CALCULATION METHODS – CONVEYOR BELTS

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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.”
# TERMINOLOGY

## Key to the abbreviations

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<tr>
<th>Designation</th>
<th>Abbreviation</th>
<th>Unit</th>
</tr>
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<td>Drum and roller width</td>
<td>b</td>
<td>mm</td>
</tr>
<tr>
<td>Belt width</td>
<td>b0</td>
<td>mm</td>
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<tr>
<td>Calculation factors</td>
<td>C</td>
<td>-</td>
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<tr>
<td>Drum and roller diameter</td>
<td>d</td>
<td>mm</td>
</tr>
<tr>
<td>Drive drum diameter</td>
<td>d1</td>
<td>mm</td>
</tr>
<tr>
<td>Rolling resistance of support rollers</td>
<td>f</td>
<td>-</td>
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<tr>
<td>Tensile force</td>
<td>F</td>
<td>N</td>
</tr>
<tr>
<td>Maximum belt pull (on the drive drum)</td>
<td>F1</td>
<td>N</td>
</tr>
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<td>Minimum belt pull (on the drive drum)</td>
<td>F2</td>
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<tr>
<td>Force of the tensioning weight</td>
<td>FR</td>
<td>N</td>
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<tr>
<td>Effective pull</td>
<td>FU</td>
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<td>Tensioning drum weight</td>
<td>FR</td>
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<td>Steady state shaft load on the drive drum</td>
<td>FWa</td>
<td>N</td>
</tr>
<tr>
<td>Initial value of the shaft load</td>
<td>FWinit</td>
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</tr>
<tr>
<td>Relaxed shaft load on the return drum</td>
<td>FWU</td>
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</tr>
<tr>
<td>Acceleration due to gravity (9.81m/s²)</td>
<td>g</td>
<td>m/s²</td>
</tr>
<tr>
<td>Difference in the drum radii (crowning)</td>
<td>h</td>
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<td>Conveying height</td>
<td>hT</td>
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<td>Relaxed belt pull at 1% elongation per unit of width</td>
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<tr>
<td>Support roller pitch on upper side</td>
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<td>mm</td>
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<td>Transition length</td>
<td>l1</td>
<td>mm</td>
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<td>Support roller pitch on return side</td>
<td>l2</td>
<td>mm</td>
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<td>Geometrical belt length</td>
<td>Lg</td>
<td>mm</td>
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<tr>
<td>Length of conveyor</td>
<td>lF</td>
<td>m</td>
</tr>
<tr>
<td>Mass of the goods conveyed over the entire length conveyed</td>
<td>m</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of the goods conveyed on the top side (total load)</td>
<td>m1</td>
<td>kg</td>
</tr>
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<td>Mass of the goods conveyed on the return side (total load)</td>
<td>m2</td>
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<tr>
<td>Mass of the belt</td>
<td>mB</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of the goods conveyed per m length conveyed on the upper</td>
<td>m1u</td>
<td>kg/m</td>
</tr>
<tr>
<td>face (line load)</td>
<td>m1u1</td>
<td>kg/m</td>
</tr>
<tr>
<td>Mass of all rotating drums, except for drive drum</td>
<td>mR</td>
<td>kg</td>
</tr>
<tr>
<td>Mechanical motor power</td>
<td>PM</td>
<td>kW</td>
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<td>Mechanical power calculated on the drive drum</td>
<td>PA</td>
<td>kW</td>
</tr>
<tr>
<td>Production tolerance</td>
<td>Tol</td>
<td>%</td>
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<td>Friction coefficient when running over roller</td>
<td>µR</td>
<td>-</td>
</tr>
<tr>
<td>Friction coefficient for accumulated conveying</td>
<td>µST</td>
<td>-</td>
</tr>
<tr>
<td>Friction coefficient when running over table support</td>
<td>µT</td>
<td>-</td>
</tr>
<tr>
<td>Belt velocity</td>
<td>V</td>
<td>m/s</td>
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<tr>
<td>Volume flow for bulk goods conveying</td>
<td>V</td>
<td>m³/h</td>
</tr>
<tr>
<td>Total take-up range</td>
<td>X</td>
<td>mm</td>
</tr>
<tr>
<td>Belt sag</td>
<td>yb</td>
<td>mm</td>
</tr>
<tr>
<td>Drum deflection</td>
<td>yD</td>
<td>mm</td>
</tr>
<tr>
<td>Margin for take-up range</td>
<td>Z</td>
<td>mm</td>
</tr>
<tr>
<td>Machine’s angle of inclination</td>
<td>α</td>
<td>°</td>
</tr>
<tr>
<td>Arc of contact on the drive drum (or snub roller)</td>
<td>β</td>
<td>°</td>
</tr>
<tr>
<td>Opening angle on the tensioning drum</td>
<td>γ</td>
<td>°</td>
</tr>
<tr>
<td>Belt elongation (pre-tensioning with weight)</td>
<td>ΔL</td>
<td>mm</td>
</tr>
<tr>
<td>Permitted angle of inclination for unit goods</td>
<td>δ</td>
<td>°</td>
</tr>
<tr>
<td>Elongation at fitting</td>
<td>ε</td>
<td>%</td>
</tr>
<tr>
<td>Maximum belt elongation</td>
<td>εmax</td>
<td>%</td>
</tr>
<tr>
<td>Drive efficiency</td>
<td>η</td>
<td>-</td>
</tr>
<tr>
<td>Bulk density of goods conveyed</td>
<td>ρs</td>
<td>kg/m³</td>
</tr>
</tbody>
</table>
UNIT GOODS CONVEYING SYSTEMS

Load examples to establish the maximum effective pull $F_U\ [N]$

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

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

$F_U = \mu_T \cdot g \cdot (m + m_2) + \mu_R \cdot g \cdot (\frac{m_b}{2} + m_l)\ [N]$  

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

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

$F_U = \mu_R \cdot g \cdot (m + m_b + m_l) \pm g \cdot m \cdot \sin \alpha\ [N]$  

$F_U = \mu_T \cdot g \cdot (m + m_2) + \mu_R \cdot g \cdot (\frac{m_b}{2} + m_l) \pm g \cdot m \cdot \sin \alpha\ [N]$  

$F_U = \mu_T \cdot g \cdot (m + m_2) + \mu_R \cdot g \cdot (\frac{m_b}{2} + m_l) + \mu_{ST} \cdot g \cdot m\ [N]$  

$F_U = \text{please enquire}\ [N]$  

$F_U = \text{please enquire}\ [N]$
If the effective pull $F_U$ cannot be calculated, $F_1$ can be established from the motor power installed $P_M$.

$$F_1 = P_M \cdot \eta \cdot C_1 \cdot 1000$$

$[N]$
In the case of perforated belts please note: calculate the load-bearing belt width $b_0$ based on the number of perforations which decrease cross sections. Staggered perforations in particular can reduce the load-bearing belt width considerably. Reduce the figure for the load-bearing belt width $b_0$ by a further 20% to take tolerances for perforations and fabric into account.

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

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.

$C_2$ is a metric indicating the belt type’s maximum tension:

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

the product data sheets include specifications on the $\varepsilon_{\text{max}}$ maximum elongations during operation. If example calculations and rough estimates without a data sheet are required, the following assumption can be made (but not guaranteed):

If subjected to high temperatures of over 100 °C, the $C_2$ factors change. Please contact us.

The table on establishing the $C_2$ factor shows product examples for the tension member concerned.
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 $\varepsilon$ of the belt, resulting from the belt load. To establish $\varepsilon$, 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.

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.

Guidelines for elongation at fitting $\varepsilon$ for head drives

The minimum elongation at fitting for head drives is:

$$\varepsilon = \frac{F_U / 2 + 2 \cdot F_z}{2 \cdot k_1 \cdot b_0} \quad \text{[\%]}$$

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 elongation at fitting $\varepsilon$ for tail drives

The minimum elongation at fitting for return side drives is:

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

Tail drive in steady state forces

Guidelines for elongation at fitting $\varepsilon$ for return-side drives

The minimum elongation at fitting for operating head drives is:

$$\varepsilon = \frac{FU (C_1 - K)}{k_{is} \cdot b_0} \quad [\%]$$

Return side drive in steady state

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 $k_{10\%}$ 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 $F_{W\text{Initial}}$ 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:

$$F_{W\text{Initial}} = F_W \cdot 1.5$$

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

Guidelines for steady state shaft load

<table>
<thead>
<tr>
<th>Typical drive drum $\beta = 180^\circ$</th>
<th>Typical end drum $\beta = 180^\circ$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{W3} = F_1 + F_2$ [N]</td>
<td>$F_{W3} = 2 \cdot F_2$ [N]</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Typical snub roller $\beta = 60^\circ$</th>
<th>Typical drive drum $\beta \neq 180^\circ$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$F_{W6} = \sqrt{2} \cdot F_2 \cdot \sin (\beta/2)$ [N]</td>
<td>$F_{Wa} = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos \beta}$ [N]</td>
</tr>
</tbody>
</table>

Typical end drum $\beta = 180^\circ$

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

Typical drive drum $\beta \neq 180^\circ$

$$F_{Wa} = \sqrt{F_1^2 + F_2^2 - 2 \cdot F_1 \cdot F_2 \cdot \cos \beta}$$ [N]
Establishing FR

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 \( \Delta L \), the production tolerance \( \text{Tol} \), the safety margin for tensioning \( Z \) and the belt selected.

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

\[
FR = 2 \cdot F_2 - F_{TR} \quad [\text{N}]
\]

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 \( \Delta L \) has to be absorbed by the take-up system. For head drives \( \Delta L \) is calculated as

\[
\Delta L = \frac{F_u A_4 + F_{TR} + F_R}{K_{TR} \cdot b_0 \cdot l_g} \quad [\text{mm}]
\]

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

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

**Establishing belt elongation \( \Delta L \)**
Longitudinal angle of inclination $\delta$

Guidelines for the longitudinal angle of inclination $\delta$ permissible in various bulk goods. The machinery’s actual angle of inclination $\alpha$ must be less than $\delta$.

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

Bulk density of some bulk goods $\rho_S$

<table>
<thead>
<tr>
<th>Goods conveyed</th>
<th>Bulk density $\rho_S [10^3 \text{ kg/m}^3]$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ash, cold, dry</td>
<td>0.7</td>
</tr>
<tr>
<td>Soil, moist</td>
<td>1.5 – 1.9</td>
</tr>
<tr>
<td>Grain (except oats)</td>
<td>0.7 – 0.85</td>
</tr>
<tr>
<td>Wood, hard</td>
<td>0.6 – 1.2</td>
</tr>
<tr>
<td>Wood, soft</td>
<td>0.4 – 0.6</td>
</tr>
<tr>
<td>Wood, chips</td>
<td>0.35</td>
</tr>
<tr>
<td>Charcoal</td>
<td>0.2</td>
</tr>
<tr>
<td>Pulses</td>
<td>0.85</td>
</tr>
<tr>
<td>Lime, lumps</td>
<td>1.0 – 1.4</td>
</tr>
<tr>
<td>Artificial fertilizer</td>
<td>0.9 – 1.2</td>
</tr>
<tr>
<td>Potatoes</td>
<td>0.75</td>
</tr>
<tr>
<td>Salt, fine</td>
<td>1.2 – 1.3</td>
</tr>
<tr>
<td>Salt, rock</td>
<td>2.1</td>
</tr>
<tr>
<td>Gypsum, pulverised</td>
<td>0.95 – 1.0</td>
</tr>
</tbody>
</table>

Volume flow $V$ for belts lying flat

The table shows the hourly volume flow (m$^3$/h) at a belt velocity of $v = 1 \text{ m/s}$. 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.
Volume flow $V$ for troughed conveyor belts

in m³/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%.

<table>
<thead>
<tr>
<th>b₀ [mm]</th>
<th>400</th>
<th>500</th>
<th>650</th>
<th>800</th>
<th>1000</th>
<th>1200</th>
<th>1400</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Troughed angle 20°</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle of surcharge 0°</td>
<td>21</td>
<td>36</td>
<td>67</td>
<td>105</td>
<td>173</td>
<td>253</td>
<td>355</td>
</tr>
<tr>
<td>Angle of surcharge 10°</td>
<td>36</td>
<td>60</td>
<td>110</td>
<td>172</td>
<td>281</td>
<td>412</td>
<td>572</td>
</tr>
<tr>
<td><strong>Troughed angle 30°</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Angle of surcharge 0°</td>
<td>30</td>
<td>51</td>
<td>95</td>
<td>149</td>
<td>246</td>
<td>360</td>
<td>504</td>
</tr>
<tr>
<td>Angle of surcharge 10°</td>
<td>44</td>
<td>74</td>
<td>135</td>
<td>211</td>
<td>345</td>
<td>505</td>
<td>703</td>
</tr>
</tbody>
</table>

In inclined conveying, the theoretical quantity of goods conveyed is slightly less. It is calculated by applying the factor $C₆$ which depends on the conveying angle $\alpha$.

<table>
<thead>
<tr>
<th>Conveying angle $\alpha$ [°]</th>
<th>2</th>
<th>4</th>
<th>6</th>
<th>8</th>
<th>10</th>
<th>12</th>
<th>14</th>
<th>16</th>
<th>18</th>
<th>20</th>
<th>22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Factor $C₆$</td>
<td>1.0</td>
<td>0.99</td>
<td>0.98</td>
<td>0.97</td>
<td>0.95</td>
<td>0.93</td>
<td>0.91</td>
<td>0.89</td>
<td>0.85</td>
<td>0.81</td>
<td>0.76</td>
</tr>
</tbody>
</table>

Additional effective pull, for example from scrapers and cleaning devices, is taken into account by including the factor $C₄$.

<table>
<thead>
<tr>
<th>lₒ [m]</th>
<th>25</th>
<th>50</th>
<th>75</th>
<th>100</th>
<th>150</th>
<th>200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Factor $C₄$</td>
<td>2</td>
<td>1.9</td>
<td>1.8</td>
<td>1.7</td>
<td>1.5</td>
<td>1.3</td>
</tr>
</tbody>
</table>
Rolling resistance for support rollers $f$

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

Establishing the mass of goods conveyed $m$

$$m = \frac{V \cdot \delta_s \cdot l \cdot 3.6}{v} \quad [\text{kg}]$$

Establishing the effective pull $F_U$

(-) downwards
(+) upwards

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. $y_B = 0.01 \cdot l_0$)
- Recommendation $l_0 \max \leq 2b_0$
- $l_0 \approx 2 - 3 \cdot l_0 \max$

$$l_0 = \sqrt{\frac{y_B \cdot 800 \cdot F}{m_0 + m_B}} \quad [\text{mm}]$$

- $l_0$ = 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 E 8/2 0/VSH S/MT black (996141) with \( k_{1\%} = 8 \text{ N/mm} \).

---

**Effective pull \( F_U [N] \)**

\[
F_U = \mu_R \cdot g \left( \frac{m}{2} \right) + \mu_T \cdot g \left( \frac{m_B}{2} \right)
\]

\[
F_U = 0.33 \cdot 9.81 \left( \frac{1200}{2} \right) + 0.033 \cdot 9.81 \left( \frac{1575}{2} \right) + 570
\]

\[
F_U = 4340 \text{ N}
\]

- \( m = 1200 \text{ kg} \)
- \( \mu_R = 0.33 \)
- \( \mu_T = 0.33 \)
- \( m_B = 157.5 \text{ kg (from 2.5 kg/m}^2 \cdot 105 \text{ m} \cdot 0.6 \text{ m}) \)

**Maximum belt pull \( F_1 [N] \)**

\[
F_U = 4350 \text{ N}
\]

\[
C_1 = 1.6
\]

\[
F_1 = F_U \cdot C_1
\]

\[
F_1 = 4350 \cdot 1.6
\]

\[
F_1 = 6960 \text{ N}
\]

---

**Checking the belt type selected**

\[
\frac{F_1}{b_0} \leq C_2
\]

\[
\frac{6960}{600} \leq 1.5 \cdot 8 \text{ N/mm}
\]

\[
11.6 \text{ N/mm} \leq 12 \text{ N/mm}
\]

The belt type has been chosen correctly.
**CALCULATION EXAMPLE FOR UNIT GOODS CONVEYING**

**Minimum drive drum diameter**

\[ F_U = 4340 \, \text{N} \]

\[ C_3 = 25 \]

\[ \beta = 180^\circ \]

\[ b_0 = 600 \, \text{mm} \]

\[ d_a = \frac{F_U \cdot C_3 \cdot 180^\circ}{b_0 \cdot \beta} \quad [\text{mm}] \]

\[ d_a = \frac{4340 \cdot 25 \cdot 180^\circ}{600 \cdot 180^\circ} \quad [\text{mm}] \]

\[ d_a = 181 \, \text{mm} \]

\[ d_a \text{ dimensioned at } 200 \, \text{mm} \]

**Power \( P_A \) on the drive drum**

\[ F_U = 4350 \, \text{N} \]

\[ v = 0.8 \, \text{m/s} \]

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

\[ P_A = \frac{4350 \cdot 0.8}{1000} \]

\[ P_A = 3.5 \, \text{kW} \]

**Motor power required \( P_M \)**

\[ P_A = 3.5 \, \text{kW} \]

\[ \eta = 0.8 \text{ (assumed)} \]

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

\[ P_M = \frac{3.5}{0.8} \]

\[ P_M = 4.4 \, \text{kW} \]

\[ P_M \text{ at } 5.5 \, \text{kW} \text{ or higher} \]

**Minimum elongation at fitting for return drive**

\[ F_U = 4350 \, \text{N} \]

\[ C_1 = 1.6 \]

\[ K = 0.62 \]

\[ k_{1\%} = 8 \, \text{N/mm for } E \, 8/2 \, 0/V5H \, S/MT \, \text{black} \]

\[ b_0 = 600 \, \text{mm} \]

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

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

\[ \varepsilon = 0.9 \, \% \]
Simplified calculation assuming $\beta = 180^\circ$

$F_1 = 6960 \text{ N}$

$F_2 = F_1 - F_U$

$F_2 = 6960 - 4350$

$F_2 = 2610 \text{ N}$

Governed by minimum belt pull $F_2$, $F_{W3}$ is calculated using the equation on page 10.

$FW_2 = 2 \cdot F_1$

$FW_2 = 2 \cdot 6960 \text{ N}$

$FW_2 \approx 13920 \text{ N}$

$FW_1 = 2 \cdot F_2$

$FW_1 = 2 \cdot 2610 \text{ N}$

$FW_1 \approx 5220 \text{ N}$

$FW_5 = F_1 + F_2$

$FW_5 = 6960 + 2610$

$FW_5 \approx 9570 \text{ N}$

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

$FW_1$ at rest $= 8640 \text{ N}$

$FW_1$ steady state $= 5220 \text{ N}$

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

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

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

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

When $\beta \neq 180^\circ$ the following applies when determining $F_W$ ($F_1 = F_2$ can be assumed at rest).
**Take-up range**

- \( \text{Tol} = \pm 0.2\% \)
- \( \epsilon = 0.9\% \)
- \( L_d = 105000 \text{ mm} \)
- \( Z = 200 \text{ mm} \)

\[
X = \frac{2 \cdot \text{Tol} \cdot L_d}{100} \cdot \frac{\epsilon \cdot L_d}{100} + Z \quad [\text{mm}]
\]

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

\[X = 210 + 473 + 200 \quad [\text{mm}]\]

\[X = 883 \text{ mm}\]

**Calculation Example for Unit Goods Conveying**
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