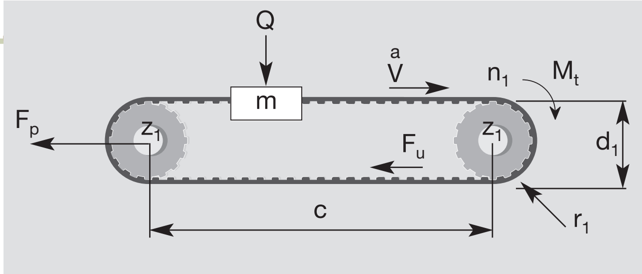
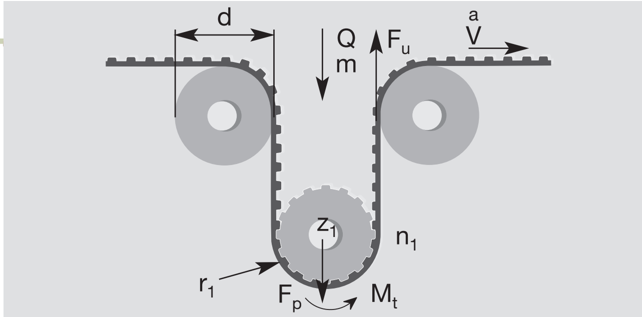
TECHNICAL CALCULATION

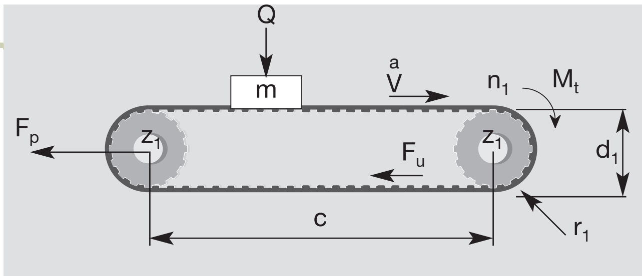
LINEAR MOTION BELT



OMEGA LINEAR MOTION BELT



CONVEYOR BELT



The following pages contain data, formulae and tables that are required to design a new belt drive.
For critical and difficult drives, it is raccomended that you contact our Application Department for advice.

Symbol	***Unit***	***Definition***	***Symbol***	***Unit***	***Definition***
a	m/s²	acceleration	**g**	m/s²	gravity (9,81)
b	mm	belt width	**μ**	–	friction coefficient
C	–	safety factor	**m**	Kg	conveyed mass
$\Delta l/_{00}$	‰	elongation	M_t	Nm	drive torque
d	mm	idler pitch diameters	n_1	1/min	revs/min (RPM) of drive sprocket 1
d_1	mm	sprocket pitch diameter	**P**	KW	drive power
F_p	N	pretension	**Q**	N	force exerted by mass (m)
F_u	N	peripheral force	**V**	m/s	belt speed
$F_{p\ spec}$	N/cm	transmittable force per tooth per unit width	Z_i		number of teeth of sprocket
MTL	N	max traction load	Z_m		number of teeth in mesh on driver sprocket (12)
BS	N	breaking strength	Z_L		number of teeth of large pulley
c	mm	centre distance	Z_s		number of teeth of small pulley
			p		belt pitch

Max traction load is maximum acceptable traction on cords.
Breaking strength is necessary load to break belt cords.
Elongation is belt elongation under load.

USEFUL FORMULAE AND CONVERSION FACTORS

$$V = \frac{d_1 \cdot n_1}{19100} \qquad n_1 = \frac{V \cdot 19100}{d_1} \qquad d_1 = \frac{V \cdot 19100}{n_1} \qquad Q = m \cdot g$$

$$P = \frac{M_t \cdot n_1}{9550} \qquad M_t = \frac{9550 \cdot P}{n_1} \qquad M_t = \frac{F_u \cdot d_1}{2000}$$

CHOICE OF BELT PITCH AND SPROCKETS

For optimum belt pitch see tables on page 10.
For optimum choice of sprocket size, it is desiderable to have as near to 12 teeth in mesh as possible.

Knowing mass →	For horizontal & conveying drives	$F_u = (m \cdot a) + (m \cdot g \cdot \mu)$
	(Note: values of μ can be found in table 1 on page 11).	
→	For vertical drives	$F_u = (m \cdot a) + (m \cdot g)$
Knowing drive torque		$F_u = 2000\ M_t / d_1$
Knowing drive power		$F_u = 19.1 \cdot 10^6 \cdot P / (d_1 \cdot n_1)$

BELT WIDTH AND PROFILE ESTIMATION

The belt width b should be calculated using the following formula

$b = (F_u \cdot c_s \cdot 10) / (F_{p\,spec} \cdot Z_m)$

C_s = safety factor from page 11 table 4
F_u = from above calculation
Z_m = number of teeth in mesh on driver sprocket

$$Z_m = [0{,}5 - \frac{4 \cdot p}{79 \cdot c}(Z_L - Z_s)] \cdot Z_s$$

= (if calculated $Z_m >= 12$ for an open-end application use $Z_m = 12$)
= (if calculated $Z_m >= 6$ for a joined application use $Z_m = 6$)

$F_{p\,spec}$ = transmittable force per tooth per unit width (see table on belt data pages)

PRE-TENSIONING

The suggested installation tension:
$F_p = 2 \cdot F_u$ for linear and omega linear movement applications
$F_p = F_u$ for conveyor applications

CORD CHECK

The maximum allowable tensile load of the belt pitch/width combination selected (see tables on belt data pages):

$$\text{max traction load of choosen belt} > \frac{F_p}{2} + (F_u \cdot C_s)$$

SPROCKET AND IDLER DIAMETER CHECK

Ensure that all selected pulley and idler diameters are equal to or greater than the minimum values specified in corresponding belt data page.

ELONGATION

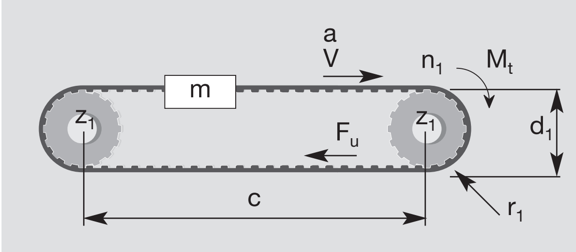
When the belt is operating there will be an elongation proportional to max traction load:

$$\Delta l/_{00} = (F_u \cdot 4) / \text{max traction load}$$

LINEAR MOTION CALCULATION EXAMPLE (OPEN-END BELT)

MACHINE DATA

C = 2.000 mm
d_1 = 76 mm
n_1 = 300 RPM
P = 1,8 KW
low fluctuating load



CHOICE OF BELT PITCH AND SPROCKETS

According to the belt pitch selection table n.1 on page 10 considering the values of P and n_1, we select RPP8 belt. Then we consider the pulley diameter nearest to the requested value and the corresponding n. of teeth (see technical information on page 65).
Therefore Z_1 = 30 teeth (with a pitch diameter of 76,4 mm).

CALCULATION OF THE EFFECTIVE TENSION

Since the drive power is known, F_u can be calculated

$$F_u = \frac{19{,}1 \bullet 10^6 \bullet P}{d_1 \bullet n_1} = \frac{19{,}1 \bullet 10^6 \bullet 1{,}8}{76{,}4 \bullet 300} = 1500\ \text{N}$$

DETERMINATION OF THE BELT WIDTH

$$b = \frac{F_u \bullet C_s \bullet 10}{F_{p\,spec} \bullet Z_m}$$

$$b = \frac{1500 \bullet 1{,}4 \bullet 10}{62 \bullet 12} = 28{,}2\ \text{mm}$$

F_u = from before (1500 N)
C_s = from page 11 table 4, for low fluctuating load C_s = 1,4
Z_m = given that driver pulley has 30 teeth and n. of teeth in mesh = 15 but max Z_m is 12, then Z_m = 12
n_1 = 300 RPM (given)
$F_{p\,spec}$ = 62N / cm (refer page 64 at 300 RPM)

Since the next closest width is 30 mm: 30 RPP8 is choosen.

PRE-TENSIONING

$F_p = 2 \bullet F_u$ $\quad$ $F_p = 3000$ N

CORD CHECK

From page 64, RPP8 pitch 30 mm wide: max traction load 4750 N

max traction load $> \frac{F_p}{2} + (F_u \bullet C_s)$ $\qquad$ $\frac{F_p}{2} + (F_u \bullet C_s) = 1500 + 1500 \bullet 1{,}4$

4750 N > 3600 N selected belt is acceptable.

SPROCKET AND IDLER DIAMETER CHECK

Ensure that all selected pulley and idler diameters are greater than or equal the minimum values specified on page 65.

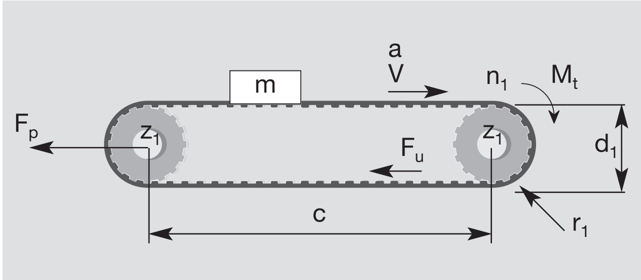
ELONGATION

$$\Delta l/_{00} = \frac{F_u \bullet 4}{\text{max traction load}} = \frac{1500 \bullet 4}{4750} = 1{,}26\ \text{mm/m}$$

In the dynamic situations you will have an elongation of 1,26 mm per meter of operating belt.

CONVEYOR BELT CALCULATION EXAMPLE (JOINED BELT)

MACHINE DATA
C = 5.000 mm
d_1 = 100 mm
V = 0,5 m/s
a = 0,5 m/s²
Guide in nylon
Q = 4500 N
low fluctuating load



CALCULATION OF THE EFFECTIVE TENSION

Since the mass is known, F_u can be calculated

$F_u = (m \cdot a) + (m \cdot g \cdot \mu)$ value of μ according to table 3 on page 11 = 0,35

$F_u = (460 \cdot 0{,}5) + (460 \cdot 9{,}81 \cdot 0{,}35) = 1810$ N

$m = Q/g = 4500 / 9{,}81 = 460$ kg

CHOICE OF BELT PITCH AND SPROCKETS

According to the belt selection table n. 2 on page 10, considering the values of F_u (for joined belts enter double of calculated F_u in table 2), we select T 10. Then we consider the pulley diameter nearest to the requested value and the corresponding n. of teeth (see technical information page 35). Therefore Z_1 = 32 teeth (with a pitch diameter of 101,86 mm).

DETERMINATION OF THE BELT WIDTH

$$b = \frac{F_u \cdot C_s \cdot 10}{F_{p\,spec} \cdot Z_m}$$

$$b = \frac{1810 \cdot 1{,}4 \cdot 10}{45 \cdot 6} = 93{,}85 \text{ mm}$$

F_u = from before (1810 N)
C_s = from page 11 table 4, for low fluctuating load C_s = 1,4
Z_m = given that driver pulley has 32 teeth and n. of teeth in mesh = 16 but max Z_m for joined belt is 6, hence, Z_m = 6
n_1 = $(Vp \cdot 60.000) / (\pi \cdot d_1) = (0{,}5 \cdot 60.000) / (\pi \cdot 101{,}86)$ as d_1 = 101,86 from before = 94 RPM
$F_{p\,spec}$ = 45 N / cm (refer page 34, at 100 RPM)

Since the next closest width is 100 mm: 100 T10 is choosen.

PRE-TENSIONING

$F_p = F_u$ so $F_p = 1810$ N

CORD CHECK

From page 34, T10 pitch 100 mm wide joined: max traction load 5415 N

max traction load > $F_p + (F_u \cdot C_s)$ $Fp + (F_u \cdot C_s) = 1810 + (1810 \cdot 1{,}4)$

5415 N > 4344 N selected belt is acceptable.

SPROCKET AND IDLER DIAMETER CHECK

Checking technical data on page 35 for pulley and idlers, it can be seen that the drive has acceptable pulley diameters.

ELONGATION

$$\Delta l/_{00} = \frac{F_u \cdot 4}{\text{max traction load}} = \frac{1810 \cdot 4}{5415} = 1{,}33 \text{ mm/m}$$

In the dynamic situations you will have an elongation of 1,33 mm per meter of operating belt.

CALCULATION PARAMETERS

BELT PITCH SELECTION

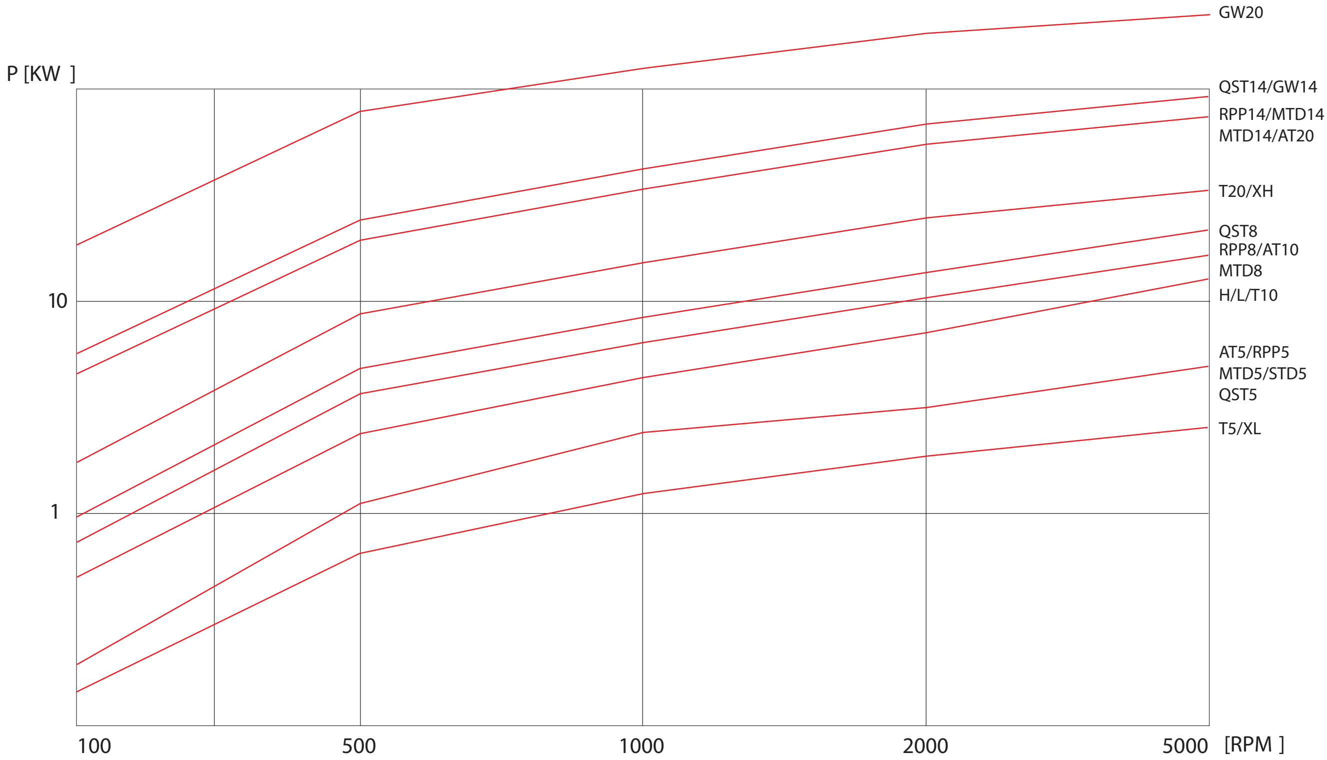


Table n. 1

BELT WIDTH SELECTION

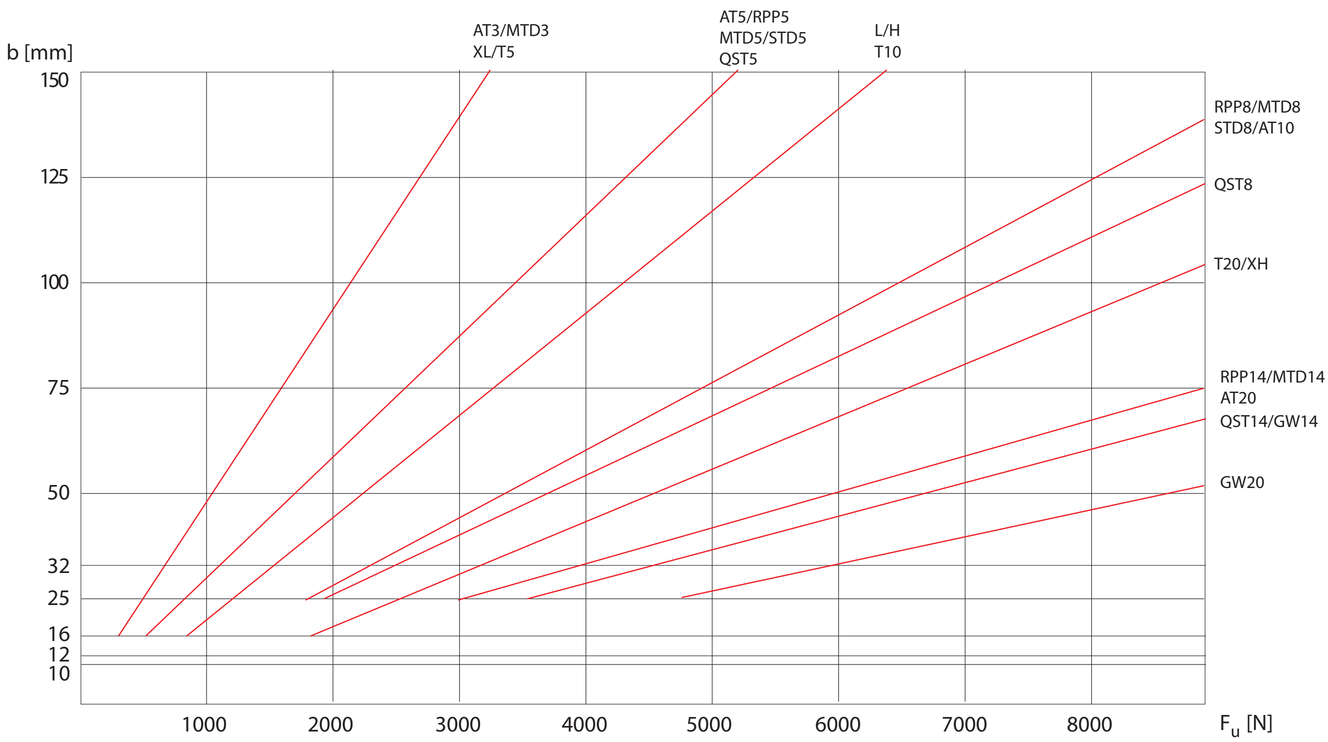


Table n. 2

Average values valid for standard steel cord. After belt selection, please check belt resistance on belt data page.

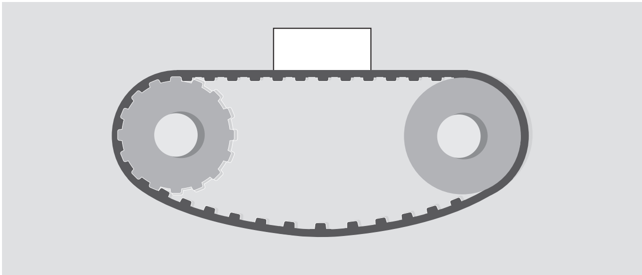
Table n. 3 - Friction coefficient	
Sliding friction on dry surface	
Polyurethane / smooth steel	μ = 0,5
Polyurethane / rough steel	μ = 0,7
Polyurethane / abrasive steel	μ = 0,9
Polyurethane NFT / smooth steel	μ = 0,25
Polyurethane NFT / rough steel	μ = 0,35
Polyurethane NFT / abrasive steel	μ = 0,6
Polyurethane / nylon	μ = 0,35
Polyurethane NFT / nylon	μ = 0,15
Polyurethane / aluminium	μ = 0,8
Polyurethane NFT / aluminium	μ = 0,45
Rolling friction on dry surface	
Bearing	μ = 0,015
Roller / PU Belt	μ = 0,03 / 0,06
Bush	μ = 0,15

Table n. 4 - Safety factor		
The choice of the Safety factor's, depends on the operating conditions. The following table shows the value to be used:		
Steady Load		1
Shock Load	Low	1,4
	Average	1,7
	High	2
Elevators, hoists		1,8
Line shafts		1,6
Paper machines:		
agitators, calenders, driers, winding frames,		1,6
willows, Jordan machines, pumps, slicers, grinders		1,8
Machines for pottery and earthenware:		
cutters, granulators,		1,7
pulping machines		2,0
Laundry machines: general		1,6
extractors, washers		1,8
Machines for rubber processing		1,8
Woodworking machines:		
lathes, band saws, cutters,		1,7
circular saws, planers, jointer		1,7
Printing machinery:		
rotary, newspaper, linotype, cutters, folders, magazine		1,6
Textile machines:		
warping machines, winders,		1,7
spinners, twisting frames, looms		1,8
Machines tools: drilling machines, lathes,		
tread cutting machines, gears cutters, boring machines		1,6
millers, planers,		1,7
grinding machines		1,7
Conveyors:		
hoists, light package		1,3
oven screw fleight		1,8
apron bucket, elevator		1,8
screw		1,8
Brick machinery		1,8

BELT INSTALLATION

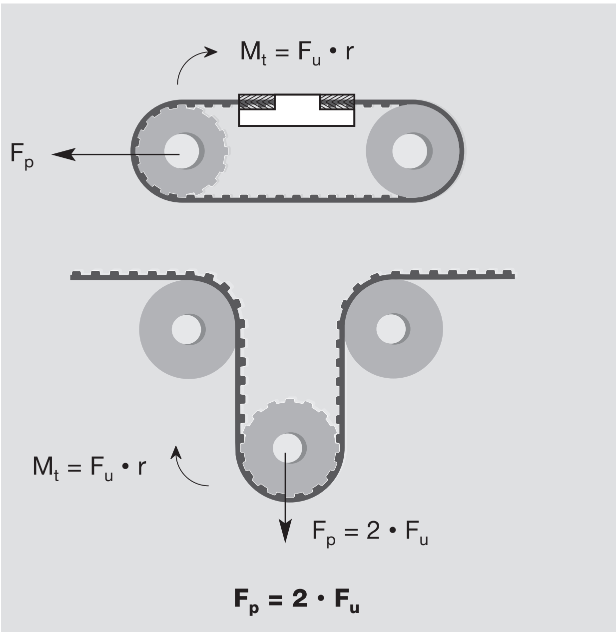
A major difficulty installing transmission belt is to achieve correct belt tension. Lifetime of support bearings and transmission belts and therefore reliability of the complete system largely depends on an optimally adjusted belt tension. Pretension is the force needed to put tension into the system to avoid the belt jumping on the pulleys as in the example below:

Not correct belt installation

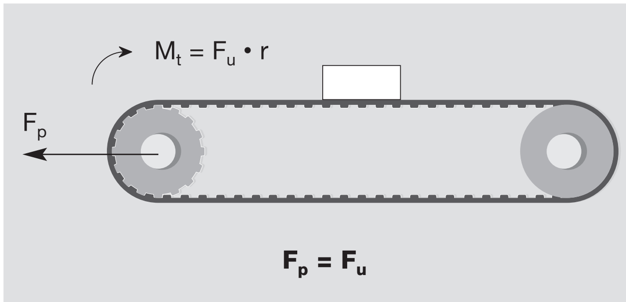


For a correct system installation, all applications with Megalinear belt can be summarised according following two sketches:

1) Linear and omega linear motion belt



2) Conveyor belt



F_p = pretension
F_u = peripheral force (see calculation pag. 8/9)
r = pulley radius

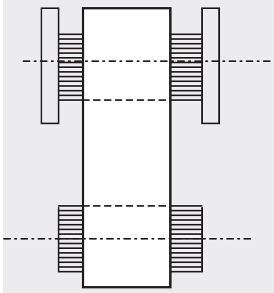
PROCEDURE TO MEASURE

The procedure to measure the tension of the belt is to use a Belt Tension Gauging Equipement. This device consists of a small sensing head which is held across the belt to be measured. The belt is then tapped to induce the belt to vibrate at its natural frequency. The vibrations are detected and the frequency of vibration is then displayed on the measuring unit. The relation between belt static tension (T_S) and frequency of vibration (f) may be calculated using the following formula:

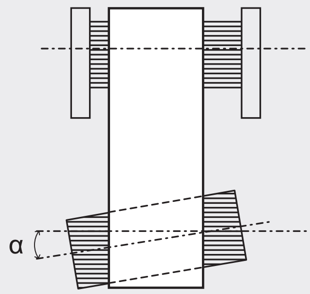
$$f = \frac{1}{2t} \cdot \sqrt{\frac{T_s}{m}} \quad \text{or} \quad T_s = 4 \cdot m \cdot t^2 \cdot f^2$$

Where :	
T_S = static tension (N)	f = Frequency of vibration in Hertz (Hz)
m = Belt mass per unit length (kg/m)	t = Free belt span length in meters (m)

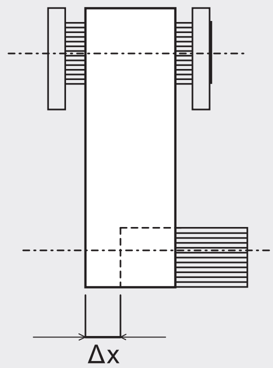
For a correct system functioning and to increase belt life, it is necessary a correct pulley installation: pulleys has to be parallel and aligned as shown in drawing 1 (correct configuration).
If pulleys are not parallel as in drawing 2, belt could fall during functioning and this can provoke damages to complete equipment.
To grant a correct belt running, α and Δx must be as smaller as possible. For more information, please contact our technical staff.



Drawing 1



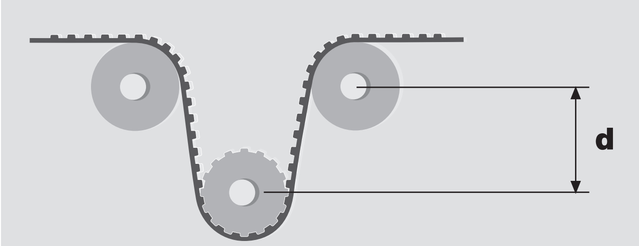
Drawing 2



Drawing 3

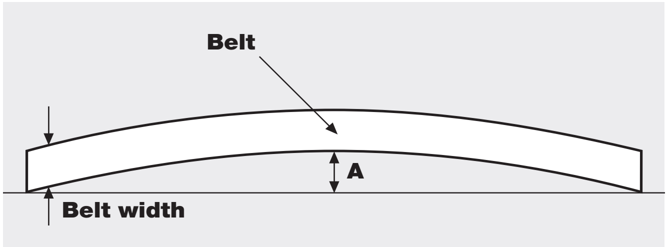
In omega application to grant good mesh between pulley and teeth and to respect belt flexibility avoiding excessive stress on cords, distance d (as drawing 4) has to be:

d = 4 • belt width
Suggested angle 120°



Drawing 4

Moreover for a good drive work, it is suggested to check belt straigthness as follows:



Belt width	Testing belt length	Maximum suggested bending (A)
Till to 20 mm	1 m	3 mm
Over 20 mm	2 m	4 mm