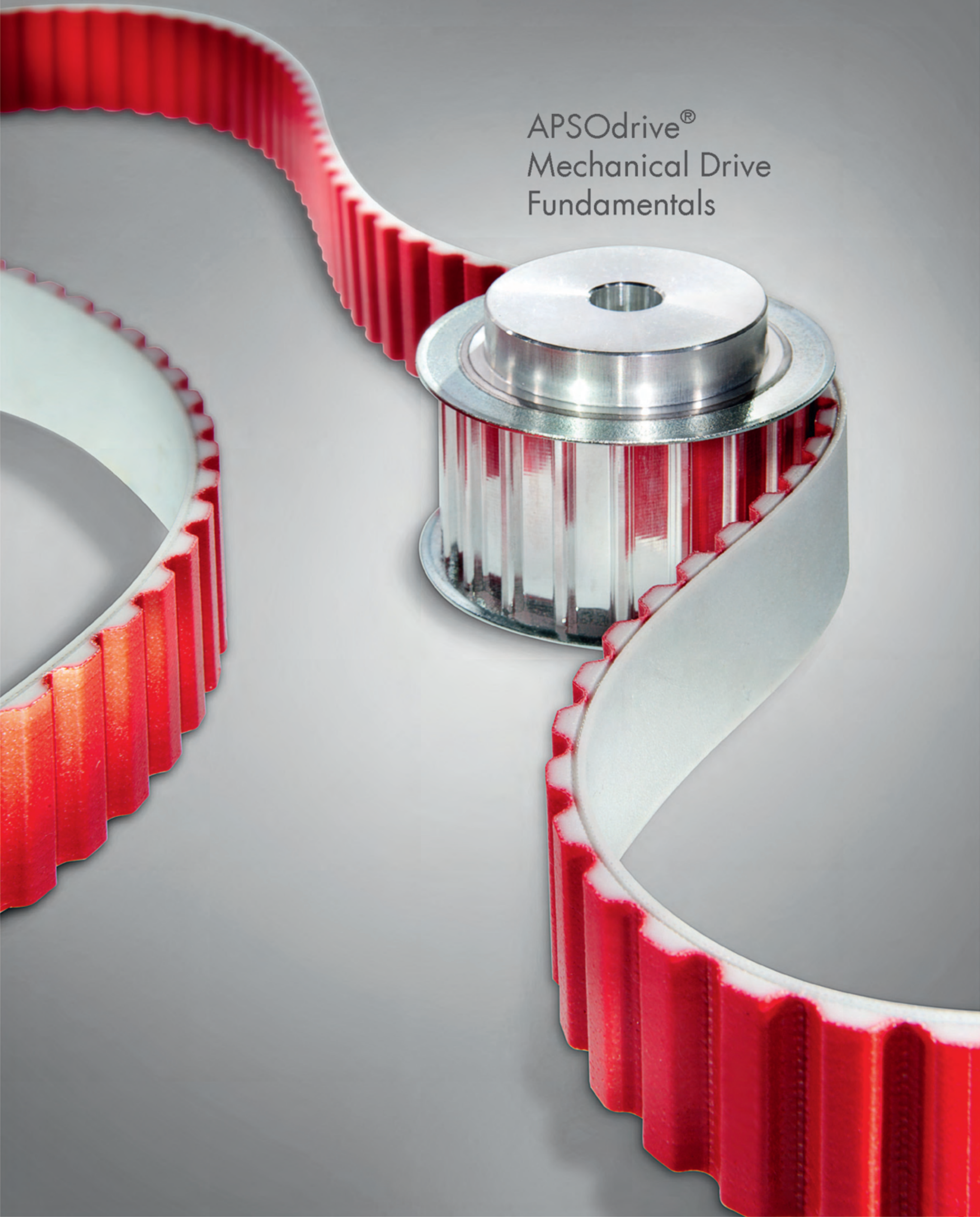
Angst+Pfister

APSOdrive®
Mechanical Drive Fundamentals



Introduction

Timing belt calculation guideline according to the TC-Calc procedure

Timing belt calculation guideline for open ended and welded belts

Characteristics, installation guidelines, tolerances

Timing belt reworking and coatings

Foreword

This Angst+Pfister drive technology manual contains an introduction to an extensive range of timing belts which are part of our stock. Nonstandard items and customized solutions can also be produced or provided swiftly. Fundamentals of calculation for belt drives, description of product properties and accessories, such as pulleys, idlers and bushings, are also included in this manual.

Traction drives

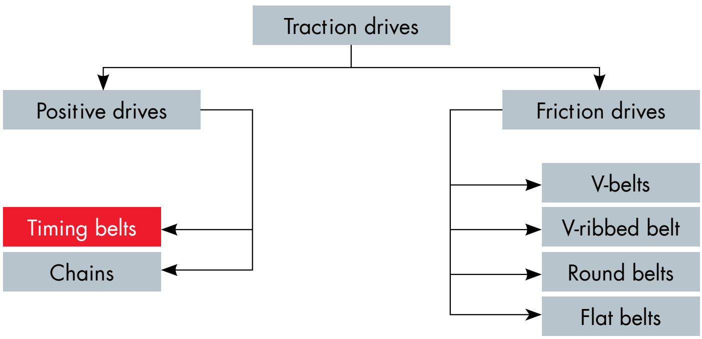
Traction drives (or commonly known as belt and chain drives) are generally used to transmit power or motion. A traction drive can also be used to move or position items, which is commonly known as transport or linear technology. Subject to the task an application has to achieve, there are several possibilities to complete the challenge. Traction drives are divided into two categories; the positive traction drives for timing belts and chains and friction traction drives for V-shaped belts as well as round and flat belts.

Positive traction drives

A positive traction drive guarantees a synchronous transmission between the pulleys, therefore it is also called synchronous drive. This kind of power transmission is gaining further importance due to its very high power ratings and striking life cycles.

Friction traction drives

Compared with positive traction drives, friction traction drives have the significant advantage of tolerating a temporary slippage due to excessive overload. It is the nature of this kind of drive that higher pre-tension forces have to be applied to ensure a flawless operation. Therefore higher bearing loads have to be accepted. Also, the belt is subject to a certain amount of constant slippage, as a result of which a perfect synchronous transmission cannot be achieved.



This manual lists many of the belts available from the Angst+Pfister Drive Technology product range.
Further information on additional components is available through your nearest Angst+Pfister sales representative.

Elastomer or polyurethane?

Timing belts are available in different materials, but the most common ones are elastomer and polyurethane. Elastomer is used as a general term for polychloroprene as well as any related elastomer compounds. The same applies to polyurethane, as different compounds from polyether or polyester are available, which are suitable for casting or extruding manufacturing processes. The commonly used abbreviation TPU stands for thermoplastic polyurethane.

Before selecting from the two materials, elastomer or polyurethane, parameters like purpose, requirements and the environment will need to be defined. All these parameters have also an impact on the reinforced tension member, which can be made of steel, glass, aramid or carbon. Any added layer on the back or tooth side of the belt needs to be considered. A solution with an elastomer belt for a power transmission is usually more economical. On the other hand, a polyurethane belt is the better solution for positioning devices.

Material properties are listed on the next two pages as well as in the belt properties.

Material properties of timing belts

Polyurethane

Standard properties

- length stability and low stretch due to steel cords
- resistant against deformation and high shear strength
- customized pulley teeth tolerances on request
- self-guiding drive belts available
- high positioning accuracy
- customized solutions available

Special properties

- various tension members available like for example for high flexibility or for high power
- made in stainless steel or aramid
- high pitch accuracy
- customized reworkings like coatings, machining or profiles (welded or screwed-on)
- special polyurethane compounds available

Elastomer

Standard properties

- good damping capabilities
- low lateral forces
- low noise emission
- low tendency for tooth skipping
- antistatic version available
- excellent price-performance ratio

Special properties

- high performance compounds
  - superior level of oil resistance
  - high temperature resistance
- PTFE refinement of the tooth fabric
- coatings

Polyurethane

Overview of standard properties

Properties	Details/additional benefits
Operating temperature	• −10°C to +80°C
Steel tension members	• precise transmission of motion • high length stability • low stretch
Shore hardness 88 to 92 ShA	• resistant to deformation and high shear strength • high resistance to abrasion
Profiles: T, AT, ATP, CTD, BAT, SFAT, V-guides, imperial profiles, HTD, RPP, STD	• narrowed gap width for reduced backlash feasible • self-guiding drive belts available
Casting, injection molding or extruding manufacturing processes	• short and long endless belts available (up to approx. 30 m) • open-end belts for open linear drives or welded transport drive belts available
Resistances	• resistant to tropical conditions • resistant to oil and gasoline • ozone resistant
Weldable with thermoplastics	• weldable up to any length • feasible to weld on cam profiles
High pitch accuracy	• for accurate positioning systems

Overview of special properties

Properties	Details/additional benefits
Operating temperature	• −30°C to +110°C
Flexible tension members	• high flexibility • pliant
Tension members in special twists	• higher rigidity • higher resistance to (reverse) bending • S/Z twist (GEN III, Brecoflex) • high pitch accuracy (Brecoflex) • low lateral running tendency
Polyamide coated teeth PAZ	• low friction • low noise emission
Polyamide coated belt back PAR	• low friction • especially for accumulating conveyors
Aramid tension members	• not magnetic • higher stretch than steel (vibration absorbing)
Stainless steel tension members	• minimized corrosion • low magnetic permeability
Various rework potential	• coatings • weld on profiles • high versatility due to screwed-on profiles (ATN): combination of different materials, easy replacement of profiles, belt lock • machining: milling, drilling/punching, water jet cutting
Coloring	• standard: white, various colors feasible
FDA conformity	• specially certified polyurethane compounds available

Elastomer

Overview of standard properties

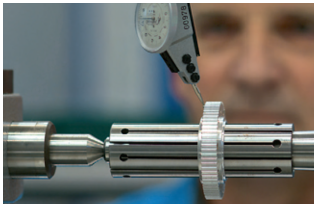
Properties	Details/additional benefits
Operating temperature	• –10°C to +100°C
Glass or aramid tension members	• excellent damping of impacts • low lateral running tendency
Shore hardness 75 to 82 ShA	• smooth running
Profiles: HTD, RPP, STD, CTD, imperial profiles	• prime meshing performance even during high-dynamic performance • smooth running • low tendency for tooth skipping
Manufactured in wide sleeves	• economically priced production
Resistances	• resistant to tropical conditions • oil resistant under certain conditions
High-strength nylon coating on teeth	• high resistance to abrasion
Antistatic	• high performance designs in accordance with ISO 9563 available
Pulleys	• wide range of standard pulleys with Taper-Lock® bushing available

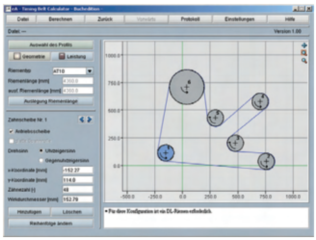
Overview of special properties

Properties	Details/additional benefits
Operating temperature	• possible up to max. +130°C
HNBR	• superior level of oil resistance
PTFE refined coating on teeth	• increased resistance to abrasion for high performance drives
Reduced noise emissions	• optimized meshing of teeth • shock absorbing material: rubber and tension members
Coatings	• vulcanized or bonded designs feasible • machined coatings available

APSOdrive® – from a standard product to a customized solution

Selecting the correct materials, components and configurations is a complex and time consuming process, but crucial for the success of a drive system. At Angst+Pfister we have more than 30 years of experience in the field of drive technology. As a customer you can benefit from this experience: APSOdrive® offers support for each individual customer to succeed with a tailor-made solution.





Engineering services: expertise all along the line
Our engineers have substantial international experience in optimizing demanding belt drives and can therefore support you with:

- technical advice for new and existing systems
- evaluating the most suitable solution
- calculating and designing mechanical drive systems
- additional use of belt drive calculation software
- commercially optimized price-performance ratio
- fast engineering and supply of customized solutions and prototypes

We trust that using standard components in combination with engineered customized parts will lead to the ultimate drive solution.

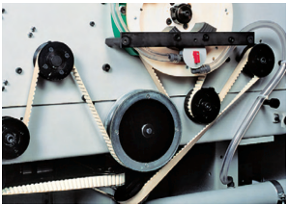
For a detailed and cost-effective calculation for your timing belt drive, we have various calculation tools available. Our technical support team will be pleased to advise you and provide you with a recommendation for the configuration and the type of belt which will suit your requirements.

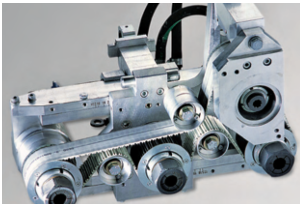
Please do not hesitate to make use of our engineers' know-how and also benefit from further application related services. Upon request, we can also organize workshops and seminars for your engineering and design team.

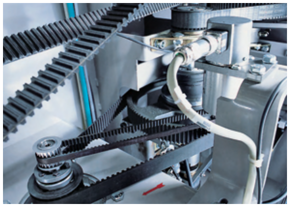
Various solutions for different applications

Whether it is a linear, transport or power transmission: we make every effort to find the most suitable and efficient solution to comply with your specific demands.

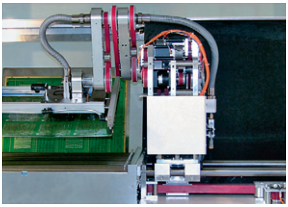
Power transmission


Saddle stitching systems

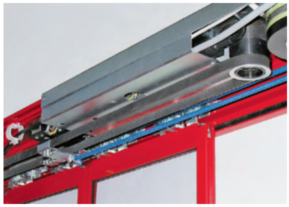

Triple spindle drilling system


Pocket spring machine

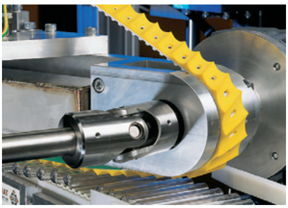
Linear drives

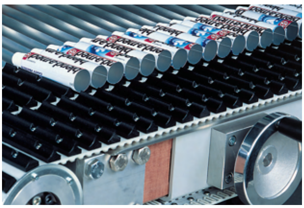

Printed circuit board transporter

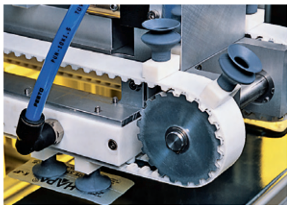

High bay rack logistics system


Automatic door system

Transport solutions


Conveying device for test tubes


Tube packaging machine

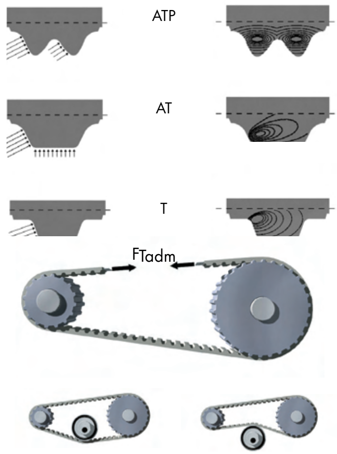

Conveying device for blister packaging

The "Teeth & Cord" (TC) calculation procedure is based on the fact that only a limited/defined number of teeth between the pulley and the belt can be in mesh at the same time. Therefore, the transmissible force/power is limited and can be calculated (calculation of tooth strength).
In order to transfer this force to a driven pulley, the timing belt needs to have adequate strength characteristics and is reinforced with cords of defined tensile strength (calculation of tensile strength of tension members). A further component to be considered in this procedure is the flexibility of the belt. This provides an important indication of the smallest pulley diameter (or belt tensioner) to be used in the belt drive.

Tooth shear strength

The shape and the material of a tooth are the two elements which define the highest force that can be transferred between the pulley and the belt. A specific tooth shear strength as a function of speed or rpm is the maximum power a tooth can bear in permanent operation. A timing belt drive is correctly designed if the transmissible power does not exceed the specific shear strength of all the teeth in mesh. An additional safety factor is usually not needed but often considered.

During the continuous and ongoing development of tooth shapes and materials, the tooth shear strength has been improved ever since. For example, an AT-Profile is larger than a T-Profile and has therefore a better distribution of the occurring forces. Furthermore, an ATP profile transmits more power than an AT profile. This is due to its optimized distribution of the transmission forces on two surfaces which results in a higher load capacity.



Drive layout with inside idler (pulsating tension)

Drive layout with backside idler (alternating tension)

Tensile strength of tension members

The circumferential force acts in proportion to the elongation of the load span; excessive slackening of the slack span is counteracted with appropriate pre-tension values. The tensile strength of the cords is the maximum allowable tensile stress of the belt, given adequate safety factors. Allowances for maximum tensile fatigue strength F_{Tadm} are listed in tables for different belts.

Flexibility

Depending on the belt model, the minimum number of teeth or diameter of the pulley must comply with the belt specification to guarantee a flawless operation. Special attention is needed for layouts with reverse bending, meaning the belt will be bent in both directions due to pulleys or idlers running on the back side of the belt. The tension members will then experience different load conditions (from pulsating to alternating). Such layouts require pulleys or rollers with a larger minimum diameter or a higher number of teeth than a layout without reverse bending.

Drive calculation

Step 1 – Evaluation of belt type
In choosing the correct belt for a drive, the field of application as well as power, rotational speed and velocity have to be considered. The smallest pulley in the whole drive needs special attention. The minimum diameter or minimum number of teeth z_{1min} will have a significant impact on the type of belt, especially for narrow drives.

P [kW]	v_{max} [m/s]	n [min^{-1}]	Field of application	Z_{1min}*	Profile
≤5	80	≤10 000	Office machinery, DIY power tools, control technology	10	T5 – XL
≤5	80	≤20 000	Small power drives, handling technology	15	AT3
≤15	80	≤10 000	Machine tools, pumps, textile machines	15	AT5
≤30	60	≤10 000	Main and auxiliary drives, machine tools, textile and printing machinery	12	T10 – L – H
≤70	60	≤10 000	Pumps, compressors, roller table drives, construction, paper and textile machinery	15	AT10 – SFAT10 – BAT10 – BATK10
≤100	60	≤10 000	Grinding machines, power drives, machine tools	15	ATP10
≤100	40	≤6 500	Heavy duty construction machinery, pumps, paper and textile machinery	15	T20 – XH
≤135	48	≤8 000	Construction machinery, pumps, compressors, paper machinery	20	SFAT15
≤140	48	≤8 000	Power drives, printing and grinding machinery	20	BAT15 – BATK15
≤160	48	≤8 000	Power drives, paper machinery, high-bay storage, hoist devices	25	ATS15
≤200	50	≤10 000	Power drives, machine tools	20	ATP15
>200	40	≤6 500	Heavy duty drives, textile and printing machinery, machine tools	18	AT20 – SFAT20

Table 1: Special timing belt designs allow the rotational speed and peripheral velocity parameters to be increased.
*Applies only to standard windings without “reverse bending” and without coating.

Step 2 – Torque
The torque is calculated from the available power. For drives which start and stop frequently, it is recommended to use the starting torque for the calculation. Starting torques for motors are usually 2.5 times higher (or sometimes more) than the rated torque.

$$M_{[Nm]} = \frac{9550 \cdot P_{[kW]}}{n_{1[min-1]}}$$

Step 3 – Circumferential force
With the known torque M and the pitch circle diameter of the driving pulley d_{01}, the circumferential force F_u can be calculated. This force must be counteracted with a correct pre-tension force to avoid a slack belt strand.

$$F_{u[N]} = \frac{2000 \cdot M_{[Nm]}}{d_{01[mm]}}$$

Step 4 – Determination of belt width
The width of the belt depends on the specific tooth shear strength F_{Tspec} which is also associated to the rotational speed. The values are listed individually by belt in the technical section. The number of teeth in mesh z_e depends on the design of the drive, but for calculation purposes only a maximum of 12 teeth can be considered to be in mesh. Excluded from this rule are some high-performance belts, which can accommodate 16 teeth in mesh (z_e is also listed in the technical section). The calculated width is usually rounded up to the upper standard belt width value.

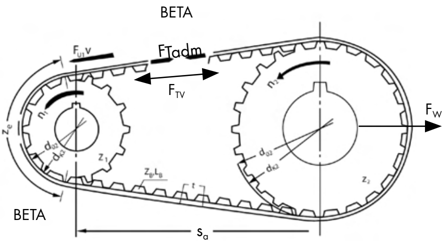
$$b_{[mm]} = \frac{10 \cdot F_{u[N]}}{z_e \cdot F_{Tspec\,[N/cm]}}$$

Step 5 – Determination of belt length
The length of a belt can only be a multiple of the chosen pitch. The pitch circle diameters d_{01} and d_{02} of both pulleys as well as the center distance s_a have to be taken into account. The calculated length L_B is rounded up to the next longer standard belt length available.

$$L_{B[mm]} \cong \frac{\pi}{2} \cdot (d_{02} - d_{01}) + 2 \cdot s_a + \frac{(d_{02} - d_{01})^2}{4 \cdot s_a}$$

By following these steps, the belt is selected for its tooth shear strength. **A further verification is now necessary for**
– tensile strength of tension members
– flexibility
– safety factors
Refer to the following chapters.

List of formulae



Determination of pre-tension force

Depending on the layout, number of teeth in mesh as well as the circumference force, the required pre-tension force in each span can now be calculated. Use the factors shown in the table to select the appropriate values for the static span force.

Configuration	Number of teeth	Pre-tension force per span
Two shaft drive	$z_B < 60$	$F_{TV} = 1/3\ F_U$
	$60 \le z_B \le 150$	$F_{TV} = 1/2\ F_U$
	$z_B > 150$	$F_{TV} = 2/3\ F_U$
Multi shaft drive	$l_{Tight\ span} \le l_{Slack\ span}$	$F_{TV} = F_U$
	$l_{Tight\ span} \le l_{Slack\ span}$	$F_{TV} > F_U$
Linear drive	all	$F_{TV} \ge F_U$

Table 2

Definition of terms

Circumferential force	F_U	[N]
Specific tooth force	F_{Tspec}	[N/cm]
Admissible tensile load	F_{Tadm}	[N]
Pre-tension force per span	F_{TV}	[N]
Static bearing load	F_W	[N]
Torque	M	[Nm]
Acceleration torque	M_B	[Nm]
Power	P	[kW]
Moment of inertia	J	[kgm²]
Density	ρ	[kg/dm³]
Velocity	v	[m/s]
Rotational speed	n	[min⁻¹]
Angular speed	ω	[s⁻¹]
Centre distance	s_a	[mm]
Belt length	L_B	[mm]
Belt width	b	[mm]
Pulley width	B	[mm]
Pulley bore diameter	d	[mm]
Pitch circle diameter	d_0	[mm]
Crown diameter	d_K	[mm]
Span length	L_T	[mm]
Pitch	t	[mm]
Arc of contact	β	[°]
Acceleration time	t_B	[s]
Number of teeth on belt	z_B	
Number of teeth if i = 1	z	
Number of teeth in mesh	z_e	
Number of teeth on small pulley	z_1	
Number of teeth on big pulley	z_2	
Ratio	i	

Basic formulae for belt configuration

Width	$b = \dfrac{10 \cdot F_U}{z_e \cdot F_{Tspec}}$	**Tooth shear strength** The belt width is calculated using the specific tooth shear strength.
Tensile strength of tension members	$F_{Tadm} \ge \dfrac{F_U}{2} + F_{TV}$	**Tensile strength of tension members** In case of too high a span force, the width of the belt needs to be increased.

Basic formulae for belt configuration

Circumferential force	$F_U = \dfrac{2 \cdot 10^3 \cdot M}{d_0}$	$F_U = \dfrac{19.1 \cdot 10^6 \cdot P}{n \cdot d_0}$	$F_U = \dfrac{10^3 \cdot P}{v}$
Torque	$M = \dfrac{d_0 \cdot F_U}{2 \cdot 10^3}$	$M = \dfrac{9.55 \cdot 10^3 \cdot P}{n}$	$M = \dfrac{d_0 \cdot P}{2 \cdot v}$
Power	$P = \dfrac{M \cdot n}{9.55 \cdot 10^3}$	$P = \dfrac{F_U \cdot d_0 \cdot n}{19.1 \cdot 10^6}$	$P = \dfrac{F_U \cdot v}{10^3}$
Belt length	$L_B = 2 \cdot s_a + \pi \cdot d_0$	$L_{B[mm]} \cong \dfrac{\pi}{2} \cdot (d_{02} + d_{01}) + 2 \cdot s_a + \dfrac{(d_{02} - d_{01})^2}{4 \cdot s_a}$	

Pitch circle diameter	$d_0 = \dfrac{z \cdot t}{\pi}$	Angular speed	$\omega = \dfrac{\pi \cdot n}{30}$
Rotational speed	$n = \dfrac{19.1 \cdot 10^3 \cdot v}{d_0}$	Circumferential speed	$v = \dfrac{d_0 \cdot n}{19.1 \cdot 10^3}$
Acceleration torque	$M_B = \dfrac{J \cdot \Delta n}{9.55 \cdot t_B}$	Moment of inertia	$J = 98.2 \cdot 10^{-15} \cdot B \cdot \rho \cdot (d_k^4 - d^4)$
Static bearing load	$F_W = 2 \cdot F_{TV} \cdot \sin \dfrac{\beta}{2}$	Ratio	$i = \dfrac{n_1}{n_2} = \dfrac{z_2}{z_1}$

Calculation example

Scope

Define a timing belt for a roller table which is used for heavy duty transportation tasks. Starting torque of the motor is 2.5 times higher than the rated operational torque.

Operating conditions are:

Given values	Power	P	=	10 kW
	Rotational speed	n	=	800 rpm
	Starting torque	M	=	2.5 times rated torque
	Ratio	i	=	1
	Number of teeth	z_1	=	z_2 = 25
	Pitch circle diameter	d_{01}	=	d_{02} = 79.58 mm
	Center distance	s_a	=	625 mm
Wanted	A suitable belt, its pitch and width.			

Solution

Step 1 – Evaluation of belt type

Based on the given values and operating conditions, an AT10 is selected from table 1 page 2.2.

Step 2 – Torque

$$M_{Nom} = \frac{9550 \cdot P}{n_1} = \frac{9550 \cdot 10\,kW}{800\,rpm} = 119\,Nm$$

Due to the start and stop function, the starting torque factor of 2.5 needs to be included in the calculation.

$$M = 2.5 \cdot M_{Nom} = 298\,Nm$$

Step 3 – Circumferential force

$$F_U = \frac{2000 \cdot M}{d_{01}} = \frac{2000 \cdot 298\,Nm}{79.58\,mm} = 7489\,N$$

Step 4 – Determination of belt width with starting torque and zero rpm (F_{Tspec} from AT10 data table)

$$b = \frac{10 \cdot F_U}{z_e \cdot F_{Tspec}} = \frac{10 \cdot 7489\,N}{12 \cdot 73.5} = 85\,mm$$

The next wider standard belt is selected b = 100 mm

Step 5 – Determination of belt length

$$L_B = 2 \cdot s_a + \pi \cdot d_{01} = 2 \times 625 + \pi \cdot 79.58 = 1500\,mm$$

Step 6 – Determination of pre-tension force

$$F_{TV} = \frac{F_U}{2} = \frac{7489\,N}{2} = 3745\,N$$

According to table 2 on page 2.3 for a two-shaft drive and150 teeth.

Step 7 – Check tensile strength of tension members (cords); F_{Tadm} from relevant AT data sheet

$$F_{Tadm} \geq \frac{F_U}{2} + F_{TV}$$

$$F_{Tadm} \geq \frac{7489\,N}{2} + 3745\,N \Rightarrow 16000 \geq 7489\,N$$

⇒ correct with enough cord safety factor

Step 8 – Check flexibility

The drive layout does not use any idler or pulley on its back side. Only alternating tension is applied to the tension members. Also the minimum number of teeth complies with the value in the AT10 data table on page 3.7.

Result

The drive is correctly dimensioned with a 100 mm wide belt. The drive should run maintenance free.

Order designation:

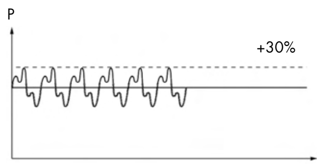
PU timing belt 100 AT10/1500

Reliability and safety

While choosing it is important to envisage the worst case scenario which can happen. That is why the values for these conditions need to be used. If the values such as teeth shear strength, tensile strength of tension members and flexibility are not exceeded, the drive will run without any maintenance.

Remarks to be considered

- Do not just use the values and ratings during operation. Attention should be given to the starting conditions. For example, a three-phase squirrel cage induction motor may produce a 2 to 2.5 times higher torque than at its operational speed – even at n = 0 rpm.
- Eventually breakaway torques as well as friction in slides have to be considered on the drive side, even at n = 0 rpm.
- Stopping or braking may cause even higher peak torques on the drive than the starting torque. Bear in mind that the torque in this case is acting in the opposite direction than during the starting phase.
- Acceleration or deceleration of inertial masses such as flywheels may have a considerable impact on the drive.
- The drive might also be subject to additional vibration and shock which have not been considered during the calculation. The sample graph on the right shows a condition where an overlaid frequency toggles +/–30% around the nominal power of the drive. Therefore the width of the belt needs to be increased by a factor of 1.3.



Speed

Apply the following safety factors for a speed increase ratio:

i = 0.66 to 1.00	S = 1.1
i = 0.40 to 0.66	S = 1.2
i = 0.46	S = 1.3

Consider in the event of a braking condition that a reverse torque occurs as well as the transmission ratio, which is changing to a speed decrease transmission.

Timing belt selection procedure

The calculation procedure LT-Calc is fundamentally focusing on the mass to be moved and the involved acceleration. As in the TC-Calc procedure, the tooth shear strength, the tensile strength of tension members and the flexibility of the belt need to be considered as well. The load in the drive is not only caused by the driving or driven pulley, but also by the forces which occur during the transport of the masses involved.

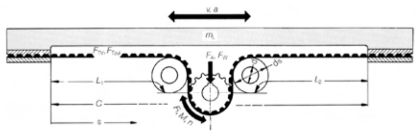
Also additional analyses need to be done which are different to the ones of a simple power drive. Properties such as positioning accuracy and eventual vibrations need to be evaluated.

The total load of a linear or transport drive involves three substantial components which need to be taken into account when calculating the maximal force on the belt:

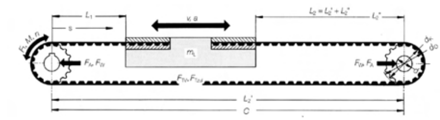
- **Acceleration force F_B**
  This is the force which is required to get all the involved masses in motion (mainly the mass to be moved, but also idler pulleys, belt etc, if their mass is significant).

- **Hoist force F_H**
  This is the required force when the motion is done against gravity. For horizontal motions $F_H = 0$.

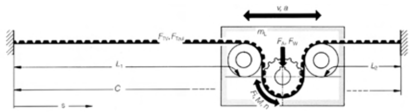
- **Friction force F_R**
  High friction forces may occur especially for transport drives where the belt runs on a support rail.



Linear table



Linear slide



Linear trolley

Design execution
All engaged assembly groups within the drive should be designed as light as possible and friction should be kept at a minimum. The surrounding structure has to be rigid. Often open-end AT and ATL timing belts are used and fixed on the linear slides by means of clamping plates. AT and ATL timing belts allow a rotational to linear translation of motion with permanent accuracy. The high pitch accuracy between timing belt and pulley results in an even load distribution on the driving pulley tooth flanks. Therefore high performance and accuracy can be achieved. The material combination between belt and pulley is exceptionally suitable for bi-directional drives. The travel distance per revolution of the driving pulley depends on the pitch and the number of teeth on the pulley. There are three common design executions for linear drives.

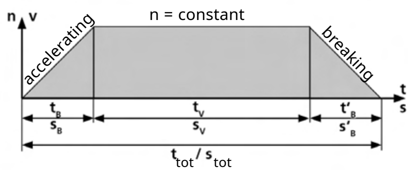
List of formulae

Used symbols		
Center distance	s_a	[mm]
Belt length	L_B	[mm]
Belt width	b	[mm]
Span length	L_1, L_2	[mm]
Pitch circle diameter	d_o	[mm]
Crown diameter	d_K	[mm]
Tension roller diameter	d_s	[mm]
Bore	d	[mm]
Useful linear distance	s_L	[mm]
Total distance of travel	s_{tot}	[mm]
Elongation	Δl	[mm]
Specific elasticity	c_{spec}	[N]
Elasticity	c	[N/mm]
Positioning deviation	Δs	[mm]
Positioning range	P_s	[mm]
Acceleration distance	s_B	[mm]
Braking distance	s'_B	[mm]
Travel distance with v = constant	s_V	[mm]
Travel time with v = constant	t_V	[s]
Overall time	t_{tot}	[s]
Acceleration time	t_B	[s]
Deceleration time	t'_B	[s]
Total distance	s_{tot}	[mm]
Number of pulley teeth	z	
Number of belt teeth	z_B	
Number of meshing teeth	z_e	
Friction force	F_R	[N]
Pitch	T	[mm]

Used symbols		
Tangential force	F_t	[N]
Acceleration force	F_B	[N]
Friction force	F_R	[N]
Hoisting force	F_H	[N]
Specific tooth force	F_{Tspec}	[N/cm]
Admissible tensile load	F_{Tadm}	[N]
Pre-tension force per span	F_{TV}	[N]
Maximum span force	F_{Tmax}	[N]
Static bearing load	F_{Sstat}	[N]
Torque	M	[Nm]
Power	P	[kW]
Mass	m	[kg]
Mass to be moved	m_{tot}	[kg]
Mass of linear slide	m_L	[kg]
Mass of timing belt	m_B	[kg]
Mass of pulley	m_Z	[kg]
Mass of idler	m_S	[kg]
Reduced pulley mass	m_{Zred}	[kg]
Reduced idler mass	m_{Sred}	[kg]
Specific belt mass	m_{Rspec}	[kg/m]
Specific weight	ρ	[kg/dm³]
Acceleration	a	[m/s²]
Gravity	g	[m/s²]
Speed	v	[m/s]
Rotational speed	n	[min⁻¹]
Angular speed	ω	[s⁻¹]
Characteristic frequency	f_e	[s⁻¹]
Excitation frequency	f_0	[s⁻¹]

Basic equations for belt definition

Tangential force	$F_t = \frac{2 \cdot 10^3 \cdot M}{d_0}$	$F_t = \frac{19.1 \cdot 10^6 \cdot P}{n \cdot d_0}$	$F_t = \frac{10^3 \cdot P}{v}$
Torque	$M = \frac{d_0 \cdot F_t}{2 \cdot 10^3}$	$M = \frac{9.55 \cdot 10^3 \cdot P}{n}$	$M = \frac{d_0 \cdot P}{2 \cdot v}$
Power	$P = \frac{M \cdot n}{9.55 \cdot 10^3}$	$P = \frac{F_t \cdot d_0 \cdot n}{19.1 \cdot 10^6}$	$P = \frac{F_t \cdot v}{10^3}$
Angular speed	$\omega = \frac{\pi \cdot n}{30}$	**Rotational speed**	$n = \frac{19.1 \cdot 10^3 \cdot v}{d_0}$
Travel time with v = constant	$t_V = \frac{s_V}{v \cdot 10^3}$	**Travel distance with v = constant**	$s_V = v \cdot t_V \cdot 10^3$
Total time with v = constant	$t_{tot} = t_B + t_V + t'_B$	**Total distance with v = constant**	$s_{tot} = s_B + s_V + s'_B$
Velocity/ Circumferential speed	$v = \frac{d_0 \cdot n}{19.1 \cdot 10^3} = \sqrt{v = \frac{2 \cdot s_B \cdot a}{1000}}$		
Acceleration time/ Breaking time	$t_B = \frac{v}{a} = \sqrt{v = \frac{2 \cdot s_B}{a \cdot 1000}}$		
Acceleration distance/ Breaking distance	$s_B = \frac{a \cdot t_B^2 \cdot 10^3}{2} = \frac{v^2 \cdot 10^3}{2 \cdot a}$		



To define the acting forces on a timing belt, all the moving and displaced masses must be considered. Therefore a reduced mass m_{Zred} of a pulley and/or tension roller is used, which is a substitute mass with equal inertia. This inertia is performing in the belt's line of action and the inertia of the rotating pulley or idler is performing on the rotational axis.

Mass of pulley	$m_z = \frac{(d_K^2 - d^2) \cdot \pi \cdot B \cdot \rho}{4 \cdot 10^6}$	**Mass of idler**	$m_s = \frac{(d_S^2 - d^2) \cdot \pi \cdot B \cdot \rho}{4 \cdot 10^6}$
Reduced mass of pulley	$m_{Zred} = \frac{m_z}{2} \cdot \left(1 + \frac{d^2}{d_K^2}\right)$	**Reduced mass of idler**	$m_{Sred} = \frac{m_s}{2} \cdot \left(1 + \frac{d^2}{d_S^2}\right)$

The static bearing load F_{Sstat} applies only in a standstill or under no load. F_{Sstat} depends on the effective circumferential force.

Static bearing load	$F_{Stat} = 2 \cdot F_{TV}$
Pitch circle diameter	$d_0 = \frac{z \cdot T}{\pi}$

Belt elongation Dl is a result of the pre-tension force F_{TV} and is spread across the whole belt length L_B. The section of the belt which is in mesh will not be stretched (see technical specification for values for c_{spec}).
The pre-tension distance for linear slide executions is only half of the belt length.

Elongation of belt	$\Delta l = \frac{F_{TV} \cdot L_B}{c_{spec}}$	**Free belt length**	$L_B = L_1 + L_2$

Linear systems have shifting spring rates which are related to the position of the slide, table or trolley. Spring rates depend on the ratio of the two lengths L_1 and L_2. The spring rate is at a minimum if L_1 and L_2 are equal.

Spring rate	$c = \frac{L_B}{L_1 \cdot L_2} \cdot c_{spec}$	**Spring rate at L_1 5 L_2**	$c_{min} = \frac{4 \cdot c_{spec}}{L_B}$

In case an external force is applied to the slide, a position deviation appears:

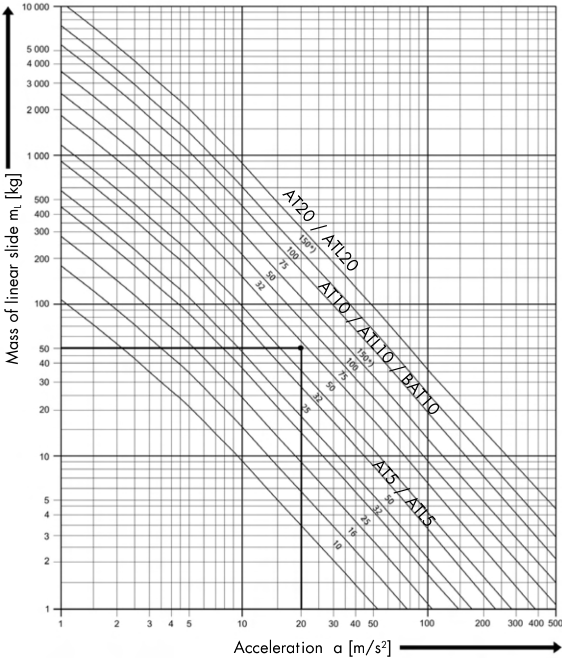
Positioning deviation	$\Delta s = \frac{F}{c}$

As a belt has a spring rate and the belt is connected to a mass, it is basically known as a spring-mass system and it is in its nature, that an impact on the system will trigger its natural oscillation. It is recommended to review the linear drive for any occurring excitation frequencies f_0 which might be in the range of the natural oscillation f_e. In case $f_e = f_0$, a design review should be considered.
Note: The natural frequency f_e of linear drives is in general much higher than any potential excitation frequency f_0 of the system, which means no resonance of the drive is to be expected. Special attention should be given when using a stepper motor as these can perform on a frequency which may cause a resonance on the belt. The countermeasure in such an event would be the use of a wider belt to alter the rigidity.

Natural oscillation	$f_e = \frac{1}{2 \cdot \pi} \cdot \sqrt{\frac{c \cdot 1000}{m_L}}$

Preliminary belt selection

Using this diagram is a fast way to find a suitable belt for a linear drive. It is only a preliminary selection and can be used as a basis for further calculations and comprehensive reviews.



Example for preliminary belt selection
Mass of linear slide m_L = 50 kg
Maximum acceleration (without deceleration) a = 20 m/s^2
Value found at the intercept point in the diagram:
Timing belt AT10/ATL10: 50 mm wide
Optional: AT20/ATL20: 32 mm wide

Recommendation
The matching driving pulley should have at least 20 teeth (for ATL10, at least 25 teeth). Should the pulley have less than 20 teeth (AT), the next wider standard belt is recommended.

Friction values

This table indicates most commonly used friction values

	Coating on teeth	Friction values μ
PUR on alumnium	–	**0.6 – 0.9**
PUR on steel	–	**0.8 – 1.3**
PUR on PTFE	–	**0.2 – 0.4**
PUR on PE-UHMW	–	**0.3 – 0.5**
PUR-PAZ on alumnium	polyamide	**0.3 – 0.4**
PUR-PAZ on steel (Rz 5 28)	polyamide	**0.3 – 0.6**
PUR-PAZ on PTFE	polyamide	**0.2 – 0.3**
PUR-PAZ on PE-UHMW	polyamide	**0.2 – 0.3**

Friction coefficients have a large tolerance; we recommend the use of a higher value. The numbers are purely indicative.

Calculation example

Task
Moving a linear slide with a mass of 50 kg. The maximal acceleration or deceleration is 20 m/s². To avoid any slackening, the belt is guided/supported by a 3 m long rail on the teeth side. Pre-tension is applied by using a movable pulley, so no idler is needed. The pulley material is AlCuMgPb (r = 2.85 kg/dm²)

Use the previously selected belt from the "Preliminary belt selection diagram"

Provided	**Value**
Mass linear slide	m_L = 50 kg
Acceleration	a = 20 m/s²
Rotational speed	n = 1500 rpm
Number of teeth	z_1 = z_2 = 30
Pitch circle diameter	d_{01} = d_{02} = 95.49 mm
Crown diameter	d_{K01} = d_{K02} = 93.67 mm
Center distance	s_a = 3500 mm
Friction	ρ = 0.5 (polyamide coated teeth on a PE guide)
Wanted	Re-calculation of the AT10, 50 mm wide belt

Solution
Step 1 – Search for all masses m_{tot} to be accelerated

Masses:

m_L $m_L = \underline{50\ kg}$

m_L $L_B = 2 \cdot s_a + \pi \cdot d_{01} = 2 \times 3500 + \pi \times 95.49 = \underline{7300\ mm}$

$$m_B = \frac{L_B}{1000} \cdot m_{Rspec} = \frac{7300\ mm}{1000} \cdot 0.29 = \underline{2.12\ kg}$$

m_{Zred}
$$m_Z = \frac{(d_K^2 - d^2) \cdot \pi \cdot B \cdot \rho}{4 \cdot 10^6} = \frac{(93.67^2 - 35^2) \cdot \pi \cdot 60 \cdot 2.85}{4 \cdot 10^6} = \underline{1.0\ kg}$$

$$m_{Zred} = \frac{m_Z}{2} \cdot \left(1 + \frac{d^2}{d_K^2}\right) = \frac{1}{2} \cdot \left(1 + \frac{35^2}{93.67^2}\right) = \underline{0.57\ kg}$$

m_{tot} $m_{tot} = m_L + m_B + m_{Zred} + m_{Sred} = 50 + 2.12 + 0.57 + 0 = \underline{\underline{52.69\ kg}}$

Step 2 – Searching for the maximal tangential force F_t

Forces:

F_B $F_B = m_{tot} \cdot a = 52.69 \cdot 20 = \underline{10\,538\ N}$

F_R Assuming that all sliding masses are supported equally. (The mass of the belt is being ignored)

$F_R = m \cdot g \cdot \mu = 50 \cdot 9.81 \cdot 0.5 = \underline{24\,525\ N}$

F_t $F_t = F_B + F_H = 10538 + 24525 = \underline{\underline{1300\ N}}$

Step 3 – Definition of pre-tension force F_{TV}

F_{TV} = $\underline{\underline{1500\ N}}$

Step 4 – Searching for the highest span force F_{Tmax}

$F_{max} = F_{TV} + F_t = 1500 + 1300 = \underline{\underline{2800\ N}}$

Step 5 – Definition of belt width

$$b = \frac{10 \cdot F_{Tmax}}{z_e \cdot F_{Tspec}} = \frac{10 \cdot 2800}{15 \cdot 44.3} = 42.14\ mm$$

b = $\underline{\underline{50\ mm}}$ (chosen belt width)

Step 6 – Review maximal permitted load on tension members F_{Tadm}

$F_{Tadm} \geq F_{Tmax}$

⇒ 8500 N ≥ 2800 N ⇒ fulfilled

Result
The drive is correctly dimensioned with a belt of 50 mm width.
The necessary power is:

$$P = \frac{F_T \cdot d_0 \cdot n}{19.1 \cdot 10^6} = \frac{130\,095.49 \cdot 1500}{19.1 \cdot 10^6} = 9.75\ kW$$

Order designation:
Open-end PU-timing belt 50 AT10/7300-PAZ-M

Characteristics of polyurethane timing belts

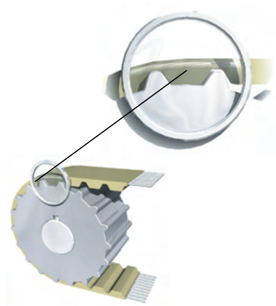
PUR timing belts, endless or open-end, are manufactured from wear resistant polyurethane and high tensile steel cord tension members. The combination of these high quality materials forms the basis for dimensionally stable and high resistance polyurethane timing belts. Polyurethane timing belts have a very high span rigidity. No post-elongation of the tension members is to be expected in continuous operation. Only under extreme load and after a short run-in time, the pre-tension of the belts might be slightly reduced by the settling of the tension members, making a single re-tensioning of the timing belt possibly necessary.

The timing belts are temperature resistant with an ambient temperature range from –30°C to +80°C. Applications close to the temperature limits (<–10°C and >+50°C), however, might require suitable dimensioning. For specific temperature ranges, various belt materials are available. Please contact the Angst+Pfister technical staff for this type of application. The production methods for timing belts are kept within tight tolerances which guarantee a uniform load distribution during power transmission. These polyurethane timing belts are suitable for the transmission of high torques as well as the precise positioning and transport of various goods.

Properties

Mechanical

- positive fit, synchronous operation
- constant length, no post fit elongation
- low noise emission
- wear resistant
- low-maintenance
- highly flexible
- positional and angular accuracy
- can be crossed (see chapter "Angular drives" on page 5.10)
- fatigue resistant, low extension steel cord tension members
- belt speed up to 80 m/s



- compact design
- excellent power-to-weight ratio
- low pre-tension
- low bearing load
- large center distances feasible
- large transmission ratios feasible
- high degree of efficiency, up to 98%

Chemical

- hydrolysis resistant
- resistant to aging
- temperature resistant from –30°C to +80°C
- resistant to tropical climate
- resistant against basic oils, greases and gasoline
- resistant to some acids and lye

For special purposes, we can produce all timing belts in materials which are appropriate for specific fields of applications and can fulfill requirements such as:

- food sector (polyurethane FDA compliant)
- low temperature range from –30°C to +5°C
- high temperature range from +20°C to +110°C
- use in a slightly aggressive environment

In addition to the standard steel cord tension members, we also offer stainless steel and aramid solutions. Should extra strong bending stress or tension load be needed, timing belts reinforced with our highly flexible E steel cord tension members can be produced.

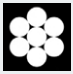
The E steel cord tension member

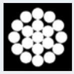
The thinner the single wire, the more flexible the entire tension member: this relation led to the development of PUR timing belts with E steel tension members.

Within the E cord, the tension is distributed more uniformly and to thinner wires, as a result the bending stress is clearly reduced in each single wire. The benefit of the E steel tension members is higher flexibility. This is an advantage for compact designs with small pulleys and idlers, where the minimum diameter or number of teeth can decrease up to 30% compared to the standard tension members.

Timing belts with E steel tension members are recommended for multi-shaft drives with alternating bending stress.

Steel tension member embedded in PUR:





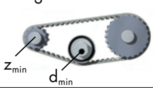
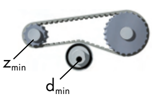
The thinner the single wire, the more flexible the entire timing belt.

Summary

- thinner single wires in the steel cord
- higher dynamic capabilities
- extremely high pulsing and alternating tensional force capabilities
- smaller pulley and idler diameters
- no correction of pulleys necessary

Remark for correct application: for applications which run on the limit of the belt's capability, please contact your nearest Angst+Pfister representative for support.

Timing belts with E steel tension members/minimum number of teeth:

Type of drive			**AT3 (Standard)**	**AT5**	**AT10 ATP10**	**T5**	**T10**	**T20**
Pulsing tension	Pulley	z_{min}	15	12	12	10	10	12
	Idler (without teeth) Running on teeth	d_{min} [mm]	20	18	50	18	50	100
Alternating tension	Pulley	z_{min}	20	20	20	12	15	22
	Idler (without teeth) Running back of the belt	d_{min} [mm]	20	50	80	18	50	120

Pre-tension

Pre-tension is intended to guarantee a minimum tensioning force at the slack span side to ensure smooth tooth meshing into the driven pulley. There are many ways of applying pre-tension to a belt, for example by adjusting the center distance between pulleys or with additional idlers.

During operation, the tension in the tight span increases while transferring the force to the driven pulley. At the same time, the tension in the slack span drops. A correct pre-tension is applied if during the maximal rated transmission of power, the belt on the slack span has just enough tension to ensure correct tooth meshing with the driven pulley.

Pre-tension should only be set as high as necessary to minimize wear on the teeth, excessive cord strain and bearing load.

Calculation of pre-tension forces
Different types of belts require different calculation procedures. The essential calculation formulae and tables are available in the calculation section.

Influence variables

Stiffness of belt
Friction forces caused by the interaction on the teeth during meshing (especially at the slack span) intensify the span forces, which increase the elongation. This may cause the teeth of the belt to climb up the teeth of the driven pulley and finally skip. Elongation is directly related to the belt stiffness; high stiffness of the steel tension members allows lower pre-tension.

Circumferential force
The circumferential force acts in proportion to the elongation of the load span, which implies excessive slackening and can be eliminated by applying a pre-tension force matching the circumferential force.

Belt length
Belt elongation due to circumferential forces and friction forces is roughly in proportion to the belt length. Therefore, the tendency of running up on teeth or skipping is basically related to the overall belt length. A short belt will only slightly stretch even under extreme circumferential and friction forces with low pre-tension force applied. Therefore, the belt barely runs up on teeth or skips. On the other hand, short timing belts can barely compensate circumferential deviations of the pulleys. This can cause heavy pre-tension variations resulting in extreme peak values.

Proportion of the span length
With multiple-shaft drives, the load span is often longer than the slack span side. In this case, a slight elongation of the load span results in a very unfavorable slack on the slack span side. Therefore, the pre-tension force of such drives should be set higher than the circumferential force.

Precise transmission of motion
If the span pre-tension forces are set equal or similar to the circumferential force, high transmission accuracy is possible in the reverse operation with PUR timing belts.

Calculation procedure

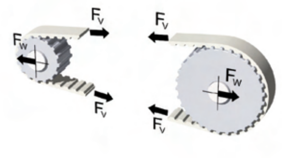
Step 1 – Selecting the type of belt Based on the mass to be moved and its acceleration, a suitable belt needs to be selected as a base for further evaluation. Find the user-friendly table on page 6.5 to select an initial type of belt.	
Step 2 – Summarizing all masses to be accelerated m_{tot} m_{tot} Summarizes all masses which will be accelerated during operation: m_L Mass of the linear table, slide or trolley to be moved m_B Mass of timing belt (see specific properties for belt mass) m_{Zred} Reduced mass of pulleys. See list of formulae for further details m_{Sred} Reduced mass of idlers. See list of formulae for further details	$m_{tot} = m_L + m_B + m_{Zred} + m_{Sred}$
Step 3 – Searching for the maximal tangential force F_t The tangential force F_t is equal to all the forces occurring on the belt. Caution: If breaking scores a higher deceleration than acceleration, use force caused by the deceleration. F_B Acceleration force F_H Hoisting force (only applies to the masses which are actually lifted) F_R Friction force (only applies to the masses which actually create forces on the belt)	$F_t = F_B + F_H + F_R$ $F_t = m_{tot} \cdot a + m \cdot g + m \cdot g \cdot \mu$
Step 4 – Definition of pre-tension force F_{TV} The pre-tension force of a linear drive is correctly applied if the maximal tangential force F_t (during acceleration and deceleration) is not causing any slack on the slack span side. Hence the minimum pre-tension force has to be at least equal to or higher than the tangential force.	$F_{TV} \geq F_t$
Step 5 – Searching for the highest span force F_{Tmax} The highest span force is expected in the load span while the pre-tension force F_{TV} is performing together with the highest (dynamic) tangential force F_t.	$F_{Tmax} = F_{TV} + F_t$
Step 6 – Definition of belt width Find the specific tooth shear strength F_{Tspec} of the belt, which is in relation with the rotational speed, in the technical chapter. The number of teeth in mesh z_e depends on the design of the drive. However, for calculation purposes, only a maximal number of 12 teeth can be taken into account (see properties in the technical chapter for z_e). Based on the result for b, the next wider standard belt is usually selected.	$b = \frac{10 \,.\, F_{Tmax}}{z_e \cdot F_{Tspec}}$
Step 7 – Review maximal permitted load on tension members F_{Tadm} The maximal permitted load on the tension members F_{Tadm} must always be higher than the maximal tangential force F_{Tmax} in the belt. A suitable safety factor must also be considered.	$F_{Tadm} \geq F_{Tmax}$

By following these steps, the belt is defined based on the tooth shear strength.

Further reviews have to be done:

- elongation
- positioning accuracy
- required power

Consequences of incorrect pre-tension



Pre-tension too low
- the teeth of the slack span side run up or skip the teeth of the driven pulley
- wear on the flanks caused by the friction force during meshing
- forced breakage by excessive elongation due to full teeth override

Excessive pre-tension
- high bearing load
- reduction of the transferable power
- wear and tear of the belt teeth

Measuring with frequency gauge
The characteristic frequency of a belt span can be measured by using a frequency meter, such as the Angst+Pfister tension meter. The pre-tension force of the span can then be calculated by using the measured characteristic frequency in the equation.

$$F_V = 4 \cdot m \cdot l_T^2 \cdot f^2 \qquad f = \sqrt{\frac{F_V}{4 \cdot m \cdot l_T^2}}$$

f:	[Hz]	Frequency
m:	[kg/m]	Mass of belt per meter
l_T:	[m]	Span length subject to vibration
F_V:	[N]	Span force

General information

Stretching

By applying pre-tension and the forces during operation, the belt will be stretched according to Hooke's law. The elongation of the belt is relative to the applied force up to the admissible tensile load F_{Tadm}. The span elongation of F_{Tadm} (see technical data) is 4 mm/m for PUR belts. For welded PUR belts it is 2 mm/m.

Design

- at least one adjustable axis is needed or, if not possible, one adjustable tension roller (not spring-loaded)
- bearings must be absolutely steady
- precise alignment of pulleys in all directions is a prerequisite

Transport/storing

- upon receipt, unpack the timing belt immediately and store in coil configuration without crimping, in a dry place at room temperature and away from direct sunlight
- do not bend or crimp during handling

Mounting

- fit timing belts loose on the pulleys without applying any force
- for fixed center distance, mount together with pulleys
- apply pre-tensioning force according to the chapter „Pre-tension"
- secure adjustable axis and tensioners against shifting or loosening
- do not clamp the timing belt between flanges on the pulley

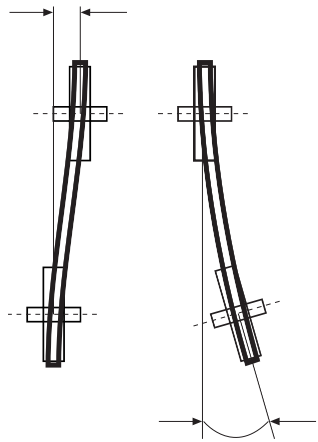
Operation

- protect the drives against dust, dirt, hot environment media as well as acids and lye
- always observe environmental temperatures
- avoid any falling object on the drive during operation

Mounting guidelines

Alignment
An immaculate alignment of the pulleys is a fundamental prerequisite for a parallel operation and a long lifespan of the belt. Extensive deviation of the parallelism between the pulleys will cause uneven distribution of tension within the belt and lateral forces will propel the belt towards the flanges on the pulley. This can cause unpleasant noise and will create heavy wear on the belt. Keeping the deviation below 0.5% of the center distance is recommended.

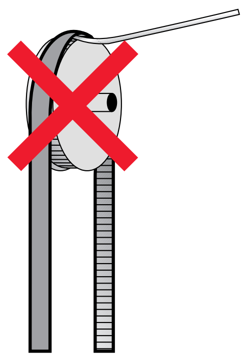
Special attention is needed for drives with extended center distances as the belt might run sideways across the pulley and run at the edge if no flanges are in place. Keeping the angular deviation between shafts below 0.25° per meter of the center distance is recommended. All shafts, pulleys and idlers must be steadily in place during operation to maintain the applied tension in the system. This is to avoid any skipping of the teeth.



Parallel deviation max. 0.5% of center distance

Angular deviation max. 0.25°/m of center distance

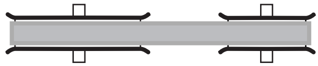
Do not use tools like tire levers and never apply high forces while mounting a belt. Shift the idler or movable pulley in such a way that it is easy to place the belt on the drive. ISO 155 provides approximate values for minimum distance required for adjustable pulleys so that a belt can be fitted on. Using force or tools while mounting a belt can initiate damages which are usually not visible but will reduce its lifespan.



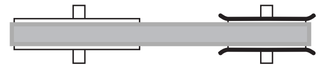
Flanges and idlers

Flanges

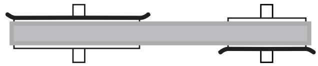
Flanges secure the belt from running laterally off the drive. Usually only the smaller pulley is equipped with flanges. Using just one flange on each pulley on opposite sides will also be suitable. Using two flanges is also possible and is often used for horizontally oriented drives. Our technical staff is at your disposal for any support needed.



Flanges on each side of both pulleys



Flanges on both sides on small pulley only



One flange per pulley on opposite sides

Idlers

Idlers are not meant to transmit any power, but to apply the required pre-tension on the drive. As tensioners are additional parts within a drive, they will also create further bending stress on the belt which reduces the lifespan. They should be made redundant whenever feasible. Idlers can be used on both sides of the belt.

Inside idler (tooth side)

Inside idlers are more favorable to the outside idlers because they create only additional pulsing tension on the tension members. As they run on the teeth of the belt, the use of a pulley is recommended instead of a flat roller. Flat rollers can be used too, but the outside diameter should be 2.5 to 3 times larger than the belt's specific minimum diameter for pulleys. These idlers should be placed relatively close to the larger pulley to minimize the reduction of the arc of contact on the smaller pulley.

Outside idler (back side)

Outside idlers create an additional and alternating bending on the tension members as they run on the back of the belt. Idlers which run on the back of the belt use flat rollers only and the diameter should be at least 1.5 times larger than the belt's specific minimum diameter for pulleys. Outside idlers should be placed closely to the smaller pulley which will then also increase the arc of contact on the smaller pulley.

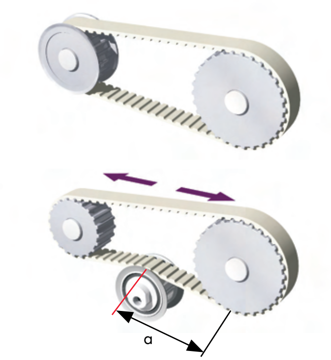
Deflection pulleys and rollers

The same rules apply to deflection rollers as they do for idlers.

Timing belt guidelines

Timing belts must be guided against the tendency to travel laterally (sideways) off the pulley. This is usually prevented by adding flanges to the pulleys. By fitting suitable guiding features, lateral forces and friction can be reduced. This can be achieved by:

- adding a guide at the end of a large free span (the length (a) of the guide should be at least 5 times the belt width);
- guidance on the driving pulley (preferably for two shaft drives with short center distance);
- guidance on pulleys with low power transmission (preferably for multi shaft drives);
- guidance on the idlers
  - located on the slack span side
  - if located on the back of the belt: consider minimum diameter due to high bending
  - located on the teeth side: at least 3 teeth in mesh
  - drives with changing rotational direction preferable in the center of the span
  - span length (a) between tension roller and pulley should be at least 5 times the belt width
- To achieve the best guiding performance all flanges and guides need to be aligned within tight tolerances. All shafts have to be precisely installed with accurate parallelism.
- It is possible to add flanges on the smaller pulley in order to optimize costs as long as the functional reliability is not impaired.



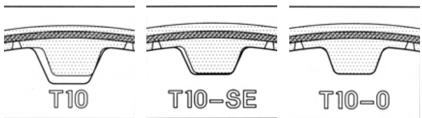
Tooth gap shoes

Timing belts are form-fitting drive elements. They work without any slippage with the corresponding synchronous pulleys. PUR timing belt drives can be improved for applications when a reduced backlash performance is required.

The standard play between the tooth on the belt and the gap on the pulley between the teeth can be reduced (SE gap) or even eliminated (Zero gap). This is usually required for precise applications. Please contact your nearest Angst+Pfister representative for technical advice.

- prerequisites for the application: matching pitch between timing belt and pulley
- influencing factors for pitch matching:
  - pre-tension force
  - meshing distance (z_e)
  - load rate (rotational speed, dynamic behavior...)
  - manufacturing tolerances

Tooth gap shapes on a T10 profile



Standard gap | SE gap (low backlash) | Zero gap (zero backlash)

Angular drives

With PUR timing belts angular drives can be designed, but they can only be twisted around the span axis, which creates additional tension in the belt. Tension members are therefore also subject to different force values.

By using a belt width/span length ratio $l_T/b \geq 20$, the drive does not require any special precaution to be taken during design and no limitation in performance is to be expected.

$l_T/b \geq 20$
b = belt width
l_T = span length

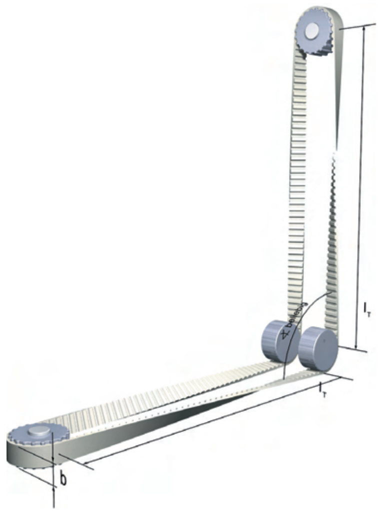


Table of tolerances for BRECOFLEX® timing belts

Length tolerances for BRECOFLEX® timing belts

Stated dimensions in mm, referred to the belt length

Belt length [mm] up to	Length tolerance [mm]
300	± 0.41
500	± 0.53
700	± 0.64
900	± 0.75
1100	± 0.85
1300	± 0.95
1500	± 1.04
1900	± 1.13
2120	± 1.22
2240	± 1.31
2360	± 1.36
2500	± 1.44
2650	± 1.49
2800	± 1.57
3000	± 1.61
3150	± 1.74
3350	± 1.82
3550	± 1.91
3750	± 2.03
4000	± 2.11
4250	± 2.24
4500	± 2.32
4750	± 2.40
5000	± 2.52
5300	± 2.64
5600	± 2.72
6000	± 2.92
6300	± 3.04
6700	± 3.19
7100	± 3.35
7500	± 3.51
8000	± 3.70
9000	± 4.09

Length tolerance for BRECO® timing belts M/V (except for ATL timing belts)	± 0.8 mm/m

Width tolerances for BRECOFLEX® and BRECO® timing belts M/V

Belt type pitch	Tolerance
T2.5	± 0.5
T5 / TK5	± 0.5
T10 / TK10	± 0.5
T20	± 1.0
AT3	± 0.5
AT5 / ATK5 / ATL5	± 0.5
AT10 / ATK10 / ATL10 / ATN10 / SFAT10 / BAT10 / BATK10	± 0.5
ATN12.7	± 0.5
ATS15 / SFAT15 / BAT15 / BATK15	± 1.0
AT20 / ATK20 / ATL20 / ATN20 / SFAT20	± 1.0
ATP10	± 0.5
ATP15	± 1.0
XL	± 0.5
L	± 0.5
H	± 0.5
XH	± 1.0

Table of tolerances for CONTI® SYNCHROFLEX timing belts

Nominal height and height tolerances for CONTI® SYNCHROFLEX timing belts

Type	Nominal height [mm]	Height tolerances [mm]
T2	1.1	± 0.15
T2.5	1.3	± 0.15
T2.5-DL	2.0	± 0.20
T5	2.2	± 0.15
T5-DL	3.4	± 0.20
T10	4.5	± 0.30
T10-DL	7.0	± 0.40
T20	8.0	± 0.45
T20-DL	13.0	± 0.60
AT3	1.9	± 0.15
AT5	2.7	± 0.15
AT10	5.0	± 0.30
ATP10	5.0	± 0.30
AT20	9.0	± 0.45

Length tolerances for standard CONTI® SYNCHROFLEX timing belts

Belt length measurement is carried out according to DIN 7721, in relation to the center distance.

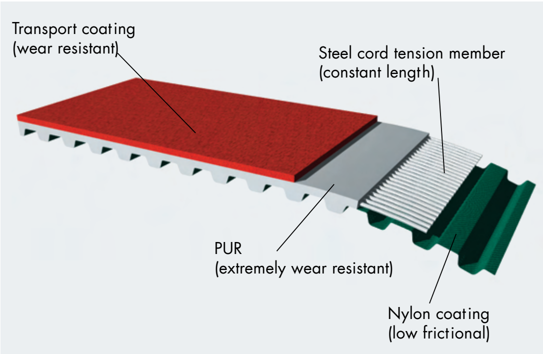
Belt length [mm]		Length tolerance in relation to center distance [mm]
over	up to	
	320	± 0.15
320	630	± 0.18
630	1000	± 0.25
1000	1960	± 0.40
1960	3500	± 0.50
3500	4500	± 0.80
4500	6000	± 1.20

Width tolerances for standard CONTI® SYNCHROFLEX polyurethane timing belts

Type/Group	Width tolerance		
	up to 50 mm [mm]	50 to 100 mm [mm]	Over 100 mm [in % of belt width]
K1	± 0.3	± 0.5	± 0.5
K1.5	± 0.3	± 0.5	± 0.5
T2	± 0.3	± 0.5	± 0.5
M (MXL)	± 0.3	± 0.5	± 0.5
T2.5	± 0.3	± 0.5	± 0.5
T5	± 0.3	± 0.5	± 0.5
T5-DL	± 0.3	± 0.5	± 0.5
T10	± 0.5	± 0.5	± 0.5
T10-DL	± 0.5	± 0.5	± 0.5
T20	± 1.0	± 1.0	± 1.0
T20-DL	± 1.0	± 1.0	± 1.0
AT3	± 0.3	± 0.5	± 0.5
AT5	± 0.5	± 0.5	± 0.5
AT10	± 1.0	± 1.0	± 1.0
ATP10/ATP15	± 1.0	± 1.0	± 1.0
AT20	± 1.0	± 1.0	± 1.0

Remarks: Tighter tolerances according to special data are possible.
Tolerance for special tension members upon request.

Introduction



Timing belt construction

BRECO® and BRECOFLEX® timing belts consist of wear resistant polyurethane (PUR) and high tensile steel cords. The coating options of the timing belts provide a variety of application possibilities in transport technology.

Correct coating selection depends on the properties of the conveyed item and the required grip. Main factors for an efficient transport application are:

- high friction for non-slip conveyance
- soft or hard coatings depending on characteristics of transported material
- low friction to reduce drag (PAZ/PAR)

Every material involved behaves according to its specific property.

To meet specific transport applications, the tooth side and/or the transport side can be mechanically re-worked. In this manner, the flexibility of the entire belt can be maintained by making incisions in thick coa-tings.

Resistance
Depending on the application, the resistance of each coating material is to be viewed separately. The material resistance depends, amongst others, on the pH value, the concentration, the temperature and the influencing time of the medium. Simple oils generally have no damaging effect on the belt. Additives in the oil and temperatures above approximately 40°C can reduce longevity.

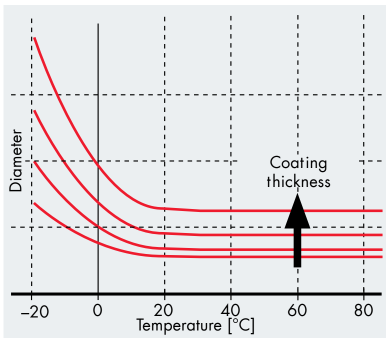
Friction
The friction of the belt on a sliding guide generates heat. This increases with the weight of the items to be transported. The guide material must be selected such that the friction of the transport belt in contact with it, results in a minimum value. The guide should guarantee good heat dissipation under high pressure forces.

The friction value changes with temperature. It increases as the temperature rises and diminishes at temperatures below zero (frost).

Information
Ask for advice on coatings over 75 mm wide and approximately 2 mm thick, due to different processing properties.

Drives with reverse bending
Coated timing belts are generally suitable for drives with reverse bending. In such applications, belts with very soft coatings (e.g., Sylomer) should be installed with reduced pretensioning. Coatings that are manufactured based on natural rubber, such as Linatex, can be used for reverse bending applications only to a limited extent. Please consult our technical staff for further information.

Temperature effect/synchronising pulley diameter
When transporting hot goods (above approx. 80°C) it must be ensured that the duration of contact is as short as possible, to avoid heating the belt substructure to above 80°C. For a short period of time, a coated belt can withstand higher thermal stress, as long as sufficient cooling is provided in the remaining cycle period. For temperatures above approximately 60°C the tooth shear strength reduces slightly. An additional safety measure is only needed if the teeth are subjected to major stress. At low ambient temperatures, the flexibility of the coating reduces. Larger diameters for the timing pulleys should therefore be selected compared to normal temperature conditions (see diagram). The flexibility of the timing belt also reduces at low temperatures. The minimum diameters serve as a guideline. They apply at an ambient temperature of 20°C and linear speed of 1 m/s, also assuming a low burden from the transported goods. If the exact application details are known, it is possible to reduce the diameters. The minimum pulley diameters shown in the following tablesfor the different coatings, apply for homogeneous coatings with an even thickness. Interruptions in the coating, e.g., due to cuts or grooves, cause significant notch effects and require considerably higher minimum diameters.



Timing pulley diameter related to the temperature

Mechanical reworking

PU timing belts can be mechanically processed to obtain specific functional characteristics. Timing belts with thicker than standard backs offer a broad range of possibilities for engineers and are available also for mechanical processing.

Available versions:
- version T
- version DR
- coated timing belt

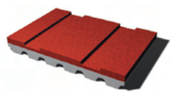
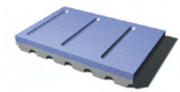
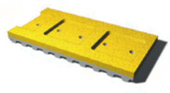
Please note that timing belts with thicker backs are less flexible and require pulleys with larger diameters. Better flexibility is achieved through transverse grooves or slits. Perforated PU timing belts are used in vacuum transport systems. The preferred version of these timing belts is manufactured with cord free zones. Flex timing belts are also available for this purpose.

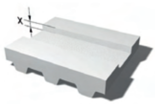
Backside cross milling
Cross grooves on the belt back enhance the flexibility of the belt. Milled grooves are, in as much as they are possible from the technical feasibility point of view, used to improve safe loading and secure positioning of the products on the belt.

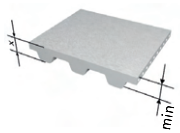
Backside longitudinal milling
Independent of the belt pitch, the belt back shaping offers a wide range of design variants for customised solutions. In this manner, belt guiding can be achieved by a trapezoidal back profile, or a round section supported and moved by means of a prism shaped cross section. Dimensions are to be indicated as depth measure x in relation to the belt back.

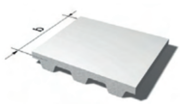
Backside grinding
The backs of all BRECOFLEX® timing belts are grinded as standard. For reasons of precision or in order to obtain a roughened surface, all other timing belts of the BRECO® range can also be grinded.

Grinded belt edges
Narrower tolerances in the belt width can be achieved by grinding the belt edges. The edges may need to be grinded especially for BRECO® timing belts guided by rails.







Removal of teeth

The removal of individual teeth or entire groups of teeth is possible and should be done for accurate interlinking purposes, for example if the remaining teeth are used to position the transported goods in a specific location.

Milling of teeth lengthwise

Timing belts with tooth profiles milled lengthwise are often used in combination with cord-free zones in vacuum transport systems.

Perforation of timing belts

The use of perforated timing belts is preferred for areas without tension members (to a limited degree also available as Flex timing belts) and areas with teeth removed in the longitudinal direction, for vacuum applications. The multitude of design possibilities for timing belts in the field of vacuum applications, ranges from the transport of delicate films up to sheet bars of several square meters in size.

Mechanical processing

Coated timing belts can be mechanically reworked for special functional characteristics, depending on the properties of the coating. Transport belts with thick coatings are less flexible. Their use therefore requires a larger diameter of toothed pulley. Transverse slits or grooves can increase the flexibility of the coating. Where technically possible from a production perspective, milled grooves are used for secure handling and better positioning of products. Perforated timing belts are used in vacuum transport systems. Flex timing belts are also available for this purpose. The preferred version of timing belts is manufactured with cord-free zones. The teeth are milled accordingly.

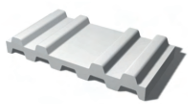
Water jet cutting

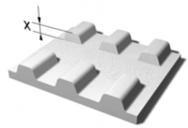
- precise
- fast
- clean
- variety of uses
- environmentally friendly

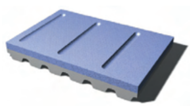
In addition to milling, drilling, stamping and grinding, timing belts can also be reworked with a water jet cutting machine. Water jet cutting offers a wide range of possibilities. A variety of cut-out contours can be obtained with high precision for special purposes. The process is also suitable for cutting flight shapes from pre-assembled polyurethane plates of different thicknesses.

Benefits

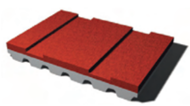
- precise cutting edges
- high cutting accuracy
- very low heat generation and no warping
- no burrs
- hardly any post-processing required



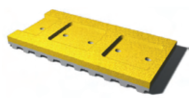




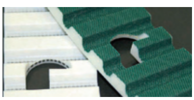
Sylomer – blue (groove milled)



Linatex (Cross milled)



PU – yellow (square milled with bore holes)

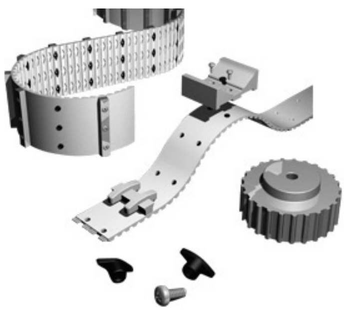


Description

The ATN timing belt is especially designed for applications in transport technology. The exchangeable profile fastening system in the belt tooth permits fast fitting and replacement of the flights individually manufactured for the specific conveying application. This flexibility provides a great variety of application possibilities, not to be realised up to now, compared to other profile fastening systems, as e.g. welding. If required, it is possible to convey different types of goods in one transport system using the same timing belt, but equipped with different profiles.

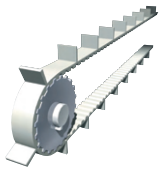
Advantages

- the belt is part of a modular design consisting of the ATN timing belt, fastening elements, the ATN timing belt lock and flights/profiles
- variable profile pitches with high accuracy
- application of various profile materials is possible (plastics, metal, ceramics...)
- high shearing forces
- fast and easy profile change when the products to be transported are changed or the profiles are worn
- no belt deinstallation for profile changes
- alternative to a chain with all advantages of a timing belt
- self-alignment of the profiles during installation
- application of standard pulleys
- high visual quality
- various fastening possibilities
- cost effective for the user:
  – standard belt with a high availability and variability
  – short machine shut-down times for profile changes
  – low test costs because of interchangeability of profiles (prototypes)

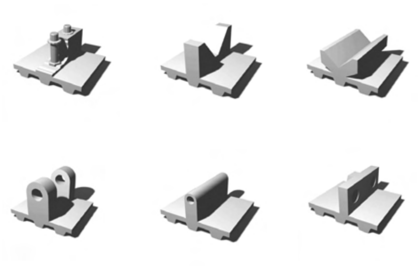


Applying profiles on belts

Welded on profiles
For whichever transport purpose timing belts are used, the belt can be fitted with any number and sequence of welded-on profiles.

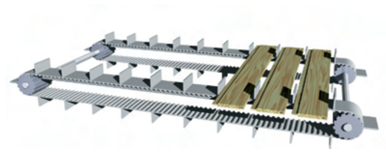


Profile selection
The profile is made out of polyurethane, the same high-quality compound as the timing belt. Depending on the transport requirement, the design can be customized according to the customer's demand. Therefore, an existing profile from our extensive stock can be used, or if needed, a semi-finished profile will be reworked accordingly. For exeptional requests and appropriate number of pieces, it is possible to manufacture new molds to achieve the required solution.



Approach
Belt length and pulley diameter are the basis for the drive selection, based on the machine configuration. Many belt types from our manufacturing range can be equipped with flights/profiles. Timing belts together with guiding surfaces allow a reduced friction operation. PAZ version timing belts are also available to further reduce friction coefficient.

Profile selection
The material to be transported and the transport purpose influence the selection of the flight.



Over 3000 standard profiles
Profiles are manufactured as polyurethane molded part. Standard profiles are available. Depending on their dimensions, standard profiles can be reworked by mechanical processes (drilling, milling). If necessary, explain design requirements by means of a drawing.

Profiles of sheet material
Depending on the quantity, flights will possibly be cut from pre-fabricated PUR sheets. The following board thicknesses are available: 1.5; 2; 3; 4; 5; 6; 7; 8; 10; 11; 15; 20 mm

Profiles from new tools
Within the framework of our production possibilities, there are practically no limitations for new design requirements as far as the shape of injection molded flights are concerned. Costs for tools and molds might apply.

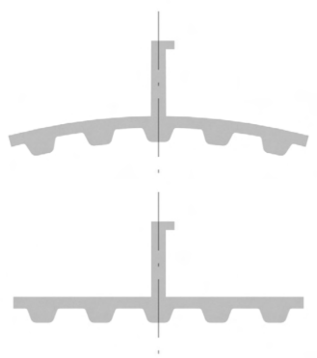
Profile compound
The profiles consist of polyurethane, the same high-quality material as the timing belts themselves.

Profile position opposite tooth
The belt flexibility of timing belts is located mainly in the tooth gap area. To retain the timing belt flexibility around the pulley, the profile position "opposite the tooth" is to be preferred.

Profile pitch, tooth pitch
We recommend to select a profile pitch which is an integral multiple of that of the tooth. Profile pitches other than the integral multiple of the tooth pitch can be supplied, it has, however, to be noted, that a uniform offset of the profile position in relation to the tooth position will accumulate.

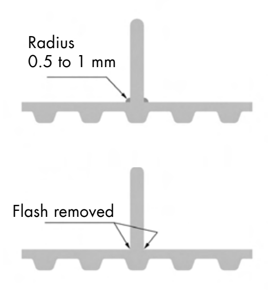
Tolerances
The reached profile position of each individual profile is ± 0.5 mm of the intended set point position. A tolerance of ± 0.5 mm is to be taken into account for the profile height.

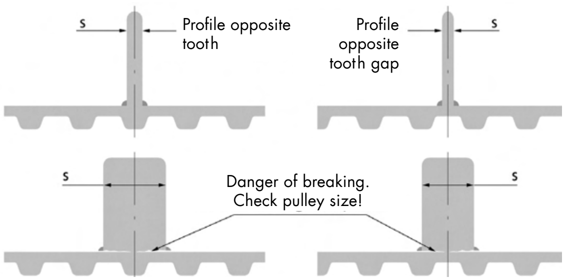
Ordering example
For the required timing belt with profiles the order should preferably be accompanied by a dimensional drawing. The timing belt with profiles can also be defined and transmitted by the order text. Example: Timing belt 50 T 10/5000 V-PAZ with welded-on profile, profile no. 2.3.2.015.008, number of profiles 100, profile pitch 50, profile position opposite the tooth.



Welding Flash

A flash builds up between flight and back of the belt. A polyurethane overhang with a 0.5 to 1 mm radius could form. Should the flash impair the intended function, ask for "flash removed" in your order information.





Profile thickness s

The timing belt flexibility can be influenced by the welded-on flight. Note as a rule that the flight thickness s is to be selected as thin as possible. The table below shows the individually recommended maximum profile thickness s [mm] in relation to the selected number of pulley teeth.

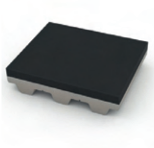
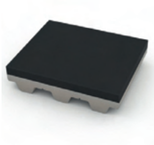
	Max. profile thickness [mm] when welded-on position is opposite tooth							**Max. profile thickness [mm] when welded-on position is opposite tooth gap**						
Number of teeth on pulley	**20**	**25**	**30**	**40**	**50**	**60**	**100**	**20**	**25**	**30**	**40**	**50**	**60**	**100**
T2.5	2.5	3	3	4	4.5	5	6	1.5	1.5	2	2	3	4	6
AT3	3	4	4	5	6	6.5	8	1.5	1.5	2	3	4	5	7
AT5/T5	5	6	6	8	9	10	12	2	2	3	4	6	8	10
AT10/T10	8	9	10	12	14	15	20	3	4	4	6	9	12	20
AT20/T20	12	13	15	18	20	23	30	5	5	6	8	12	20	30
MXL	2	2.5	2.5	3.5	4	4.5	5	1	1	1.5	1.5	2	3	5
XL	5	6	6	8	9	10	12	2	2	3	4	6	8	10
L	6	7	8	10	12	13	16	3	3	4	5	7	10	16
H	8	9	10	12	14	15	20	4	5	6	7	10	12	20
XH	13	14	15	18	20	23	30	5	5	6	8	12	20	30

Example for the calculation of the profile thickness s for a timing belt with pitch T10, which is running around a pulley with 20 teeth:

- when the profile position is «opposite the tooth», profile thickness $s \le 8$ mm
- when the profile position is «opposite the tooth gap», profile thickness $s \le 3$ mm

Remark: We recommend to select the next smaller size as profile thickness when there are intermediate sizes (e.g., 22 teeth).

Timing belt coatings

No.	Product	Color	Hardness	Working temperature	Toler. (timing belt + coating)	Degree of grip	Abrasion resistance
1	Linatex HM 	red	38 Shore A	–40°C to +70°C	1/+1.8 mm	Medium-high	Medium-low
2	Linard 60 	red	60 Shore A	–20°C to +110°C	1/+1.8 mm	Medium-high	Medium-low
3	Linatrile 	orange	55 Shore A	–20°C to +110°C	1/+1.8 mm	Medium-high	Medium
4	Linagard OZ 	black	39 Shore A	–40°C to +75°C	–	Medium-high	Medium-low
5	Linaplus FG FDA (Natural rubber) 	white	38 Shore A	–40°C to +70°C	1/+1.8 mm	Medium-high	Medium-low
6	NBR 65/EPDM 	black	65 Shore A	–35°C to +70°C	±0.6 mm	Medium-high	Medium

(thk) Available thickness/ (Ø) minimum pulley diameter [mm]

1 Linatex HM

thk	2	3	4	5	6	8	10
Ø	60	80	80	80	100	100	120

2 Linard 60

thk	3	6	12	20
Ø	60	60	120	120

3 Linatrile

thk	1.5	3	5	6	10
Ø	40	60	60	80	100

4 Linagard OZ

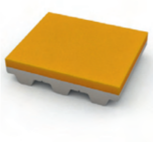
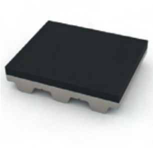
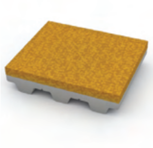
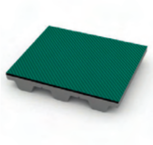
thk	1.5	2	3	5	6	8	10	12
Ø	40	40	60	60	80	100	100	120

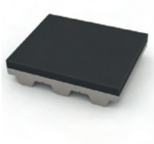
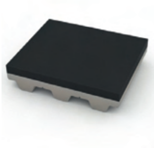
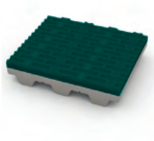
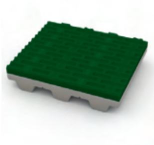
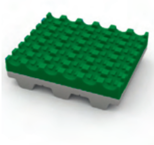
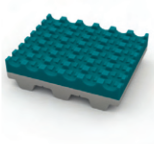
5 Linaplus FG FDA (Natural rubber)

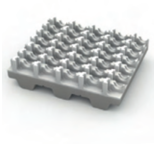
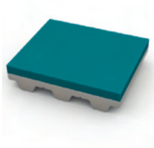
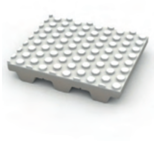
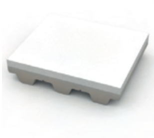
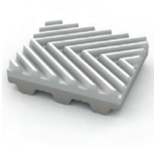
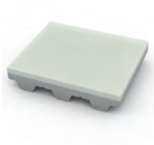
thk	1.5	2	3	5	6	8	10	12
Ø	40	40	60	60	80	100	100	120

6 NBR 65/EPDM

thk	1	2	3	4	5	6	8	10	12	15
Ø	60	60	80	80	80	100	100	100	120	130

No.	Product	Color	Degree of grip	Hardness	Abrasion resistance	Working temperature	Toler. (timing belt + coating)	(thk) Available thickness/ (Ø) minimum pulley diameter [mm]
7	NBR 60 white FDA 	white	Medium-high	60 Shore A	Medium	–30°C to +80°C	–	thk 1, 2, 3, 4, 5, 6, 8, 10 / Ø 60, 60, 80, 80, 80, 100, 100, 100
8	RP400 	yellow	Medium-high	39 Shore A	Medium	–10°C to +120°C	±0.7 mm	thk 2, 3, 4, 5, 6 / Ø 40, 50, 50, 70, 70
9	CM280 	black	Medium-high	175 kg/m³	Medium-low	–50°C to +95°C	–	thk 2, 3, 4, 5, 6, 7, 8, 9, 10 / Ø 60, 60, 80, 80, 80, 100, 100, 100, 100
10	RG250 	orange	Medium-high	160 kg/m³	Low	–40°C to +80°C	–	thk 10, 15 / Ø 120, 150
11	Hamid 	Top layer green, bottom layer black	Medium-high	65 Shore A	Medium-high	–30°C to +60°C	±0.5 mm	thk 1.4 / Ø 20
12	Correx 	light brown	Medium-high	36 Shore A	Medium	–15°C to +70°C	±0.7 mm	thk 6, 10 / Ø 80, 120

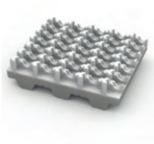
No.	Product	Color	Hardness	Working temperature	Toler. (timing belt + coating)
		Degree of grip	**Abrasion resistance**	**(thk) Available thickness/ (Ø) minimum pulley diameter [mm]**	
13	Porol	black	180 kg/m³	–40°C to +75°C	±0.7 mm
		High	Low	thk: 3 | 5 | 10 Ø: 40 | 60 | 80	
14	Viton	black	75 Shore A	–10°C to +190°C	±0.6 mm
		High	Medium-high	thk: 2 | 4 Ø: 80 | 100	
15	MiniGrip Blue	blue	50 Shore A	–15°C to +90°C	±0.5 mm
		High	Medium	thk: 1.5 Ø: 30	
16	MiniGrip green	green	65 Shore A	–10°C to +110°C	±0.5 mm
		High	Medium	thk: 1.5 Ø: 30	
17	SuperGrip green	green	40 Shore A	–15°C to +90°C	±0.5 mm
		High	Medium	thk: 4 Ø: 60	
18	SuperGrip blue	blue	40 Shore A	–15°C to +90°C	±0.5 mm
		High	Medium	thk: 4 Ø: 60	

No.	Product	Color	Hardness	Working temperature	Toler. (timing belt + coating)	Degree of grip	Abrasion resistance	(thk) Available thickness / (Ø) minimum pulley diameter [mm]
19	Supergrip white FDA 	white	55 Shore A	–15°C to +90°C	±0.5 mm	High	Medium	thk 4; Ø 60
20	PVC film blue 	blue	65 Shore A	–15°C to +90°C	±0.5 mm	High	Medium-high	thk 1, on request (2; 3; 4; 5; 6); Ø 30
21	PVC dots white FDA 	white	60 Shore A	–10°C to +110°C	±0.5 mm	High	Medium	thk 1.5; Ø 60
22	PVC film white FDA 	white	48 Shore A	–10°C to +110°C	±0.5 mm	High	Medium	thk 1.5, on request (1; 3 ;4 ;5 ;6); Ø 40
23	PVC herringbone FDA 	white	65 Shore A	–10°C to +110°C	±0.5 mm	High	Medium	thk 3; Ø 60
24	T-version (extruded) PU thick back 	transparent	85 Shore A	–20°C to +80°C	±0.5 mm	Medium-low	High	thk 1.5 (for 5 mm pitch), 2 (rest); Ø 80, 80

No.	Product	Color / Degree of grip	Hardness / Abrasion resistance	Working temperature	Toler. (timing belt + coating)
25	PU 385	transparent / Medium-low	85 Shore A / High	–20°C to +80°C	±0.4 mm
26	PU 60	transparent / Medium-low	60 Shore A / High	–20°C to +80°C	±0.4 mm
27	HV film	Transparent glossy / Medium-low	85 Shore A / High	–20°C to +80°C	±0.4 mm
28	HV film FDA	Transparent glossy / Medium-low	85 Shore A / High	–20°C to +80°C	±0.4 mm
29	T-groove TR1 & TR2 - PU with longitudinal grooves	transparent / Medium-low	85 Shore A / High	–20°C to +80°C	±0.4 mm
30	WM 385	transparent / Medium-low	85 Shore A / High	–20°C to +80°C	±0.4 mm

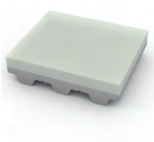
(thk) Available thickness/ (Ø) minimum pulley diameter [mm]

25 PU 385



thk	2	3	4	5	6
Ø	80	80	120	150	180

26 PU 60

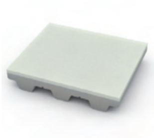


thk	2	3	4	5	6
Ø	80	80	120	150	180

27 HV film

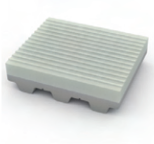
thk	1	2
Ø	60	80

28 HV film FDA



thk	1	2
Ø	60	80

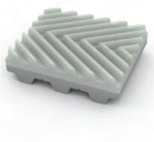
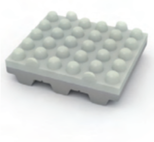
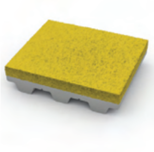
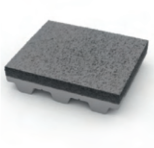
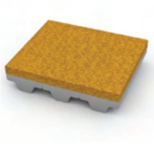
29 T-groove TR1 & TR2 - PU with longitudinal grooves

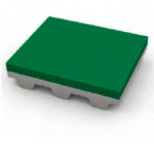
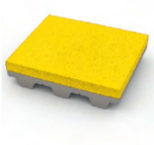
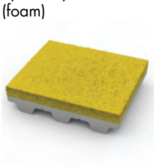
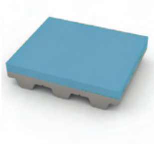
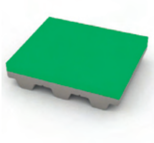


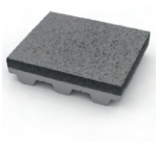
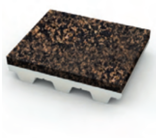
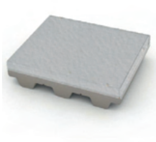
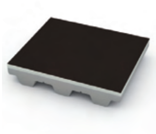
thk	2.4	2.5
Ø	60	80

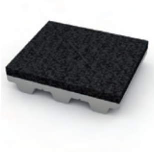
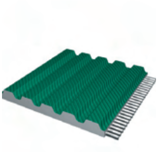
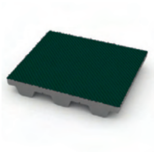
30 WM 385

thk	4
Ø	120

No.	Product	Color / Degree of grip	Hardness / Abrasion resistance	Working temperature	Toler. (timing belt + coating)	(thk) Available thickness/ (Ø) minimum pulley diameter [mm]
31	FG 385 	Transparent Medium-low	85 Shore A High	–20°C to +80°C	±0.4 mm	thk: 4 Ø: 120
32	NP 385 	transparent Medium-low	85 Shore A High	–20°C to +80°C	±0.4 mm	thk: 4 Ø: 120
33	PU Yellow 	Yellow Medium	55 Shore A Medium	–10°C to +70°C	±0.4 mm	thk: 2, 3, 4, 5, 6, 8 Ø: 70, 70, 90, 110, 110, 110 thk: 10 Ø: 130
34	PU Grey 	Grey Medium	55 Shore A Medium	–10°C to +70°C	±0.4 mm	thk: 2, 3, 4, 5, 6, 8 Ø: 70, 70, 90, 110, 110, 110 thk: 10 Ø: 130
35	Polyurethane D15 	yellowish-trans-parent Medium	60 Shore A High	–20°C to +80°C	±0.6 mm	thk: 2, 3, 4, 5, 6, 8 Ø: 60, 80, 80, 100, 100, 100
36	Celloflex 	yellowish-brown Medium	350 kg/m³ Medium-low	–30°C to +80°C	±0.7 mm	thk: 2, 3, 4, 5, 6, 8 Ø: 40, 60, 60, 80, 80, 100 thk: 10 Ø: 120

No.	Product	Color	Hardness	Working temperature	Toler. (timing belt + coating)
		Degree of grip	**Abrasion resistance**	**(thk) Available thickness/ (Ø) minimum pulley diameter [mm]**	
37	Sylodyn green	green	600 kg/m^3	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6 Ø: 100	
38	Sylodyn yellow	yellow	450 kg/m^3	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6 Ø: 80	
39	Sylomer yellow (foam)	yellow	150 kg/m^3	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6 / 12 Ø: 80 / 80	
40	Sylomer blue (foam)	blue	220 kg/m^3	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6 / 12 Ø: 60 / 80	
41	Sylomer green (foam)	green	300 kg/m^3	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6 / 12 Ø: 60 / 80	
42	Sylomer brown (foam)	brown	400 kg/m^3	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6 / 12 Ø: 60 / 80	

No.	Product	Color Degree of grip	Hardness Abrasion resistance	Working temperature (thk) Available thickness/ (Ø) minimum pulley diameter [mm]	Toler. (timing belt + coating)
43	Sylomer red (foam)	red	510 kg/m³	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6, 12 Ø: 80, 100	
44	Sylomer grey (foam)	grey	680 kg/m³	–30°C to +70°C	±0.7 mm
		Medium-high	Low	thk: 6, 12 Ø: 80, 100	
45	APSOcork HWR	brown-black	750 kg/m³	0°C to +100°C	–
		Medium	Medium-high	thk: 1, 1.5, 2, 3, 4, 5 Ø: 40, 40, 40, 60, 60, 60 thk: 6 Ø: 80	
46	ECOVIB	black with colorful particles	600 kg/m³	–30°C to +100°C	–
		Medium	Medium-high	thk: 3, 6, 8, 10, 12, 15 Ø: 60, 80, 100, 100, 120, 120	
47	Chrome Leather	grey	–	–10°C to +120°C	±0.7 mm
		Medium	Medium	thk: 2, 3 Ø: 100, 120	
48	Teflon	black	–	–200°C to +260°C	–
		Low	Low	thk: 0.3 Ø: 90	

No.	Product	Color	Hardness	Working temperature	Toler. (timing belt + coating)
		Degree of grip	Abrasion resistance	(thk) Available thickness/ (Ø) minimum pulley diameter [mm]	
49	TT60 / Novoflies	black	–	–10°C to +120°C	±0.5 mm
		Medium-low	Medium	thk: 2; Ø: 120	
50	PAZ	green	–	–20°C to +50°C	±0.2 mm
		very-low	Medium-high	thk: 0.5, 0.8; Ø: 15, 25	
51	PAR	green	–	–20°C to +50°C	±0.2 mm
		very-low	Medium-high	thk: 0.5, 0.8; Ø: 15, 25	
52	PAZ-PAR	green	–	–20°C to +50°C	±0.2 mm
		very-low	Medium-high	thk: 0.5, 0.8; Ø: 15, 25	

Coefficients of friction

Nr.	Material	Coefficient friction – Polyethylene – value	Coefficient friction – Polyethylene – angle	Coefficient friction – Aluminium – value	Coefficient friction – Aluminium – angle	Coefficient friction – Steel – value	Coefficient friction – Steel – angle	Coefficient friction – Glass – value	Coefficient friction – Glass – angle
1	Linatex HM	1.56 µ	57°	1.41 µ	55°	1.26 µ	52°	1.63 µ	58°
2	Linard 60	1.56 µ	57°	1.41 µ	55°	1.26 µ	52°	1.63 µ	58°
3	Linatrile	1.26 µ	52°	1.48 µ	56°	1.19 µ	50 °	1.63 µ	58°
4	Linagard OZ	0.96 µ	44°	1.26 µ	52°	1.04 µ	46°	1.48 µ	56°
5	Linaplus FG FDA (Natural rubber)	0.96 µ	44°	1.26 µ	52°	1.04 µ	46°	1.48 µ	56°
6	NBR 65/EPDM	1.56 µ	57°	1.41 µ	55°	1.26 µ	52°	1.63 µ	58°
7	NBR 60 white FDA	1.56 µ	57°	1.41 µ	55°	1.26 µ	52°	1.63 µ	58°
8	RP430	1.2 µ	50°	1.2 µ	50°	1.2 µ	50°	1.5 µ	57°
9	CM280	1.26 µ	52°	1.63 µ	58°	1.19 µ	50°	1.56 µ	57°
10	RG250	1.63 µ	58°	1.63 µ	58°	1.63 µ	58°	1.63 µ	58°
11	Hamid	0.89 µ	42°	1.04 µ	46°	0.96 µ	44°	1.19 µ	50°
12	Correx	1.63 µ	58°	1.63 µ	58°	1.63 µ	58°	1.63 µ	58°
13	Porol	1.63 µ	58°	1.63 µ	58°	1.63 µ	58°	1.63 µ	58°
14	Viton	0.52 µ	27°	0.74 µ	37°	0.74 µ	37°	0.74 µ	37°
15	MiniGrip blue	1.24 µ	51°	1.08 µ	47°	1.05 µ	46°	0.98 µ	44°
16	MiniGrip green	1.24 µ	51°	1.08 µ	47°	1.05 µ	46°	0.98 µ	44°
17	SuperGrip green	1.24 µ	51°	1.15 µ	49°	1.05 µ	46°	1.04 µ	46°
18	SuperGrip blue	1.24 µ	51°	1.15 µ	49°	1.05 µ	46°	1.04 µ	46°
19	Supergrip white FDA	0.95 µ	43°	0.93 µ	43°	0.81 µ	39°	1.33 µ	53°
20	PVC film blue	1.04 µ	46°	0.89 µ	42°	0.96 µ	44°	0.89 µ	42°
21	PVC dots white FDA	0.74 µ	37°	1.19 µ	50°	0.89 µ	42°	1.33 µ	53°
22	PVC film white FDA	0.96 µ	44°	0.81 µ	39°	0.89 µ	42°	0.81 µ	39°
23	PVC herringbone FDA	0.59 µ	31°	0.96 µ	44°	0.96 µ	44°	1.63 µ	58°
24	T-version (extruded) PU thick back	1.19 µ	50°	1.19 µ	50°	1.19 µ	50°	1.56 µ	57°
25	PU 385 (85° Shore A)	1.19 µ	50°	1.19 µ	50°	1.19 µ	50°	1.56 µ	57°
26	PU 60 (60° Shore A)	1.19 µ	50°	1.19 µ	50°	1.19 µ	50°	1.56 µ	57°

Nr.	Material	Coefficient friction – Polyethylene – value	Coefficient friction – Polyethylene – angle	Coefficient friction – Aluminium – value	Coefficient friction – Aluminium – angle	Coefficient friction – Steel – value	Coefficient friction – Steel – angle	Coefficient friction – Glass – value	Coefficient friction – Glass – angle
27	HV film	1.63 µ	58°	1.41 µ	55°	1.41 µ	55°	1.63 µ	58°
28	HV film FDA	1.63 µ	58°	1.41 µ	55°	1.41 µ	55°	1.63 µ	58°
29	T-groove TR1 & TR2 – PU with longitudinal grooves	1.19 µ	50°	1.19 µ	50°	1.19 µ	50°	1.56 µ	57°
30	WM 385	0.52 µ	27°	0.67 µ	34°	0.74 µ	37°	0.89 µ	42°
31	FG 385	1.63 µ	58°	1.41 µ	55°	1.41 µ	55°	1.63 µ	58°
32	NP 385	1.52 µ	56°	1.39 µ	55°	1.24 µ	52°	1.60 µ	58°
33	PU yellow	0.74 µ	37°	0.74 µ	37°	0.96 µ	44°	1.11 µ	48°
34	PU grey	0.74 µ	37°	0.74 µ	37°	0.96 µ	44°	1.11 µ	48°
35	Polythane D15	0.89 µ	42°	0.96 µ	44°	0.89 µ	42°	1.04 µ	46°
36	Celloflex	0.74 µ	37°	0.74 µ	37°	0.89 µ	42°	0.96 µ	44°
37	Sylodyn green	1.26 µ	52°	1.63 µ	58°	1.19 µ	50°	1.56 µ	57°
38	Sylodyn yellow	1.26 µ	52°	1.63 µ	58°	1.19 µ	50°	1.56 µ	57°
39	Sylomer yellow (foam)	1.26 µ	52°	1.63 µ	58°	1.19 µ	50°	1.56 µ	57°
40	Sylomer blue (foam)	1.33 µ	53°	1.63 µ	58°	1.26 µ	52°	1.63 µ	58°
41	Sylomer green (foam)	1.26 µ	52°	1.48 µ	56°	1.19 µ	50°	1.63 µ	58°
42	Sylomer brown (foam)	1.33 µ	53°	1.63 µ	58°	1.48 µ	56°	1.63 µ	58°
43	Sylomer red (foam)	1.41 µ	55°	1.63 µ	58°	1.41 µ	55°	1.63 µ	58°
44	Sylomer grey (foam)	1.33 µ	53°	1.63 µ	58°	1.41 µ	55°	1.63 µ	58°
45	APSOcork HWR	1.56 µ	57°	1.41 µ	55°	1.26 µ	52°	1.63 µ	58°
46	ECOVIB	1.56 µ	57°	1.41 µ	55°	1.26 µ	52°	1.63 µ	58°
47	Chrome Leather	0.44 µ	24°	0.89 µ	42°	0.59 µ	31°	1.04 µ	46°
48	Teflon	0.15 µ	9°	0.30 µ	17°	0.37 µ	20°	0.37 µ	20°
49	TT60/Novoflies	0.15 µ	9°	0.30 µ	17°	0.37 µ	20°	0.37 µ	20°
50	Polyamide fabric (PAZ)	0.22 µ	12°	0.30 µ	17°	0.30 µ	17°	0.30 µ	17°
51	Polyamide fabric (PAR)	0.22 µ	12°	0.30 µ	17°	0.30 µ	17°	0.30 µ	17°
52	Polyamide fabric (PAZ-PAR)	0.22 µ	12°	0.30 µ	17°	0.30 µ	17°	0.30 µ	17°

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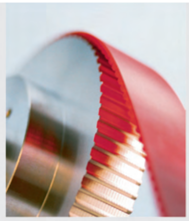

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