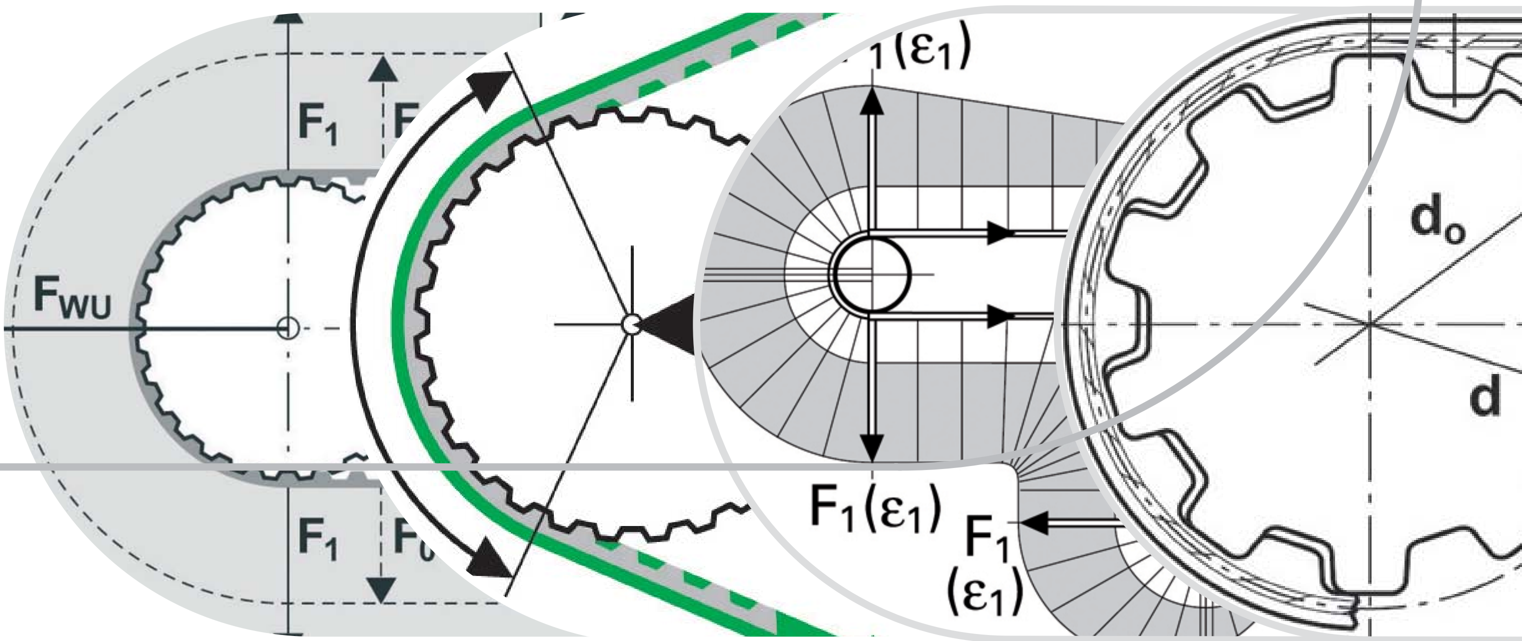
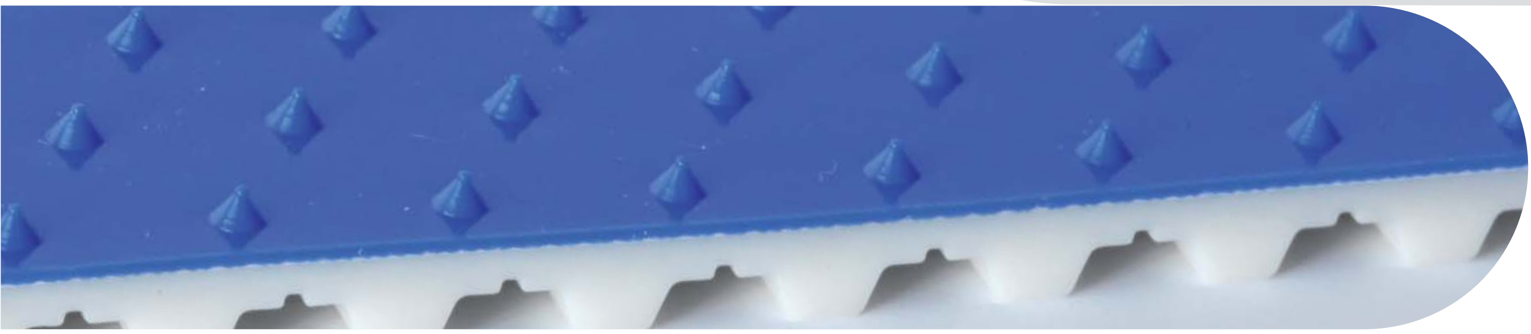
Services
Media No. 6018

HabaSYNC®
Timing Belts

Engineering Guide

Habasit – Solutions in motion





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Introduction

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.

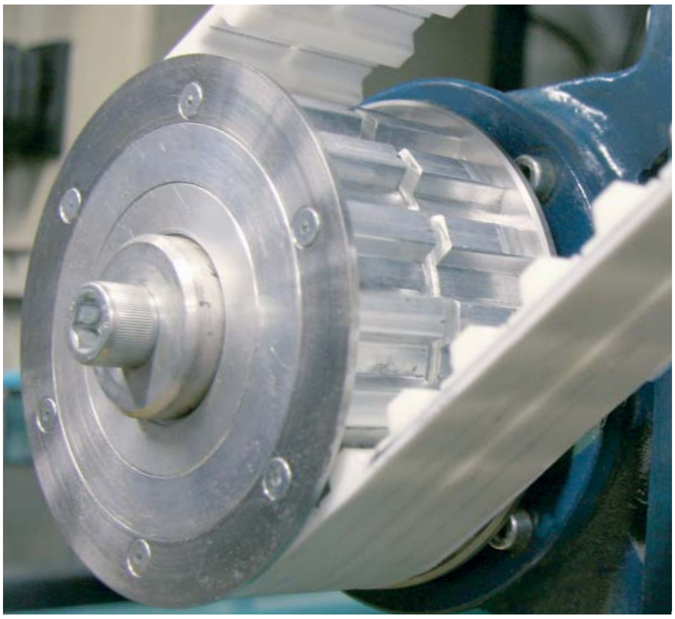
Habasit is your partner
With more than 60 years of belting experience in elastomers and textiles, Habasit and its more than 3,300 well-trained and committed employees are situated at strategic locations worldwide to be close to you. Our sales and production activities are supported by a dedicated research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

Features of HabaSYNC® timing belts

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.

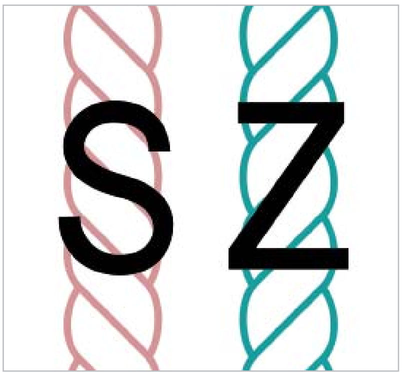
Where applications require exceptional hydrolysis resistance, we manufacture HabaSYNC® timing belts using a **polyether** polyurethane. Our standard polyether material is cobalt blue with a Shore A value of 90.

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

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.



Timing belt range

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.

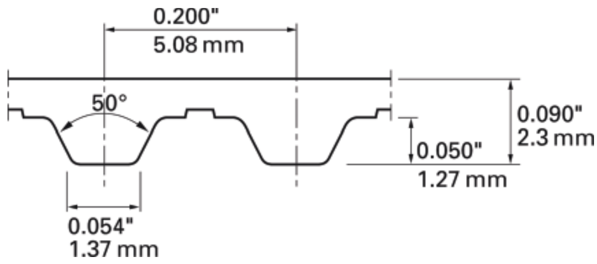
Timing belt teeth are generally formed in either a trapezoid or curvilinear design. Both tooth designs will yield good results in general conveying applications. The trapezoidal-shaped (T) timing belts are the common choice for standard conveying tasks and in cases with "counter flection" due to their backbending properties. The modified trapezoid AT series is used in bidirectional and critical product-positioning applications where zero backlash (in combination with respectively toleranced 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.

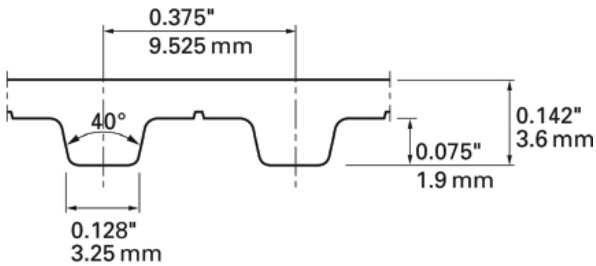
Polyamide facings are available on either the tooth side, conveying side, or on both sides.

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

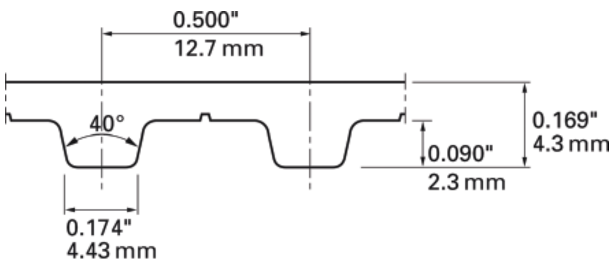
Imperial pitch belts can only be used with the respective imperial pitch timing belt pulleys.



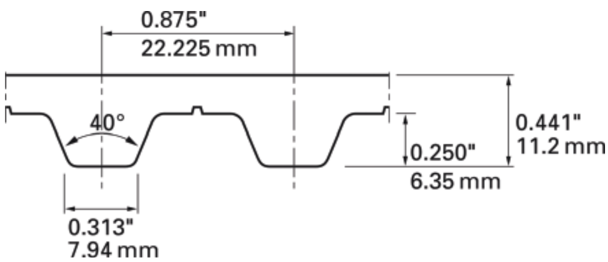
XL



L



H



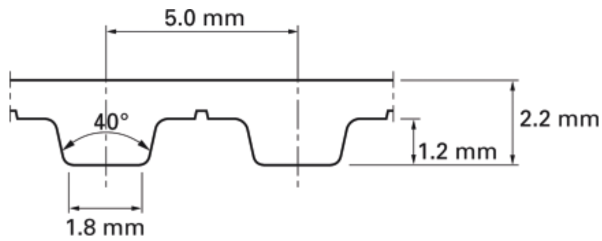
XH

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.

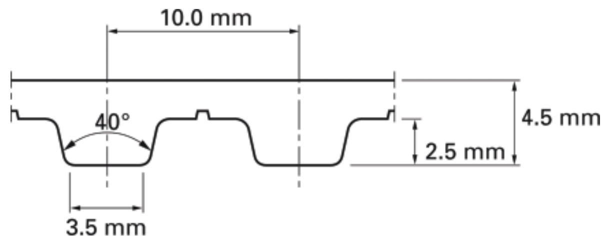
Polyamide facings are available on either the tooth side, conveying side, or on both sides.

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

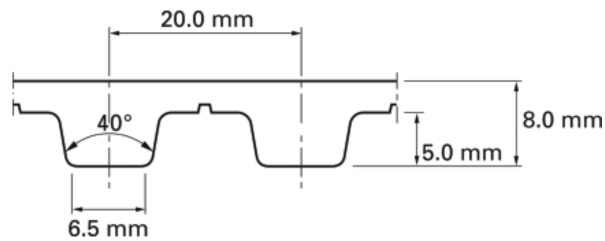
Metric pitch belts can only be run with standard metric pitch timing belt pulleys.



T5



T10



T20

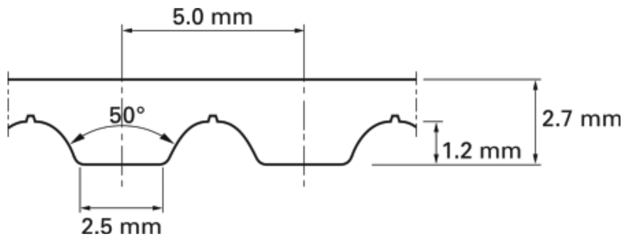
Timing belt range

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.

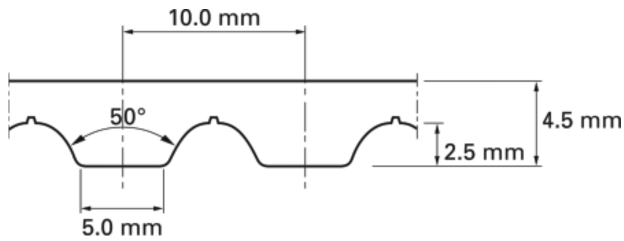
Polyamide facings are available on either the tooth side, conveying side, or on both sides.

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

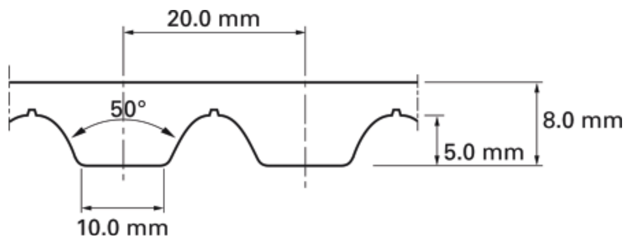
AT metric pitch belts can only be run with AT metric pitch timing belt pulleys.



AT5



AT10



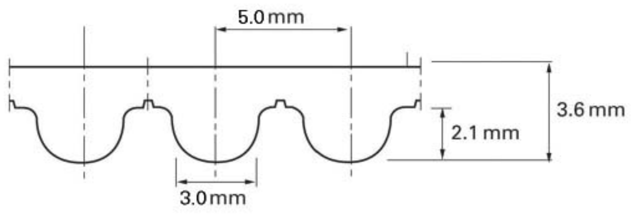
AT20

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

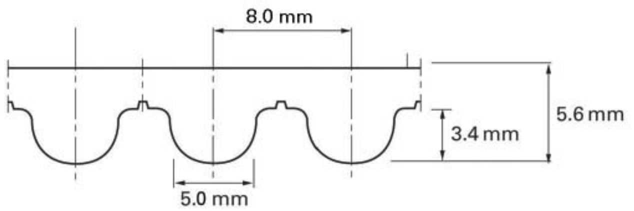
Polyamide facings are available on either the tooth side, conveying side, or on both sides.

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

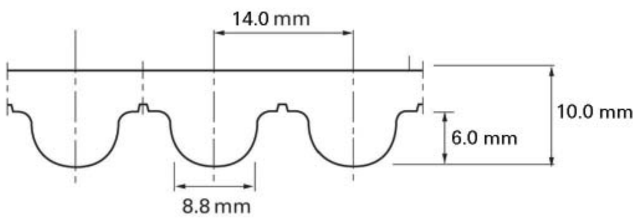
Metric pitch HTD belts can only be run with metric pitch HTD timing belt pulleys.



HTD5



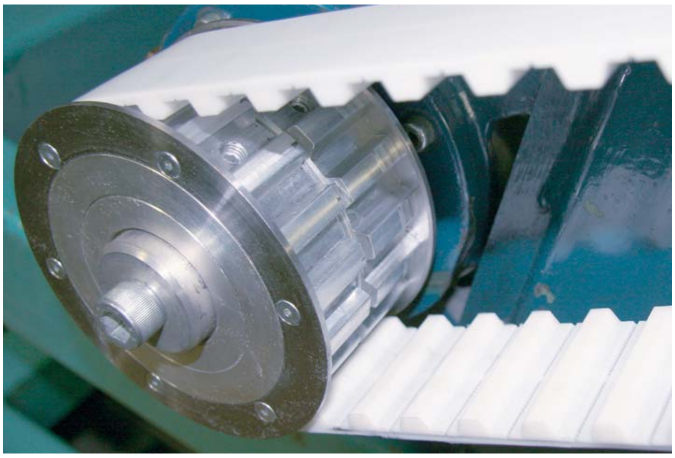
HTD8



HTD14

Timing belt nomenclature

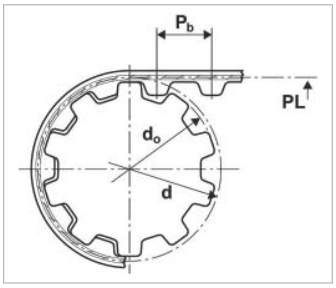
HabaSYNC® timing belts are made out of several key component parts. Each must complement the other precisely in order to provide a highly effective synchronous-drive solution.



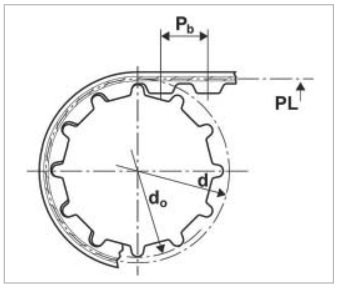
Teeth

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.



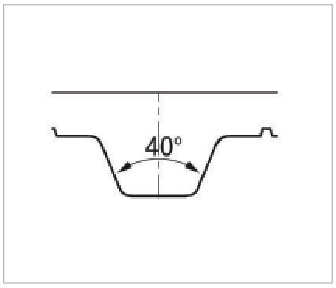
Trapezoid



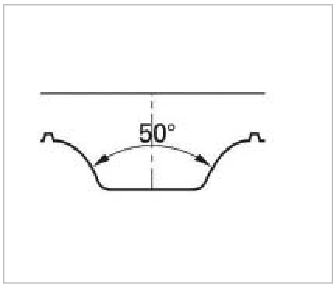
Modified trapezoid

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.



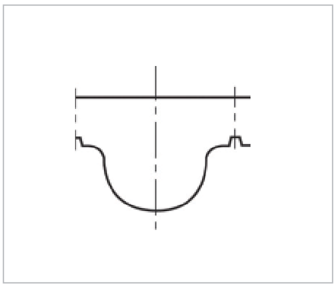
Metric T and imperial L, H, and XH types



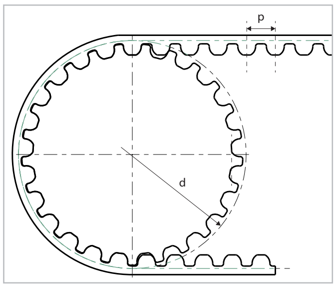
Metric AT and imperial XL types

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.



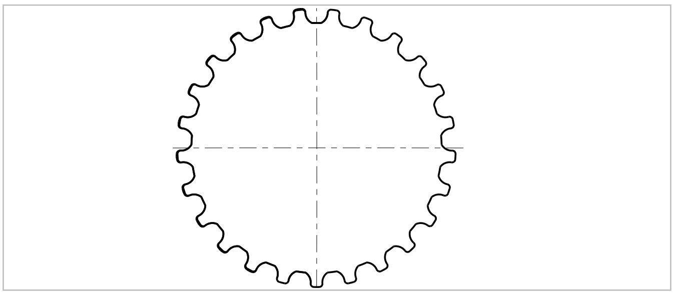
HTD type



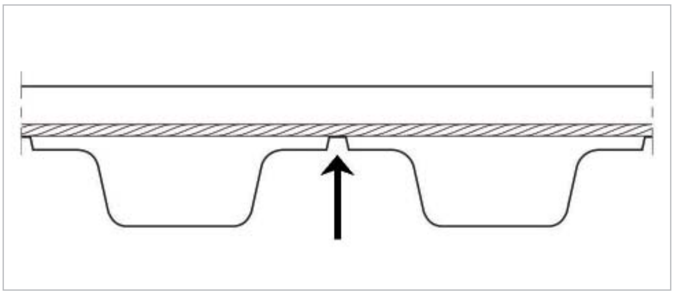
Curvilinear shape

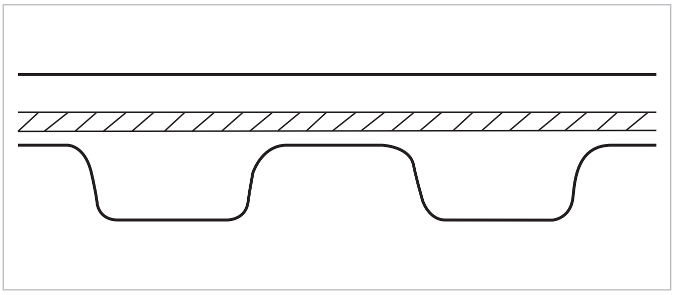
Timing belt nomenclature

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.

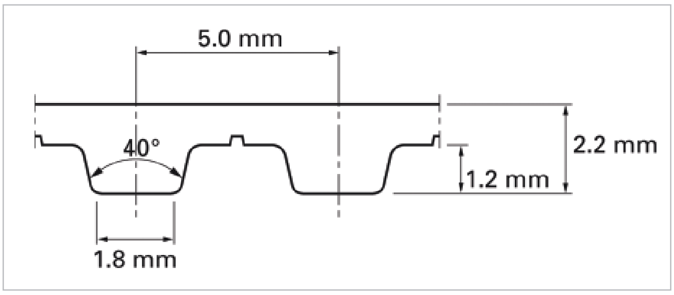


Flight
The flight is the noticeable line between the teeth. It is created as a result of a machined point on the forming wheel that is designed to locate cord placement. This critical position for cord resting ensures that the belt will mesh smoothly. It yields low drive noise and delivers vibration-free 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.

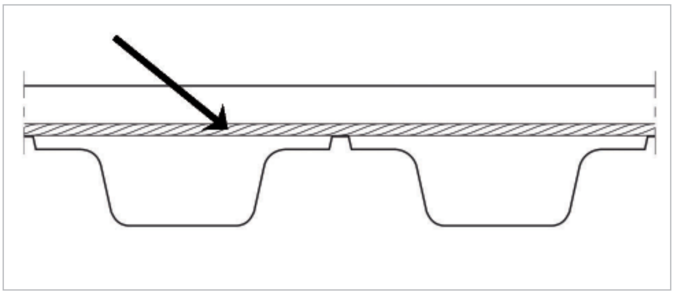




Tooth pitch
The tooth pitch is the accurate measurement of the distance from the vertical centerline of one tooth to the vertical centerline of the next tooth. Metric pitch belts are measured in millimeters, imperial pitch belts are measured in inches.



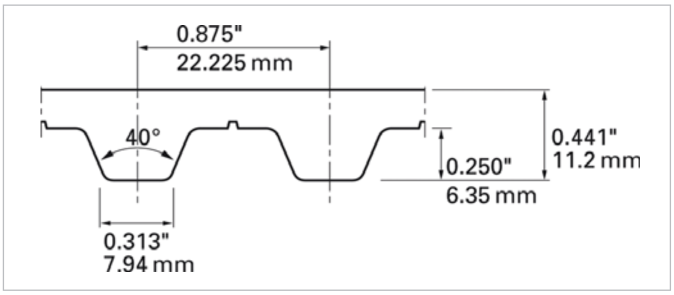
Pitch line
The pitch line is the centerline of the cord measured around the entire belt length. The measurement of the cord around the entire belt is the result of the cord resting on each flight as the belt is made. The belt length is calculated from the pitch line (= neutral layer).



Timing belt nomenclature

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.

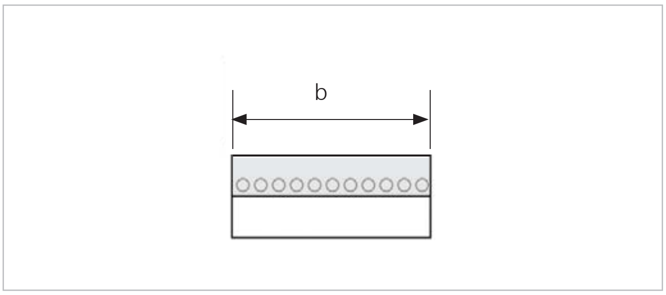
Imperial pitch belts are typically measured in inches; metric pitch belts are measured in millimeters.

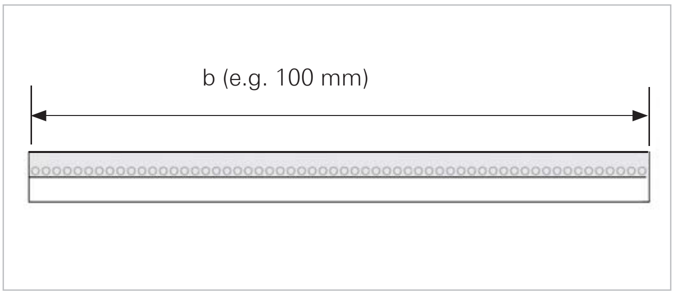


Belt width
This is the actual measured dimension of the belt width.

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.

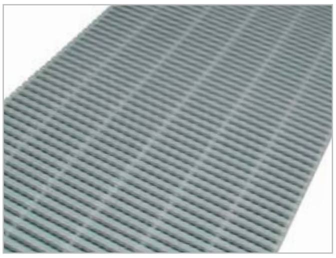
Imperial timing belts are measured in inches and are noted to 3 digits. For example “200” is a *2.00 inch* belt width and “075” is a *0.75 (3/4) inch* belt width.

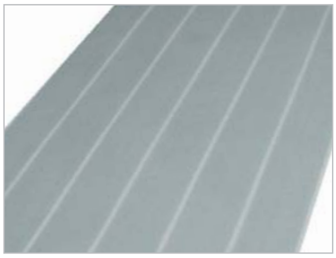




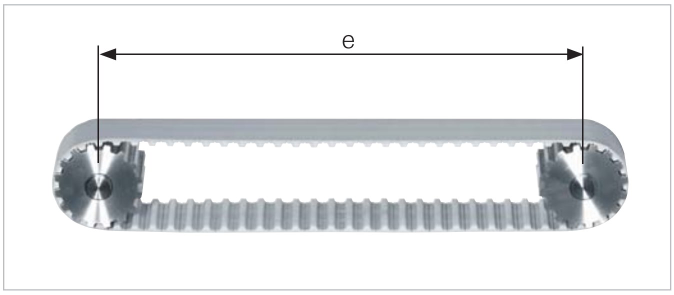
Slitting lane
Habasit offers HabaSYNC® timing belts in all industry standard widths.

Special widths are available upon request.

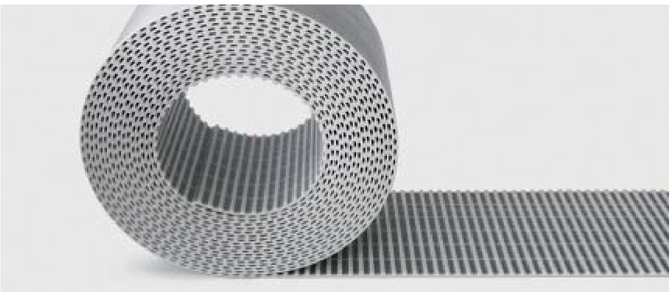


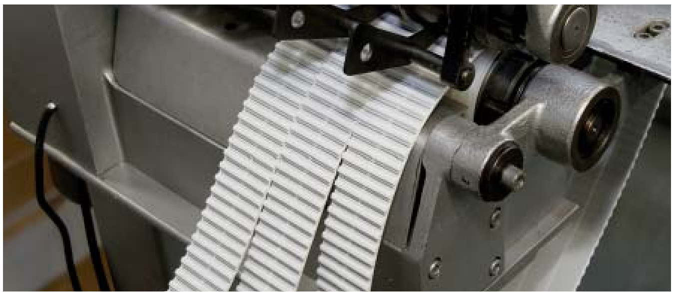


Center distance
The center distance e of a two-pulley belt drive is measured from the center of one pulley to the center of the next.



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

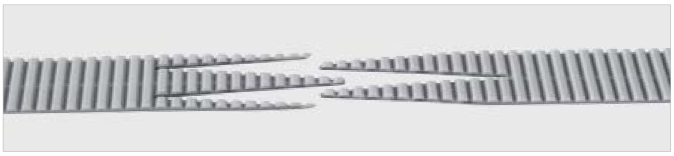
Joining by heat
Synthetic belts with thermoplastic matrix material like HabaSYNC® can be cut to length, then joined together by a hot-welding process. The TPU in the joining area is melted and cooled again to create an endless belt.

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.

Joining methods

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.

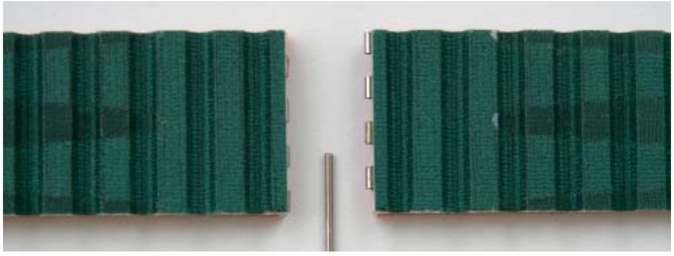


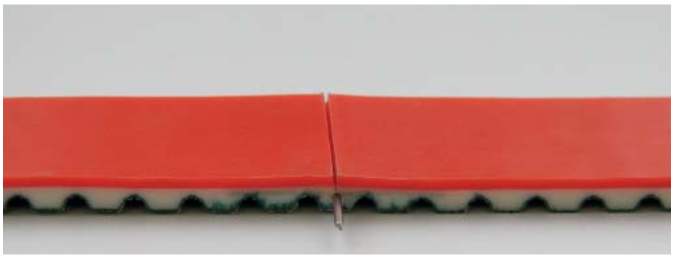
Step 2: Interlocking the fingers into joining plates
After fingers have been cut into both ends of the belt, the belt ends are interlocked into a HabaSYNC® fixed width joining plate.

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.

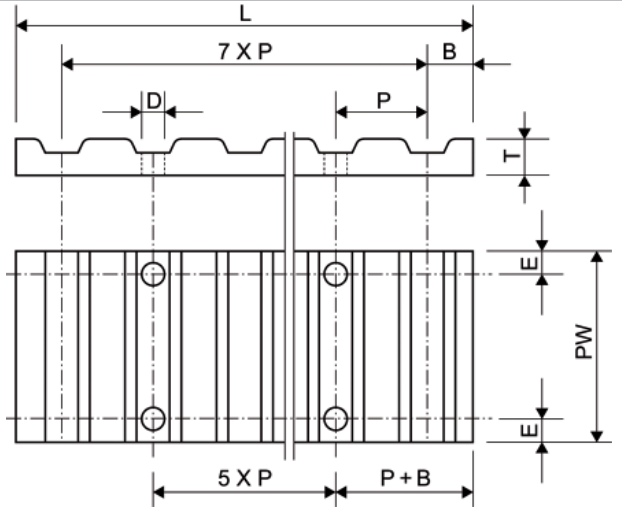
The stainless steel pivot parts are fully embedded within the cut-to-length belt ends. These are just brought together and a metal pin interlocks the hinges. And the belt is ready to run. Therefore, neither machine disassembling nor cumbersome joining procedures are required to install spare timing belts.

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:

Belt types applicable	T10, AT10, H and HTD8
Possible belt widths	16, 25, 32, 50, 75, 100 and 150 mm (other widths are available on request)
Minimum belt length	900 mm
Pulley diameter with counter flection	120 mm
Pulley diameter without counter flection	100 mm
Maximum tensile strength	750 N / 25 mm belt width

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.



Sketch of basic shape

Clamping plates

Pitch (P)		E (in)	D (in)	B (in)	L (in)	T (in)
XL	*(0.200")*	*0.24*	*0.22*	*0.14*	*1.67*	*0.31*
L	*(0.375")*	*0.31*	*0.35*	*0.2*	*3.02*	*0.59*
H	*(0.500")*	*0.39*	*0.43*	*0.35*	*4.21*	*0.87*

Pitch (P)		E (mm)	D (mm)	B (mm)	L (mm)	T (mm)
T5	(5 mm)	6.0	5.5	3.2	41.4	8.0
T10	(10 mm)	8.0	9.0	5.0	80.0	15.0
T20	(20 mm)	10.0	11.0	10.0	160.0	20.0
AT5	(5 mm)	6.0	5.5	3.2	41.4	8.0
AT10	(10 mm)	8.0	9.0	5.0	80.0	15.0
AT20	(20 mm)	10.0	11.0	10.0	160.0	20.0
HTD5	(5 mm)	6.0	5.5	3.2	41.4	8.0
HTD8	(8 mm)	8.0	9.0	5.0	66.0	15.0
HTD14	(14 mm)	10.0	11.0	9.0	116.0	22.0

Plate widths (PW)

Belt width in inches	*0.375*	*0.500*	*0.750*	*1.000*	*1.500*	*2.000*	*3.000*	*4.000*
XL	*1.12*	–	–	–	–	–	–	–
L	–	*1.54*	*1.77*	*2.03*	*2.52*	*3.03*	–	–
H	–	*1.77*	*2.00*	*2.26*	*2.75*	*3.26*	*4.25*	*5.27*

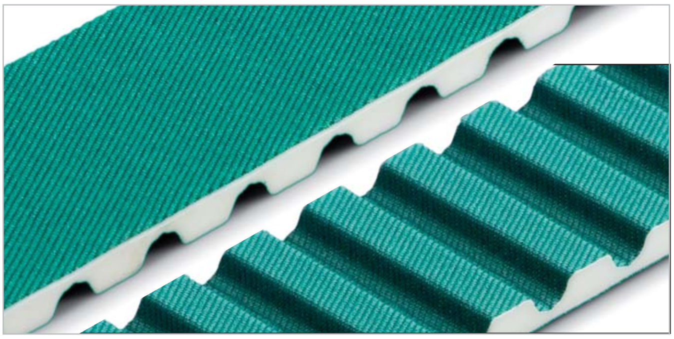
Belt width in mm	16	25	32	50	75	100
T5	35	44	51	–	–	–
T10	41	50	57	75	100	125
T20	–	56	63	81	106	132
AT5	35	44	51	–	–	–
AT10	41	50	57	75	100	125
AT20	–	56	63	81	106	132

Belt width in mm	10	15	20	25	30	50	55
HTD5	–	34	–	44	–	–	–
HTD8	35	40	45	50	55	75	–
HTD14	–	–	–	56	–	–	86

Belt surfaces

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.

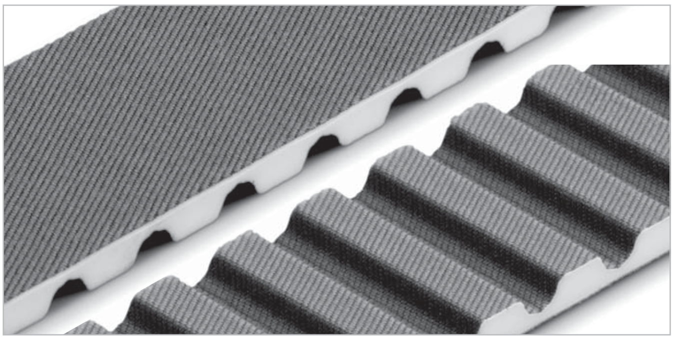
Fabric facings
Polyamide
Polyamide fabric added to the tooth side (PT), the conveying side (PC) or both the tooth and conveying side (PTC) reduces frictional drag of the belt and provides the benefit of lower noise. Polyamide fabric on the tooth side of a timing belt reduces the energy consumption by 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

Urethane vs. pickled steel	0.5 to 0.7	Polyamide vs. pickled steel	0.2 to 0.4
Urethane vs. PE-UHMW	0.3 to 0.5	Polyamide vs. PE-UHMW	0.1 to 0.3

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.

Belt surfaces

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

Mostly, these are combinations of:

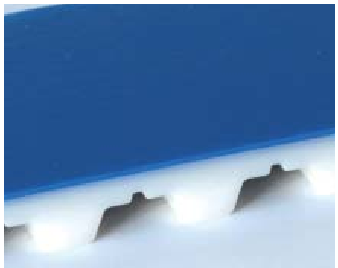
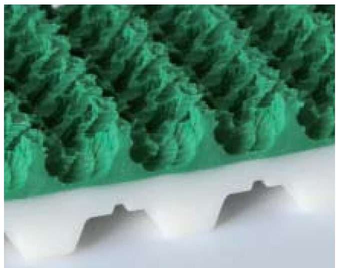
- Abrasion resistance
- Chemical resistance
- Compressibility
- Electrostatic discharge – antistatic features
- Excellent release properties
- Gentle movement
- Heat resistance
- High or low coefficient of friction
- Shock absorption

Commonly used material options include:

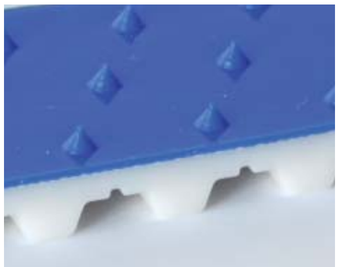
- TPU
- PU
- Natural rubber
- NBR
- EPDM
- Nitrile
- PVC
- EVA foam
- PU foam
- Rubber foam
- Artificial leather
- Aramide
- Polyester fleece

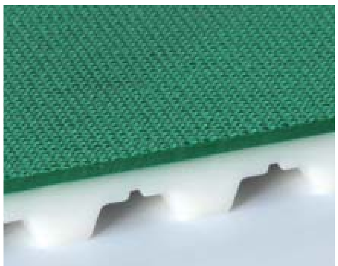
Each material with its particular property characteristics offers very specific inherent features. Also varied structure options provide a broad range of possibilities.

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





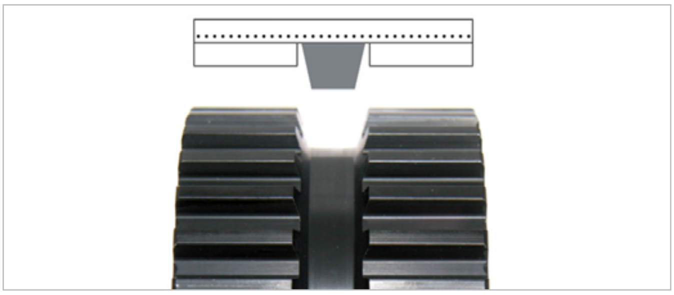

Tracking guides
Tracking guides are added to the tooth side of the HabaSYNC® timing belt. They are used on long-center-distance conveyors where true belt tracking is critical and where pulley flanges would interfere with the product being conveyed. They are also used where cross loading or unloading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

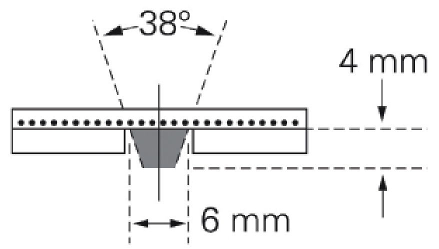
Tracking guides can also be used on linear-positioning and -conveyor applications where the belt is run in a vertical position rather than lying flat on a conveyor surface.

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

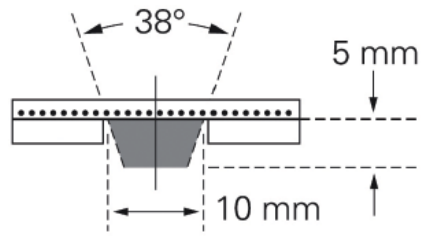
Tracking guides are typically notched to allow maximum flexibility of the belt when running around pulleys.

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



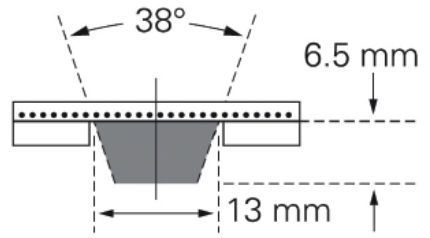


G6



G10

T10 with tracking guide on tooth side.



G13

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

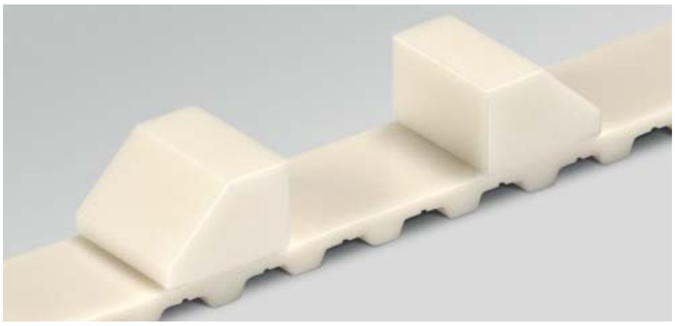
Habasit profiles are available in several Shore A hardness values. The thermoplastic polyurethane bonds securely to the conveying side of the HabaSYNC® timing belt using processes that include thermal bonding, vibration and high-frequency technology.

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.

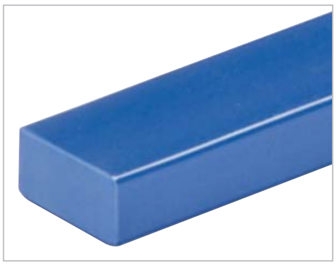
HabaSYNC® profiles can be produced in three ways. Manufacturing processes include:
- Machining
- Extrusion
- Injection molding

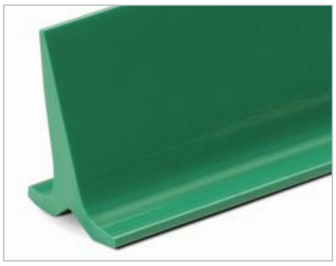
Machining
Machined profiles are produced with CNC equipment designed to machine plastic. We hold material in square or rectangular shapes in 85 and 92 Shore A hardness in stock, which can be machined quickly to provide any HabaSYNC® standard design.

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.

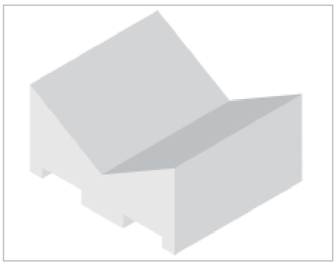


Extrusion
Where larger quantities of profiles are needed, extrusion can be an economical option. Habasit's extruded thermoplastic conveyor belt profiles may also be considered as an option if a softer material hardness is required. These profiles are in the range of 85 Shore A hardness.

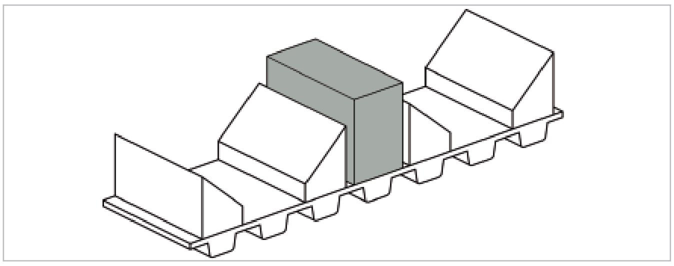




Molding
Profiles are injection-molded if the profile design must be exact or is complex.
HabaSYNC® injection-molded profiles can be produced in the same material as the base belt, and in many cases, up to maximum width to match the widest standard belt produced by Habasit.



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:

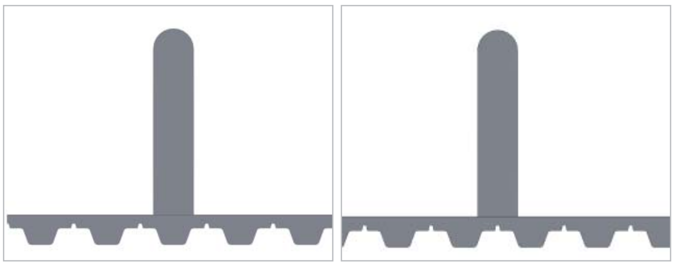
Minimum number of pulley teeth for profiles over a tooth

Profile base thickness in	1/16	1/8	3/16	1/4	5/16	3/8	7/16	1/2	5/8	3/4
mm	1.6	3	5	6	8	10	11	13	16	19
XL	10	10	18	25	40	50	60	100	–	–
L	12	12	12	28	30	40	50	60	100	–
H	14	14	14	14	18	25	35	45	80	100
XH	18	18	18	18	18	18	18	20	35	50
T5	12	12	18	25	40	50	60	100	–	–
AT5, HTD5	15	15	18	25	40	50	60	100	–	–
T10, AT10, HTD8	16	16	16	16	18	25	35	45	80	100
T20, AT20, HTD14	18	18	18	18	18	18	18	20	35	50

Minimum number of pulley teeth for profiles NOT over a tooth

Profile base thickness in	1/16	1/8	3/16	1/4	5/16	3/8	7/16	1/2	5/8	3/4
mm	1.6	3	5	6	8	10	11	13	16	19
XL	12	30	45	50	60	100	–	–	–	–
L	12	20	40	45	55	60	70	80	100	–
H	14	14	25	30	45	50	55	65	80	100
XH	18	18	20	30	40	45	50	54	58	60
T5	12	30	45	50	60	100	–	–	–	–
AT5, HTD5	15	30	45	50	60	100	–	–	–	–
T10, AT10, HTD8	18	20	30	40	45	50	55	65	80	100
T20, AT20, HTD14	18	18	20	30	40	45	50	54	58	60

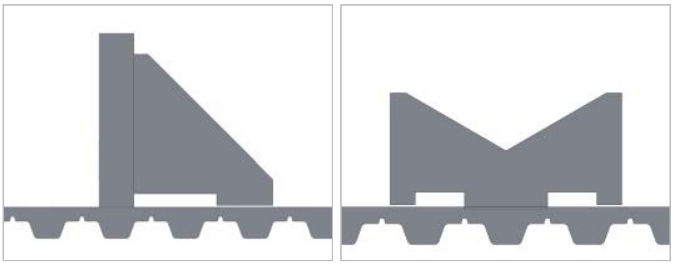
Sample sketches to show the differences between profile welding **over** a tooth or **between** the teeth.



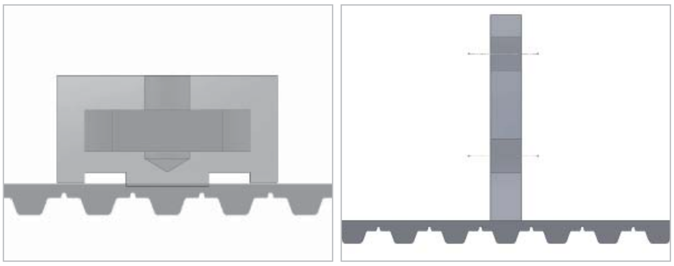
Profile strength
The strength of the profile weld is a direct factor of the dimension of the base weld. When reviewing profile strength, it is vital to consider (besides the size of the welded profile foot) the location and direction of force on the profile.

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.



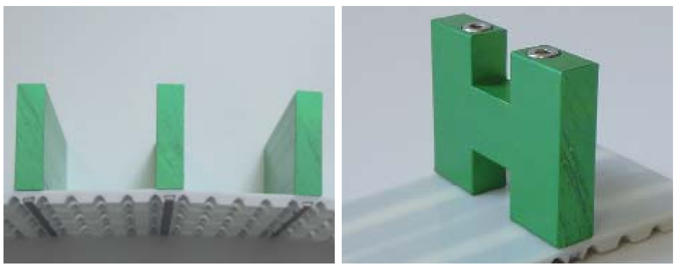
Profiles prepared for mounting attachments
In some cases a profile will be designed to allow an attachment to be fixed onto the profile. Typically these attachments are difficult or costly to mold and can be obtained in other materials with ease. To enable an attachment to be fixed HabaSYNC® profiles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

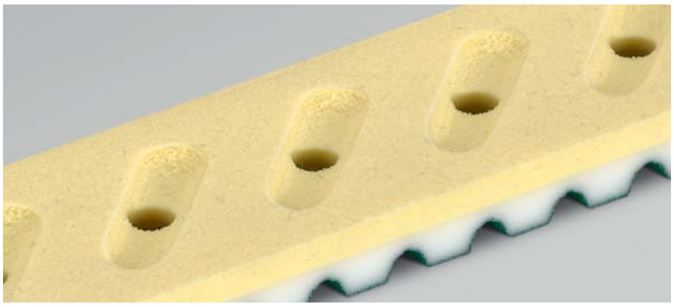
In many applications, particularly those in general conveying, modifications may be made to enhance product movement performance of a timing belt.

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.

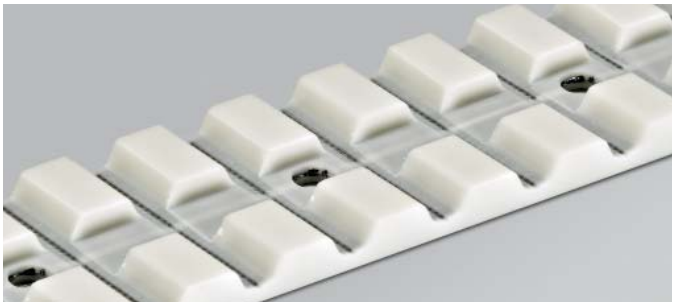


Lateral machining
On thick applied covers, HabaSYNC® timing belts can be machined across the belt conveying surface to create slots for holding the product in place on incline or vertical conveyors.

Modifications

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.

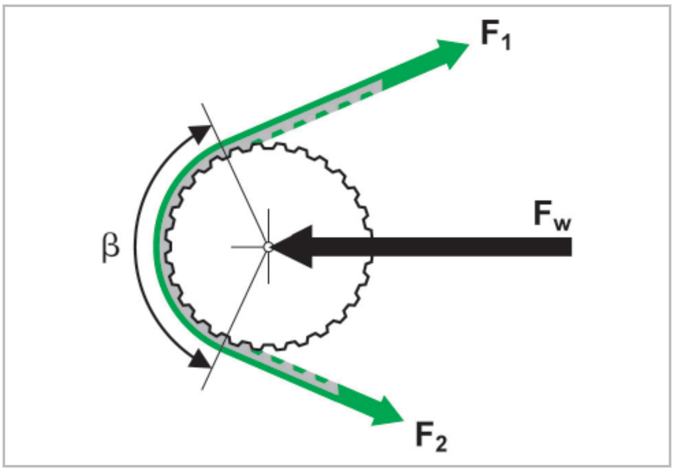
Modifications are typically designed for:
- Vacuum or hold-down conveyors
- Product capture points
- Sizing and separation of material conveyed
- Attachment ports for metal clamps or profiles
- Applications where precision thickness tolerances are required

Modifications are largely dependent on application circumstances. Please contact your Habasit representative to discuss your specific needs.

Design guide
Belt tension

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°, the required shaft load F_W on the drive pulley should be about 1.2 times the peripheral force F_U.



$F_W = 1.2 \cdot F_U$ [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

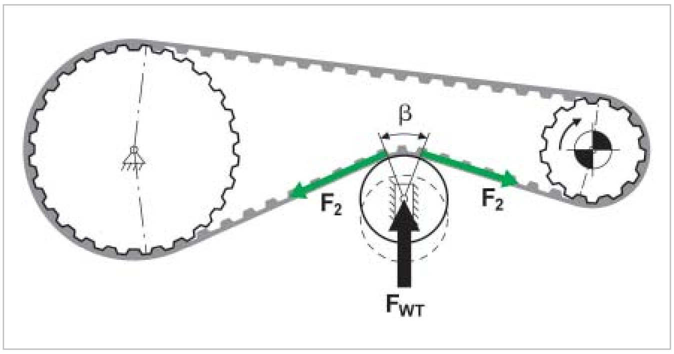
For an arc of contact $\beta \neq 180°$, the respective shaft load can be determined by the following approximation method:

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

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

Design guide
Tensioning devices

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.

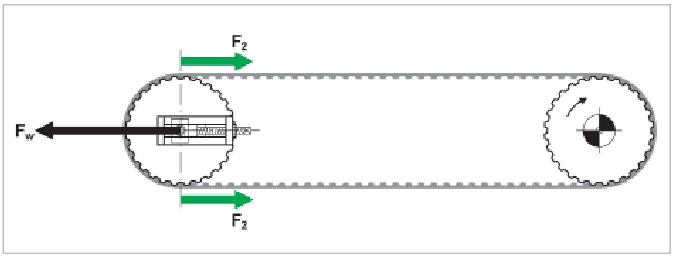


$$F_2 = \frac{F_{WT}}{2 \cdot \sin\left(\frac{\beta}{2}\right)} \quad [N]$$

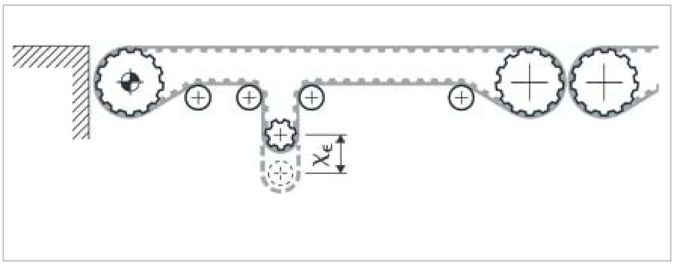
F_{WT} = Pressure force of tension roller
F_2 = Tensile force in the slack side of the belt
β = Arc of contact on tension roller

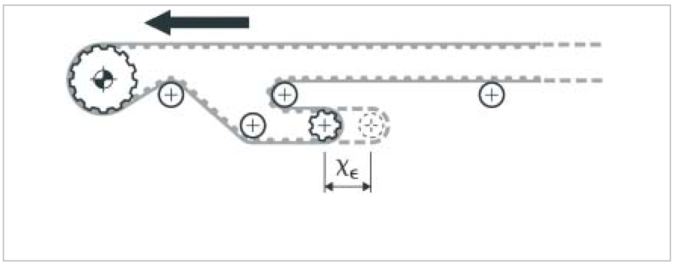
Drives with a fixed center-to-center distance
Fixed tensioning devices are used in applications where there is no need to compensate for variations in belt length or belt extension during operation.

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



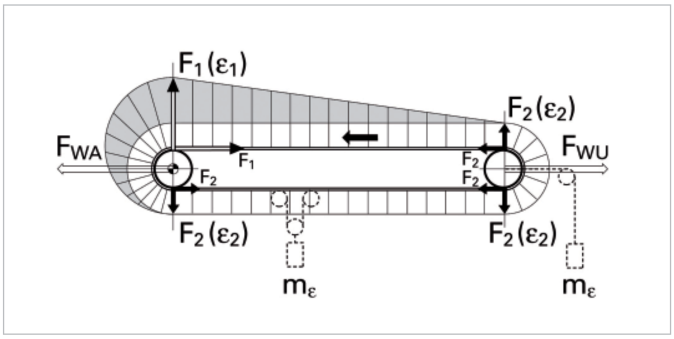
When the center distance between the head and tail rollers may not be changed, e.g. with intermediate or transition conveyors, the tension station is incorporated in the return side.



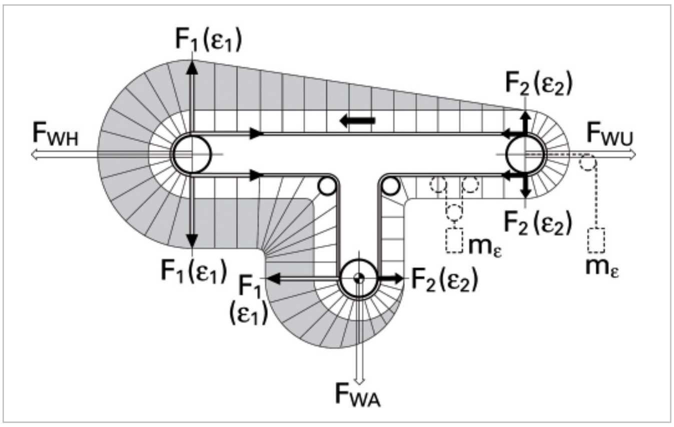


Position of drive
In order to calculate the initial belt extension, the position of the drive is extremely important.

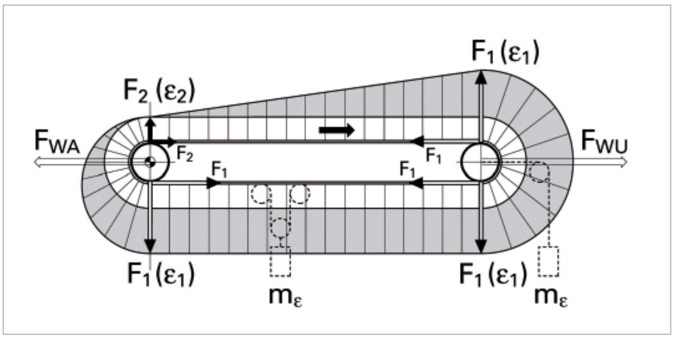
Head drive
This illustration indicates how the tensile force in the belt continuously increases due to the conveying of a mass. Since in this example the drive is placed at the head of the conveyor (on the left side of the illustration), the belt length with the higher tensile force level (F_1) is much 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

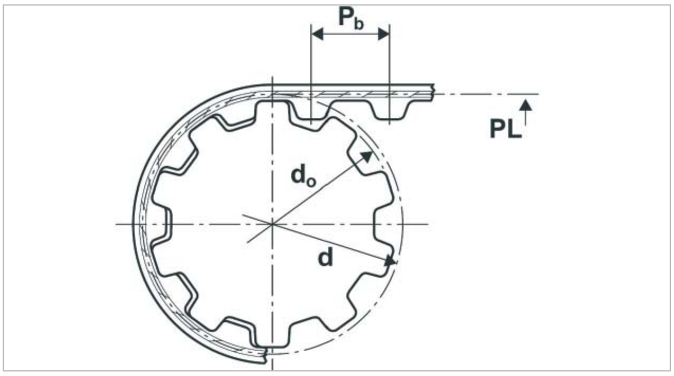
Belt evaluation
The evaluation of the optimal timing belt for a specific application is primarily a question of requirements. Initial questions include:
- Minimum pulley diameters
- Coefficient of friction of surfaces
- Properties of materials (suitable for food applications, chemical resistance, surface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

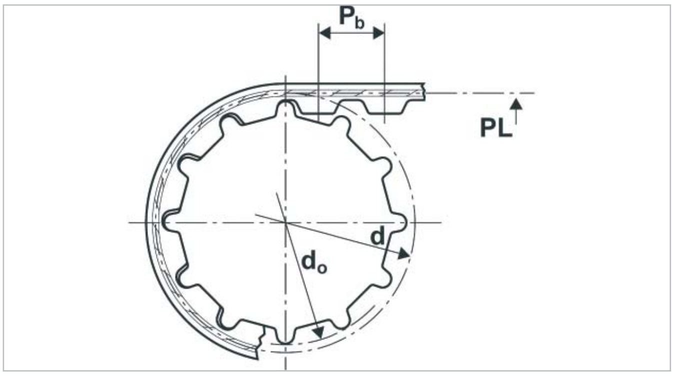
Belt series with trapezoid tooth shape
- T5 (5 mm pitch)
- T10 (10 mm pitch)
- T20 (20 mm pitch)
- XL (*1/5*" pitch / 5.08 mm)
- L (*3/8*" pitch / 9.525 mm)
- H (*1/2*" pitch / 12.7 mm)
- XH (*7/8*" pitch / 22.225 mm)



Trapezoid tooth shape

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

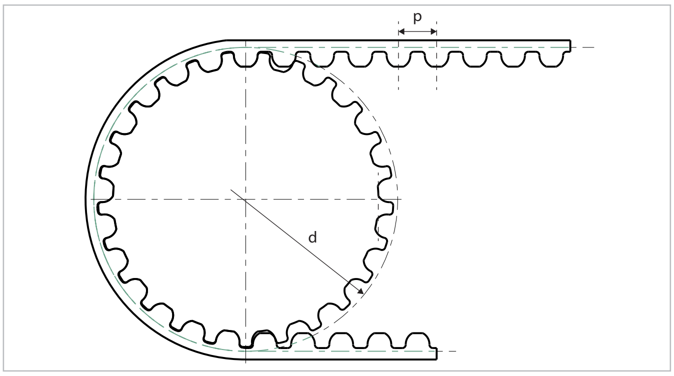
Belt series with modified trapezoid tooth shape
- AT5 (5 mm pitch)
- AT10 (10 mm pitch)
- AT20 (20 mm pitch)



Modified trapezoid tooth shape

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)



Curvilinear tooth shape

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}:

$$F_{US} = g \cdot m \cdot \mu_G \qquad [N]$$

g = Acceleration of gravity = 9.81 m/s²
m = Total mass to be carried over the slider bed [kg]
μ_G = Coefficient of friction between the belt and slider bed

For linear-positioning applications the friction force F_f 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):

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

g = Acceleration of gravity = 9.81 m/s²
h_T = Elevating height [mm]
l_T = Conveying length [mm]

- In applications where a mass is accelerated (actuator, stop-and-go operation), there is force F_{Ua} required for the acceleration of the carried goods:

$$F_{Ua} = m \cdot a \qquad [N]$$

m = Mass of carried goods on total conveying length (total load) [kg]
a = Acceleration [m/s²]

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

v = Belt speed [m/s]
t = 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:

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

Design guide
Evaluation of tooth and pitch

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

No. of teeth in mesh z_m	Tooth-in-mesh factor t_m
1	0.20
2	0.40
3	0.55
4	0.70
5	0.85
> 5	1.00

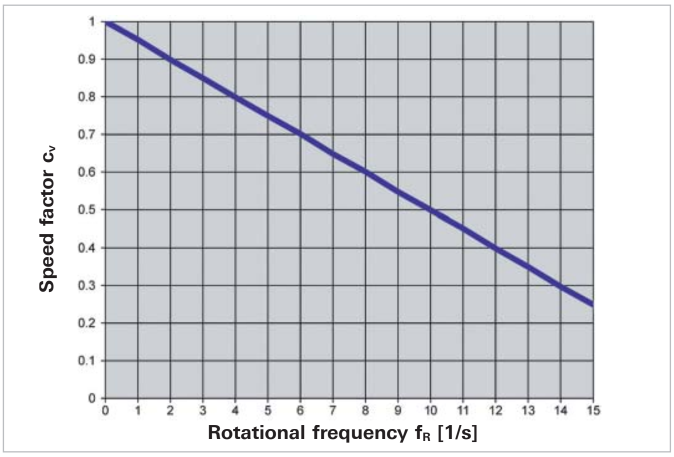
Open-ended belts (without joint)

No. of teeth in mesh z_m	Tooth-in-mesh factor t_m
1	0.15
2	0.30
3	0.40
4	0.50
5	0.60
6	0.70
7	0.80
8	0.85
9	0.90
10	0.95
11	0.97
> 11	1.00

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 (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

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

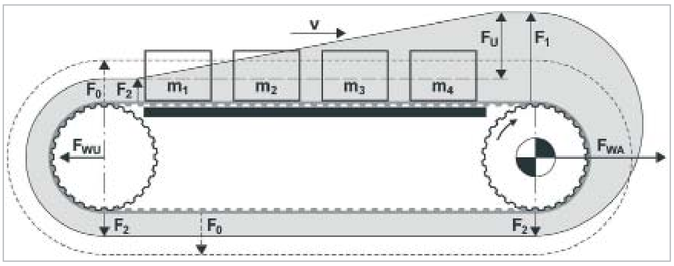
The peripheral force F_U at the drive pulley is the sum of all individual forces resisting the belt motion. The individual loads contributing to the peripheral force F_U must be identified and calculated based on the loading conditions and drive configuration. However, some loads cannot be calculated 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:

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

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



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

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

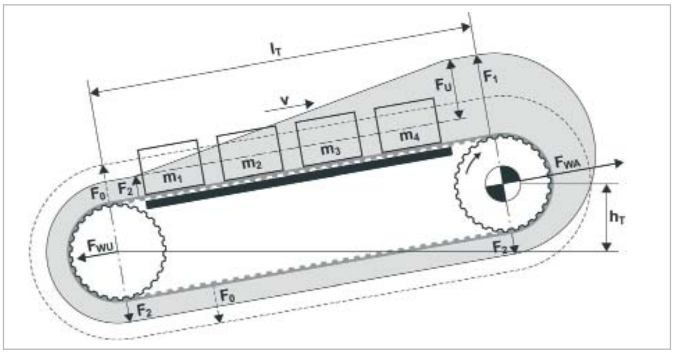
$$m_{tot} = m + m_B = m + \frac{l_T \cdot m'}{1000} \quad [kg]$$

m = Mass of carried goods on total conveying length (total load) [kg]
m_B = Mass of the belt moved over the slider bed [kg]
m' = Mass of belt per meter [kg/m]
l_T = Conveying length [mm]

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.

Calculation guide
Step 1 – Peripheral force

Force required to elevate the carried goods F_{Ui} (2nd addend)
F_{Ui} is the force required to elevate the mass m of the carried goods (not required in horizontal drives).



Formula for inclined transportation

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

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_{Ua} (3rd addend)
Force F_{Ua} required for the acceleration of the total mass:

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

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/s²]

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

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

v = Belt speed [m/s]
t = Time required to accelerate up to speed [s]

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

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

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.

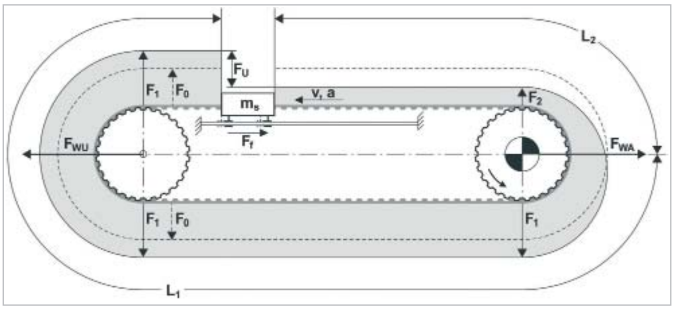
F_U for a linear-positioning application is primarily the sum of the following addends resisting the belt motion:

- 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:

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

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.



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

m_S = Mass of the slider plus maximum load [kg]
a = Acceleration [m/s²]

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

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

Δv = Speed difference (final speed minus initial speed) [m/s]
t = Time required to accelerate up to speed [s]

Friction force F_f (2nd addend)
The friction force F_f 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.

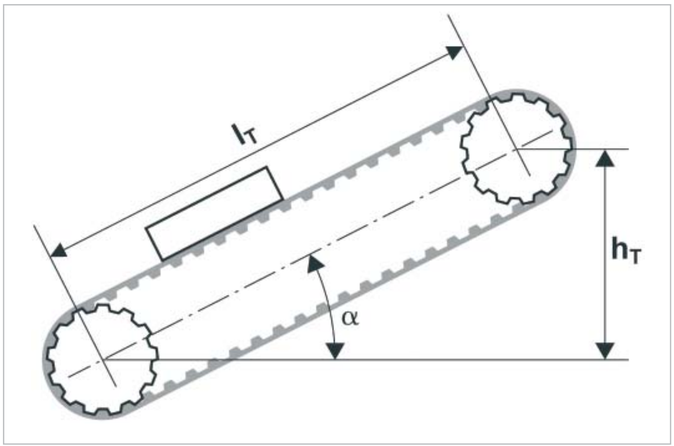
Externally applied working load F_E (3rd addend)
If existing, externally applied working load F_E cannot be ignored. It is possible, for example, that an actuator pulls a mass over a table. The respective friction force has to be considered as an "externally applied working load."

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
α = Angle of inclination [°]
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:

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

Calculation guide
Step 2 – Belt selection/Step 3 – Pulley definition

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:

$$d = \frac{P_b \cdot z_P}{\pi} \quad \text{[mm]}$$

Calculation guide
Step 4 – Center distance and belt length

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:

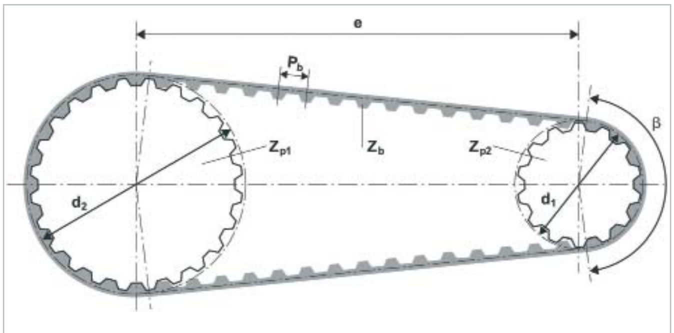
Use the preliminary center distance e desired for the design envelope to determine a preliminary number of belt teeth z_b:

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

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:

$$z_b = \frac{2 \cdot e}{P_b} + \frac{z_{P1} + z_{P2}}{2} + \frac{P_b}{4e}\left[\frac{z_{P2} - z_{P1}}{\pi}\right]^2$$



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).

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

$$l_0 = z_b \cdot P_b$$

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:

$$e = \frac{l_0 - \frac{\pi \cdot (d_2 + d_1)}{2} \sqrt{ + \left[l_0 - \frac{\pi \cdot (d_2 + d_1)}{2}\right]^2 - 2\,(d_2 + d_1)^2}}{4}$$

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]

Calculation guide
Step 5 – Teeth in mesh

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:

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

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:

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

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

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

Joined endless belts

No. of teeth in mesh z_m	Tooth-in-mesh factor t_m
1	0.20
2	0.40
3	0.55
4	0.70
5	0.85
> 5	1.00

Open-ended belts (without joint)

No. of teeth in mesh z_m	Tooth-in-mesh factor t_m
1	0.15
2	0.30
3	0.40
4	0.50
5	0.60
6	0.70
7	0.80
8	0.85
9	0.90
10	0.95
11	0.97
> 11	1.00

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:

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

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

or expressed in elongation:

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

ε_2 = Minimal belt elongation in the slack side
ε_U = Belt elongation generated by peripheral force F_U

$$\varepsilon_U = \frac{F_U}{k_{1\%}} \qquad [\%]$$

$k_{1\%}$ = Tensile force for 1% elongation [N]

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:

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

F_{WT} = Pressure force of slack-side tensioning idler [N]
F_U = Peripheral force [N]
β_T = 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 ε_0 is:

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

ε_0 = Initial belt elongation [%]
ε_2 = Minimal belt elongation in the slack side [%]
ε_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**:

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

Tail drives**:

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

Center drives**:

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

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

Calculation guide
Step 7 – Elongation and forces

Step 7
Calculation of elongations and forces in the tight and slack side

The belt elongation ε_1 in the **tight** belt strand is obtained by (for fixed center distances):

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

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

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

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 ε_2 in the **slack** belt strand is obtained by (for fixed center distances):

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

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

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

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 = Tensile force due to initial tension = $\varepsilon_0 \cdot k_{1\%}$ [N]
F_1 = Maximum tensile force in the tight belt strand [N]
F_2 = Minimum tensile force in the slack belt strand [N]
F_U = Peripheral force [N] ($F_U = F_1 - F_2$)
ε_0 = Initial belt elongation [%]
ε_1 = Maximal belt elongation in the tight side [%]
ε_2 = Minimal belt elongation in the slack side [%]
ε_U = Belt elongation generated by peripheral force F_U [%] ($\varepsilon_U = \varepsilon_1 - \varepsilon_2$)
l_0 = Belt length = $l_1 + l_2$ [mm]
l_1 = Length of the tight belt strand [mm]
l_2 = Length of the slack belt strand [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

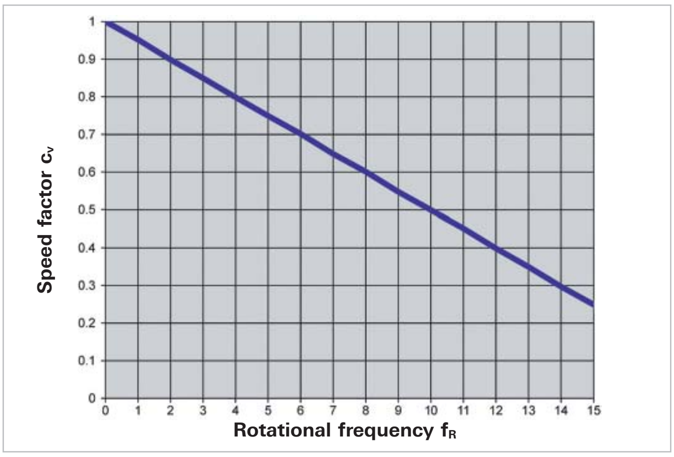
B To determine the admissible load on teeth, specify the tooth-in-mesh factor t_m for joined or endless belts (see step 5).

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.

Calculation guide
Step 9 – Shaft load

Step 9
Calculation of shaft loads
For an arc of contact of 180° the shaft load F_W is:

$$F_W = F_1 + F_2 \quad [N]$$

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

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

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

Determine the static shaft load F_{WAs} and dynamic shaft load F_{WAd} of the **drive** pulley:

$$F_{WAs} = 2 \cdot F_0 \cdot \sin\left(\frac{\beta}{2}\right) \quad [N]$$

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

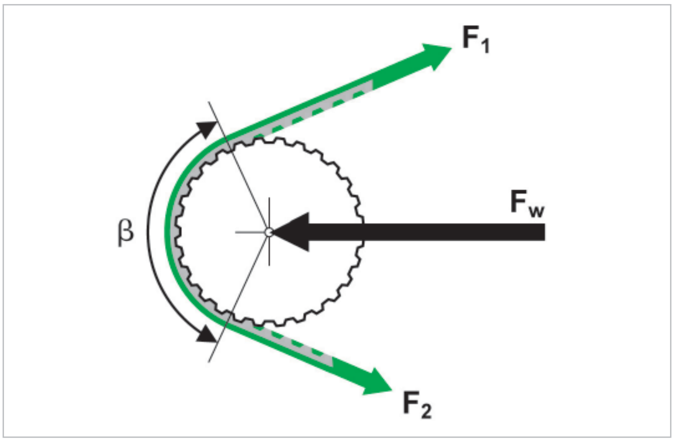
Determine the static shaft load F_{WUs} and dynamic shaft load F_{WUd} of the **tail** pulley:

$$F_{WUs} = 2 \cdot F_0 \cdot \sin\left(\frac{\beta}{2}\right) \quad [N]$$

F_W = Shaft load [N]
F_{WAs} = Static shaft load of the drive pulley [N]
F_{WAd} = Dynamic shaft load of the drive pulley [N]
F_{WUs} = Static shaft load of the tail pulley [N]
F_{WUd} = Dynamic shaft load of the tail pulley [N]
F_0 = Tensile force due to initial tension ($F_0 = \varepsilon_0 \cdot k_{1\%}$) [N]
F_1 = Maximum tensile force in the tight belt strand [N]
F_2 = Minimum tensile force in the slack belt strand [N]

Since in linear-positioning applications the highest shaft load of the tail pulley F_{WUd} 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.

$$F_{WUd} = 2 \cdot F_1 \cdot \sin\left(\frac{\beta}{2}\right) \quad [N]$$



Arc of contact β		sin β/2
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

Step 10
Calculation of drive power and required motor power
The required power on the drive pulley is:

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

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

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:

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

The respective torque M_a on the drive pulley shaft is:

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

P_M = Power of the motor [kW]
η = 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 η = 75% if the exact figure is unknown.

Calculation guide
Step 11 – Positioning error

Step 11
Calculation of positioning error
Positioning errors have to be distinguished in terms of

- random positioning error Δx_R (tolerance when many positioning procedures are compared with each other)
- systematic positioning error Δx_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 Δx the accuracy factor of the specific belt [%] times the maximum covered distance of the slide has to be added to the random positioning error.

$$\Delta x = \Delta x_R + \Delta x_S = \Delta x_R + \frac{l_T \cdot af}{100} \qquad [mm]$$

Δx = Positioning error [mm]
l_T = Maximal covered distance of the slide [mm]
af = Accuracy factor of belt [%]
Δx_R = Random positioning error
Δx_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 Δx_R is the sum of the following three partial errors:

A Belt elongation due to elasticity of the belt Δx_1
B Deformation of teeth in mesh on the drive pulley Δx_2
C Backlash due to the clearance between the belt teeth and the pulley grooves Δx_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 ΔF

The partial error Δx_1 of a slider in a determined position is:

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

Δx_1 = Maximal possible deviation of slider position caused by belt elongation [mm]
ΔF = 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.

Calculation guide
Step 11 – Positioning error

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:

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

Δx_2 = Maximal possible deviation of the slider position caused by the deformation of belt teeth [mm]
ΔF = Highest possible variation of force component on the positioned slider [N]
df = Deformation factor
t_m = 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)

P_b = Belt pitch [mm]
$k_{1\%}$ = Tensile force for 1% elongation [N]

C The backlash (the clearance between the belt teeth and the pulley grooves) may be negligible if the positioning is always done from the same side with a similar braking procedure.

If the braking procedure varies from case to case, or if the positioning of the slide is done from both sides, we recommend adding a clearance value Δx_3 to define the random positioning error.
Since this clearance value Δx_3 is determined by both the belt and by the tolerances of the pulley, in principle it is impossible to define a value for a specific belt, but only for a belt and pulley combination.

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

Δx_3 = Maximal clearance between the belt teeth and the pulley grooves [mm]
P_b = Belt pitch [mm]

For demanding applications where minimal backlash is required, use zero backlash pulleys. If such pulleys are used, it is not necessary to consider Δx_3.

Resulting positioning error consists of

random error:

$$\Delta x_R = \Delta x_1 + \Delta x_2 + \Delta x_3 \quad [mm]$$

and systematic error:

$$\Delta x_S = \frac{l_T \cdot af}{100} \quad [mm]$$

l_T = Maximum covered distance of the slide [mm]
af = Accuracy factor of belt [%]

The total error (absolute) is:

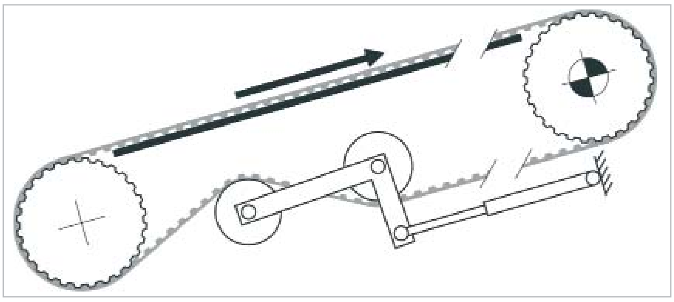
$$\Delta x = \Delta x_R + \Delta x_S \quad [mm]$$

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

Belt series	Metric pitch, trapezoid tooth shape
Conveying length	3,000 mm
Elevating height	800 mm
Total load on belt	900 kg (450 kg per belt)
Position of drive	head
Arc of contact on drive pulley	180°
Arc of contact on pressure roller	60°
Conveyor bed	Slider bed (UHMW PE)
Diameter of drive pulley	≈ 150 mm
Diameter of tension pulleys	as small as possible
Belt speed	40 m/min

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 μ_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:

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

⇒ 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:

$$v\,[m/s] = \frac{v\,[m/min]}{60}$$

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:

$$f_R = \frac{v \cdot 1000}{l_0} \approx \frac{0.67 \cdot 1000}{7000} \approx 0.1 \quad 1/s$$

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 $\mathbf{t_m}$ **= 1.0** and $\mathbf{c_v}$ **= 1.0).**

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

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:

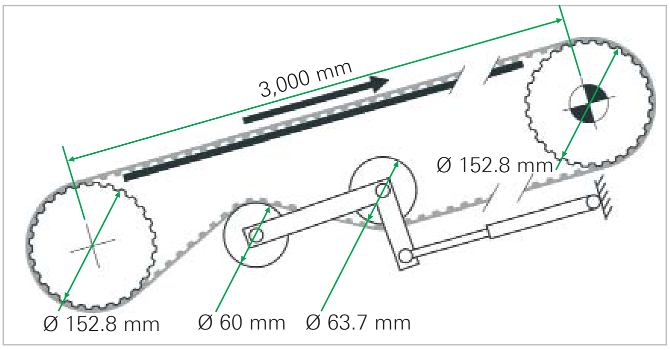
$$d = \frac{P_b \cdot z_P}{\pi} = \frac{10 \cdot 48}{3.14} = 152.8 \text{ mm}$$

For the tensioner, the minimum pulley diameter for counter flection is found on the T10 product data sheet: d_T = 60 mm

Idler: for forward flection 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 $\mathbf{t_m = 1.0}$).

Step 6
Determine the minimal tensile force in the slack belt strand
Peripheral force F_U = 2,943 N

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

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

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

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

For drives with controlled slack-side tension
Arc of contact of the belt on the tensioning idler β_T = 60°

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

$$F_{WT} = 0.4 \cdot F_U \cdot \sin\left(\frac{\beta_T}{2}\right) = 0.4 \cdot 2943 \cdot \sin\left(\frac{60}{2}\right) \approx 589 \text{ N}$$

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 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.

Arc of contact on tail pulley β = 210°

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

Arc of contact β		sin β/2
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

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:

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

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 η = 75%, which is a recommended value if the correct figure is not known, the required motor power P_M is:

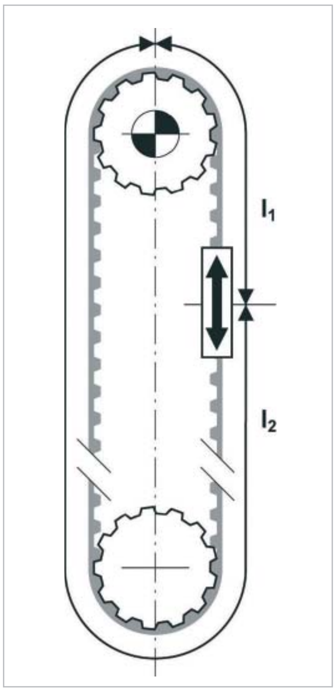
$$P_M = \frac{P \cdot 100}{\eta} = \frac{1.96 \cdot 100}{75} = 2.61 \text{ kW}$$

Calculation examples
Linear-positioning drive

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).

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



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:

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

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

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

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_{Ua} required for the acceleration of the mass:

$$F_{Ua} = m \cdot a \quad [N]$$

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

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

- 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:

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

The estimated peripheral force F_U is 3,323 N.

Calculation examples
Linear-positioning drive

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:

$$f_R = \frac{v \cdot 1000}{l_0} \approx \frac{0.6 \cdot 1000}{7200} \approx 0.83 \quad 1/s$$

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 $\mathbf{t_m = 1.0}$ and $\mathbf{c_V = 1.0}$).

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 mm
Belt pitch	P_b = 10 mm

Number of belt teeth z_b:

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

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:

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

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

Peripheral force F_U = 3,323 N

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

$k_{1\%}$ (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:

$$\varepsilon_2 \approx 0.2 \cdot \varepsilon_U = 0.2 \cdot 0.19 \approx 0.0380\%$$

Initial belt elongation ε_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	≈ 3,300 mm
Belt length $l_0 = l_1 + l_2$	= 7,250 mm
(Length of the slack belt strand l_2	≈ 3,950 mm)
Belt elongation ε_U generated by peripheral force F_U	= 0.190%

$$\varepsilon_0 = \varepsilon_2 + \varepsilon_U \cdot \frac{l_1}{l_0} = 0.038 + 0.19 \cdot \frac{3300}{7250} = 0.124\%$$

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:

$$F_1 = F_0 + F_U \cdot \frac{l_2}{l_0} = 2170 + 3323 \cdot \frac{3950}{7250} = 3{,}980 \text{ N}$$

The belt elongation in the slack belt strand ε_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 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:

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

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:

$$M_a = \frac{F_U \cdot d_a}{2000} = \frac{3323 \cdot 79.6}{2000} = 132 \text{ Nm}$$

Step 11
Calculate the positioning error
The random positioning error Δx_R is the sum of the following three partial errors:

A Belt elongation due to elasticity of the belt Δx_1
B Deformation of the teeth in mesh on the drive pulley Δx_2
C Backlash due to the clearance between the belt teeth and the pulley grooves Δx_3

A The partial error Δx_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 Δm is therefore 280 kg.

The possible variation of the force ΔF on the slider is:

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

Acceleration of gravity g	$= 9.81$ m/s^2
Length of the tight belt strand l_1	$\approx$ 3,300 mm
Length of the slack belt strand l_2	$\approx$ 3,950 mm
Belt length $l_0 = l_1 + l_2$	= 7,250 mm

$$\Delta x_1 = \frac{\Delta F \cdot l_1 \cdot (l_0 - l_1)}{l_0 \cdot k_{1\%} \cdot 100} = \frac{2747 \cdot 3300 \cdot (7250 - 3300)}{7250 \cdot 17500 \cdot 100} = 2.82 \text{ mm}$$

B Deformation of the teeth in mesh on the drive pulley Δx_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 Δx_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 mm

Total error (absolute)

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

Total error (relative)

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

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

Types	Up to 50 mm (2") width	Over 50 mm to 100 mm *(2" to 4")* width	Over 100 mm to 200 mm* *(4" to 8")* width
T5	± 0.50 mm	± 0.75 mm	– 1.00 mm / + 1.00 mm
T10	± 0.50 mm	± 0.75 mm	– 1.00 mm / + 1.00 mm
T20	± 0.75 mm	± 1.00 mm	– 1.00 mm / + 1.00 mm
AT5	± 0.50 mm	± 0.75 mm	– 1.00 mm / + 1.00 mm
AT5P	± 0.50 mm	± 0.75 mm	– 1.00 mm / + 1.00 mm
AT10	± 0.75 mm	± 1.00 mm	– 1.00 mm / + 1.00 mm
AT10P	± 1.00 mm	± 1.50 mm	– 1.00 mm / + 1.50 mm
AT20	± 1.00 mm	± 1.50 mm	– 1.00 mm / + 1.50 mm
HTD5	± 0.50 mm	± 0.75 mm	– 1.00 mm / + 1.00 mm
HTD8	± 0.75 mm	± 1.00 mm	– 1.00 mm / + 1.00 mm
HTD14	± 1.00 mm	± 1.50 mm	– 1.00 mm / + 1.50 mm
XL	± 0.51 mm ± *.020"*	± 0.76 mm ± *.030"*	± 1.02 mm ± *.040"*
L	± 0.51 mm ± *.020"*	± 0.76 mm ± *.030"*	± 1.02 mm ± *.040"*
H	± 0.51 mm ± *.020"*	± 0.76 mm ± *.030"*	± 1.02 mm ± *.040"*
XH	± 1.02 mm ± *.040"*	± 1.02 mm ± *.040"*	± 1.02 mm ± *.040"*

* 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

Up to 50 mm *(2")* width	Over 50 mm to 100 mm *(2" to 4")* width	Over 100 mm to 150 mm* *(4" to 6")* width
– 0.5 mm	– 1.0 mm	– 1.5 mm

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.

Appendix
Tolerances

Standard belt thickness tolerance

Types	Normal thickness	Tolerance
T5	2.2 mm	± 0.15 mm
T10	4.5 mm	± 0.30 mm
T20	8.0 mm	± 0.45 mm
AT5	2.7 mm	± 0.20 mm
AT5P	2.7 mm	± 0.20 mm
AT10	4.5 mm	± 0.30 mm
AT10P	4.8 mm	± 0.30 mm
AT20	8.0 mm	± 0.45 mm
AT20P	8.4 mm	± 0.50 mm
HTD5	3.6 mm	± 0.20 mm
HTD8	5.6 mm	± 0.30 mm
HTD14	10.0 mm	± 0.50 mm
XL	2.3 mm *0.090"*	± 0.15 mm *± .006"*
L	3.6 mm *0.142"*	± 0.20 mm *± .008"*
H	4.3 mm *0.169"*	± 0.30 mm *± .012"*
XH	11.2 mm *0.441"*	± 0.50 mm *± .020"*

Thickness tolerance of covers

Cover	Thickness up to 2 mm	Over 2 mm to 5 mm	Over 5 mm
Solid PU cover	± 0.20 mm	± 0.40 mm	± 0.60 mm
Structured surface	± 0.50 mm	± 1.00 mm	± 1.50 mm
Rubber	± 0.30 mm	± 0.50 mm	± 0.75 mm
Foam	± 0.30 mm	± 0.50 mm	± 0.75 mm
Grinded surface	± 0.30 mm	± 0.50 mm	± 0.50 mm

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 groves 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.

Standard material	Code	Hardness	Properties	Cord	Temperature
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.

Appendix
Diameters of cords

Steel

Type	Cord diameter [mm]
T5	0.33
T10	0.63
T20	0.91
AT5	0.49
AT10	0.91
AT20	1.21
HTD5	0.49
HTD8	0.91
HTD14	1.21
XL	0.33
L	0.49
H	0.63
XH	0.91

Aramide

Type	Cord diameter [mm]
T5	0.34
T10	0.75
T20	1.25
AT5	0.59
AT10	1.25
AT20	1.25
HTD5	0.59
HTD8	–
HTD14	–
TT5	0.52
XL	0.34
L	0.59
H	0.75
XH	1.25

Stainless steel

Type	Cord diameter [mm]
T5	–
T10	0.63
T20	0.90
AT5	–
AT10	0.90
AT20	–
HTD5	
HTD8	0.9
HTD14	–
XL	–
L	–
H	0.63
XH	0.90

P Steel

Type	Cord diameter [mm]
AT5P	0.63
AT10P	1.21
AT20P	1.60

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*.

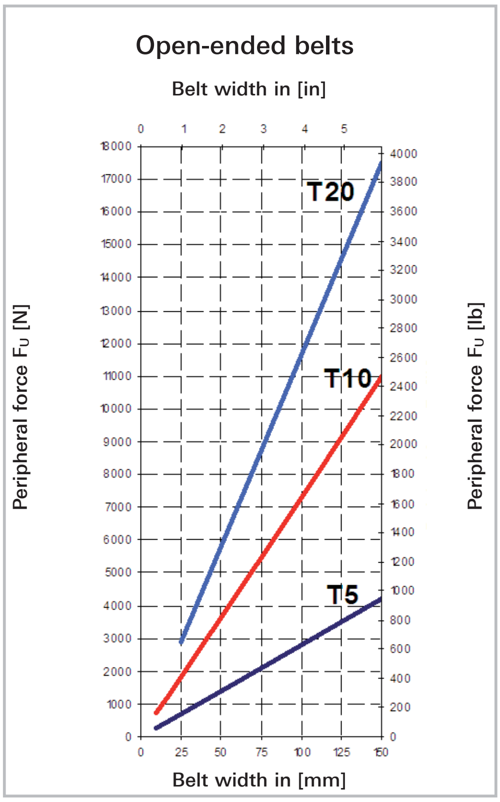
Please note that users still need to make qualification tests in order to ensure suitability for your application. For additional details, please contact your local Habasit representative.

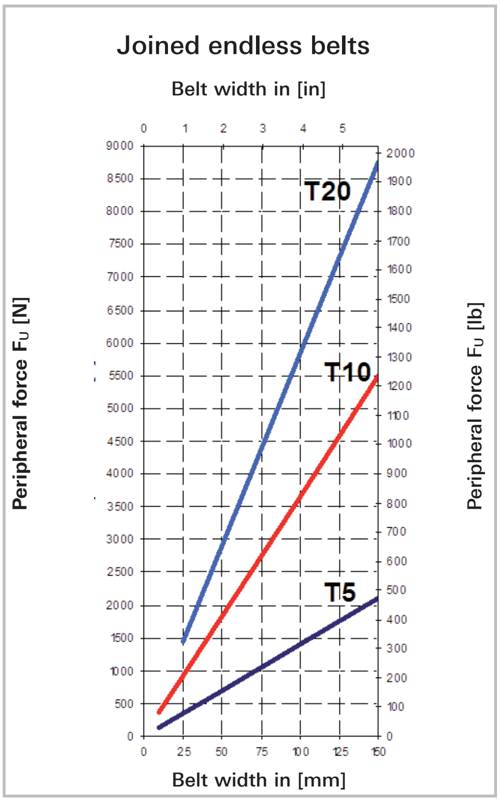
Code: ■ = good resistance ▼ = conditionally / sometimes resistant ❑ = not resistant (do not use)

Designation of chemical	Polyurethane
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	▼
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	■
Nitrous 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	■
Toluene	❑
Turpentine	❑
Vegetable oils	■
Vinegar	❑
Vinyl acetate	❑
Water – deionized	■
Xylene	❑
Zinc acetate	❑

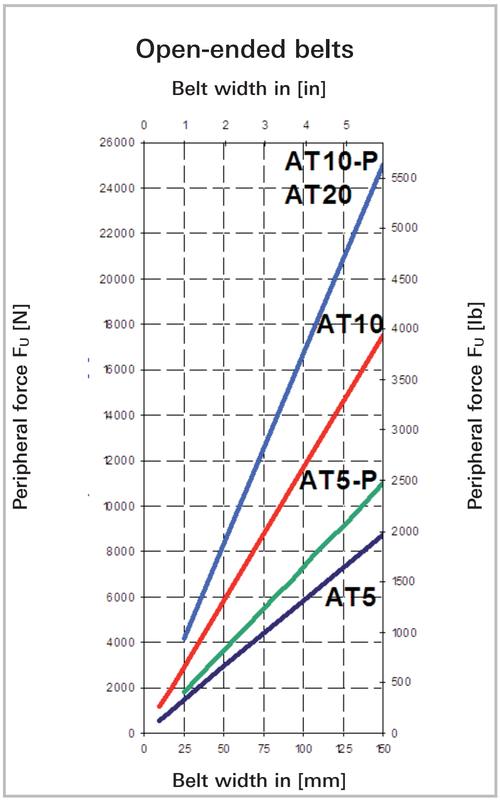
Appendix
Evaluation of tooth and pitch

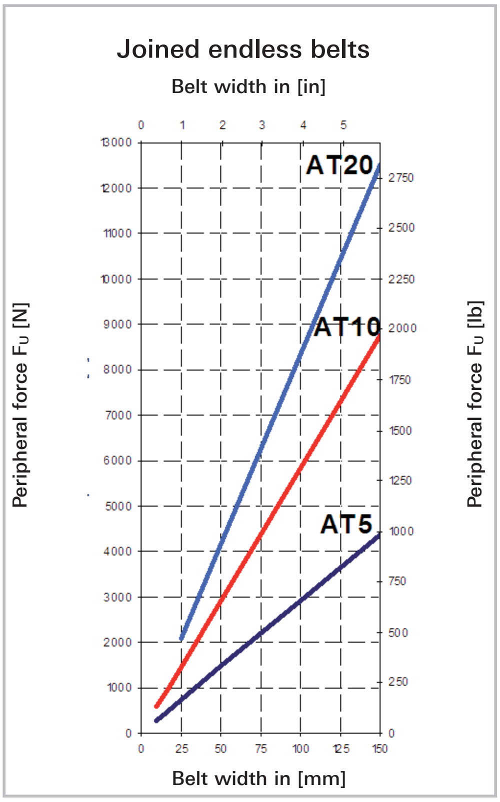
Pitch selection for T series belts





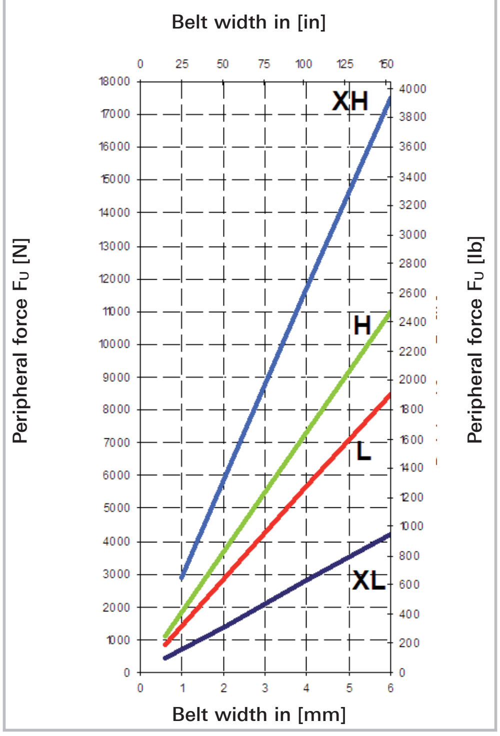
Pitch selection for AT series belts



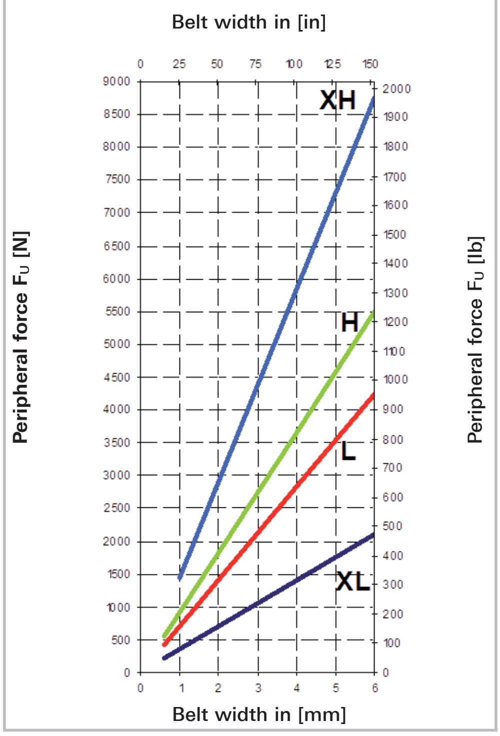


Pitch selection for belts of series with imperial pitches

Open-ended belts

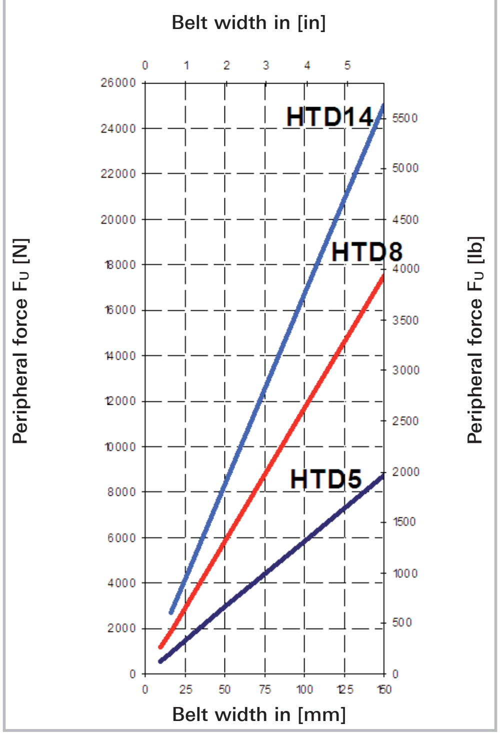


Joined endless belts

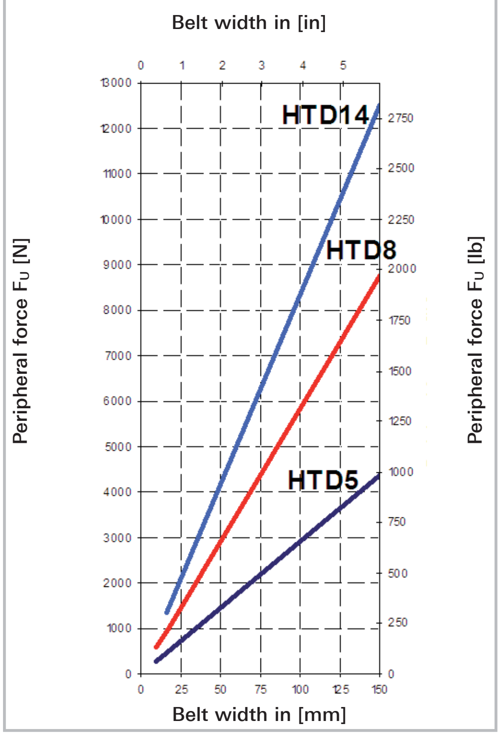


Pitch selection for HTD series belts

Open-ended belts



Joined endless belts



Appendix
List of abbreviations

Term	Symbol	Metric value	Imperial value
Acceleration	a	m/s^2	ft/s^2
Accuracy factor of belt	af	%	%
Admissible tensile force	F_{adm}	N	lb
Angle of inclination	α	°	°
Arc of contact on drive pulley	β_a	°	°
Arc of contact on pulley	β	°	°
Arc of contact on tensioning idler	β_T	°	°
Backlash due to pulley groove clearance	Δx_3	mm	inch
Belt elongation due to elasticity of belt	Δx_1	mm	inch
Belt elongation due to peripheral force	ε_U	%	%
Belt elongation in the slack belt strand	ε_2	%	%
Belt elongation in the tight belt strand	ε_1	%	%
Belt length	l_0	mm	inch
Belt pitch	P_b	mm	inch
Belt speed	v	m/s	ft/s
Belt width	b_0	mm	inch
Center-to-center distance	e	mm	inch
Coefficient of friction belt/slider bed	μ_G	–	–
Conveying length	l_T	mm	inch
Deformation factor	df	–	–
Deformation of teeth in mesh	Δx_2	mm	inch
Dynamic shaft load of drive pulley	F_{WAd}	N	lb
Dynamic shaft load of tail pulley	F_{WUd}	N	lb
Efficiency of drive (gearbox, etc.)	η	%	%
Elevating height	h_T	mm	inch
Externally applied working load	F_E	N	lb
Friction force of linear bearing	F_f	N	lb
Highest admissible operating temperature (continuous)	T_{max}	°C	°F
Highest possible variation of force on positioned slider	ΔF	N	lb
Initial belt extension	ε_0	%	%
Length of slack belt strand	l_2	mm	inch
Length of tight belt strand	l_1	mm	inch
Lowest admissible operating temperature (continuous)	T_{min}	°C	°F
Mass of belt carried over the slider bed	m_B	kg	lb
Mass of belt per meter (weight of belt/m; weight of belt/ft)	m'	kg/m	lb/ft
Mass of carried goods on total conveying length	m	kg	lb
Mass of slider plus load on slider	m_s	kg	lb
Maximal covered distance of linear drive	l_T	mm	inch
Mechanical power on drive pulley	P	kW	HP
Minimum required belt width	b_{req}	mm	inch
Number of belt teeth	z_b	–	–
Number of pulley teeth	z_p	–	–
Number of pulley teeth of drive pulley	z_a	–	–

Appendix
List of abbreviations

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

Appendix
Conversion of units

Metric units		Factor to convert to imperial units			Factor to convert to metric units		
Length							
mm	millimeter	0.0394	*in*	*inch*	25.4	mm	millimeter
m	meter	3.281	*ft*	*foot*	0.3048	m	meter
Area							
mm^2	square millimeter	0.00155	*in^2*	*square inch*	645.2	mm^2	square millimeter
m^2	square meter	10.764	*ft^2*	*square foot*	0.0929	m^2	square meter
Speed							
m/s	meter/second	3.281	*ft/s*	*foot/second*	0.3048	m/s	meter/second
m/min	meter/minute	3.281	*ft/min*	*foot/minute*	0.3048	m/min	meter/minute
Mass							
kg	kilogram	2.205	*lb*	*pound weight*	0.4536	kg	kilogram
kg/m	kilogram/meter	0.672	*lb/ft*	*pound/foot*	1.4882	kg/m	kilogram/meter
kg/m^2	kilogram/sqm	0.205	*lb/ft^2*	*pound/square foot*	4.882	kg/m^2	kilogram/sqm
Force and strength							
N	Newton	0.225	*lb*	*pound force*	4.448	N	Newton
N/mm	Newton/millimeter	5.7102	*lb/in*	*pound/inch*	0.17513	N/mm	Newton/millimeter
N/m	Newton/meter	0.0685	*lb/ft*	*pound/foot*	14.6	N/m	Newton/meter
Power							
kW	kilowatt	1.341	*hp*	*horsepower*	0.7457	kW	kilowatt
Torque							
Nm	Newton meter	8.85	*in-lb*	*inch pound*	0.113	Nm	Newton meter
Temperature							
°C	Celsius	9 · (°C / 5) + 32°	*°F*	*Fahrenheit*	5/9 · (°F – 32°)	°C	Celsius

Appendix
Glossary of terms

Term	Explanation	Habasit symbol
Accessories	Objects or devices commonly used for belt applications (e.g. guide strips, pulleys, tensioners, belt clamps, etc.)	
Admissible tensile force	Admissible belt tensile force allowed in the tightest belt section under process conditions	F_{adm}
Admissible tensile force, joined belt	Admissible belt tensile force allowed in the tightest belt section under process conditions for joined belts (only valid for master joint)	F_{adm} joined endless
Admissible tensile force, open belt	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)	F_{adm} open ended
Aramide	High modulus synthetic fiber (Kevlar, Technora, Twaron)	
Balanced cords	The cord twist of the tensile member alternates from cord to cord (S twist/Z twist/S twist and so on)	
Belt length	Length of belt measured along the neutral layer (length of traction member)	l_0
Belt options	Nonstandard surfaces, materials, colors, etc.	
Belt pitch	Distance from the center of a tooth to the center of the next tooth	P_b
Belt width	Geometric width of the belt from edge to edge	P_0
Bidirectional drive	Driving concept allowing to run the belt forward and backward	
Center distance	Linear distance between two pulleys	
Center drive	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	
Coefficient of friction	Ratio of frictional force and contact force acting between two material surfaces	μ
COF	Abbreviation for coefficient of friction	
Conveying length	Conveying length measured between the centers of the head and tail pulleys	l_T
Conveying side	Opposite side to toothed belt side (belt side which commonly carries the conveyed goods)	
Conveying side cover	Cover material (surface material) on the conveying side	
Cord	Tensile member	
Counter flection	Belt is bent over pulley(s) on the conveying side, e.g. when tension pulley is used	
Cover	Cover material (surface material) on the conveying side	
Elastomer	Comparatively soft synthetic material like rubber (thermoset elastomer) or thermoplastic polyurethane (thermoplastic elastomer)	
Family	T, AT, and HTD are the metric pitch families; L, XL, H, and XH belts are the imperial pitch family	
FDA	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	**FDA**
Flex belt	Truly endless timing belt made of TPU extruded over helically wound cords	
Flight	Small groove in the tooth root required for cord positioning in the belt production process	
Head drive	Driven head pulley (preferred design) However, for bidirectional belt runs a center drive is recommended	
Head pulley	Pulley at the end of the conveyor (refers to the belt running direction)	
Height of belt	Overall thickness of the timing belt	h_s
HTD High Torque Drive	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	
Indexing	Feeding or conveying of goods synchronously with the beat of a process. Indexing conveyors run often in stop-and-go mode	
Joined endless	Joined endless belt	**J**
Joining code	A code which describes the preparation of belt ends for ordered belts (open ended, prepared ends or joined endless)	
Linear positioning	Linear drives (actuators) which accurately position a mass or which precisely move along a predefined curve	
Mass of belt	Belt weight in kg per m; or in *lb* per *ft*	

Appendix
Glossary of terms

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

Appendix
Index

The Habasit solution

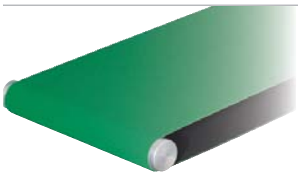
Customers first
At Habasit we understand that our success depends on your success. This is why we offer solutions, not just products; partnership, not just sales.

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

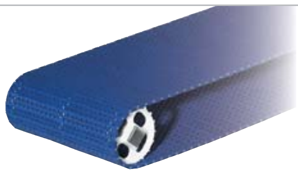
Committed to innovation
Because our customers' belting challenges and needs never cease, we consistently dedicate a substantial percentage of our employees and resources to the research and development of new products and solutions.

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.

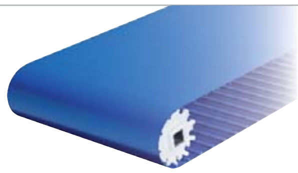
Worldwide leading product range
Habasit offers the largest selection of belting, conveying, processing and complementary products in the industry. Our response to any request is nothing less than a specific, tailor-made solution.



Fabric-based conveyor and processing belts
HabaFLOW®



Plastic modular belts
HabasitLINK®/KVP®



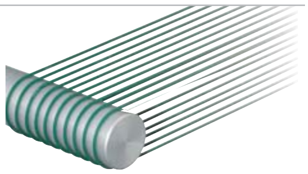
Positive drive conveyor and processing belts
Habasit Cleandrive™

Power transmission belts
HabaDRIVE®

Timing belts
HabaSYNC®

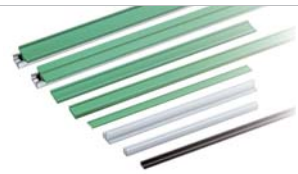
Chains (slat and conveyor chains)
HabaCHAIN®

Machine tapes



Round belts

Seamless belts



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HabiPLAST®

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