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.”

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## Terminology

<table>
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<th>Designation</th>
<th>Abbreviation</th>
<th>Unit</th>
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<tbody>
<tr>
<td>Drum and roller width</td>
<td>b</td>
<td>mm</td>
</tr>
<tr>
<td>Belt width</td>
<td>b₀</td>
<td>mm</td>
</tr>
<tr>
<td>Calculation factors</td>
<td>C.</td>
<td>–</td>
</tr>
<tr>
<td>Drum and roller diameter</td>
<td>d</td>
<td>mm</td>
</tr>
<tr>
<td>Drive drum diameter</td>
<td>dₐ</td>
<td>mm</td>
</tr>
<tr>
<td>Rolling resistance of support rollers</td>
<td>f</td>
<td>–</td>
</tr>
<tr>
<td>Tensile force</td>
<td>Fₘ</td>
<td>N</td>
</tr>
<tr>
<td>Maximum belt pull (on the drive drum)</td>
<td>Fₘ₁</td>
<td>N</td>
</tr>
<tr>
<td>Minimum belt pull (on the drive drum)</td>
<td>Fₘ₂</td>
<td>N</td>
</tr>
<tr>
<td>Force of the tensioning weight</td>
<td>Fₘₙ₁</td>
<td>N</td>
</tr>
<tr>
<td>Effective pull</td>
<td>Fₘₚ₁</td>
<td>N</td>
</tr>
<tr>
<td>Tensioning drum weight</td>
<td>Fₘₚ₂</td>
<td>N</td>
</tr>
<tr>
<td>Steady-state shaft load on the drive drum</td>
<td>Fₛₚ₂</td>
<td>N</td>
</tr>
<tr>
<td>Initial value of the shaft load</td>
<td>Fₛₚ₁</td>
<td>N</td>
</tr>
<tr>
<td>Relaxed shaft load on the return drum</td>
<td>Fₛₚ₃</td>
<td>N</td>
</tr>
<tr>
<td>Acceleration due to gravity (9.81 m/s²)</td>
<td>g</td>
<td>m/s²</td>
</tr>
<tr>
<td>Difference in the drum radii (crowning)</td>
<td>h</td>
<td>mm</td>
</tr>
<tr>
<td>Conveying height</td>
<td>hₐ</td>
<td>m</td>
</tr>
<tr>
<td>Relaxed belt pull at 1% elongation per unit of width</td>
<td>kₘₘ₁</td>
<td>N/mm</td>
</tr>
<tr>
<td>Support roller pitch on upper side</td>
<td>lₗ</td>
<td>mm</td>
</tr>
<tr>
<td>Support roller pitch on return side</td>
<td>lₜ</td>
<td>mm</td>
</tr>
<tr>
<td>Geometrical belt length</td>
<td>Lₐ</td>
<td>mm</td>
</tr>
<tr>
<td>Length of conveyor</td>
<td>Lₜ</td>
<td>m</td>
</tr>
<tr>
<td>Mass of the goods conveyed over the entire length conveyed (total load)</td>
<td>mₘ₁</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of the goods conveyed on the top side (total load)</td>
<td>mₘ₂</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of the goods conveyed on the return side (total load)</td>
<td>mₘ₃</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of the belt</td>
<td>mₘ₄</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of the goods conveyed per m length conveyed on the upper face (line load)</td>
<td>mₘ₅</td>
<td>kg/m</td>
</tr>
<tr>
<td>Mass of all rotating drums, except for drive drum</td>
<td>mₘ₆</td>
<td>kg</td>
</tr>
<tr>
<td>Mass of the goods conveyed per m length conveyed on the return side (line load)</td>
<td>mₘ₇</td>
<td>kg/m</td>
</tr>
<tr>
<td>Mechanical motor power</td>
<td>Pₘ</td>
<td>kW</td>
</tr>
<tr>
<td>Mechanical power calculated on the drive drum</td>
<td>Pₘₚ₂</td>
<td>kW</td>
</tr>
<tr>
<td>Production tolerance</td>
<td>Tol</td>
<td>%</td>
</tr>
<tr>
<td>Friction coefficient when running over roller</td>
<td>µₘ</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>µₛₜₚ₁</td>
<td>–</td>
</tr>
<tr>
<td>Belt velocity</td>
<td>v</td>
<td>m/s</td>
</tr>
<tr>
<td>Volume flow for bulk goods conveying</td>
<td>Vₘ</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>yₘ</td>
<td>mm</td>
</tr>
<tr>
<td>Drum deflection</td>
<td>yₚₘ</td>
<td>mm</td>
</tr>
<tr>
<td>Margin for take-up range</td>
<td>Zₘ</td>
<td>mm</td>
</tr>
<tr>
<td>Machine's angle of inclination</td>
<td>αₘ</td>
<td>°</td>
</tr>
<tr>
<td>Arc of contact on the drive drum (or snub roller)</td>
<td>βₘ</td>
<td>°</td>
</tr>
<tr>
<td>Opening angle on the tensioning drum</td>
<td>γₘ</td>
<td>°</td>
</tr>
<tr>
<td>Belt elongation (pre-tensioning with weight)</td>
<td>Δₘ</td>
<td>mm</td>
</tr>
<tr>
<td>Permitted angle of inclination for unit goods</td>
<td>ðₘ</td>
<td>°</td>
</tr>
<tr>
<td>Elongation at fitting</td>
<td>εₘ</td>
<td>%</td>
</tr>
<tr>
<td>Maximum belt elongation</td>
<td>εₘₘ₈</td>
<td>%</td>
</tr>
<tr>
<td>Drive efficiency</td>
<td>ηₘ</td>
<td>–</td>
</tr>
<tr>
<td>Bulk density of goods conveyed</td>
<td>ρₘ</td>
<td>kg/m³</td>
</tr>
</tbody>
</table>
Unit goods conveying systems

\[ m = l \cdot \text{Weight of conveyed goods per metre} \]

\[ F_U = \mu_l \cdot g \cdot (m + m_B + m_R) \quad [N] \]

\[ F_U = \mu_l \cdot g \cdot (m + \frac{m_B}{2}) + \mu_R \cdot g \left( \frac{m_R}{2} + m_B \right) \quad [N] \]

\[ F_U = \mu_l \cdot g \cdot (m_1 + m_2 + m_R) \quad [N] \]

Direction conveyed upwards:
\[ F_U = \mu_l \cdot g \cdot (m + m_B + m_R) + g \cdot m \cdot \sin \alpha \quad [N] \]

Direction conveyed downwards:
\[ F_U = \mu_l \cdot g \cdot (m + m_B + m_R) - g \cdot m \cdot \sin \alpha \quad [N] \]

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

\[ m = l \cdot T \cdot \text{Weight of conveyed goods per metre} \]

\[ F_U = \mu_t \cdot g \cdot (m + m_B + m_R + m_ST) \quad [N] \]
Friction coefficients $\mu_S$ for various coatings (guidelines)

<table>
<thead>
<tr>
<th></th>
<th>0, A0, E0, T, U0, P</th>
<th>NOVO</th>
<th>U1, V1, VH</th>
<th>UH, V2H, U2H, E0, A0, V5H, V10H</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu_S$ (table)</td>
<td>0.33</td>
<td>0.33</td>
<td>0.5</td>
<td>0.5</td>
</tr>
<tr>
<td>$\mu_S$ (roller)</td>
<td>0.033</td>
<td>0.033</td>
<td>0.033</td>
<td>0.033</td>
</tr>
<tr>
<td>$\mu_S$ (accumulated)</td>
<td>0.33</td>
<td>0.33</td>
<td>0.5</td>
<td>0.5</td>
</tr>
</tbody>
</table>

Maximum belt pull $F_1$

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

If effective pull $F_U$ can be calculated

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

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

Factor $C_1$

(applies to the drive drums)

| Siegling Transilon | V3, V5, U2, A5, E3 | V1, U1, UH, U2H | V2H, V5H | 0, U0, NOVO, E0, A0, T, P |
|--------------------|---------------------|----------------|----------------|-----------------
| Arc of contact $\beta$ | 180° | 210° | 240° | 180° | 210° | 240° | 180° | 210° | 240° |
| Smooth steel drum | 1.5 | 1.4 | 1.3 | 1.8 | 1.6 | 1.5 | 2.1 | 1.9 | 1.7 |
| wet             | 3.7 | 3.2 | 2.9 | 5.0 | 4.0 | 3.0 | not recommended |
| Lagged drum | 1.4 | 1.3 | 1.2 | 1.6 | 1.5 | 1.4 | 1.5 | 1.4 | 1.3 |
| wet             | 1.8 | 1.6 | 1.5 | 3.7 | 3.2 | 2.9 | 2.1 | 1.9 | 1.7 |

Factor $C_2$

Checking the Transilon type selected

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

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):

<table>
<thead>
<tr>
<th>Tension member Type</th>
<th>Polyester (key letter “E”)</th>
<th>Polyester (key letter “AE”)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>E 2/1, E 3/1, E 4/2, E 6/1, NOVO, E 8/2, E 10/2, E 15/2, E 15/4/M, E 18/3, E 20/3/M, E 30/3, E 44/3</td>
<td>AE 48/H, AE 80/3, AE 100/3, AE 140/H, AE 140/3</td>
</tr>
<tr>
<td>Examples of type classes $k_{max}$</td>
<td>2.0</td>
<td>0.8</td>
</tr>
</tbody>
</table>

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.
\[ d_A = \frac{F_U \cdot C_3 \cdot 180}{D_0 \cdot \beta} \quad [\text{mm}] \]

<table>
<thead>
<tr>
<th>Siegling Transilon Underside coating</th>
<th>V3, V5, U2, A5, E3</th>
<th>V1, U1, UH</th>
<th>O, U0, NOVO, T, P</th>
</tr>
</thead>
<tbody>
<tr>
<td>Smooth steel drum</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>dry</td>
<td>25</td>
<td>30</td>
<td>40</td>
</tr>
<tr>
<td>wet</td>
<td>50</td>
<td>Not recommended</td>
<td>Not recommended</td>
</tr>
<tr>
<td>Lagged drum</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>dry</td>
<td>25</td>
<td>25</td>
<td>30</td>
</tr>
<tr>
<td>wet</td>
<td>30</td>
<td>40</td>
<td>40</td>
</tr>
</tbody>
</table>

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

Factor \( C_3 \) (applies to the drive drums)

Minimum diameter of the drive drums \( d_A \)

Mechanical capacity calculated on the drive drum \( P_A \)

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

Mechanical capacity required \( P_M \)
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.

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 ε for head drives

The minimum elongation at fitting for head drives is:

\[ \varepsilon = \frac{F_U/2 + 2 \cdot F_2}{2 \cdot k_{196} \cdot b_0} \]  

[N]

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.
Tail drive in steady state forces

\[ F_2 = F_1 - F_U \]

Guidelines for elongation at fitting \( \varepsilon \) for tail drives

The minimum elongation at fitting for return side drives is:

\[ \varepsilon = \frac{F_U/2 + 2 \cdot F_2 + F_U}{2 \cdot k_{W1} \cdot b_0} \] [%]

Guidelines for elongation at fitting \( \varepsilon \) for return-side drives

The minimum elongation at fitting for operating head drives is:

\[ \varepsilon = \frac{F_U \cdot (C_1 - k)}{k_{W1} \cdot b_0} \] [%]

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

Guidelines for steady state shaft load

Typical drive drum \( \beta = 180^\circ \)

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

Typical end drum \( \beta = 180^\circ \)

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

Typical snub roller \( \beta = 60^\circ \)

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

Shaft load when tensioning belts

Tension members made of synthetic materials display significant relaxation behaviour. As a result, the relaxed \( k_{W1} \) 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.
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 $\Delta L$, the production tolerance $Tol$, the safety margin for tensioning $Z$ and the belt selected.

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

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

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

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

Establishing belt elongation $\Delta L$

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

$$\Delta L = \frac{F_U/4 + F_{TR} + F_R}{k_{1\%} \cdot b_0} \cdot L_g \quad [\text{mm}]$$
Bulk goods conveying systems

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 $\rho_S$

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

in $\text{m}^3/\text{h}$ 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>$b_0$ (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>

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 $\alpha$.

<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>
</tr>
</thead>
<tbody>
<tr>
<td>Factor $C_6$</td>
<td>1.0</td>
<td>0.99</td>
<td>0.97</td>
<td>0.95</td>
<td>0.93</td>
<td></td>
</tr>
</tbody>
</table>

Conveying angle $\alpha$ [°] | 14  | 16  | 18  | 20  | 22  |
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Factor $C_6$</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 $C_4$.

<table>
<thead>
<tr>
<th>$l_T$ (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>$C_4$</td>
<td>2.0</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 $m$

$$m = \frac{\dot{V} \cdot \delta_s \cdot l_T \cdot 3.6}{\dot{V}} \quad [\text{kg}]$$
If maximum sag of 1% is permitted, (i.e. \( y_B = 0.01 \cdot l_0 \))

Recommendation \( l_0 \text{ max} \leq 2b_0 \)

\( l_0 \approx 2 - 3 \cdot l_0 \text{ max} \)

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

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

Calculation as for unit goods

\[
F_U = g \cdot C_4 \cdot f \frac{m + m_B}{m_R} \pm g \cdot m \cdot \sin \alpha \quad \text{[N]}
\]

Establishing the effective pull \( F_U \)

(−) downwards
(+) upwards

Support roller pitches

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

\[
l_0 = \frac{8 \cdot F}{m_0 + m_B} \quad \text{[mm]}
\]

\( l_0 = \) 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
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 $k_{1b} = 8 \text{ N/mm}$.

Effective pull $F_U [\text{N}]$

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

$F_U = 0.33 \cdot 9.81 \left( \frac{1200}{2} + \frac{570}{2} \right) + 0.033 \cdot 9.81 \left( \frac{575}{2} \right)$

$F_U \approx 4340 \text{ N}$

Maximum belt pull $F_1 [\text{N}]$

$F_1 = 4350 \text{ N}$

$C_1 = 1.6$

$F_1 = F_U \cdot C_1$

$F_1 = 4350 \cdot 1.6$

$F_1 \approx 6960 \text{ N}$

Checking the belt type selected

$F_1 = 6960 \text{ N}$

$b_0 = 600 \text{ mm}$

$k_{1b} = 8 \text{ N/mm}$

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

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

$11.6 \text{ N/mm} \leq 16 \text{ N/mm}$

The belt type has been chosen correctly.
\[ F_U = 4340 \text{ N} \]
\[ C_1 = 25 \]
\[ \beta = 180^\circ \]
\[ b_0 = 600 \text{ mm} \]

\[ d_A = \frac{F_U \cdot C_1 \cdot 180^\circ}{b_0 \cdot \beta} \quad \text{[mm]} \]

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

\[ d_A = 181 \text{ mm} \]

\[ d_A \text{ dimensioned at 200 mm} \]

\[ F_U = 4350 \text{ N} \]
\[ v = 0.8 \text{ m/s} \]

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

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

\[ P_A \approx 3.5 \text{ kW} \]

\[ P_A = 3.5 \text{ kW} \]
\[ \eta = 0.8 \text{ (assumed)} \]

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

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

\[ P_M \approx 4.4 \text{ kW} \]

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

\[ F_U = 4350 \text{ N} \]
\[ C_1 = 1.6 \]
\[ K = 0.62 \]
\[ k_{1N} = 8 \text{ N/mm for E8/2 U0/VSH black} \]
\[ b_0 = 600 \text{ mm} \]

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

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

\[ \varepsilon = 0.9 \% \]

\[ \text{Minimum drive drum diameter} \]

\[ \text{Power } P_A \text{ on the drive drum} \]

\[ \text{Motor power required } P_M \]

\[ \text{Minimum elongation at fitting for return drive} \]
Shaft load in steady state drum 2 (return drum)

Simplified calculation assuming $\beta = 180^\circ$

$F_1 = 6960$ N

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

$F_{W2} = 2 \cdot F_1$
$F_{W2} = 2 \cdot 6960$ N
$F_{W2} \approx 13920$ N

Shaft load in steady state 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 (return 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 $\varepsilon$. The tensile force $F$ is calculated according to:

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

[ 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).

\[ F_W = 2 \cdot F \]
\[ F_W = 2 \cdot 0.9 \cdot 8 \cdot 600 \]
\[ F_W \approx 8640 \text{ N} \]

\[ F_W = \sqrt{F_{1W}^2 + F_{2W}^2 - 2 \cdot F_{1W} \cdot F_{2W} \cdot \cos \beta} \]

\[ F_W = [\text{N}] \]

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

\[ F_{W1} \text{ at rest} = 8640 \text{ N} \]
\[ F_{W1} \text{ steady state} = 5220 \text{ N} \]

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

**Shaft load at rest**

**Take-up range**

\[ \varepsilon = 0.9 \% \]
\[ L_g = 105000 \text{ mm} \]
\[ Z = 200 \text{ mm} \]

\[ \text{Tol} = \pm 0.2 \% \]

\[ X = 210 + 473 + 200 \]
\[ X \approx 883 \text{ mm} \]
Because our products are used in so many applications and because of the individual factors involved, our operating instructions, details and information on the suitability and use of the products are only general guidelines and do not absolve the ordering party from carrying out checks and tests themselves. When we provide technical support on the application, the ordering party bears the risk of the machinery functioning properly.

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