610$ N

$F_{W5} = F_1 + F_2$

$F_{W5} = 6960 + 2610$

$F_{W5} \approx 9570$ N

Shaft load in steady drum 3 (snub roller)

Governed by minimum belt pull F_2, F_{W3} is calculated using the equation on page 7.

At rest, tensile forces are defined on the top and underside by elongation at fitting ε. The tensile force F is calculated according to:

$F = \varepsilon\,[\%] \cdot k_{1\%} \cdot b_0$ [N]

Example for a drum with β = 180°
Arc of contact
(In our example, this force is exerted equally on drums 1, 5 and 6 because of the 180° arc of contact).

$F_W = 2 \cdot F$
$F_W = 2 \cdot 0.9 \cdot 8 \cdot 600$
$F_W \approx 8640$ N

When β ≠ 180° the following applies when determining F_W ($F_1 = F_2$ can be assumed at rest).

$F_W = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos\beta}$

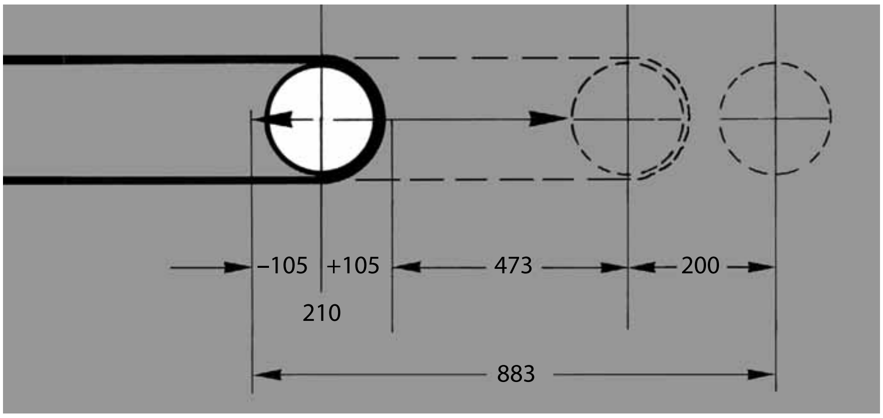
$F_W = $ [N]

Shaft load at rest

To compare rest and steady state modes, please observe the different shaft loads in drum 1.

F_{W1} at rest = 8640 N
F_{W1} steady state = 5220 N

Note:
When designing machinery, both modes must be taken into account.



Take-up range

Tol = ± 0.2 %
ε = 0.9 %
L_g = 105000 mm
Z = 200 mm

$$X = \frac{\frac{2 \cdot Tol \cdot L_g}{100} + \frac{\varepsilon \cdot L_g}{100}}{2} + Z \quad \text{[mm]}$$

$$X = \frac{\frac{2 \cdot 0.2 \cdot 105000}{100} + \frac{0.9 \cdot 105000}{100}}{2} + 200 \quad \text{[mm]}$$

$X = 210 + 473 + 200$ [mm]

$X \approx 883$ mm

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