HabaSYNC®
Timing Belts

Engineering Guide
This disclaimer is made by and on behalf of Habasit and its affiliated companies, directors, employees, agents and contractors (hereinafter collectively "HABASIT") with respect to the products referred to herein (the "Products").

SAFETY WARNINGS SHOULD BE READ CAREFULLY AND ANY RECOMMENDED SAFETY PRECAUTIONS BE FOLLOWED STRICTLY! Please refer to the Safety Warnings herein, in the Habasit catalogue as well as installation and operating manuals.

All indications / information as to the application, use and performance of the Products are recommendations provided with due diligence and care, but no representations or warranties of any kind are made as to their completeness, accuracy or suitability for a particular purpose. The data provided herein are based on laboratory application with small-scale test equipment, running at standard conditions, and do not necessarily match product performance in industrial use. New knowledge and experience may lead to re-assessments and modifications within a short period of time and without prior notice.

EXCEPT AS EXPLICITLY WARRANTED BY HABASIT, WHICH WARRANTIES ARE EXCLUSIVE AND IN LIEU OF ALL OTHER WARRANTIES, EXPRESS OR IMPLIED, THE PRODUCTS ARE PROVIDED "AS IS". HABASIT DISCLAIMS ALL OTHER WARRANTIES, EITHER EXPRESS OR IMPLIED, INCLUDING, BUT NOT LIMITED TO, IMPLIED WARRANTIES OF MERCHANTABILITY, FITNESS FOR A PARTICULAR PURPOSE, NON-INFRINGEMENT, OR ARISING FROM A COURSE OF DEALING, USAGE, OR TRADE PRACTICE, ALL OF WHICH ARE HEREBY EXCLUDED TO THE EXTENT ALLOWED BY APPLICABLE LAW, BECAUSE CONDITIONS OF USE IN INDUSTRIAL APPLICATION ARE OUTSIDE OF HABASIT'S CONTROL, HABASIT DOES NOT ASSUME ANY LIABILITY CONCERNING THE SUITABILITY AND PROCESS ABILITY OF THE PRODUCTS, INCLUDING INDICATIONS ON PROCESS RESULTS AND OUTPUT.
## Contents

**Introduction** 4  
Features of HabaSYNC® timing belts 5  
Timing belt range 7  
Timing belt nomenclature 11  
Joining methods 14  
Mechanical clamping 17  
Belt surfaces 18  
Tracking guides 20  
Profiles 21  
Modifications 24  

**Design guide**  
Belt tension 26  
Tensioning devices 27  
Drive concept 28  
Evaluation of tooth and pitch 291  

**Calculation guide**  
Belt calculation procedure 32  
Peripheral force 33  
Belt selection/Pulley definition 37  
Center distance and belt length 38  
Teeth in mesh 39  
Belt tension 40  
Elongation and forces 41  
Belt width 42  
Shaft load 43  
Drive power 44  
Positioning error 45  

**Calculation examples**  
Conveying 47  
Linear-positioning drive 51  

**Appendix**  
Tolerances 56  
Material properties 58  
Diameters of cords 60  
Chemical resistance 61  
Evaluation of tooth and pitch 62  
List of abbreviations 64  
Conversion of units 66  
Glossary of terms 67  
Index 69  
The Habasit solution 71  
Contacts 72
Timing belts designed for conveying and linear movement are a logical extension of Habasit’s product portfolio. Produced at our state-of-the-art extrusion facility, HabaSYNC® timing belts are made of thermoplastic polyurethane and reinforced with steel and aramide tensile members. The belts are designed to provide a synchronous option in conveying and linear-movement applications where precision positioning is instrumental to performance.

Habasit offers complete base- and special-fabrication capabilities to meet the specific need of your application. In addition to making belts endless to your desired dimension, we can add covers, attachments and special modifications to customize HabaSYNC® to be the exact answer to your requirements.

This Engineering Guide provides design assistance to follow for selection determination with examples for the calculation of your most appropriate HabaSYNC® timing belt.

Habasit is your partner
With more than 60 years of belting experience in elastomers and textiles, Habasit and its more than 3,300 well-trained and committed employees are situated at strategic locations worldwide to be close to you. Our sales and production activities are supported by a dedicated research and development team, modern production and test facilities and a strong ambition to be the best.

At Habasit we also believe in strong partnerships. This is especially true with our HabaSYNC® product line where the possibility of design options in many cases is only limited by imagination. With a thorough understanding of your application and with our knowledge of material and manufacturing we can propose the right design.

Find more information about our product and joining data sheets, our calculation program “SeleCalc” on www.habasit.com.
Due to the high-strength cords, securely encapsulated in thermoplastic polyurethane (TPU), our HabaSYNC® timing belts provide precise indexing and accurate positioning in conveying and linear movements. Accurately formed teeth and belt body insure efficient mesh between belt and pulley. They offer positive synchronization that yields low noise and reduces vibration.

TPU is the best choice of elastomers due to its high strength, and chemical and wear resistance. TPU also allows the belt to be finished to any length with a thermal-welding process.

**HabaSYNC® timing belt benefits include**

- High-strength cords for longitudinal stability and low elongation
- Exact tooth shapes mean high positional accuracy, and no belt slipping
- Excellent abrasion-resistance
- Truly encapsulated cord

**In application these benefits yield**

- Quiet running performance
- Efficient operation
- Structural flexibility for streamlined design
- Oil and ozone resistance
- Low-installed tension meaning low-bearing loads

**Additional features include**

- Polyamide facings that deliver a low coefficient of friction and excellent abrasion resistance. This allows slider-bed applications or accumulation of goods without product damage.
- Well-developed joining technology that delivers excellent length of life and low bending fatigue.
Features of HabaSYNC® timing belts

**Belt matrix material**

Our timing belts are manufactured with thermoplastic polyurethane according to the table shown below.

The standard material polyester polyurethane, mostly in 92 Shore A hardness, provides very good wear resistance. As a result the belt teeth have less deflection, which means more efficient belt-to-pulley meshing, and efficient energy use.

<table>
<thead>
<tr>
<th>Code</th>
<th>Material</th>
<th>Hardness</th>
<th>Properties</th>
<th>Color</th>
<th>Temperature range</th>
<th>Cords used</th>
</tr>
</thead>
<tbody>
<tr>
<td>01</td>
<td>Polyester urethane</td>
<td>92 Shore A</td>
<td>High abrasion resistance</td>
<td>white</td>
<td>-20 to 80 °C (+4 to 176 °F)</td>
<td>S = Steel, A = Aramide, P = Performance</td>
</tr>
<tr>
<td>02</td>
<td>Polyester urethane</td>
<td>88 Shore A</td>
<td>FDA/EU approved for dry applications</td>
<td>transparent</td>
<td>-20 to 70 °C (+4 to 150 °F)</td>
<td>A = Aramide</td>
</tr>
<tr>
<td>03</td>
<td>Polyester urethane</td>
<td>88 Shore A</td>
<td>Good abrasion resistance</td>
<td>green</td>
<td>-20 to 70 °C (+4 to 150 °F)</td>
<td>A = Aramide</td>
</tr>
<tr>
<td>04</td>
<td>Carbonate urethane</td>
<td>92 Shore A</td>
<td>Good microbial resistance</td>
<td>white</td>
<td>-20 to 80 °C (+4 to 176 °F)</td>
<td>A = Aramide</td>
</tr>
<tr>
<td>05</td>
<td>Polyether urethane</td>
<td>90 Shore A</td>
<td>FDA/EU approved Good hydrolysis features</td>
<td>cobalt blue</td>
<td>-20 to 80 °C (+4 to 176 °F)</td>
<td>A = Aramide</td>
</tr>
<tr>
<td>06</td>
<td>Polyester urethane</td>
<td>92 Shore A</td>
<td>High abrasion resistance</td>
<td>black</td>
<td>-20 to 80 °C (+4 to 176 °F)</td>
<td>S = Steel</td>
</tr>
</tbody>
</table>

Where applications require exceptional hydrolysis resistance, we manufacture HabaSYNC® timing belts using a polyether polyurethane. Our standard polyether material is cobalt blue with a Shore A value of 90.

**HabaSYNC® cords**

Our timing belt tensile members are designed to provide high strength and flexibility which deliver the positional accuracy and excellent structural flexibility required in most linear timing belt applications.

Habasit offers standard steel (S) and aramide (A) cords in all pitches. High-performance (P) steel cords are available in AT5, AT10 and AT20 pitch. Contact Habasit for your pitch requests.

Performance cords deliver higher strength than the standard steel cords and are the suggested tensile member for linear-positioning applications such as vertical lifts and mechanical actuators.

HabaSYNC® cord lay incorporates both S and Z twists. Cords of each type are laid down in an alternating pattern to build a balanced construction. This assures that the belts track straight.
HabaSYNC® timing belts are highly effective in conveying- and linear-movement applications offering 98–99% performance efficiency. Homogeneously formed teeth run in matching pulleys under low-installed loads to provide the synchronization required to locate a product or position a component accurately.

Timing belt teeth are generally formed in either a trapezoid or curvilinear design. Both tooth designs will yield good results in general conveying applications. The trapezoidal-shaped (T) timing belts are the common choice for standard conveying tasks and in cases with “counter flexion” due to their backbending properties. The modified trapezoid AT series is used in bidirectional and critical product-positioning applications where zero backlash (in combination with respectively tolerated pulleys) is important. It provides high tooth strength in combination with a reduction of meshing impacts, which results in low noise emissions. With its big tooth area in contact with slider beds, AT belts are ideal for supported conveying applications. The curvilinear tooth shape HTD offers high strength in combination with reduced meshing impacts leading to lower noise development. In conjunction with special pulleys with reduced backlash, a high positioning accuracy can be accomplished.

**Imperial pitch belts (trapezoid design)**

Imperial pitch sizes include: XL, L, H and XH. Imperial pitch sizes are available with either steel or aramide cords.

Polyamide facings are available on either the tooth side, conveying side, or on both sides.

For details see the respective product data sheet on www.habasync.com.

Imperial pitch belts can only be used with the respective imperial pitch timing belt pulleys.
**Metric T belts (trapezoid design)**

Trapezoid metric T pitch sizes include: T5, T10 and T20. Metric T pitch sizes are available with either steel or aramide cords.

Polyamide facings are available on either the tooth side, conveying side, or on both sides.

For details see the respective product data sheet on www.habasync.com.

Metric pitch belts can only be run with standard metric pitch timing belt pulleys.
**Metric AT belts (modified trapezoid)**
Modified trapezoid metric pitches include: AT5, AT10 and AT20. Metric AT pitch sizes are available with steel, performance steel, or aramide cords.

Polyamide facings are available on either the tooth side, conveying side, or on both sides.

For details see the respective product data sheet on www.habasync.com.

AT metric pitch belts can only be run with AT metric pitch timing belt pulleys.
**Metric HTD belts (curvilinear)**

Curvilinear metric HTD pitches include: HTD5, HTD8 and HTD14. Metric HTD pitch sizes are available with steel cords.

Polyamide facings are available on either the tooth side, conveying side, or on both sides.

For details see the respective product data sheet on www.habasync.com.

Metric pitch HTD belts can only be run with metric pitch HTD timing belt pulleys.
HabaSYNC® timing belts are made out of several key component parts. Each must complement the other precisely in order to provide a highly effective synchronous-drive solution.

**Teeth**

The teeth on a timing belt are responsible for the intermeshing action that occurs when a timing belt and pulley are engaged. HabaSYNC® teeth are homogeneously formed through extrusion. They mesh with matching pulleys to yield accurate positioning of the belt, allowing the component or product being conveyed to be in the right place at the right time.

The teeth on HabaSYNC® standard belts are designed with a trapezoid form or curvilinear shape. Both, the trapezoid (T) and the modified trapezoid (AT) have straight-line dimensions. The rounded tooth pitch of the HTD curvilinear-shaped design allows deep engagement of belt into the pulley. This offers excellent linear and rotary positioning, as well as power transmission applications.

**Tooth angle (T and AT)**

The tooth angle identifies the necessary geometry for the belt. The matching pulley of the trapezoid shape belt must be designed to mesh with the belt to operate at optimum. A perfectly formed tooth angle will intermesh with matching pulleys and deliver high accuracy. This is a key factor that assures accurate positional placement in synchronous-conveying and linear-movement applications.

**Curvilinear shape (HTD)**

The round tooth shape of HTD types offers considerably increased torque capacity compared to trapezoid or modified trapezoid shape. Due to that shape the force distribution of the meshing teeth is more favorable and guarantees smooth running without backlash. The matching pulley of the curvilinear-shaped belt must correspond for optimum results.
Timing belt nomenclature

**Pulley**
Timing belt pulleys can be ordered either in metric or in imperial pitches and according to your tolerance requirements. They are plain, flanged, with or without hub, or with key way, set screw, or machined for mechanical mounting.

**Flight**
The flight is the noticeable line between the teeth. It is created as a result of a machined point on the forming wheel that is designed to locate cord placement. This critical position for cord resting ensures that the belt will mesh smoothly. It yields low drive noise and delivers vibration-free interaction with the pulley. The flight is a key part of the mold design. It is also an important factor for determining the pitch length of a belt.

**Tooth pitch**
The tooth pitch is the accurate measurement of the distance from the vertical centerline of one tooth to the vertical centerline of the next tooth. Metric pitch belts are measured in millimeters, imperial pitch belts are measured in inches.

**Pitch line**
The pitch line is the centerline of the cord measured around the entire belt length. The measurement of the cord around the entire belt is the result of the cord resting on each flight as the belt is made. The belt length is calculated from the pitch line (= neutral layer).
**Belt thickness**
The total height of a (single-sided) belt is the measurement from the tip of the tooth to the conveying surface of the belt. The tooth height is indicated on the drawing as well.

Imperial pitch belts are typically measured in inches; metric pitch belts are measured in millimeters.

**Belt width**
This is the actual measured dimension of the belt width.

Metric timing belts are measured in millimeters and noted to the actual width. For example, a “25” is used to specify 25 mm width and “100” is used to specify 100 mm width.

Imperial timing belts are measured in inches and are noted to 3 digits. For example “200” is a 2.00 inch belt width and “075” is a 0.75 (3/4) inch belt width.

**Slitting lane**
Habasit offers HabaSYNC® timing belts in all industry standard widths.

Special widths are available upon request.

**Center distance**
The center distance e of a two-pulley belt drive is measured from the center of one pulley to the center of the next.
Joining methods

HabaSYNC® open-end timing belt construction allows belts to be joined endless to any length. The joining process provides a multitude of belt length options when designing a new conveyor system. The belts can be made endless in the workshop and then installed or they can be installed and joined directly on the application.

HabaSYNC® timing belts are manufactured in open-end length. They need to be cut to the width required in the application. Slitting is done along the predesigned slitting lanes on the coil to create rolls of belt.

Joining process
Depending on the technical requirements belts can be:
- Joined endless by heat
- Hinge joint
- Clamped endless
- Joined with mechanical fasteners

Joining by heat
Synthetic belts with thermoplastic matrix material like HabaSYNC® can be cut to length, then joined together by a hot-welding process. The TPU in the joining area is melted and cooled again to create an endless belt.

It’s your choice whether to buy a fabricated or prefabricated belt. You can even buy a coil of belt material to make up yourself. To do this job effectively and efficiently, Habasit offers a range of tools which make on-site installation quick and reliable. Valid and always up-to-date joining data sheets can be found on www.habasit.com/timing-belts.htm.
Several steps are required to make an endless splice:

**Step 1: Preparing the belt ends – finger cutting**
In order to prepare the open-end belt to be joined endless, it is cut using HabaSYNC®’s finger geometry to create prepared ends for the joining process. Dedicated finger geometry can be obtained using HabaSYNC® cutting dies.

**Step 2: Interlocking the fingers into joining plates**
After fingers have been cut into both ends of the belt, the belt ends are interlocked into a HabaSYNC® fixed width joining plate.

**Step 3: Hot pressing of belt ends**
After the fingers are interlocked in the joining plate, the plate is placed in the PF-150C hot-pressing device.

When the pressing procedure is completed, the fingers are properly interlocked and the belt is joined endless. The joining area is hardly recognizable and balanced-finger integrity assures smooth action as the belt rotates around the pulley. The spliced belt provides approximately 50–60% of the open-end tensile force. See the respective product data sheet for details. This should be considered when designing your application.
Hinge Joint – the mechanical timing belt fastener

In many synchronous-conveying applications, timing belts must be frequently replaced. Habasit’s new mechanical Hinge Joint fastening system makes the job easy.

For the fast and easy exchange of installed timing belts, the patent-pending HabaSYNC® Hinge Joint has been developed. Assembly and disassembly with this simple and quick method cuts down standstill periods due to belt replacement.

The stainless steel pivot parts are fully embedded within the cut-to-length belt ends. These are just brought together and a metal pin interlocks the hinges. And the belt is ready to run. Therefore, neither machine disassembling nor cumbersome joining procedures are required to install spare timing belts.

When preparing the belt ends for this type of joining all teeth remain unaffected and no voids are visible. Just a small cut can be seen on the conveying side of the timing belt afterwards. Furthermore, frictional covers in a large variety of sizes and characteristics can be added. Also modifications or the application of profiles is possible as usual.

HabaSYNC® Hinge Joints can be fabricated with the following technical data:

<table>
<thead>
<tr>
<th>Specification</th>
<th>Data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Belt types applicable</td>
<td>T10, AT10, H and HTD8</td>
</tr>
<tr>
<td>Possible belt widths</td>
<td>16, 25, 32, 50, 75, 100 and 150 mm (other widths are available on request)</td>
</tr>
<tr>
<td>Minimum belt length</td>
<td>900 mm</td>
</tr>
<tr>
<td>Pulley diameter with counter flexion</td>
<td>120 mm</td>
</tr>
<tr>
<td>Pulley diameter without counter flexion</td>
<td>100 mm</td>
</tr>
<tr>
<td>Maximum tensile strength</td>
<td>750 N / 25 mm belt width</td>
</tr>
</tbody>
</table>
Clamping plates are an alternative joining mechanism

Belt clamps are used where the belt moves in a bidirectional fashion. In these cases the belt joint never rotates around the pulley. It simply moves backwards and forwards. Mechanical clamping plates are typically found in linear-movement applications. Such clamps are usually made out of aluminum. The graph and data tables show design insight with all the data needed for the manufacturing of the toothed clamping plates in the HabaSYNC® pitches offered.

---

**Clamping plates**

<table>
<thead>
<tr>
<th>Pitch (P)</th>
<th>E (in)</th>
<th>D (in)</th>
<th>B (in)</th>
<th>L (in)</th>
<th>T (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>XL</td>
<td>0.24</td>
<td>0.22</td>
<td>0.14</td>
<td>1.67</td>
<td>0.31</td>
</tr>
<tr>
<td>L</td>
<td>0.31</td>
<td>0.35</td>
<td>0.2</td>
<td>3.02</td>
<td>0.59</td>
</tr>
<tr>
<td>H</td>
<td>0.39</td>
<td>0.43</td>
<td>0.35</td>
<td>4.21</td>
<td>0.87</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pitch (P)</th>
<th>E (mm)</th>
<th>D (mm)</th>
<th>B (mm)</th>
<th>L (mm)</th>
<th>T (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>T5</td>
<td>6.0</td>
<td>5.5</td>
<td>3.2</td>
<td>41.4</td>
<td>8.0</td>
</tr>
<tr>
<td>T10</td>
<td>8.0</td>
<td>9.0</td>
<td>5.0</td>
<td>80.0</td>
<td>15.0</td>
</tr>
<tr>
<td>T20</td>
<td>10.0</td>
<td>11.0</td>
<td>10.0</td>
<td>160.0</td>
<td>20.0</td>
</tr>
<tr>
<td>AT5</td>
<td>6.0</td>
<td>5.5</td>
<td>3.2</td>
<td>41.4</td>
<td>8.0</td>
</tr>
<tr>
<td>AT10</td>
<td>8.0</td>
<td>9.0</td>
<td>5.0</td>
<td>80.0</td>
<td>15.0</td>
</tr>
<tr>
<td>AT20</td>
<td>10.0</td>
<td>11.0</td>
<td>10.0</td>
<td>160.0</td>
<td>20.0</td>
</tr>
<tr>
<td>HTD5</td>
<td>6.0</td>
<td>5.5</td>
<td>3.2</td>
<td>41.4</td>
<td>8.0</td>
</tr>
<tr>
<td>HTD8</td>
<td>8.0</td>
<td>9.0</td>
<td>5.0</td>
<td>66.0</td>
<td>15.0</td>
</tr>
<tr>
<td>HTD14</td>
<td>10.0</td>
<td>11.0</td>
<td>9.0</td>
<td>116.0</td>
<td>22.0</td>
</tr>
</tbody>
</table>

**Plate widths (PW)**

<table>
<thead>
<tr>
<th>Belt width in inches</th>
<th>0.375</th>
<th>0.500</th>
<th>0.750</th>
<th>1.000</th>
<th>1.500</th>
<th>2.000</th>
<th>3.000</th>
<th>4.000</th>
</tr>
</thead>
<tbody>
<tr>
<td>XL</td>
<td>1.12</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>L</td>
<td>–</td>
<td>1.54</td>
<td>1.77</td>
<td>2.03</td>
<td>2.52</td>
<td>3.03</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>H</td>
<td>–</td>
<td>1.77</td>
<td>2.00</td>
<td>2.26</td>
<td>2.75</td>
<td>3.26</td>
<td>4.25</td>
<td>5.27</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Belt width in mm</th>
<th>16</th>
<th>25</th>
<th>32</th>
<th>50</th>
<th>75</th>
<th>100</th>
</tr>
</thead>
<tbody>
<tr>
<td>T5</td>
<td>35</td>
<td>44</td>
<td>51</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>T10</td>
<td>41</td>
<td>50</td>
<td>57</td>
<td>75</td>
<td>100</td>
<td>125</td>
</tr>
<tr>
<td>T20</td>
<td>–</td>
<td>56</td>
<td>63</td>
<td>81</td>
<td>106</td>
<td>132</td>
</tr>
<tr>
<td>AT5</td>
<td>35</td>
<td>44</td>
<td>51</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>AT10</td>
<td>41</td>
<td>50</td>
<td>57</td>
<td>75</td>
<td>100</td>
<td>125</td>
</tr>
<tr>
<td>AT20</td>
<td>–</td>
<td>56</td>
<td>63</td>
<td>81</td>
<td>106</td>
<td>132</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Belt width in mm</th>
<th>10</th>
<th>15</th>
<th>20</th>
<th>25</th>
<th>30</th>
<th>50</th>
<th>55</th>
</tr>
</thead>
<tbody>
<tr>
<td>HTD5</td>
<td>–</td>
<td>34</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>HTD8</td>
<td>35</td>
<td>40</td>
<td>45</td>
<td>50</td>
<td>55</td>
<td>75</td>
<td>–</td>
</tr>
<tr>
<td>HTD14</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>56</td>
<td>–</td>
<td>–</td>
<td>86</td>
</tr>
</tbody>
</table>

---

Sketch of basic shape
Sometimes conveying- and bidirectional-movement applications necessitate the need for lower frictional or antistatic properties.

The addition of fabric facings on HabaSYNC® timing belts offers performance benefits for these applications, where special conditions within product movement and positioning is sought. Applying a cover of thermoplastic, rubber or foam layer to the conveying side of the timing belt expands tremendously the array of application possibilities.

**Fabric facings**

**Polyamide**

Polyamide fabric added to the tooth side (PT), the conveying side (PC) or both the tooth and conveying side (PTC) reduces frictional drag of the belt and provides the benefit of lower noise. Polyamide fabric on the tooth side of a timing belt reduces the energy consumption by means of the lower coefficient of friction. As the belt meshes with the pulley teeth, the smooth engagement extends wear resistance and lowers noise emission. On the conveying surface of the belt it is used for accumulating product conveyors to overcome friction buildup and to allow easier product slip. With a polyamide conveying surface on the belt, the product can slip in place while belt motion continues.

**Coefficient of friction on the tooth side**

<table>
<thead>
<tr>
<th>Material Comparison</th>
<th>Coefficient of Friction</th>
</tr>
</thead>
<tbody>
<tr>
<td>Urethane vs. pickled steel</td>
<td>0.5 to 0.7</td>
</tr>
<tr>
<td>Polyamide vs. pickled steel</td>
<td>0.2 to 0.4</td>
</tr>
<tr>
<td>Urethane vs. PE-UHMW</td>
<td>0.3 to 0.5</td>
</tr>
<tr>
<td>Polyamide vs. PE-UHMW</td>
<td>0.1 to 0.3</td>
</tr>
</tbody>
</table>

**Antistatic fabric**

The addition of an antistatic fabric helps dissipate static generation on the belt, protecting sensitive products conveyed.

Belts made of synthetic material continuously in firm contact with pulleys, rollers, slider beds, etc. and again separated at high speed, create an electrostatic charge, detectable as an electrical field. This means synthetic belts would theoretically provide ideal conditions for the generation of electrostatic charge.

The generation of electrostatic buildup, however, is not desired because it could have an adverse effect on the product conveyed. Therefore, it is prevented by specific measures on the belt itself by selection of suitable materials and a special belt design. The antistatic property is provided by a conductive element as part of the belt: the antistatic fabric.

HabaSYNC® antistatic belts are tested according to ISO 21179. This standard measures three separate values: the potential of a running belt, the surface resistance of a stationary belt and the volume resistance of a stationary belt. The voltage and the surface potential are measured on the tooth side and the conveying side. For each value a specific target is given, and the measured value must be below this target.

---

**Urethane vs. Pickled Steel**

- Urethane vs. Pickled Steel: 0.5 to 0.7
- Polyamide vs. Pickled Steel: 0.2 to 0.4

**Polyamide vs. PE-UHMW**

- Urethane vs. PE-UHMW: 0.3 to 0.5
- Polyamide vs. PE-UHMW: 0.1 to 0.3
Belt surfaces

Covering layers
Additionally applied cover materials provide surface features needed for specific applications.

Mostly, these are combinations of:
- Abrasion resistance
- Chemical resistance
- Compressibility
- Electrostatic discharge – antistatic features
- Excellent release properties
- Gentle movement
- Heat resistance
- High or low coefficient of friction
- Shock absorption

Commonly used material options include:
- TPU
- PU
- Natural rubber
- NBR
- EPDM
- Nitrile
- PVC
- EVA foam
- PU foam
- Rubber foam
- Artificial leather
- Aramide
- Polyester fleece

Each material with its particular property characteristics offers very specific inherent features. Also varied structure options provide a broad range of possibilities.

See brochure HabaSYNC® timing belt covers (4282) or the HabaSYNC® website for specific details, or ask your local representative for further information.
Tracking guides

Tracking guides are added to the tooth side of the HabaSYNC® timing belt. They are used on long-center-distance conveyors where true belt tracking is critical and where pulley flanges would interfere with the product being conveyed. They are also used where cross loading or unloading of the product conveyed could cause a lateral load that forces the belt to one side of the conveyor.

Tracking guides can also be used on linear-positioning and conveyor applications where the belt is run in a vertical position rather than lying flat on a conveyor surface.

HabaSYNC® tracking guides are available in G6, G10, and G13 sizes. Our standard TPU hardness is 92 Shore A in white.

Tracking guides are typically notched to allow maximum flexibility of the belt when running around pulleys.

HabaSYNC® tracking guides must run in timing belt pulleys designed with a matching groove to fit the tracking-guide dimension.

T10 with tracking guide on tooth side.
Profiles are placed on the conveying side of HabaSYNC® timing belts. Profiles provide a simple solution for conveying products that require indexing and separation.

Habasit profiles are available in several Shore A hardness values. The thermoplastic polyurethane bonds securely to the conveying side of the HabaSYNC® timing belt using processes that include thermal bonding, vibration and high-frequency technology.

Profiles can be easily added to the TPU timing belts with both manual and automated equipment. The choice of equipment is typically related to the quantity and complexity of the profile design.

HabaSYNC® profiles can be produced in three ways. Manufacturing processes include:
- Machining
- Extrusion
- Injection molding

Machining
Machined profiles are produced with CNC equipment designed to machine plastic. We hold material in square or rectangular shapes in 85 and 92 Shore A hardness in stock, which can be machined quickly to provide any HabaSYNC® standard design.

Typically, machined profiles are chosen when small to medium production quantities are required, for example for prototypes where several variations in design must be evaluated before molds or dies can be justified.

Extrusion
Where larger quantities of profiles are needed, extrusion can be an economical option. Habasit’s extruded thermoplastic conveyor belt profiles may also be considered as an option if a softer material hardness is required. These profiles are in the range of 85 Shore A hardness.

Molding
Profiles are injection-molded if the profile design must be exact or is complex. HabaSYNC® injection-molded profiles can be produced in the same material as the base belt, and in many cases, up to maximum width to match the widest standard belt produced by Habasit.

Custom made-to-order profiles
In many applications a standard profile design will not suffice. Habasit can design custom profiles to meet the exact needs of your design. Please consult your local Habasit representative to discuss the details.

The drawing shows a simplified example of a custom profile designed for a battery-conveying application. In this case, the batteries are securely held between the profile openings.
Guidelines for profile design

- **Profile spacing:** We suggest that the spacing of profiles should be a multiple of the belt pitch being used. This provides for a whole number of profiles on the belt, and easily considers tolerances from one profile to the next.
- **Dimension of the profile base:** Ideally the base of the profile should be as thin as possible to ensure maximum flexibility. The profile should be welded directly over the tooth of the belt to assure maximum flexibility.

As the thickness of the profile base increases, so does the need for larger pulleys:

**Minimum number of pulley teeth for profiles over a tooth**

<table>
<thead>
<tr>
<th>Profile base thickness</th>
<th>1/16</th>
<th>1/8</th>
<th>3/16</th>
<th>1/4</th>
<th>5/16</th>
<th>3/8</th>
<th>7/16</th>
<th>1/2</th>
<th>5/8</th>
<th>3/4</th>
</tr>
</thead>
<tbody>
<tr>
<td>mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>XL</td>
<td>10</td>
<td>10</td>
<td>18</td>
<td>25</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td>100</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>L</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>28</td>
<td>30</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td>100</td>
<td>–</td>
</tr>
<tr>
<td>H</td>
<td>14</td>
<td>14</td>
<td>14</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>20</td>
<td>35</td>
<td>50</td>
</tr>
<tr>
<td>XH</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>20</td>
<td>35</td>
<td>50</td>
</tr>
<tr>
<td>T5</td>
<td>12</td>
<td>12</td>
<td>12</td>
<td>25</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td>100</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>AT5, HTD5</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>18</td>
<td>25</td>
<td>40</td>
<td>50</td>
<td>60</td>
<td>100</td>
<td>–</td>
</tr>
<tr>
<td>T10, AT10, HTD8</td>
<td>16</td>
<td>16</td>
<td>16</td>
<td>16</td>
<td>18</td>
<td>25</td>
<td>35</td>
<td>45</td>
<td>80</td>
<td>100</td>
</tr>
<tr>
<td>T20, AT20, HTD14</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>18</td>
<td>20</td>
<td>35</td>
<td>50</td>
</tr>
</tbody>
</table>

**Minimum number of pulley teeth for profiles NOT over a tooth**

<table>
<thead>
<tr>
<th>Profile base thickness</th>
<th>1/16</th>
<th>1/8</th>
<th>3/16</th>
<th>1/4</th>
<th>5/16</th>
<th>3/8</th>
<th>7/16</th>
<th>1/2</th>
<th>5/8</th>
<th>3/4</th>
</tr>
</thead>
<tbody>
<tr>
<td>mm</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>XL</td>
<td>12</td>
<td>30</td>
<td>45</td>
<td>50</td>
<td>60</td>
<td>100</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>L</td>
<td>12</td>
<td>20</td>
<td>40</td>
<td>45</td>
<td>55</td>
<td>60</td>
<td>70</td>
<td>80</td>
<td>100</td>
<td>–</td>
</tr>
<tr>
<td>H</td>
<td>14</td>
<td>14</td>
<td>25</td>
<td>30</td>
<td>45</td>
<td>50</td>
<td>55</td>
<td>65</td>
<td>80</td>
<td>100</td>
</tr>
<tr>
<td>XH</td>
<td>18</td>
<td>18</td>
<td>20</td>
<td>30</td>
<td>40</td>
<td>45</td>
<td>50</td>
<td>54</td>
<td>58</td>
<td>60</td>
</tr>
<tr>
<td>T5</td>
<td>12</td>
<td>30</td>
<td>45</td>
<td>50</td>
<td>60</td>
<td>100</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>AT5, HTD5</td>
<td>15</td>
<td>30</td>
<td>45</td>
<td>50</td>
<td>60</td>
<td>100</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>T10, AT10, HTD8</td>
<td>18</td>
<td>20</td>
<td>30</td>
<td>40</td>
<td>45</td>
<td>50</td>
<td>55</td>
<td>65</td>
<td>80</td>
<td>100</td>
</tr>
<tr>
<td>T20, AT20, HTD14</td>
<td>18</td>
<td>18</td>
<td>20</td>
<td>30</td>
<td>40</td>
<td>45</td>
<td>50</td>
<td>54</td>
<td>58</td>
<td>60</td>
</tr>
</tbody>
</table>

Sample sketches to show the differences between profile welding over a tooth or between the teeth.
Profile strength
The strength of the profile weld is a direct factor of the dimension of the base weld. When reviewing profile strength, it is vital to consider (besides the size of the welded profile foot) the location and direction of force on the profile.

Tolerances
- Thickness, height, length: ± 0.5 mm (± 0.02")
- Profile distance
  1. Profile located over tooth (profile distance is multiple of belt pitch): ± 0.5 mm (± 0.02")
  2. Profile located between teeth (profile distance is not a multiple of belt pitch): ± 0.8 mm (± 0.03")

Wide-base profiles
In many cases, the profile will be welded to a belt leaving one side of the base to float. In other words, part of the profile is not welded to the belt surface. This provides maximum flexibility over the pulley.

Profiles prepared for mounting attachments
In some cases a profile will be designed to allow an attachment to be fixed onto the profile. Typically these attachments are difficult or costly to mold and can be obtained in other materials with ease. To enable an attachment to be fixed HabaSYNC® profiles will be either drilled or molded including the hole pattern required.

Welding bead
In some cases a bead of molten urethane can develop between the belt and profile bond. This bead can be removed if it affects the performance of the product to be conveyed or due to appearance reasons.

Please contact your local Habasit representative to discuss your application and required tolerances.

False teeth
This mechanical mounting system for ATM10 and ATM20 timing belts allows for quick and easy exchange of individually designed cleats. The belt can still run smoothly around pulleys because the threaded stainless steel inserts are embedded within the milled teeth.
Modifications

In many applications, particularly those in general conveying, modifications may be made to enhance product movement performance of a timing belt.

Modifications are changes or finishes made to the base belt and possibly to the attachments placed on the belt to provide special features to either the conveying-side or tooth-side surface.

**Modifications include:**
- Surface grinding
- Profile grinding/routing
- Lateral and longitudinal machining
- Slotting and hole punching

**Surface grinding** is an option to improve thickness tolerance needs or to increase quality of the surface finish.

**Profile grinding or routing** includes the addition of slots and pockets to the belt surface. These are used for, e.g. proper positioning of products conveyed, precise placement of profiles for actuation or indexing tasks.

**Lateral machining**
On thick applied covers, HabaSYNC® timing belts can be machined across the belt conveying surface to create slots for holding the product in place on incline or vertical conveyors.
Longitudinal machining
A trough running the whole length on the tooth or conveying side of the belt can be used. A machined groove down the length of the belt can also be used to locate or enhance application performance.

Perforations, also known as hole punching, are used in many vacuum-conveying applications. Additionally, holes may be punched into HabaSYNC® timing belts to allow profiles to be mechanically attached to their surface.

In all cases, modification requests should be accompanied with a drawing clearly specifying dimensions and tolerances.

 Modifications are typically designed for:
- Vacuum or hold-down conveyors
- Product capture points
- Sizing and separation of material conveyed
- Attachment ports for metal clamps or profiles
- Applications where precision thickness tolerances are required

Modifications are largely dependent on application circumstances. Please contact your Habasit representative to discuss your specific needs.
Transmitting the peripheral force \( F_U \) from the periphery of the driving pulley to the timing belt requires a certain belt tension. The tensile force needed is determined by a calculation.

If the belt wraps the drive pulley with an angle of about \( 180^\circ \), the required shaft load \( F_W \) on the drive pulley should be about 1.2 times the peripheral force \( F_U \).

\[
F_W = 1.2 \cdot F_U \quad [N]
\]

- \( F_W \) = Shaft load \( (F_W = F_1 + F_2) \)
- \( F_1 \) = Tensile force in the tight side of the belt
- \( F_2 \) = Tensile force in the slack side of the belt

For an arc of contact \( \beta \neq 180^\circ \), the respective shaft load can be determined by the following approximation method:

\[
F_W = 1.2 \cdot F_U \cdot \sin \left( \frac{\beta}{2} \right) \quad [N]
\]

For nondriven pulleys (tension pulley, idlers, etc.) the forces \( F_1 \) and \( F_2 \) are the same.
Drives with controlled belt tension
Since HabaSYNC® timing belts have a very high stress-strain ratio, it is highly recommended (at least for belt lengths below 6 m/20 ft) to use a tensioning device to provide controlled belt tension. Typically, a constant shaft-load or slack-side tension is incorporated by using pneumatic cylinders, spring-loaded or gravity tensioners, etc. Such tensioning devices provide the advantage of reduced maintenance and minimized maximum belt tension. Both have a positive influence on the overall life of the belt.

\[
F_2 = \frac{F_{WT}}{2 \cdot \sin \left( \frac{\beta}{2} \right)} \quad [N]
\]

Drives with a fixed center-to-center distance
Fixed tensioning devices are used in applications where there is no need to compensate for variations in belt length or belt extension during operation.

The simplest solution for tensioning is to use the tail roller to tension and lock down.

When the center distance between the head and tail rollers may not be changed, e.g. with intermediate or transition conveyors, the tension station is incorporated in the return side.
Position of drive
In order to calculate the initial belt extension, the position of the drive is extremely important.

Head drive
This illustration indicates how the tensile force in the belt continuously increases due to the conveying of a mass. Since in this example the drive is placed at the head of the conveyor (on the left side of the illustration), the belt length with the higher tensile force level ($F_1$) is much shorter than the belt section with lower tensile force ($F_2$). Therefore, a lower initial belt extension is required. This configuration is recommended if the belt is running in one direction.

Center drive
This illustration shows that the belt section with high tensile force ($F_1$) has more or less the same length as the section with low force ($F_2$). This symmetrical situation is an advantage in bidirectional applications. Therefore, this configuration is recommended if the belt's running direction changes.

Tail drive
In contrast to the head drive, the tail-driven belt is exposed to a high tensile force $F_1$ in the return side. As a result, the belt length with the lower tensile force level ($F_2$) is much shorter than the length of the belt section with high tensile force ($F_1$). Therefore, higher initial belt extension is required. For this reason, this configuration should be avoided whenever possible.
Design guide
Evaluation of tooth and pitch

Belt evaluation
The evaluation of the optimal timing belt for a specific application is primarily a question of requirements. Initial questions include:
- Minimum pulley diameters
- Coefficient of friction of surfaces
- Properties of materials (suitable for food applications, chemical resistance, surface suitable for applying attachments, etc.)

Secondly, the chosen belt type must be dimensioned in terms of required forces and possible belt width. For the evaluation of pitch and belt width, the peripheral force on the drive pulley and the maximum load on the teeth must be considered (see Calculation Guide chapter). In some cases, not every detail of the drive can be considered. In very rare cases, it is possible that the final calculation will indicate that the belt selected according to these guidelines does not meet the requirements. In such cases, a second belt evaluation and calculation is necessary.

Evaluation of belt family
The first step is to choose whether a trapezoid or a modified trapezoid (AT series) is preferable.

Trapezoid tooth shape (T series)
Advantages:
- Optimal for standard drive tasks
- Greater flexibility in drives with counterflections

Belt series with trapezoid tooth shape
- T5 (5 mm pitch)
- T10 (10 mm pitch)
- T20 (20 mm pitch)
- XL (1/5" pitch / 5.08 mm)
- L (3/8" pitch / 9.525 mm)
- H (1/2" pitch / 12.7 mm)
- XH (7/8" pitch / 22.225 mm)

Modified trapezoid tooth shape (AT series)
Advantages:
- Higher tooth strength
- Stronger tension members
- Superior backlash control
- Reduction of meshing impacts (lower noise and vibration)
- Larger tooth area in contact with slider bed

Belt series with modified trapezoid tooth shape
- AT5 (5 mm pitch)
- AT10 (10 mm pitch)
- AT20 (20 mm pitch)

Curvilinear tooth shape (HTD series)
Advantages:
- Deeper tooth shape = higher torque
- Ideal for power transmission
- Low noise and vibration level
- Smooth rolling action in and out of pulleys

Belt series with curvilinear tooth shape
- HTD5 (5 mm pitch)
- HTD8 (8 mm pitch)
- HTD14 (14 mm pitch)
Design guide
Evaluation of tooth and pitch

Belt options
In addition to specific requirements like those for food applications and chemical resistance, another factor in belt selection is the coefficient of friction required on the belt surfaces (tooth side and conveying side).

The belt surface of the unprocessed standard types is wear-resistant polyurethane with hardnesses of 88, 90, or 92 Shore A values.

This material provides a coefficient of friction that is high enough to provide a good grip, without being too high. It performs well when running over slider beds or in applications with the accumulation of lightweight goods.

If a higher coefficient of friction (grip) is required (e.g. for steep transportation, etc.) we recommend the use of belts with special covers and surface structures, such as profiles or modifications on the conveying side. In order to select the optimal belt surface we recommend that you seek the support of your local Habasit representative.

If a low coefficient of friction is required (e.g. if a belt with a high load runs over a slider bed, or if there is a relative movement between the belt and heavy goods), we recommend using a belt with polyamide facing. Polyamide fabric is available on the tooth side (PT), conveying side (PC), or on both sides (PTC).

Further advantages of polyamide facing are:
- Improved wear resistance
- Reduced peripheral force when running over a slider bed or when goods are accumulated. Therefore, less drive power and less belt width are required
- Low noise properties

Evaluation of belt pitch
For the evaluation of pitch and belt width the peripheral force on the drive pulley and the maximum load on the teeth need to be considered.

How to determine the peripheral force
The peripheral force $F_U$ at the drive pulley is the sum of all individual forces resisting the belt motion. The individual loads contributing to the peripheral force $F_U$ must be identified and calculated based on the loading conditions and drive configuration. However, some loads cannot be calculated until the layout has been decided. To determine the peripheral force $F_U$, use the following methods for either conveying or linear positioning:

- The friction force $F_{US}$:
  \[
  F_{US} = g \cdot m \cdot \mu_G \quad \text{[N]}
  \]
  \[
  g = \text{Acceleration of gravity} = 9.81 \text{ m/s}^2
  \]
  \[
  m = \text{Total mass to be carried over the slider bed [kg]}
  \]
  \[
  \mu_G = \text{Coefficient of friction between the belt and slider bed}
  \]

  For linear-positioning applications the friction force $F_I$ of the slide needs to be considered. If this force is not defined by the supplier of the linear bearings, it must be determined experimentally (e.g. by means of a spring scale).

- Force required to elevate the carried goods $F_{UI}$ (not required for horizontal conveyors):
  \[
  F_{UI} = g \cdot m \cdot \frac{h_T}{l_T} \quad \text{[N]}
  \]
  \[
  g = \text{Acceleration of gravity} = 9.81 \text{ m/s}^2
  \]
  \[
  h_T = \text{Elevating height [mm]}
  \]
  \[
  l_T = \text{Conveying length [mm]}
  \]

- In applications where a mass is accelerated (actuator, stop-and-go operation), there is force $F_{Us}$ required for the acceleration of the carried goods:
  \[
  F_{Us} = m \cdot a \quad \text{[N]}
  \]
  \[
  m = \text{Mass of carried goods on total conveying length (total load) [kg]}
  \]
  \[
  a = \text{Acceleration [m/s}^2\text{]}\]

  \[
  a = \frac{\nu}{t} \quad \text{[m/s}^2\text{]}
  \]

  \[
  \nu = \text{Belt speed [m/s]}
  \]
  \[
  t = \text{Time required to run the conveyor up to speed [s]}
  \]

Therefore, the peripheral force $F_U$ at the drive pulley is primarily the sum of the following forces resisting the belt motion:

\[
F_U = F_{US} + F_{UI} + F_{Us} \quad \text{[N]}
\]
In applications with less than 5 teeth in mesh on the drive pulley (less than 11 teeth in mesh for open-ended belts), the $F_U$ value has to be corrected with the tooth-in-mesh factor $t_m$.

### Joined endless belts

<table>
<thead>
<tr>
<th>No. of teeth in mesh $z_m$</th>
<th>Tooth-in-mesh factor $t_m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.20</td>
</tr>
<tr>
<td>2</td>
<td>0.40</td>
</tr>
<tr>
<td>3</td>
<td>0.55</td>
</tr>
<tr>
<td>4</td>
<td>0.70</td>
</tr>
<tr>
<td>5</td>
<td>0.85</td>
</tr>
<tr>
<td>&gt; 5</td>
<td>1.00</td>
</tr>
</tbody>
</table>

Since a high rotational frequency of the belt may lead to high stress on the belt teeth (due to buildup of heat on the drive pulley), the speed factor $c_V$ has to be considered if the belt rotates more than once per second. In order to find this speed factor, the rotational frequency $f_R$ of the belt has to be defined:

$$f_R = \frac{v \cdot 1000}{l_0} \quad [1/s]$$

$v = $ Belt speed [m/s]

$l_0 = $ Belt length [mm]

Therefore the corrected peripheral force $F_{U \text{ corrected}}$ with the tooth-in-mesh factor $t_m$ and the speed factor $c_V$ is:

$$F_{U \text{ corrected}} = \frac{F_U}{t_m \cdot c_V} \quad [N]$$

$F_U = $ Peripheral force [N]

$t_m = $ Tooth-in-mesh factor

$c_V = $ Speed factor

### Open-ended belts (without joint)

<table>
<thead>
<tr>
<th>No. of teeth in mesh $z_m$</th>
<th>Tooth-in-mesh factor $t_m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td>2</td>
<td>0.30</td>
</tr>
<tr>
<td>3</td>
<td>0.40</td>
</tr>
<tr>
<td>4</td>
<td>0.50</td>
</tr>
<tr>
<td>5</td>
<td>0.60</td>
</tr>
<tr>
<td>6</td>
<td>0.70</td>
</tr>
<tr>
<td>7</td>
<td>0.80</td>
</tr>
<tr>
<td>8</td>
<td>0.85</td>
</tr>
<tr>
<td>9</td>
<td>0.90</td>
</tr>
<tr>
<td>10</td>
<td>0.95</td>
</tr>
<tr>
<td>11</td>
<td>0.97</td>
</tr>
<tr>
<td>&gt; 11</td>
<td>1.00</td>
</tr>
</tbody>
</table>
Calculation guide
Belt calculation procedure

A timing belt used in conveying applications typically operates well below its rated nominal tensile strength. For many applications the belt is selected according to the dimensional requirements of the drive system (pulley diameter, size of conveying load, required belt features, etc.) without considering a belt calculation. In such cases where the transmission of power is of minor importance we recommend using the smallest belt pitch possible. For these applications we recommend operating with an initial belt elongation of about 0.1% (= 1‰).

For applications where belts need to be selected according to their load capacity, we highly recommend a belt calculation like that described below or using Habasit’s SYNC-“SeleCalc”. For details see www.habasit.com/en/belt-calculation.htm

Belt calculation procedure

Peripheral force has to be evaluated
Whether for a conveying or linear-positioning application, the first step is to determine the peripheral force \( F_U \) at the drive pulley (this is the sum of all individual forces resisting the belt motion). All individual loads contributing to the peripheral force \( F_U \) must be identified and calculated based on the loading conditions and drive configuration. In some cases, however, certain loads cannot be calculated until the layout has been determined.

Evaluation of belt and pitch
In order to determine the belt pitch and width the peripheral force on the drive pulley and the maximum load on the teeth have to be considered.

Please see the Design Guide chapter to learn how to determine peripheral force and how to evaluate the belt type.

Calculation of installation parameters
Required belt width, required belt tension, shaft loads, and safety (utilized tensile force) are the common results of calculations for conveying, indexing conveyors and linear-drive applications.

For linear drive applications the accuracy of positioning (possibly for different masses or positions) has to be ascertained.

Belt selection and calculation for timing belt applications requires the following steps:
1. Determination of peripheral force
   a) For conveying or indexing conveyors
   b) For linear-positioning applications
2. Selection of belt, belt width and pitch
3. Definition of pulley diameters / number of pulley teeth
4. Definition of center distance and belt length
5. Calculation of the number of teeth in mesh on the drive pulley
6. Determination of minimal tensile force in the slack belt strand
7. Calculation of elongations and forces in the tight and slack side
8. Calculation of required belt width
9. Calculation of shaft loads
10. Calculation of the drive power and required motor power

For the calculation of linear drives an additional calculation is often required:
11. Calculation of the positioning error
Step 1a
For conveying or indexing conveyors

The peripheral force $F_U$ at the drive pulley is the sum of all individual forces resisting the belt motion. The individual loads contributing to the peripheral force $F_U$ must be identified and calculated based on the loading conditions and drive configuration. However, some loads cannot be calculated until the layout has been decided. Therefore, in some cases a correction of belt width or pitch is needed, and revision of the calculation will be required.

$F_U$ for a conveying application is primarily the sum of the following addends resisting the belt motion:

- Resistance due to friction between the belt and the slider bed ($F_{US}$)
- Force to elevate carried goods ($F_{Ui}$)
- Acceleration forces ($F_{Ua}$)
- Other contributing friction forces ($F_{Uau}$)

The peripheral force $F_U$ at the drive pulley is therefore the sum of these forces:

$$F_U = F_{US} + F_{Ui} + F_{Ua} + F_{Uau} \ [N]$$

Friction force $F_{US}$ (1st addend)
The friction force $F_{US}$ is the resistance due to friction between the belt and the slider bed.

$$F_{US} = g \cdot m_{tot} \cdot \mu_G \ [N]$$

- $g$ = Acceleration of gravity = 9.81 m/s$^2$
- $m_{tot}$ = Total mass to be moved across the slider bed [kg]
- $\mu_G$ = Coefficient of friction between the belt and the slider bed [-]

The total mass to be carried over the slider bed ($m_{tot}$) consists of the mass of the carried goods ($m = m_1 + m_2 + ... + m_n$) and the mass of the belt moving across the slider bed ($m_B$). The conveying length is identical with the shaft center distance.
**Force required to elevate the carried goods**

\[ F_{Ui} \text{ (2nd addend)} \]

\( F_{Ui} \) is the force required to elevate the mass \( m \) of the carried goods (not required in horizontal drives).

**Other contributing factors to the friction force**

\[ F_{Uau} \text{ (4th addend)} \]

Other contributing factors to the friction force \( F_{Uau} \) are:

- Resistance due to bearing friction of the rollers or idlers
- Resistance due to friction between the belt and the conveyed goods due to accumulation or diversion
- Resistance due to friction from auxiliary elements such as tracking devices (profiles), belt-cleaning devices, etc.

In most cases these resistances are negligible or not relevant for timing belt conveyors. However, in rare cases they become relevant and have to be considered.

**The peripheral force \( F_U \) at the drive pulley is therefore the sum of the above forces:**

\[ F_U = F_{US} + F_{Ui} + F_{Usa} + F_{Uau} \] [N]

**Formula for inclined transportation**

\[ F_{Ui} = g \cdot m \cdot \frac{h_T}{l_T} \] [N]

- \( h_T \) = Elevating height [mm]
- \( l_T \) = Conveying length [mm]

For declining conveyor applications the elevating height \( h_T \) becomes negative and therefore the force component \( F_{Ui} \) will be negative.

**Force required for the acceleration of the total mass**

\[ F_{Usa} \text{ (3rd addend)} \]

Force \( F_{Usa} \) required for the acceleration of the total mass:

\[ F_{Usa} = \left( \frac{m + m' \cdot l_0}{1000} \right) \cdot a \] [N]

- \( m \) = Mass of carried goods on total conveying length (total load) [kg]
- \( m' \) = Mass of belt per meter [kg/m]
- \( l_0 \) = Belt length [mm]
- \( a \) = Acceleration [m/s\(^2\)]

The average acceleration is equal to the belt velocity per unit of time required to accelerate up to speed.

\[ a = \frac{v}{t} \] [m/s\(^2\)]

- \( v \) = Belt speed [m/s]
- \( t \) = Time required to accelerate up to speed [s]
Step 1b
For linear-positioning applications

The peripheral force $F_U$ at the drive pulley is the sum of all individual forces resisting the belt motion. The individual loads contributing to the peripheral force $F_U$ must be identified and calculated based on the loading conditions and drive configuration. However, some loads cannot be calculated until the layout has been decided. Therefore, in some cases a correction of belt width or pitch is needed, and a revision of the calculation will be required.

$F_U$ for a linear-positioning application is primarily the sum of the following addends resisting the belt motion:
- Force required for the acceleration of a loaded slide ($F_{Ua}$)
- Friction force of the slider against the linear rail ($F_f$)
- Externally applied working load ($F_E$)
- Force required to elevate the mass $m_S$ of the slide and the load ($F_{Ui}$)

The peripheral force $F_U$ at the drive pulley is therefore the sum of these forces:

$$F_U = F_{Ua} + F_f + F_E + F_{Ui} \ [\text{N}]$$

**Force required for the acceleration of a loaded slider $F_{Ua}$ (1st addend)**

Force $F_{Ua}$ required for the acceleration of a loaded slide with mass $m_S$.

$$F_{Ua} = m_S \cdot a \ [\text{N}]$$

$m_S$ = Mass of the slider plus maximum load [kg]
$a$ = Acceleration [m/s$^2$]

The average acceleration is equal to the change in velocity per unit time.

$$a = \frac{\Delta v}{t} \ [\text{m/s}^2]$$

$\Delta v$ = Speed difference (final speed minus initial speed) [m/s]
$t$ = Time required to accelerate up to speed [s]
**Friction force \( F_r \) (2nd addend)**
The friction force \( F_r \) of the slider against the linear rail may be provided by the supplier of the linear bearing. If it is not, it needs to be determined experimentally. Friction force from bearing losses of rollers or idlers must be considered as part of the investigation.

**Externally applied working load \( F_E \) (3rd addend)**
If existing, externally applied working load \( F_E \) cannot be ignored. It is possible, for example, that an actuator pulls a mass over a table. The respective friction force has to be considered as an “externally applied working load.”

**Force required to elevate the mass \( F_{Ui} \) (4th addend)**
\( F_{Ui} \) is the force required to elevate the mass of the slide and the load (not required in horizontal drives).

**Formula for inclining actuation**

\[
F_{Ui} = g \cdot m \cdot \sin \alpha \quad [N]
\]

For declining actuation \( \sin \alpha \) becomes negative and therefore the force component \( F_{Ui} \) will be negative.

\[
sin \alpha = \frac{h_T}{l_T} \Rightarrow F_{Ui} = g \cdot m \cdot \frac{h_T}{l_T}
\]

\( g \) = Acceleration of gravity = 9.81 m/s\(^2\)
\( m \) = Sum of slider mass and load
\( \alpha \) = Angle of inclination [\(^\circ\)]
\( h_T \) = Elevating height [mm]
\( l_T \) = Conveying length [mm]

The peripheral force \( F_U \) at the drive pulley is therefore the sum of the above forces:

\[
F_U = F_{Us} + F_r + F_E + F_{Ui} \quad [N]
\]
Step 2
Selection of belt, belt width and pitch
To select the belt pitch please follow the instructions in the Design Guide chapter. This chapter will help you safely evaluate the tooth and select the belt pitch $P_b$ according to the peripheral force $F_U$.
The graphs also provide an estimate of the required belt width.

Step 3
Definition of pulley diameters / number of pulley teeth
Use the preliminary pulley diameter $d$ desired for the design envelope and the selected pitch $t$ to determine the preliminary number of pulley teeth.

$$z_P = \frac{d \cdot \pi}{P_b}$$

- $z_P$ = Number of pulley teeth
- $d$ = (Effective) Pulley diameter [mm]
- $P_b$ = Belt pitch [mm]

Round off to a whole number of pulley teeth $z_P$. Give preference to common pulley diameters. Check against the minimum number of pulley teeth $z_{min}$ for the selected belt type given in the product data sheets.

Finally, determine the effective pitch diameter $d$ according to the number of pulley teeth $z_P$ chosen:

$$d = \frac{P_b \cdot z_P}{\pi} \text{ [mm]}$$
Step 4
Definition of center distances and belt length

For applications with more than two pulleys the design envelope is commonly calculated on a CAD system or manually.

For two-pulley applications use the following procedure:

Use the preliminary center distance $e$ desired for the design envelope to determine a preliminary number of belt teeth $z_b$:

$$z_b = \frac{2 \cdot e}{P_b} + z_P$$

$z_b$ = Number of belt teeth [-]
$z_P$ = Number of pulley teeth [-]
$e$ = Center-to-center distance [mm]
$P_b$ = Belt pitch [mm]

For unequal pulley diameters:

$$z_b = \frac{2 \cdot e}{P_b} + \frac{z_{F1} + z_{F2}}{2} + \frac{P_b}{4e} \left(\frac{z_{F2} - z_{F1}}{\pi}\right)^2$$

$\bf{Determine the belt length $l_0$ according to the number of belt teeth chosen:}$$

$$l_0 = z_b \cdot P_b$$

$\bf{Determine the center-to-center distance $e$ corresponding to the chosen belt length.}$

For equal-diameter pulleys:

$$e = \frac{l_0 - d \cdot \pi}{2}$$

For unequal-diameter pulleys:

$$e = \frac{l_0 - \frac{\pi \cdot (d_2 + d_1)}{2} + \frac{l_0 - \frac{\pi \cdot (d_2 + d_1)}{2}}{2} - 2 \frac{(d_2 + d_1)^2}{4}}{2}$$

$l_0$ = Belt length [mm]
$z_b$ = Number of belt teeth
$P_b$ = Belt pitch [mm]
$e$ = Center-to-center distance [mm]
$d_1, d_2$ = Pitch diameter of pulley [mm]

Round off to a whole number of belt teeth $z_b$. If your application requires profiles, consider the profile spacing when selecting the number of belt teeth. Please note that the ideal profile design locates the profile over the tooth (not between the teeth).
Step 5
Calculation of the number of teeth in mesh on the drive pulley

Calculate the number of teeth in mesh $z_m$ using the appropriate formula.

For two equal-diameter pulleys:

$$z_m = \frac{z_a}{2}$$

- $z_a$ = Number of pulley teeth of the drive pulley
- $\beta$ = Arc of contact on the respective pulley [°]
- $d_1$, $d_2$ = Pitch diameter of pulley [mm]
- $e$ = Center-to-center distance [mm]

For two unequal-diameter pulleys:

$$z_m = z_a \left[ 0.5 - \frac{d_2 - d_1}{2\pi \cdot e} \right]$$

For pulleys with a known arc of contact:

$$z_m = \frac{z_a \cdot \beta}{360}$$

Determine the tooth-in-mesh factor $t_m$ according to these tables:

**Joined endless belts**

<table>
<thead>
<tr>
<th>No. of teeth in mesh $z_m$</th>
<th>Tooth-in-mesh factor $t_m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.20</td>
</tr>
<tr>
<td>2</td>
<td>0.40</td>
</tr>
<tr>
<td>3</td>
<td>0.55</td>
</tr>
<tr>
<td>4</td>
<td>0.70</td>
</tr>
<tr>
<td>5</td>
<td>0.85</td>
</tr>
<tr>
<td>&gt; 5</td>
<td>1.00</td>
</tr>
</tbody>
</table>

**Open-ended belts (without joint)**

<table>
<thead>
<tr>
<th>No. of teeth in mesh $z_m$</th>
<th>Tooth-in-mesh factor $t_m$</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.15</td>
</tr>
<tr>
<td>2</td>
<td>0.30</td>
</tr>
<tr>
<td>3</td>
<td>0.40</td>
</tr>
<tr>
<td>4</td>
<td>0.50</td>
</tr>
<tr>
<td>5</td>
<td>0.60</td>
</tr>
<tr>
<td>6</td>
<td>0.70</td>
</tr>
<tr>
<td>7</td>
<td>0.80</td>
</tr>
<tr>
<td>8</td>
<td>0.85</td>
</tr>
<tr>
<td>9</td>
<td>0.90</td>
</tr>
<tr>
<td>10</td>
<td>0.95</td>
</tr>
<tr>
<td>11</td>
<td>0.97</td>
</tr>
<tr>
<td>&gt; 11</td>
<td>1.00</td>
</tr>
</tbody>
</table>
Step 6

**Determination of minimal tensile force in the slack belt strand**

The tensile force in the slack belt side $F_2$ prevents jumping off the pulley teeth during belt operation. Based on experience, timing belts perform best with slack-side tension in the range 0.1 to 0.3 times the peripheral force $F_U$. Therefore:

$$F_2 = 0.2 \cdot F_U \quad [N]$$

or expressed in elongation:

$$\varepsilon_2 = 0.2 \cdot \varepsilon_U \quad [%]$$

$F_2$ = Tensile force in the slack belt strand [N]
$F_U$ = Peripheral force [N]

The initial elongation for belt applications with fixed center distance can also be approximated using the following formulas:

**Drives with controlled belt tension**

Since HabaSYNC® timing belts have a very high stress-strain ratio, it is highly recommended (at least for belt lengths below 6 m/20 ft) to use a tensioning device to provide controlled belt tension. Typically, a constant shaft load or slack-side tension is incorporated by using pneumatic cylinders, spring-loaded or gravity tensioners, etc.

Such tensioning devices provide the advantage of reduced maintenance and minimized maximum belt tension, both of which have a positive influence on belt life.

Since the minimum tensile force in the slack-side should be about 0.2 times the peripheral force $F_U$, the pressure force of a tensioning idler $F_{WT}$ can be calculated as follows:

$$F_{WT} = 0.4 \cdot F_U \cdot \sin \left( \frac{\beta_r}{2} \right) \quad [N]$$

$F_{WT}$ = Pressure force of slack-side tensioning idler [N]
$F_U$ = Peripheral force [N]
$\beta_r$ = Arc of contact of the belt on the tensioning idler
(see table in calculation step 9)

**Drives with a fixed center-to-center distance**

Drives with fixed center distances typically incorporate an adjustable shaft locked after pretensioning the belt. Assuming tight- and slack-side tensions are constant over the respective belt lengths, and a minimum slack-side elongation in the range of the above relationship, the initial belt tension $\varepsilon_0$ is:

$$\varepsilon_0 = \varepsilon_2 + \varepsilon_U \cdot \frac{l_1}{l_0} \quad [%]$$

$\varepsilon_0$ = Initial belt elongation [%]
$\varepsilon_2$ = Minimal belt elongation in the slack side [%]
$\varepsilon_U$ = Belt elongation generated by peripheral force $F_U$ [%]
$l_0$ = Belt length = $l_1 + l_2$ [mm]
$l_1$ = Length of the tight belt strand [mm]

The initial elongation for belt applications with fixed center distance can also be approximated using the following formulas:

**Head drives**

$$\varepsilon_0 = 0.5 \cdot \varepsilon_U \quad [%]$$

**Tail drives**

$$\varepsilon_0 = \varepsilon_U \quad [%]$$

**Center drives**

$$\varepsilon_0 = 0.75 \cdot \varepsilon_U \quad [%]$$

* See Design Guide, Tensioning devices on page 27
** See Design Guide, Drive concept on page 28
**Step 7**

**Calculation of elongations and forces in the tight and slack side**

The belt elongation $\varepsilon_1$ in the **tight** belt strand is obtained by (for fixed center distances):

\[ \varepsilon_1 = \varepsilon_0 + \varepsilon_U \cdot \frac{l_2}{l_0} \] [%]

The respective force $F_1$ in the **tight** side is obtained by: (for fixed center distances)

\[ F_1 = F_0 + F_U \cdot \frac{l_2}{l_0} \] [N]

**The expression $\frac{l_2}{l_0}$ is commonly substituted by:**

- 0.75 for the head drive
- 0.5 for the center drive
- 0.25 for the tail drive

The belt elongation $\varepsilon_2$ in the **slack** belt strand is obtained by (for fixed center distances):

\[ \varepsilon_2 = \varepsilon_0 - \varepsilon_U \cdot \frac{l_1}{l_0} \] [%]

The respective force $F_2$ in the **slack** side is obtained by:

\[ F_2 = F_0 - F_U \cdot \frac{l_1}{l_0} \] [N]

**The expression $\frac{l_1}{l_0}$ is commonly substituted by:**

- 0.25 for the head drive
- 0.5 for the center drive
- 0.75 for the tail drive

For drives with constant slack-side tension the force $F_2$ in the slack side is defined by the tensioning device and the force in the tight side is: $F_1 = F_2 + F_U$.

- $F_0 = \text{Tensile force due to initial tension} = \varepsilon_0 \cdot k_{1\%}$ [N]
- $F_1 = \text{Maximum tensile force in the tight belt strand} [\text{N}]$
- $F_2 = \text{Minimum tensile force in the slack belt strand} [\text{N}]$
- $F_U = \text{Peripheral force} [\text{N}] (F_U = F_1 - F_2)$
- $\varepsilon_0 = \text{Initial belt elongation} [%]$
- $\varepsilon_1 = \text{Maximal belt elongation in the tight side} [%]$
- $\varepsilon_2 = \text{Minimal belt elongation in the slack side} [%]$
- $\varepsilon_U = \text{Belt elongation generated by peripheral force} F_U [%] (\varepsilon_U = \varepsilon_1 - \varepsilon_2)$
- $l_0 = \text{Belt length} = l_1 + l_2 \text{ [mm]}$
- $l_1 = \text{Length of the tight belt strand} \text{ [mm]}$
- $l_2 = \text{Length of the slack belt strand} \text{ [mm]}$
Step 8
Calculation of required belt width

The determination of the required belt width has to include two independent criteria; required belt width in terms of:

- A admissible tensile force
- B admissible load on teeth

A  Determine the admissible tensile force $F_{adm}$ of the selected pitch given in the data sheets.
   Note that $F_{adm}$ is different for open-ended and joined endless belts.

Since a high rotational frequency of the belt may lead to high stress on the belt teeth (due to buildup of heat on the drive pulley), the speed factor $c_v$ has to be considered if the belt rotates more than once per second.

To find this speed factor the rotational frequency $f_R$ of the belt has to be calculated:

$$ f_R = \frac{v \cdot 1000}{l_0} \quad [1/s] $$

$v$ = Belt speed [m/s]
$l_0$ = Belt length [mm]

With the rotational frequency $f_R$ the speed factor $c_v$ can be derived by means of the graph below or mathematically:

$$ c_v = 1 - \frac{50 \cdot v}{l_0} $$

Determine the required belt width $b_{req}$ in terms of admissible tensile force and speed factor:

$$ b_{req} = \frac{F_1 \cdot b_0}{F_{adm} \cdot c_v} \quad [mm] $$

$b_{req}$ = Minimum required belt width [mm]
$F_1$ = Maximum tensile force in the tight belt strand [N]
$b_0$ = Estimated belt width [mm]
$F_{adm}$ = Admissible tensile force (different values for open and joined belts!) [N]
$c_v$ = Speed factor

B  To determine the admissible load on teeth, specify the tooth-in-mesh factor $t_m$ for joined or endless belts (see step 5).

Determine the required belt width $b_{req}$ in terms of tooth strength:

$$ b_{req} = \frac{F_U \cdot b_0}{F_{adm} \cdot t_m \cdot c_v} \quad [mm] $$

$b_{req}$ = Minimum required belt width [mm]
$F_U$ = Peripheral force [N]
$b_0$ = Estimated belt width [mm]
$F_{adm}$ = Admissible tensile force (different values for open and joined belts!) [N]
$t_m$ = Tooth-in-mesh factor (table step 5)
$c_v$ = Speed factor

Select the standard belt width that satisfies the last two conditions.

The forces contributing to $F_U$ which in step 1 were estimated can now be calculated accurately. Evaluate the contribution of these forces to the peripheral force $F_U$ and, if necessary, recalculate $F_U$ and repeat steps 6, 7 and 8.

For conveyors, the dimensions of the transported products will normally determine the belt width.
Step 9

Calculation of shaft loads

For an arc of contact of 180° the shaft load \( F_W \) is:

\[
F_W = F_1 + F_2 \quad [\text{N}]
\]

For pulleys and rollers with an arc of contact \( \beta \neq 180° \), the shaft load can be determined using the following approximation method:

\[
F_W = (F_1 + F_2) \cdot \sin \left( \frac{\beta}{2} \right) \quad [\text{N}]
\]

For nondriven pulleys (tail pulley, idlers, etc.) the forces \( F_1 \) and \( F_2 \) are the same.

Determine the static shaft load \( F_{W_{As}} \) and dynamic shaft load \( F_{W_{Ad}} \) of the drive pulley:

\[
F_{W_{As}} = 2 \cdot F_0 \cdot \sin \left( \frac{\beta}{2} \right) \quad [\text{N}]
\]

\[
F_{W_{Ad}} = (F_1 + F_2) \cdot \sin \left( \frac{\beta}{2} \right) \quad [\text{N}]
\]

Determine the static shaft load \( F_{W_{Us}} \) and dynamic shaft load \( F_{W_{Ud}} \) of the tail pulley:

\[
F_{W_{Us}} = 2 \cdot F_0 \cdot \sin \left( \frac{\beta}{2} \right) \quad [\text{N}]
\]

\[
F_{W_{Ud}} = 2 \cdot F_1 \cdot \sin \left( \frac{\beta}{2} \right) \quad [\text{N}]
\]

Arc of contact \( \beta \) | \( \sin \frac{\beta}{2} \)
---|---
10° | 0.176
20° | 0.283
30° | 0.405
40° | 0.522
50° | 0.633
60° | 0.707
70° | 0.770
80° | 0.819
90° | 0.866
100° | 0.906
110° | 0.940
120° | 0.966
130° | 0.985
140° | 0.996
150° | 0.996
160° | 0.996
170° | 0.996
180° | 1.000

Since in linear-positioning applications the highest shaft load of the tail pulley \( F_{W_{Ud}} \) occurs during acceleration when the load moves away from the drive pulley, the tension of both belt strands of the tail pulley is equivalent to \( F_1 \).
Step 10
Calculation of drive power and required motor power

The required power on the drive pulley is:

\[ P = \frac{F_U \cdot v}{1000} \]  [kW] or

\[ P = \frac{F_U \cdot d_a \cdot \pi \cdot n_1}{60000} \]  [kW]

- \( F_U \) = Peripheral force [N]
- \( v \) = Belt speed [m/s]
- \( d_a \) = Pitch diameter of driving pulley [mm]
- \( n_1 \) = Number of revolutions of driving pulley [1/min]

When considering the efficiency of the gearbox placed between the drive pulley and the motor, the required power of the motor \( P_M \) is:

\[ P_M = \frac{P \cdot 100}{\eta} \]  [kW]

The respective torque \( M_a \) on the drive pulley shaft is:

\[ M_a = \frac{F_U \cdot d_a}{2000} \]  [Nm]

- \( P_M \) = Power of the motor [kW]
- \( \eta \) = Efficiency of gearbox [%]*
- \( M_a \) = Torque on drive pulley shaft [Nm]

* For an application with a normal motor/gearbox unit we recommend using the default value of \( \eta = 75\% \) if the exact figure is unknown.
Step 11  
Calculation of positioning error

Positioning errors have to be distinguished in terms of:

- random positioning error $\Delta x_R$ (tolerance when many positioning procedures are compared with each other)
- systematic positioning error $\Delta x_S$ (referring to the tolerance of the belt pitch)

The total tolerance (tolerance referring to an angle of rotation of the drive pulley) is the sum of the above partial addends.

In both cases the random positioning error has to be calculated. To define the total error $\Delta x$ the accuracy factor of the specific belt [%] times the maximum covered distance of the slide has to be added to the random positioning error.

$$\Delta x = \Delta x_R + \Delta x_S = \Delta x_R + \frac{l_T \cdot af}{100}$$

$\Delta x =$ Positioning error [mm]  
$l_T =$ Maximal covered distance of the slide [mm]  
$af =$ Accuracy factor of belt [%]  
$\Delta x_R =$ Random positioning error  
$\Delta x_S =$ Systematic positioning error

HabaSYNC® timing belts commonly have a pitch tolerance of 0.04%, thus the accuracy factor af is:

$af = 0.04$

This value has been carefully evaluated through thorough measurements.

The random positioning error $\Delta x_R$ is the sum of the following three partial errors:

- **A** Belt elongation due to elasticity of the belt $\Delta x_1$
- **B** Deformation of teeth in mesh on the drive pulley $\Delta x_2$
- **C** Backlash due to the clearance between the belt teeth and the pulley grooves $\Delta x_3$

A When positioning the mass, a force component generates a belt elongation which causes a positioning error. This force is caused by resistance of the bearings or by external forces at the slider (e.g. mass on an inclined linear-positioning drive).

This positioning error is influenced by:

- Position of the slider (length of tight and slack belt strand)
- Belt strength
- The possible variation of the force on the slider $\Delta F$

The partial error $\Delta x_1$ of a slider in a determined position is:

$$\Delta x_1 = \frac{\Delta F \cdot l_1 \cdot (l_0 - l_1)}{l_0 \cdot k_{1\%} \cdot 100}$$

$\Delta x_1 =$ Maximal possible deviation of slider position caused by belt elongation [mm]  
$\Delta F =$ Highest possible variation of force component on the positioned slider [N]  
$l_0 =$ Belt length [mm]  
$l_1 =$ Length of tight belt strand if the slider is in critical position [mm]*  
$k_{1\%} =$ Tensile force for 1% elongation [N]

* In most cases the critical position of the slider means the maximum distance from the drive pulley.
B The deformation of teeth in mesh on the drive pulley is in most cases negligible. However, in highly demanding applications it has to be considered. Since an exact calculation of this deformation is very complex, we have developed a simplified estimation:

$$\Delta x_2 = \frac{\Delta F \cdot df}{t_m} \quad [\text{mm}]$$

$$\Delta x_2 = \text{Maximal possible deviation of the slider position caused by the deformation of belt teeth} \quad [\text{mm}]$$

$$\Delta F = \text{Highest possible variation of force component on the positioned slider} \quad [\text{N}]$$

$$df = \text{Deformation factor}$$

$$t_m = \text{Tooth-in-mesh factor (see step 5)}$$

Since the deformation factor $df$ is dependent on the tooth load and tooth shape, we recommend using the following approximations:

$$df = 0.125 \cdot \frac{P_b}{k_{1\%}}$$

This is valid for belts with a trapezoid or curvilinear tooth shape (T5, T10, T20, XL, L, H, XH, HTD5, HTD8, HTD14)

$$df = 0.075 \cdot \frac{P_b}{k_{1\%}}$$

Valid for belts with a modified trapezoid tooth shape (AT5, AT10, AT20)

If the respective tolerances are not mentioned and common pulleys are used, we recommend using a general factor of 0.05* times the belt pitch.

* Since AT type belts generally have fewer backlashes, a factor of 0.03 is usually sufficient for belts with a modified tooth shape

$$\Delta x_3 = \text{Maximal clearance between the belt teeth and the pulley grooves} \quad [\text{mm}]$$

$$P_b = \text{Belt pitch} \quad [\text{mm}]$$

For demanding applications where minimal backlash is required, use zero backlash pulleys. If such pulleys are used, it is not necessary to consider $\Delta x_3$.

**Resulting positioning error consists of**

- **random error:**
  $$\Delta x_R = \Delta x_1 + \Delta x_2 + \Delta x_3 \quad [\text{mm}]$$

- **systematic error:**
  $$\Delta x_S = \frac{l_T \cdot af}{100} \quad [\text{mm}]$$

  $$l_T = \text{Maximum covered distance of the slide} \quad [\text{mm}]$$

  $$af = \text{Accuracy factor of belt} \quad [\%]$$

The total error (absolute) is:

$$\Delta x = \Delta x_R + \Delta x_S \quad [\text{mm}]$$

and total error (relative) is:

$$x = \frac{\Delta x \cdot 100}{l_T} \quad [\%]$$
**Calculation example**

An inclined conveyor with two timing belts is used to transport heavy containers. The belt is supported by HabiPLAST® guide strips made out of ultrahigh molecular weight PE (UHMW PE). A gas spring provides constant belt tension in the slack side.

**Technical data and parameters**

<table>
<thead>
<tr>
<th>Belt series</th>
<th>Metric pitch, trapezoid tooth shape</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conveying length</td>
<td>3,000 mm</td>
</tr>
<tr>
<td>Elevating height</td>
<td>800 mm</td>
</tr>
<tr>
<td>Total load on belt</td>
<td>900 kg (450 kg per belt)</td>
</tr>
<tr>
<td>Position of drive</td>
<td>head</td>
</tr>
<tr>
<td>Arc of contact on drive pulley</td>
<td>180°</td>
</tr>
<tr>
<td>Arc of contact on pressure roller</td>
<td>60°</td>
</tr>
<tr>
<td>Conveyor bed</td>
<td>Slider bed (UHMW PE)</td>
</tr>
<tr>
<td>Diameter of drive pulley</td>
<td>≈ 150 mm</td>
</tr>
<tr>
<td>Diameter of tension pulleys</td>
<td>as small as possible</td>
</tr>
<tr>
<td>Belt speed</td>
<td>40 m/min</td>
</tr>
</tbody>
</table>
**Evaluation of tooth and pitch according to the Design Guide**

In order to evaluate the tooth, the belt pitch and width, the peripheral force \( F_U \) at the drive pulley needs to be estimated first.

The peripheral force \( F_U \) for a conveying application is primarily the sum of the following partial forces resisting the belt motion:

- Friction force \( F_{US} \) [N]
- Force required to elevate the carried goods \( F_{Ui} \) [N]

Total mass to be carried over the slider bed = 900 kg (450 kg per belt)

Coefficient of friction between the belt and the slider bed \( \mu_G = 0.4 \) according to the product data sheet for the T belt series. So the friction force \( F_{Us} \) is:

\[
F_{US} = g \cdot m \cdot \mu_G = 9.81 \cdot 450 \cdot 0.4 = 1,766 \, \text{N}
\]

Conveying length = 3000 mm
Elevating height = 800 mm

With these data the elevation force \( F_{Ui} \) is:

\[
F_{Ui} = g \cdot m \cdot \frac{h_T}{l_T} = 9.81 \cdot 450 \cdot \frac{800}{3000} = 1,177 \, \text{N}
\]

**Therefore, the estimated peripheral force \( F_U \) is:**

\[
F_U = F_{US} + F_{Ui} = 2,943 \, \text{N}
\]

The graphic in the Design Guide for T series joined belts indicates that for this peripheral force a T10 with a width of 100 mm is required.

Therefore, the 150 mm drive pulley with a 10 mm pitch is required, with the following number of teeth:

\[
z_P = \frac{d \cdot \pi}{P_b} = \frac{150 \cdot 3.14}{10} \approx 47
\]

\( \Rightarrow \) Chosen \( z_P = 48 \) (common pulley diameter)

\( d \) = Effective pulley diameter [mm]
\( P_b \) = Belt pitch [mm]

Following the Design Guide, it is obvious that for a drive pulley with 48 teeth and an arc of contact of 180°, there will be more than 5 teeth in mesh.

To define the speed factor we have to proceed as follows:

\[
\nu \,[\text{m/s}] = \frac{\nu \,[\text{m/min}]}{60}
\]

The indicated belt speed of 40 m/min corresponds to 0.67 m/s.

To define the belt length, a rough approximation is enough. Since the belt is a little longer than twice the conveying length, we will consider a belt length of about 7,000 mm.

Accordingly, the rotational frequency \( f_R \) is:

\[
f_R = \frac{\nu \cdot 1000}{l_0} = \frac{0.67 \cdot 1000}{7000} \approx 0.1 \, \text{1/s}
\]

Since \( f_R \) is well below 1 rotation per second, no speed factor has to be considered.

Therefore, the consideration of a tooth-in-mesh or speed factor is not required (which means that \( t_{m} = 1.0 \) and \( c_{v} = 1.0 \)).

The preselected belts are therefore two T10 belts with a width of 100 mm each.
**Calculation examples**

**Conveying**

**Calculation according to the Calculation Guide**

**Step 1**

**Determination of peripheral force**

For an accurate determination of the peripheral force $F_U$ at the drive pulley, it is now possible to also consider the belt mass. However, since the transported mass of 900 kg is so much greater than the mass of the belts, the consideration of the belt mass to define the friction force on the slider bed is not required.

Therefore, the already estimated peripheral force $F_U$ of 2,943 N is accurate enough for the final calculation.

**Step 2**

**Selection of the belt type, belt width and pitch**

Selected belt according to Design Guide: T10, with 10 mm pitch 100 mm wide

**Step 3**

**Pulley diameters / number of pulley teeth**

To define the design envelope around all pulleys the effective pulley diameters have to be defined.

Since the number of teeth for the drive and tail pulley is already defined, the respective effective diameter according to the chosen number of pulley teeth $z_P$ is:

$$d = \frac{P_b \cdot z_P}{\pi} = \frac{10 \cdot 48}{3.14} = 152.8 \text{ mm}$$

For the tensioner, the minimum pulley diameter for counter friction is found on the T10 product data sheet: $d_T = 60 \text{ mm}$

Idler: for forward friction the minimum number of pulley teeth is 20. Using this the respective effective diameter can be defined by:

$$d = \frac{P_b \cdot z_P}{\pi} = \frac{10 \cdot 20}{3.14} = 63.7 \text{ mm}$$

**Step 4**

**Define the center distance and belt length**

If all pulley diameters are known, the belt length of 6,540 mm (654 teeth) can be specified manually or by using a CAD tool.

**Step 5**

**Calculate the number of teeth in mesh on the drive pulley**

Following the Calculation Guide it is obvious that for the drive pulley with 48 teeth and an arc of contact of 180°, there will be more than 5 teeth in mesh. Therefore, consideration of a tooth-in-mesh factor is not required (which means that $t_m = 1.0$).

**Step 6**

**Determine the minimal tensile force in the slack belt strand**

Peripheral force $F_U = 2,943 \text{ N}$

$$F_2 = 0.2 \cdot F_U = 0.2 \cdot 2943 = 589 \text{ N}$$

$k_{1\%}$ (stress-strain ratio per unit of width) = 22,000 N

$$\varepsilon_U = \frac{F_U}{k_{1\%}} = \frac{2943}{22000} = 0.134\%$$

$$\varepsilon_2 = 0.2 \cdot \varepsilon_U = 0.2 \cdot 0.134 = 0.0268\%$$

**For drives with controlled slack-side tension**

Arc of contact of the belt on the tensioning idler $\beta_T = 60°$

Pressure force of a tensioning idler $F_{WT}$ is:

$$F_{WT} = 0.4 \cdot F_U \cdot \sin \left( \frac{180°}{2} \right) = 0.4 \cdot 2943 \cdot \sin \left( \frac{60°}{2} \right) = 589 \text{ N}$$
Step 7
Calculate the elongations and forces in the tight and slack-sides
For drives with constant slack-side tension the force in the slack side \( F_2 \) is defined by the tensioning device and the force in the tight side is:

\[
F_1 = F_2 + F_U = 589 + 2,943 = 3,532 \text{ N}
\]

Step 8
Calculate the required belt width
Determine the required belt width \( b_{req} \) in terms of admissible tensile force:

Admissible tensile force joined belt \( F_{adm} = 4,400 \text{ N} \)

\[
b_{req} = \frac{F_1 \cdot b_0}{F_{adm} \cdot c_V} = \frac{3532 \cdot 100}{4400 \cdot 1} = 80.2 \text{ mm}
\]

Determine the required belt width \( b_{req} \) in terms of tooth strength:

\[
b_{req} = \frac{F_U \cdot b_0}{F_{adm} \cdot t_m \cdot c_V} = \frac{2943 \cdot 100}{4400 \cdot 1 \cdot 1} = 67 \text{ mm}
\]

Step 9
Calculate the shaft loads

Drive pulley
For the arc of contact of 180° the dynamic shaft load \( F_{WAd} \) is:

\[
F_{WAd} = F_1 + F_2 = 3532 + 589 = 4121 \text{ N}
\]

Since the belt has a constant slack-side tension, the tension in the tight side is at the level of the slack-side tension if the conveyor is switched off or if no load is on the conveyor. Therefore, the static shaft load \( F_{WAs} \) is:

\[
F_{WAs} = 2 \cdot F_2 = 2 \cdot 589 = 1178 \text{ N}
\]

Tail pulley
On the nondriven tail pulley both belt strands are loaded with the tensile force controlled by the slack-side tensioning device. Therefore, the static and dynamic shaft loads \( F_{WUs} \) and \( F_{WUD} \) are equal.

\[
F_{WUs} = F_{WUD} = 2 \cdot F_2 \cdot \sin \left( \frac{\beta}{2} \right) = 2 \cdot 589 \cdot 0.966 = 1,137 \text{ N}
\]

Arc of contact on tail pulley \( \beta = 210° \)

\[
\begin{array}{|c|c|}
\hline
\text{Arc of contact } \beta & \sin \beta/2 \\
\hline
10° & 350° & 0.087 \\
20° & 340° & 0.174 \\
30° & 330° & 0.259 \\
40° & 320° & 0.342 \\
50° & 310° & 0.423 \\
60° & 300° & 0.500 \\
70° & 290° & 0.574 \\
80° & 280° & 0.643 \\
90° & 270° & 0.707 \\
100° & 260° & 0.766 \\
110° & 250° & 0.819 \\
120° & 240° & 0.866 \\
130° & 230° & 0.906 \\
140° & 220° & 0.940 \\
150° & 210° & 0.966 \\
160° & 200° & 0.985 \\
170° & 190° & 0.996 \\
180° & & 1.000 \\
\hline
\end{array}
\]

Step 10
Calculate the drive power and required motor power
The belt speed is given as 40 m/min. To define the power on the drive pulley the belt speed in m/s has to be calculated:

\[
v[m/s] = \frac{v[m/min]}{60} = \frac{40}{60} = 0.667 \text{ m/s}
\]

The power \( P \) on the drive pulley is:

\[
P = \frac{F_U \cdot v}{1000} = \frac{2943 \cdot 0.667}{1000} = 1.96 \text{ kW}
\]

Considering the efficiency of the gearbox of \( \eta = 75\% \), which is a recommended value if the correct figure is not known, the required motor power \( P_M \) is:

\[
P_M = \frac{P \cdot 100}{\eta} = \frac{1.96 \cdot 100}{75} = 2.61 \text{ kW}
\]
Calculation example

A timing-belt-driven vertical actuator is positioning a mass. The belt is pretensioned with a fixed center-to-center distance.

Technical data and parameters

No belt joint is required (belt ends are mechanically clamped on the slide).

<table>
<thead>
<tr>
<th>Belt series</th>
<th>Metric pitch</th>
</tr>
</thead>
<tbody>
<tr>
<td>Maximum covered distance of slide</td>
<td>3,000 mm</td>
</tr>
<tr>
<td>Elevating height</td>
<td>3,000 mm</td>
</tr>
<tr>
<td>Center-to-center distance</td>
<td>3,500 mm</td>
</tr>
<tr>
<td>Total load (slide plus load)</td>
<td>300 kg</td>
</tr>
<tr>
<td>Weight of slide</td>
<td>20 kg</td>
</tr>
<tr>
<td>Belt speed</td>
<td>0.6 m/s</td>
</tr>
<tr>
<td>Acceleration time</td>
<td>0.5 s</td>
</tr>
<tr>
<td>Position of drive</td>
<td>top</td>
</tr>
<tr>
<td>Arc of contact on pulleys</td>
<td>180°</td>
</tr>
<tr>
<td>Diameter of pulleys</td>
<td>&lt; 80 mm</td>
</tr>
<tr>
<td>Friction force of slide</td>
<td>20 N</td>
</tr>
</tbody>
</table>

Evaluation of tooth and pitch according to the Design Guide

Determination of peripheral force $F_U$:
The peripheral force $F_U$ at the drive pulley is the sum of all individual forces resisting the belt motion:

$$F_U = F_{Ui} + F_{Us} + F_f$$

- Force required to elevate the carried good (mass) $F_{Ui}$:

$$F_{Ui} = g \cdot m \cdot \frac{h_T}{l_T}$$

For vertical applications the elevating height $h_T$ and conveying length $l_T$ is identical.

$$F_{Ui} = g \cdot m \cdot 1 = 9.81 \cdot 300 = 2,943 \text{ N}$$

- Force $F_{Us}$ required for the acceleration of the mass:

$$F_{Us} = m \cdot a$$

$$a = \frac{v}{t} = \frac{0.6}{0.5} = 1.2 \text{ m/s}^2$$

$$F_{Us} = m \cdot a = 300 \cdot 1.2 = 360 \text{ N}$$

- Since the friction force of the slide $F_f$ is known, it can be considered.

$$F_f = 20 \text{ N}$$

The peripheral force $F_U$ at the drive pulley is primarily the sum of the following forces resisting the belt motion:

$$F_U = F_{Ui} + F_{Us} + F_f = 2943 + 360 + 20 = 3,323 \text{ N}$$

The estimated peripheral force $F_U$ is 3,323 N.
Following the Design Guide we can assume that for an arc of contact of 180°, more than eleven teeth are in mesh. Therefore, considering a tooth-in-mesh factor may not be required.

To define the speed factor we have to proceed as follows:

The belt speed is given with \( v = 0.6 \) m/s.

To define the belt length, a rough approximation is enough. Since the belt is slightly longer than twice the center-to-center distance, we will consider a belt length of 7,200 mm.

Accordingly, the rotational frequency \( f_R \) is:

\[
\frac{v \cdot 1000}{l_0} = \frac{0.6 \cdot 1000}{7200} = 0.83 \text{ 1/s}
\]

Since \( f_R \) is below 1 rotation per second, no speed factor needs to be considered.

**Therefore, the consideration of a tooth-in-mesh or speed factor is not required** (which means that \( t_m = 1.0 \) and \( c_V = 1.0 \)).

The graphic in the Design Guide for AT series open-ended belts shows that for this peripheral force an AT5 in a width of 75 mm or an AT10 in a width of 50 mm are required.

If small pulleys and precise positioning have higher priority, the AT5 is the right choice. If the priority is for a small belt width, AT10 should be selected.

In our calculation example we have given priority to a smaller belt width. Therefore, we have chosen **AT10 in a width of 50 mm**.

Using this information, we can make further calculations based on the Calculation Guide.
Calculation according to the Calculation Guide

**Step 1**
**Determination of peripheral force**
For an accurate determination of the peripheral force $F_U$ at the drive pulley, no additional forces have to be considered relating to the estimation according to the Design Guide.

The already estimated peripheral force $F_U$ of 3,323 N is the correct value for the final calculation.

**Step 2**
**Selection of the belt, belt width and pitch**
Selected belt according to the Design Guide: AT10, 50 mm wide

**Step 3**
**Define pulley diameters /number of pulley teeth**
According to the product data sheet for AT10 Steel the minimum number of pulley teeth is 25. Thus, the pitch diameter $d$ according to the chosen number of pulley teeth $z_p$ is:

$$d = \frac{P_b \cdot z_p}{\pi} = \frac{10 \cdot 25}{3.14} = 79.6 \text{ mm}$$

**Step 4**
**Define the center distances and belt length**
Number of pulley teeth $z_p = 25$
Center-to-center distance $e = 3,500 \text{ mm}$
Belt pitch $P_b = 10 \text{ mm}$

Number of belt teeth $z_b$:

$$z_b = \frac{2 \cdot e + z_p}{P_b} = \frac{2 \cdot 3500 + 25}{10} = 725$$

Determine the belt length $l_0$ according to the chosen number of belt teeth:

$$l_0 = z_b \cdot P_b = 725 \cdot 10 = 7,250 \text{ mm}$$

Determine the center-to-center distance $e$ corresponding to the chosen belt length (for equal diameters):

$$e = \frac{l_0 - d \cdot \pi}{2} = \frac{7250 - 79.6 \cdot 3.14}{2} = 3,500 \text{ mm}$$

**Step 5**
**Calculate the number of teeth in mesh on the drive pulley**
For two equal pulley diameters:

$$z_m = \frac{z_p}{2} = \frac{25}{2} = 12.5$$

No tooth-in-mesh factor to consider (more than 11 teeth in mesh).

**Step 6**
**Determine the minimal tensile force in the slack belt strand and initial belt extension**
Peripherial force $F_U = 3,323 \text{ N}$

$$F_2 \approx 0.2 \cdot F_U = 0.2 \cdot 3323 = 665 \text{ N}$$

$k_{1\%}$ (tensile force for 1% elongation) = 17,500 N for a AT10 with 50 mm belt width.

Belt elongation generated by peripheral force $F_U$ is:

$$\varepsilon_U = \frac{F_U}{k_{1\%}} = \frac{3323}{17500} = 0.190\%$$

Minimal belt elongation in the slack side:

$$\varepsilon_2 = 0.2 \cdot \varepsilon_U = 0.2 \cdot 0.19 = 0.0380\%$$
**Initial belt elongation $\varepsilon_0$ for drives with fixed center distance**

To determine the initial belt tension the critical position of the slide has to be rated. The critical position of the slide means the maximum length of the tight belt strand (usually the case when the slide is at the maximum distance from the drive pulley). In our case this is the situation with the mass in the lowest position.

The lowest position of the slide is about 3,250 mm beyond the drive pulley. Therefore the tight belt strand has a maximal length of about 3,300 mm.

Length of the tight belt strand $l_1 \approx 3,300 \text{ mm}$

Belt length $l_0 = l_1 + l_2 = 7,250 \text{ mm}$

(Length of the slack belt strand $l_2 \approx 3,950 \text{ mm}$)

Belt elongation $\varepsilon_U$ generated by peripheral force $F_U$

Thus the tensile force due to initial tension $F_0$ is:

$$F_0 = \varepsilon_0 \cdot k_1\% = 0.124 \cdot 17500 = 2170 \text{ N}$$

**Step 7**

**Calculate the elongations and forces in the tight and slack sides**

The force in the tight side $F_1$ is obtained by:

$$F_1 = F_0 + F_U \cdot \frac{l_2}{l_0} = 2170 + 3323 \cdot \frac{3950}{7250} = 3980 \text{ N}$$

The belt elongation in the slack belt strand $\varepsilon_2$ is obtained by:

$$\varepsilon_2 = \varepsilon_0 - \varepsilon_U \cdot \frac{l_1}{l_0} = 0.124 - 0.19 \cdot \frac{3300}{7250} = 0.0375\%$$

The respective force in the slack side $F_2$ is obtained by:

$$F_2 = F_0 - F_U \cdot \frac{l_1}{l_0} = 2170 - 3323 \cdot \frac{3300}{7250} = 657 \text{ N}$$

**Step 8**

**Calculate the required belt width**

Required belt width $b_{req}$ in terms of admissible tensile force:

The admissible tensile force $F_{adm}$ of an open AT10 belt $= 7000 \text{ N}$

$$b_{req} = \frac{F_1 \cdot b_0}{F_{adm}} = \frac{3980 \cdot 50}{7000} = 28.4 \text{ mm}$$

Required belt width $b_{req}$ in terms of tooth strength:

$$b_{req} = \frac{F_0 \cdot b_0}{F_{adm} \cdot t_m} = \frac{3323 \cdot 50}{7000 \cdot 1} = 23.7 \text{ mm}$$

The selected belt width of 50 mm satisfies these requirements.

**Step 9**

**Calculate the shaft loads**

For an arc of contact of 180° the dynamic shaft load $F_{WAd}$ on the drive pulley is:

$$F_{WAd} = F_1 + F_2 = 3980 + 657 = 4637 \text{ N}$$

On the nondriven pulley the forces of both belt strands are the same. The highest load on the pulley shaft occurs if no load is on the slide (static conditions). In this case, both belt strands have a tensile force due to initial tension $F_0$. The respective static shaft load $F_{Was}$ is:

$$F_{Was} = 2 \cdot F_0 = 2 \cdot \varepsilon_0 \cdot k_1\% = 2 \cdot 0.124 \cdot 17500 = 4340 \text{ N}$$

**Step 10**

**Calculate the drive power and respective torque**

The required power $P$ on the drive pulley is:

$$P = \frac{F_U \cdot v}{1000} = \frac{3323 \cdot 0.6}{1000} = 1.99 \text{ kW}$$

The respective torque $M_a$ on the drive pulley shaft is:

$$M_a = \frac{F_U \cdot d_m}{2000} = \frac{3323 \cdot 79.6}{2000} = 132 \text{ Nm}$$
Step 11
Calculate the positioning error

The random positioning error $\Delta x_R$ is the sum of the following three partial errors:

A Belt elongation due to elasticity of the belt $\Delta x_1$

B Deformation of the teeth in mesh on the drive pulley $\Delta x_2$

C Backlash due to the clearance between the belt teeth and the pulley grooves $\Delta x_3$

A The partial error $\Delta x_1$ is considered for the already mentioned critical position of the slide (maximum distance from the drive pulley). In our case this is when the mass is in the lowest position. In this position the slide may be loaded (max. weight 300 kg) or not (weight of slide 20 kg). The mass variation $\Delta m$ is therefore 280 kg.

The possible variation of the force $\Delta F$ on the slider is:

$$\Delta F = g \cdot \Delta m = 9.81 \cdot 280 = 2,747 \text{ N}$$

Acceleration of gravity $g = 9.81 \text{ m/s}^2$

Length of the tight belt strand $l_1 = 3,300 \text{ mm}$

Length of the slack belt strand $l_2 = 3,950 \text{ mm}$

Belt length $l_0 = l_1 + l_2 = 7,250 \text{ mm}$

$$\Delta x_1 = \frac{\Delta F \cdot l_1 \cdot (l_0 - l_1)}{l_0 \cdot k_1 \% \cdot 100} = \frac{2747 \cdot 3300 \cdot (7250 - 3300)}{7250 \cdot 1.0 \cdot 100} = 2.82 \text{ mm}$$

B Deformation of the teeth in mesh on the drive pulley $\Delta x_2$

We use the estimated deformation factor for the AT series:

$$df = 0.075 \cdot \frac{P_b}{k_{1\%}} = 0.075 \cdot \frac{10}{17500} = 0.000043$$

The maximal possible deviation of the slide position caused by the deformation of belt teeth $\Delta x_2$ is:

$$\Delta x_2 = \frac{\Delta F \cdot df}{t_m} = \frac{2747 \cdot 0.000043}{1.0} = 0.12 \text{ mm}$$

Tooth-in-mesh factor $t_m = 1.0$

C The backlash due to the clearance between the belt teeth and the pulley grooves is negligible since the weight of the slide is greater than the respective friction force. Therefore, the backlash of the pulley has no influence.

Resulting positioning error

Random error

$$\Delta x_R = \Delta x_1 + \Delta x_2 = 2.82 + 0.12 = 2.94 \text{ mm}$$

Systematic error

Since HabaSYNC® timing belts have an accuracy factor of $af = 0.04$ and the maximum covered distance of the slide is 3,000 mm:

$$\Delta x_S = \frac{l_T \cdot af}{100} = \frac{3000 \cdot 0.04}{100} = 1.2 \text{ mm}$$

Maximum covered distance of slide $l_T = 3,000 \text{ mm}$

Total error (absolute)

$$\Delta x = \Delta x_R + \Delta x_S = 2.94 + 1.2 = 4.14 \text{ mm}$$

Total error (relative)

$$x = \frac{\Delta x \cdot 100}{l_T} = \frac{4.14 \cdot 100}{3000} = 0.14\%$$
Appendix

Tolerances

**Pitch tolerance**
The pitch length tolerance is ± 0.80 mm per meter belt length.

**Length tolerance**
The length of a timing belt is given by the pitch times number of teeth. The effective length of a timing belt is also depending on the pitch tolerance.

Timing belt roll stock can have – 0.5% / + 2% in length.

**Width tolerance**

**Standard tolerance for the width of timing belts**

<table>
<thead>
<tr>
<th>Types</th>
<th>Up to 50 mm (2&quot;) width</th>
<th>Over 50 mm to 100 mm (2&quot; to 4&quot;) width</th>
<th>Over 100 mm to 200 mm* (4&quot; to 8&quot;) width</th>
</tr>
</thead>
<tbody>
<tr>
<td>T5</td>
<td>± 0.50 mm</td>
<td>± 0.75 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>T10</td>
<td>± 0.50 mm</td>
<td>± 0.75 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>T20</td>
<td>± 0.75 mm</td>
<td>± 1.00 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>AT5</td>
<td>± 0.50 mm</td>
<td>± 0.75 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>AT5P</td>
<td>± 0.50 mm</td>
<td>± 0.75 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>AT10</td>
<td>± 0.75 mm</td>
<td>± 1.00 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>AT10P</td>
<td>± 1.00 mm</td>
<td>± 1.50 mm</td>
<td>– 1.00 mm / + 1.50 mm</td>
</tr>
<tr>
<td>AT20</td>
<td>± 1.00 mm</td>
<td>± 1.50 mm</td>
<td>– 1.00 mm / + 1.50 mm</td>
</tr>
<tr>
<td>HTD5</td>
<td>± 0.50 mm</td>
<td>± 0.75 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>HTD8</td>
<td>± 0.75 mm</td>
<td>± 1.00 mm</td>
<td>– 1.00 mm / + 1.00 mm</td>
</tr>
<tr>
<td>HTD14</td>
<td>± 1.00 mm</td>
<td>± 1.50 mm</td>
<td>– 1.00 mm / + 1.50 mm</td>
</tr>
<tr>
<td>XL</td>
<td>± 0.51 mm ± .020&quot;</td>
<td>± 0.76 mm ± .030&quot; ± 1.02 mm ± .040&quot;</td>
<td></td>
</tr>
<tr>
<td>L</td>
<td>± 0.51 mm ± .020&quot;</td>
<td>± 0.76 mm ± .030&quot; ± 1.02 mm ± .040&quot;</td>
<td></td>
</tr>
<tr>
<td>H</td>
<td>± 0.51 mm ± .020&quot;</td>
<td>± 0.76 mm ± .030&quot; ± 1.02 mm ± .040&quot;</td>
<td></td>
</tr>
<tr>
<td>XH</td>
<td>± 1.02 mm ± .040&quot;</td>
<td>± 1.02 mm ± .040&quot; ± 1.02 mm ± .040&quot;</td>
<td></td>
</tr>
</tbody>
</table>

* This width tolerance is equal to the manufacturing width tolerance.

**Width in joining area**
In the joining area it can happen that the width varies compared to the rest of the belt.

**Tolerances for the joining area**

<table>
<thead>
<tr>
<th>Up to 50 mm (2&quot;) width</th>
<th>Over 50 mm to 100 mm (2&quot; to 4&quot;) width</th>
<th>Over 100 mm to 150 mm* (4&quot; to 6&quot;) width</th>
</tr>
</thead>
<tbody>
<tr>
<td>– 0.5 mm</td>
<td>– 1.0 mm</td>
<td>– 1.5 mm</td>
</tr>
</tbody>
</table>

Pitch length tolerance ± 0.80 mm

For any tolerances over 200 mm (8") belt width, please contact your Habasit representative to discuss your specific needs.
Standard belt thickness tolerance

<table>
<thead>
<tr>
<th>Types</th>
<th>Normal thickness</th>
<th>Tolerance</th>
</tr>
</thead>
<tbody>
<tr>
<td>T5</td>
<td>2.2 mm</td>
<td>± 0.15 mm</td>
</tr>
<tr>
<td>T10</td>
<td>4.5 mm</td>
<td>± 0.30 mm</td>
</tr>
<tr>
<td>T20</td>
<td>8.0 mm</td>
<td>± 0.45 mm</td>
</tr>
<tr>
<td>AT5</td>
<td>2.7 mm</td>
<td>± 0.20 mm</td>
</tr>
<tr>
<td>AT5P</td>
<td>2.7 mm</td>
<td>± 0.20 mm</td>
</tr>
<tr>
<td>AT10</td>
<td>4.5 mm</td>
<td>± 0.30 mm</td>
</tr>
<tr>
<td>AT10P</td>
<td>4.8 mm</td>
<td>± 0.30 mm</td>
</tr>
<tr>
<td>AT20</td>
<td>8.0 mm</td>
<td>± 0.45 mm</td>
</tr>
<tr>
<td>AT20P</td>
<td>8.4 mm</td>
<td>± 0.50 mm</td>
</tr>
<tr>
<td>HTD5</td>
<td>3.6 mm</td>
<td>± 0.20 mm</td>
</tr>
<tr>
<td>HTD8</td>
<td>5.6 mm</td>
<td>± 0.30 mm</td>
</tr>
<tr>
<td>HTD14</td>
<td>10.0 mm</td>
<td>± 0.50 mm</td>
</tr>
<tr>
<td>XL</td>
<td>2.3 mm 0.090&quot;</td>
<td>± 0.15 mm ± .006&quot;</td>
</tr>
<tr>
<td>L</td>
<td>3.6 mm 0.142&quot;</td>
<td>± 0.20 mm ± .008&quot;</td>
</tr>
<tr>
<td>H</td>
<td>4.3 mm 0.169&quot;</td>
<td>± 0.30 mm ± .012&quot;</td>
</tr>
<tr>
<td>XH</td>
<td>11.2 mm 0.441&quot;</td>
<td>± 0.50 mm ± .020&quot;</td>
</tr>
</tbody>
</table>

Thickness tolerance of covers

<table>
<thead>
<tr>
<th>Cover</th>
<th>Thickness up to 2 mm</th>
<th>Over 2 mm to 5 mm</th>
<th>Over 5 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Solid PU cover</td>
<td>± 0.20 mm</td>
<td>± 0.40 mm</td>
<td>± 0.60 mm</td>
</tr>
<tr>
<td>Structured surface</td>
<td>± 0.50 mm</td>
<td>± 1.00 mm</td>
<td>± 1.50 mm</td>
</tr>
<tr>
<td>Rubber</td>
<td>± 0.30 mm</td>
<td>± 0.50 mm</td>
<td>± 0.75 mm</td>
</tr>
<tr>
<td>Foam</td>
<td>± 0.30 mm</td>
<td>± 0.50 mm</td>
<td>± 0.75 mm</td>
</tr>
<tr>
<td>Grinded surface</td>
<td>± 0.30 mm</td>
<td>± 0.50 mm</td>
<td>± 0.50 mm</td>
</tr>
</tbody>
</table>

Holes
The tolerance of punched holes is ± 0.5 mm to a reference point in width and ± 0.3 mm in length direction to the pitch.

Longitudinal grooves and V-guides
The tolerance for longitudinal grooves and for the position of V-guides is ± 0.5 mm to a reference point.

Pockets and transverse grooves
The tolerance for machined pockets and cross grooves is ± 0.5 mm for the dimensions and for the position to a reference point.

Profile position
The tolerance for welded profiles over tooth is ± 0.5 mm to the pitch. If the position of the profile is not over tooth and/or the spacing is not a multiple of the pitch, the tolerance is ± 0.8 mm.

Special tolerances
Standard tolerances can be achieved without additional efforts. Beside the standard tolerances we can achieve tighter tolerances. Those special tolerances require additional production expenditure and have to be calculated separately.
Appendix

Material properties

**HabaSYNC® timing belts are manufactured from various primary materials**
Thermoplastic polyurethane is the elastomer, with a tensile-cord reinforcement that can be either steel\(^1\) or aramide.

Our standard products are manufactured from thermoplastic polyurethane in 88, 90 or 92 Shore A hardness polyurethane, which is white, blue or transparent in color.

Polyurethane is the preferred choice of elastomer due to its high strength and application performance. Thermoplastic polyurethane also allows the belt to be finished to any length by using a thermal-welding process.

**HabaSYNC® thermoplastic polyurethane (TPU) advantages include:**
- Excellent dimensional stability
- Excellent wear resistance
- Excellent chemical resistance
- High tear resistance
- High tooth-shear strength
- Runs with no lubrication; no maintenance
- Precision-formed teeth
- High linear- and angular-positioning precision
- Good temperature range
- Good structural flexibility

**Thermoplastic polyurethane**
HabaSYNC® timing belts are highly resistant to abrasion. They are ideal for applications that require extremely clean running conditions. Our polyurethane provides greater stiffness than less hard materials such as rubber or softer urethanes. As a result, our teeth have less deflection, which provides more efficient belt-to-pulley meshing.

\(^1\) Some product types are available with stainless steel tensile member. Please contact your Habasit representative to check availability.
## Appendix

### Material properties

<table>
<thead>
<tr>
<th>Standard material</th>
<th>Code</th>
<th>Hardness</th>
<th>Properties</th>
<th>Cord</th>
<th>Temperature</th>
</tr>
</thead>
</table>
| White polyester thermoplastic polyurethane    | 01 TPU | 92 Shore A | Highly resistant to abrasion  
Long shelf life, no aging  
Resistant to ozone, oils and grease | S = Steel  
A = Aramide  
P = Performance  
I = Stainless steel | -20 to +80 °C  
-4 to +176 °F |
| Transparent polyester thermoplastic polyurethane | 02 TPU | 88 Shore A | Resistant to oils, solvents and UV radiation | S = Steel  
A = Aramide  
I = Stainless steel | -20 to +70 °C  
-4 to +150 °F |
| Blue polyether TPU                             | 05 TPU | 90 Shore A | Hydrolysis resistant  
High oil, grease and solvent resistance | A = Aramide  
I = Stainless steel | -20 to +80 °C  
-4 to +176 °F |
| Nonstandard material                           | Code  | Hardness | Properties                                                                 | Cord           | Temperature          |
| Green polyester polyurethane                  | 03 TPU | 88 Shore A | Good friction  
Superior resistance to oils | A = Aramide | -20 to +70 °C  
-4 to +150 °F |
| White carbonate urethane                      | 04 TPU | 92 Shore A | Highly resistant to abrasion  
Resistant to hydrolytic influence | A = Aramide | -20 to +80 °C  
-4 to +176 °F |
| Black polyurethane                            | 06 TPU | 92 Shore A | Highly resistant to abrasion | S = Steel | -20 to +80 °C  
-4 to +176 °F |

### Steel or aramide cords

Both steel and aramide cord tensile members offer significant, but still flexible stiffness. This is important in linear-drive and precision-conveying applications where minimal creep is needed, with structural flexibility required to yield precise bidirectional movement and accurate positional product placement.

Our tensile cords yield low elongation that delivers high positional accuracy and excellent flexibility. All this means long life with little or no retensioning required.

### Truly encapsulated cord reinforcement

Accurately machined tooling and a state-of-the-art tension control system allow for precise placement of steel or aramide cords in the body of each pitch belt. In our standard product, predesigned slit lanes are engineered to ensure slitting does not cut into and expose the cord reinforcement.
### Diameters of cords

#### Steel

<table>
<thead>
<tr>
<th>Type</th>
<th>Cord diameter [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T5</td>
<td>0.33</td>
</tr>
<tr>
<td>T10</td>
<td>0.63</td>
</tr>
<tr>
<td>T20</td>
<td>0.91</td>
</tr>
<tr>
<td>AT5</td>
<td>0.49</td>
</tr>
<tr>
<td>AT10</td>
<td>0.91</td>
</tr>
<tr>
<td>AT20</td>
<td>1.21</td>
</tr>
<tr>
<td>HTD5</td>
<td>0.49</td>
</tr>
<tr>
<td>HTD8</td>
<td>0.91</td>
</tr>
<tr>
<td>HTD14</td>
<td>1.21</td>
</tr>
<tr>
<td>XL</td>
<td>0.33</td>
</tr>
<tr>
<td>L</td>
<td>0.49</td>
</tr>
<tr>
<td>H</td>
<td>0.63</td>
</tr>
<tr>
<td>XH</td>
<td>0.91</td>
</tr>
</tbody>
</table>

#### Aramide

<table>
<thead>
<tr>
<th>Type</th>
<th>Cord diameter [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T5</td>
<td>0.34</td>
</tr>
<tr>
<td>T10</td>
<td>0.75</td>
</tr>
<tr>
<td>T20</td>
<td>1.25</td>
</tr>
<tr>
<td>AT5</td>
<td>0.59</td>
</tr>
<tr>
<td>AT10</td>
<td>1.25</td>
</tr>
<tr>
<td>AT20</td>
<td>1.25</td>
</tr>
<tr>
<td>HTD5</td>
<td>0.59</td>
</tr>
<tr>
<td>HTD8</td>
<td>—</td>
</tr>
<tr>
<td>HTD14</td>
<td>—</td>
</tr>
<tr>
<td>TT5</td>
<td>0.52</td>
</tr>
<tr>
<td>XL</td>
<td>0.34</td>
</tr>
<tr>
<td>L</td>
<td>0.59</td>
</tr>
<tr>
<td>H</td>
<td>0.75</td>
</tr>
<tr>
<td>XH</td>
<td>1.25</td>
</tr>
</tbody>
</table>

#### Stainless steel

<table>
<thead>
<tr>
<th>Type</th>
<th>Cord diameter [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>T5</td>
<td>—</td>
</tr>
<tr>
<td>T10</td>
<td>0.63</td>
</tr>
<tr>
<td>T20</td>
<td>0.90</td>
</tr>
<tr>
<td>AT5</td>
<td>—</td>
</tr>
<tr>
<td>AT10</td>
<td>0.90</td>
</tr>
<tr>
<td>AT20</td>
<td>—</td>
</tr>
<tr>
<td>HTD5</td>
<td>—</td>
</tr>
<tr>
<td>HTD8</td>
<td>0.9</td>
</tr>
<tr>
<td>HTD14</td>
<td>—</td>
</tr>
<tr>
<td>XL</td>
<td>—</td>
</tr>
<tr>
<td>L</td>
<td>—</td>
</tr>
<tr>
<td>H</td>
<td>0.63</td>
</tr>
<tr>
<td>XH</td>
<td>0.90</td>
</tr>
</tbody>
</table>

#### P Steel

<table>
<thead>
<tr>
<th>Type</th>
<th>Cord diameter [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>AT5P</td>
<td>0.63</td>
</tr>
<tr>
<td>AT10P</td>
<td>1.21</td>
</tr>
<tr>
<td>AT20P</td>
<td>1.60</td>
</tr>
</tbody>
</table>
Appendix

Chemical resistance

The data presented in the chart below is based on information provided by our raw-materials manufacturers and suppliers. The data is presented for ambient conditions at 20 °C and 70 °F. Please note that users still need to make qualification tests in order to ensure suitability for your application. For additional details, please contact your local Habasit representative.

Designation of chemical | Poly-urethane
--- | ---
Acetic acid | ❑
Acetone | ❑
Alkyl alcohol | ■
Acetyl chloride | ❑
Alkyl benzene | ❑
Alkyl chloride | ▼
Aluminum acetate | ❑
Aluminum chloride | ■
Aluminum nitrate | ▼
Ammonia anhydrous | ❑
Ammonia gas – cold | ■
Ammonia gas – hot | ❑
Ammonium chloride | ■
Ammonium hydroxide | ❑
Amyl acetate | ❑
Animal fat | ▼
Antifreeze | ❑
Antimony pentachloride | ❑
Argon | ■
Aromatic fuels | ❑
Aromatic hydrocarbons | ▼
Aromatic vinegar | ■
Baking soda | ■
Barium fluoride | ■
Barium nitrate | ■
Benzene | ❑
Bleach | ❑
Blood | ■
Boric acid | ■
Butadiene | ❑
Butyric acid | ❑
Calcium carbonate | ■
Calcium nitrate | ■
Calcium phosphate | ■
Calcium sulfate | ■
Carbon monoxide | ■
Carbonated beverages | ■
Carbonic acid | ■
Castor oil | ■
Chlorine water | ❑
Chloroethane | ▼
Chloroform | ■
Chromic acid | ■
Citric acid | ■
Coconut oil | ▼
Copper sulphate | ❑
Cottonseed oil | ■
Creosote | ▼
Degreasing agents | ■
Detergent | ■
Dichlorethylene | ▼

Designation of chemical | Poly-urethane
--- | ---
Dichloroethane | ▼
Diesel oil | ▼
Dimethyl formamide | ❑
Dry-cleaning fluids | ❑
Ethyl hexyl alcohol | ❑
Ethylene alcohol | ▼
Ethylene chloride | ❑
Ferric sulfate | ■
Fish oil | ■
Fluorine | ❑
Freon | ❑
Gallic acid | ■
Gasoline – premium | ■
Gelatin | ❑
Glue | ■
Glycerin | ❑
Honey | ▼
Hydrogen | ■
Hydrogen peroxide | ▼
Iodine | ❑
Isobutyl alcohol | ❑
Isopropanol | ■
Lactic acid | ■
Magnesium acetate | ■
Magnesium salts | ■
Mercury | ■
Methane | ▼
Methanol | ❑
Methyl butyl ketone | ❑
Methyl chloride | ■
Methyl ethyl ketone | ❑
Nicotine | ■
Nitrogen | ■
Nitric oxide | ■
Oleic acid | ■
Ozone | ■
Peanut oil | ■
Pectin | ■
Phosphoric acid | ▼
Pine oil | ■
Potassium acid sulfate | ■
Radiation | ▼
Salt | ■
Salt water | ❑
Silicone grease | ■
Silver nitrate | ■
Soap | ▼
Soybean oil | ■
Steam | ❑
Sugar cane liquor | ■
Tannic acid | ■

Designation of chemical | Poly-urethane
--- | ---
Toluene | ❑
Turpentine | ❑
Vegetable oils | ■
Vinegar | ❑
Vinyl acetate | ❑
Water – deionized | ■
Xylene | ❑
Zinc acetate | ■
Appendix
Evaluation of tooth and pitch

Pitch selection for T series belts

Open-ended belts
Belt width in [in]

Joined endless belts
Belt width in [in]

Pitch selection for AT series belts

Open-ended belts
Belt width in [in]

Joined endless belts
Belt width in [in]
Appendix
Evaluation of tooth and pitch

Pitch selection for belts of series with imperial pitches

Pitch selection for HTD series belts
## Appendix

### List of abbreviations

<table>
<thead>
<tr>
<th>Term</th>
<th>Symbol</th>
<th>Metric value</th>
<th>Imperial value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Acceleration</td>
<td>( a )</td>
<td>m/s(^2)</td>
<td>ft/s(^2)</td>
</tr>
<tr>
<td>Accuracy factor of belt</td>
<td>( a_f )</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Admissible tensile force</td>
<td>( F_{adm} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Angle of inclination</td>
<td>( \alpha )</td>
<td>°</td>
<td>°</td>
</tr>
<tr>
<td>Arc of contact on drive pulley</td>
<td>( \beta_a )</td>
<td>°</td>
<td>°</td>
</tr>
<tr>
<td>Arc of contact on pulley</td>
<td>( \beta )</td>
<td>°</td>
<td>°</td>
</tr>
<tr>
<td>Arc of contact on tensioning idler</td>
<td>( \beta_T )</td>
<td>°</td>
<td>°</td>
</tr>
<tr>
<td>Backlash due to pulley groove clearance</td>
<td>( \Delta x_3 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Belt elongation due to elasticity of belt</td>
<td>( \Delta x_1 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Belt elongation due to peripheral force</td>
<td>( \varepsilon_U )</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Belt elongation in the slack belt strand</td>
<td>( \varepsilon_2 )</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Belt elongation in the tight belt strand</td>
<td>( \varepsilon_1 )</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Belt length</td>
<td>( l_0 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Belt pitch</td>
<td>( P_0 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Belt speed</td>
<td>( v )</td>
<td>m/s</td>
<td>ft/s</td>
</tr>
<tr>
<td>Belt width</td>
<td>( b_0 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Center-to-center distance</td>
<td>( e )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Coefficient of friction belt/slider bed</td>
<td>( \mu_G )</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Conveying length</td>
<td>( l_T )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Deformation factor</td>
<td>( df )</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Deformation of teeth in mesh</td>
<td>( \Delta x_2 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Dynamic shaft load of drive pulley</td>
<td>( F_{WA4} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Dynamic shaft load of tail pulley</td>
<td>( F_{WA4} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Efficiency of drive (gearbox, etc.)</td>
<td>( \eta )</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Elevating height</td>
<td>( h_T )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Externally applied working load</td>
<td>( F_E )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Friction force of linear bearing</td>
<td>( F_T )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Highest admissible operating temperature (continuous)</td>
<td>( T_{max} )</td>
<td>°C</td>
<td>°F</td>
</tr>
<tr>
<td>Highest possible variation of force on positioned slider</td>
<td>( \Delta F )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Initial belt extension</td>
<td>( \varepsilon_0 )</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Length of slack belt strand</td>
<td>( l_2 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Length of tight belt strand</td>
<td>( l_1 )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Lowest admissible operating temperature (continuous)</td>
<td>( T_{min} )</td>
<td>°C</td>
<td>°F</td>
</tr>
<tr>
<td>Mass of belt carried over the slider bed</td>
<td>( m_b )</td>
<td>kg</td>
<td>lb</td>
</tr>
<tr>
<td>Mass of belt per meter (weight of belt/m; weight of belt/ft)</td>
<td>( m' )</td>
<td>kg/m</td>
<td>lb/ft</td>
</tr>
<tr>
<td>Mass of carried goods on total conveying length</td>
<td>( m )</td>
<td>kg</td>
<td>lb</td>
</tr>
<tr>
<td>Mass of slider plus load on slider</td>
<td>( m_s )</td>
<td>kg</td>
<td>lb</td>
</tr>
<tr>
<td>Maximal covered distance of linear drive</td>
<td>( l_T )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Mechanical power on drive pulley</td>
<td>( P )</td>
<td>kW</td>
<td>HP</td>
</tr>
<tr>
<td>Minimum required belt width</td>
<td>( b_{req} )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Number of belt teeth</td>
<td>( z_b )</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Number of pulley teeth</td>
<td>( z_p )</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Number of pulley teeth of drive pulley</td>
<td>( z_a )</td>
<td>–</td>
<td>–</td>
</tr>
</tbody>
</table>
### List of abbreviations

<table>
<thead>
<tr>
<th>Term</th>
<th>Symbol</th>
<th>Metric value</th>
<th>Imperial value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peripheral force component due to friction on slider bed</td>
<td>( F_{US} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Peripheral force component due to mass acceleration</td>
<td>( F_{Ua} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Peripheral force component due to mass elevation</td>
<td>( F_{Ub} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Peripheral force component due to other factors</td>
<td>( F_{Uau} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Peripheral force on drive pulley</td>
<td>( F_{U} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Pitch diameter (effective diameter) of drive pulley</td>
<td>( d_a )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Pitch diameter (effective diameter) of pulley</td>
<td>d</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Positioning error (absolute)</td>
<td>( \Delta x )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Positioning error (relative)</td>
<td>x</td>
<td>%</td>
<td>%</td>
</tr>
<tr>
<td>Pressure force of slack-side tensioning idler</td>
<td>( F_{WT} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Random positioning error</td>
<td>( \Delta x_R )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Required motor power, motor output</td>
<td>( P_M )</td>
<td>kW</td>
<td>HP</td>
</tr>
<tr>
<td>Shaft load</td>
<td>( F_W )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Speed difference (final speed minus initial speed)</td>
<td>( \Delta v )</td>
<td>m/s</td>
<td>ft/s</td>
</tr>
<tr>
<td>Static shaft load of drive pulley</td>
<td>( F_{Wa_s} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Static shaft load of tail pulley</td>
<td>( F_{Wa_s} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Systematic positioning error</td>
<td>( \Delta x_S )</td>
<td>mm</td>
<td>inch</td>
</tr>
<tr>
<td>Teeth in mesh</td>
<td>( z_m )</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Tensile force due to initial belt extension</td>
<td>( F_0 )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Tensile force for 1% elongation</td>
<td>( k_{1%} )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Tensile force in the slack belt strand</td>
<td>( F_2 )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Tensile force in the tight belt strand</td>
<td>( F_1 )</td>
<td>N</td>
<td>lb</td>
</tr>
<tr>
<td>Time required to accelerate up to speed</td>
<td>t</td>
<td>s</td>
<td>s</td>
</tr>
<tr>
<td>Tooth-in-mesh factor</td>
<td>( t_m )</td>
<td>–</td>
<td>–</td>
</tr>
<tr>
<td>Total mass to be carried over the slider bed</td>
<td>( m_{tot} )</td>
<td>kg</td>
<td>lb</td>
</tr>
</tbody>
</table>
## Appendix

### Conversion of units

<table>
<thead>
<tr>
<th>Metric units</th>
<th>Factor to convert to imperial units</th>
<th>Factor to convert to metric units</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Length</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>mm</td>
<td>0.0394 in (inch)</td>
<td>25.4 mm</td>
</tr>
<tr>
<td>m</td>
<td>3.281 ft (foot)</td>
<td>0.3048 m</td>
</tr>
<tr>
<td><strong>Area</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>mm²</td>
<td>0.00155 in² (square inch)</td>
<td>645.2 mm²</td>
</tr>
<tr>
<td>m²</td>
<td>10.764 ft² (square foot)</td>
<td>0.0929 m²</td>
</tr>
<tr>
<td><strong>Speed</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>m/s</td>
<td>3.281 ft/s (foot/second)</td>
<td>0.3048 m/s</td>
</tr>
<tr>
<td>m/min</td>
<td>3.281 ft/min (foot/minute)</td>
<td>0.3048 m/min</td>
</tr>
<tr>
<td><strong>Mass</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>kg</td>
<td>2.205 lb (pound weight)</td>
<td>0.4536 kg</td>
</tr>
<tr>
<td>kg/m</td>
<td>0.672 lb/ft (pound/foot)</td>
<td>1.4882 kg/m</td>
</tr>
<tr>
<td>kg/m²</td>
<td>0.205 lb/ft² (pound/square foot)</td>
<td>4.882 kg/m²</td>
</tr>
<tr>
<td><strong>Force and strength</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>N</td>
<td>0.225 lb (pound force)</td>
<td>4.448 N</td>
</tr>
<tr>
<td>N/mm</td>
<td>5.7102 lb/in (pound/inch)</td>
<td>0.17513 N/mm</td>
</tr>
<tr>
<td>N/m</td>
<td>0.0685 lb/ft (pound/foot)</td>
<td>14.6 N/m</td>
</tr>
<tr>
<td><strong>Power</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>kW</td>
<td>1.341 hp (horsepower)</td>
<td>0.7457 kW</td>
</tr>
<tr>
<td><strong>Torque</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Nm</td>
<td>8.85 in-lb (inch pound)</td>
<td>0.113 Nm</td>
</tr>
<tr>
<td><strong>Temperature</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>°C</td>
<td>9 · (°C / 5) + 32° °F</td>
<td>5/9 · (°F – 32°) °C</td>
</tr>
</tbody>
</table>
## Glossary of terms

<table>
<thead>
<tr>
<th>Term</th>
<th>Explanation</th>
<th>Habasit symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Accessories</td>
<td>Objects or devices commonly used for belt applications (e.g. guide strips, pulleys, tensioners, belt clamps, etc.)</td>
<td></td>
</tr>
<tr>
<td>Admissible tensile force</td>
<td>Admissible belt tensile force allowed in the tightest belt section under process conditions</td>
<td>F&lt;sub&gt;adm&lt;/sub&gt;</td>
</tr>
<tr>
<td>Admissible tensile force, joined belt</td>
<td>Admissible belt tensile force allowed in the tightest belt section under process conditions for joined belts (only valid for master joint)</td>
<td>F&lt;sub&gt;adm&lt;/sub&gt; joined endless</td>
</tr>
<tr>
<td>Admissible tensile force, open belt</td>
<td>Admissible belt tensile force allowed in the tightest belt section under process conditions for unjoined belts (or for belts where the joint is never under load)</td>
<td>F&lt;sub&gt;adm&lt;/sub&gt; open ended</td>
</tr>
<tr>
<td>Aramide</td>
<td>High modulus synthetic fiber (Kevlar, Technora, Twaron)</td>
<td></td>
</tr>
<tr>
<td>Balanced cords</td>
<td>The cord twist of the tensile member alternates from cord to cord (S twist/Z twist/S twist and so on)</td>
<td></td>
</tr>
<tr>
<td>Belt length</td>
<td>Length of belt measured along the neutral layer (length of traction member)</td>
<td>I&lt;sub&gt;0&lt;/sub&gt;</td>
</tr>
<tr>
<td>Belt options</td>
<td>Nonstandard surfaces, materials, colors, etc.</td>
<td></td>
</tr>
<tr>
<td>Belt pitch</td>
<td>Distance from the center of a tooth to the center of the next tooth</td>
<td>P&lt;sub&gt;b&lt;/sub&gt;</td>
</tr>
<tr>
<td>Belt width</td>
<td>Geometric width of the belt from edge to edge</td>
<td>P&lt;sub&gt;0&lt;/sub&gt;</td>
</tr>
<tr>
<td>Bidirectional drive</td>
<td>Driving concept allowing to run the belt forward and backward</td>
<td></td>
</tr>
<tr>
<td>Center distance</td>
<td>Linear distance between two pulleys</td>
<td></td>
</tr>
<tr>
<td>Center drive</td>
<td>Position of the drive that provides the same length of tight and slack belt strands (under process conditions). Preferred design for a bidirectional belt run</td>
<td></td>
</tr>
<tr>
<td>Coefficient of friction</td>
<td>Ratio of frictional force and contact force acting between two material surfaces</td>
<td>µ</td>
</tr>
<tr>
<td>Conveying length</td>
<td>Conveying length measured between the centers of the head and tail pulleys</td>
<td>I&lt;sub&gt;T&lt;/sub&gt;</td>
</tr>
<tr>
<td>Conveying side</td>
<td>Opposite side to toothed belt side (belt side which commonly carries the conveyed goods)</td>
<td></td>
</tr>
<tr>
<td>Conveying side cover</td>
<td>Cover material (surface material) on the conveying side</td>
<td></td>
</tr>
<tr>
<td>Cord</td>
<td>Tensile member</td>
<td></td>
</tr>
<tr>
<td>Counter flection</td>
<td>Belt is bent over pulley(s) on the conveying side, e.g. when tension pulley is used</td>
<td></td>
</tr>
<tr>
<td>Cover</td>
<td>Cover material (surface material) on the conveying side</td>
<td></td>
</tr>
<tr>
<td>Elastomer</td>
<td>Comparatively soft synthetic material like rubber (thermoset elastomer) or thermoplastic polyurethane (thermoplastic elastomer)</td>
<td></td>
</tr>
<tr>
<td>Family</td>
<td>T, AT, and HTD are the metric pitch families; L, XL, H, and XH belts are the imperial pitch family</td>
<td></td>
</tr>
<tr>
<td>FDA</td>
<td>Food and Drug Administration is a federal agency of the US which regulates materials that may come into contact with food, but FDA is also the synonym for the regulation itself</td>
<td>FDA</td>
</tr>
<tr>
<td>Flex belt</td>
<td>Truly endless timing belt made of TPU extruded over helically wound cords</td>
<td></td>
</tr>
<tr>
<td>Flight</td>
<td>Small groove in the tooth root required for cord positioning in the belt production process</td>
<td></td>
</tr>
<tr>
<td>Head drive</td>
<td>Driven head pulley (preferred design)</td>
<td></td>
</tr>
<tr>
<td>Head pulley</td>
<td>Pulley at the end of the conveyor (refers to the belt running direction)</td>
<td></td>
</tr>
<tr>
<td>Height of belt</td>
<td>Overall thickness of the timing belt</td>
<td>h&lt;sub&gt;s&lt;/sub&gt;</td>
</tr>
<tr>
<td>HTD</td>
<td>Curvilinear shape tooth design, where the taller tooth with round bottom shape allows for belt/pulley mesh far from the pitch line, thus producing minimal noise and vibration</td>
<td></td>
</tr>
<tr>
<td>High Torque Drive</td>
<td>Indexing</td>
<td></td>
</tr>
<tr>
<td>Indexing</td>
<td>Feeding or conveying of goods synchronously with the beat of a process. Indexing conveyors run often in stop-and-go mode</td>
<td></td>
</tr>
<tr>
<td>Joined endless</td>
<td>Joined endless belt</td>
<td>J</td>
</tr>
<tr>
<td>Joining code</td>
<td>A code which describes the preparation of belt ends for ordered belts (open ended, prepared ends or joined endless)</td>
<td></td>
</tr>
<tr>
<td>Linear positioning</td>
<td>Linear drives (actuators) which accurately position a mass or which precisely move along a predefined curve</td>
<td></td>
</tr>
<tr>
<td>Mass of belt</td>
<td>Belt weight in kg per m; or in lb per ft</td>
<td></td>
</tr>
</tbody>
</table>
## Glossary of terms

<table>
<thead>
<tr>
<th>Term</th>
<th>Explanation</th>
<th>Habasit symbol</th>
</tr>
</thead>
<tbody>
<tr>
<td>Minimum clamping length</td>
<td>In applications where belt ends are clamped, a minimum clamping length must be considered to prevent the belt being torn out of the clamp</td>
<td></td>
</tr>
<tr>
<td>Minimum length of joined belt</td>
<td>Minimum belt length which can be joined (according to tools used)</td>
<td></td>
</tr>
<tr>
<td>Minimum number of teeth</td>
<td>Minimum number of teeth of the smallest timing belt pulley</td>
<td></td>
</tr>
<tr>
<td>Minimum number of teeth of joined belt</td>
<td>The minimum belt length which can be joined defines the respective minimum number of teeth</td>
<td></td>
</tr>
<tr>
<td>Minimum pulley diameter</td>
<td>Minimum diameter of smallest flat pulley</td>
<td>( d_{\text{min}} )</td>
</tr>
<tr>
<td>Modified trapezoid tooth shape</td>
<td>Wider trapezoid tooth with a 50° tooth angle</td>
<td></td>
</tr>
<tr>
<td>Open ended</td>
<td>Belt ends are just cut and not prepared for joining</td>
<td>O</td>
</tr>
<tr>
<td>Option, belt option</td>
<td>Nonstandard surfaces, materials, colors, etc.</td>
<td></td>
</tr>
<tr>
<td>Outside pulley diameter</td>
<td>Diameter of a timing belt pulley measured over the tips of teeth</td>
<td>( d_o )</td>
</tr>
<tr>
<td>Pitch diameter</td>
<td>Effective diameter of a timing belt pulley which defines the position of the tensile member (cords) of the belt</td>
<td>( \bar{d} )</td>
</tr>
<tr>
<td>Pitch line</td>
<td>Neutral layer of the belt (line that stays the same length when the belt is bent). The cords lie exactly in the pitch line</td>
<td>PL</td>
</tr>
<tr>
<td>Polyamide fabric facing on both sides</td>
<td>Both surfaces (tooth side and conveying side) are coated with a wear-resistant polyamide fabric with a low coefficient of friction</td>
<td>PTC</td>
</tr>
<tr>
<td>Polyamide fabric facing on conveying side</td>
<td>Conveying side surface is coated with a wear-resistant polyamide fabric with a low coefficient of friction</td>
<td>PC</td>
</tr>
<tr>
<td>Polyamide fabric facing on tooth side</td>
<td>Tooth side surface is coated with a wear-resistant polyamide fabric with a low coefficient of friction</td>
<td>PT</td>
</tr>
<tr>
<td>Polygon effect</td>
<td>Pulsation of the belt velocity caused by the polygon shape of the driving pulley, with rise and fall of the belt surface</td>
<td></td>
</tr>
<tr>
<td>Prepared ends</td>
<td>Open-ended belt with prepared belt ends for joining</td>
<td>P</td>
</tr>
<tr>
<td>Required take-up</td>
<td>Length of take-up device required to realize the initial belt extension</td>
<td>( x_c )</td>
</tr>
<tr>
<td>Series</td>
<td>Group of belts according to standardized timing belt geometries (T5, T10, T20, L, XL, etc.)</td>
<td></td>
</tr>
<tr>
<td>Slider bed</td>
<td>Belt support plate to carry the running belt with low friction and wear</td>
<td></td>
</tr>
<tr>
<td>Standard color of elastomer</td>
<td>The color of elastomers is standardized in order to indicate special belt options (suitable for food applications, aramide cords, etc.)</td>
<td></td>
</tr>
<tr>
<td>Tail drive</td>
<td>Driven tail pulley (should be avoided whenever possible)</td>
<td></td>
</tr>
<tr>
<td>Tail pulley</td>
<td>Pulley at the beginning of the conveyor (refers to belt running direction)</td>
<td></td>
</tr>
<tr>
<td>Take-up</td>
<td>Tensioning device for adjustment of belt tensile force. Screw type, gravity type or spring-loaded type</td>
<td></td>
</tr>
<tr>
<td>Tensile force for 1% elongation</td>
<td>Force which theoretically would be required for 1% belt extension. It describes the stress-strain behavior of the timing belt and must not be confused with &quot;admissible elongation.&quot;</td>
<td>( k_{1%} )</td>
</tr>
<tr>
<td>Tensile member</td>
<td>High modulus layer (steel cords, aramide cords, etc.) responsible for longitudinal belt strength</td>
<td></td>
</tr>
<tr>
<td>Timing belt</td>
<td>Synchronous belt as described in ISO 5296</td>
<td></td>
</tr>
<tr>
<td>Timing belt pulleys</td>
<td>Toothed pulleys for synchronous belt drives as described in ISO 5294</td>
<td></td>
</tr>
<tr>
<td>Tooth side</td>
<td>Toothed belt side (opposite the belt side which commonly supports the conveyed goods)</td>
<td></td>
</tr>
<tr>
<td>Trapezoid tooth shape</td>
<td>Trapezoid timing belt tooth with a 40° tooth angle according to DIN7721</td>
<td>E</td>
</tr>
<tr>
<td>Truly endless</td>
<td>Endless-produced belts (no joint)</td>
<td>U</td>
</tr>
<tr>
<td>Unprocessed</td>
<td>Belt produced without belt options such as fabric facings, etc.</td>
<td></td>
</tr>
<tr>
<td>Without counter flexion</td>
<td>Belt is only bent over pulleys on tooth side</td>
<td></td>
</tr>
</tbody>
</table>
The Habasit solution

Customers first
At Habasit we understand that our success depends on your success. This is why we offer solutions, not just products; partnership, not just sales.

Since our foundation in 1946, Habasit has brought this understanding of customer needs to life every day and for every application. That’s why we’re the No. 1 in belting today. Worldwide. Learn more on www.habasit.com

Committed to innovation
Because our customers’ belting challenges and needs never cease, we consistently dedicate a substantial percentage of our employees and resources to the research and development of new products and solutions.

Certified for quality
We deliver the highest quality standards not only in our products and solutions, but also in our employees’ daily work processes. Habasit AG is certified according to ISO 9001:2008.

Worldwide leading product range
Habasit offers the largest selection of belting, conveying, processing and complementary products in the industry. Our response to any request is nothing less than a specific, tailor-made solution.

Fabric-based conveyor and processing belts
HabaFLOW®

Plastic modular belts
HabasitLINK®/KVP®

Positive drive conveyor and processing belts
Habasit Cleandrive™

Power transmission belts
HabaDRIVE®

Timing belts
HabaSYNC®

Chains (slat and conveyor chains)
HabaCHAIN®

Machine tapes

Round belts

Seamless belts

Profiles, Guides, Wear strips
HabiPLAST®

Fabrication tools (joining tools)

Gearmotors
Electric motors
Motion control

Certified for quality

Committed to innovation

Customers first

Worldwide leading product range