Many conveying timing belts operate at low speeds and minimal loads. This eliminates the need for extensive calculations and a simplified approach to belt selection can be used. For these lightly loaded applications, the belt can be selected according to the dimensional requirements of the system, product size, desired pulley diameter, conveyor length, etc.

The belt width $b$ is often determined according to the size of the product conveyed, and as a rule, the smallest available belt pitch is used. For proper operation, the pre-tension $T_i$ should be set as follows:

$$T_i = 0.3 \cdot b \cdot T_{\text{all}}$$

where: $T_i = \text{belt pre-tension}$
$T_{\text{all}} = \text{max allowable belt tension for}$
1” or 25mm wide belt (see Table 1 or Table 2)

U.S. customary units: $T_i$ [lb], $T_{\text{all}}$ [lb/in], $b$ [in]
Metric units: $T_i$ [N], $T_{\text{all}}$ [N/25mm], $b$ [mm].

For all applications where the loads are significant, the following step-by-step procedure should be used for proper belt selection.

**Step 1. Determine Effective Tension**

The effective tension $T_e$ at the driver pulley is the sum of all individual forces resisting the belt motion. The individual loads contributing to the effective tension must be identified and calculated based on the loading conditions and drive configuration. However, some loads cannot be calculated until the layout has been decided.

To determine the effective tension $T_e$ use one of the following methods for either conveying or linear positioning.

**Conveying**

$T_e$ for conveying application is primarily the sum of the following forces (see Figs. 1 and 2).

1. The friction force $F_f$ between the belt and the slider bed resulting from the weight of the conveyed material.

$$F_f = \mu \cdot w_m \cdot L_m \cdot \cos \beta$$

where: $\mu = \text{coefficient of friction between the slider bed and the belt}$
and the belt (see Table 1A)
$w_m = \text{load weight per unit length over conveying length}$
$L_m = \text{conveying length}$
$\beta = \text{angle of conveyor incline}$

U.S. customary units: $F_f$ [lb], $w_m$ [lb/ft], $L_m$ [ft].
Metric units: $F_f$ [N], $w_m$ [N/m], $L_m$ [m].

2. The gravitational load $F_g$ to lift the material being transported on an inclined conveyor.

$$F_g = w_m \cdot L_m \cdot \sin \beta$$

**Fig. 1**
5. The force $F_{ai}$ required to accelerate the idler.

$$F_{ai} = \frac{J_i \cdot \alpha}{r_0^2} = \frac{m_i \cdot r_0^2}{2 \cdot r_0^2} \cdot a = \frac{m_i \cdot a}{2}$$

where:
- $J_i = \frac{m_i \cdot r_0^2}{2}$ = inertia of the idler
- $m_i$ = mass of the idler
- $r_0$ = idler outer radius
- $\alpha = \frac{a}{r_0^2}$ = angular acceleration

In the formula above, the mass of the idler $m_i$ is approximated by the mass of a full disk.

$$m_i = \rho \cdot b_i \cdot \pi \cdot r_0^2$$

where:
- $\rho$ = density of idler material
- $b_i$ = width of the idler

U.S. units: $\rho$ [lb*ft^2/ft^4], $b_i$ and $r_0$ [ft].
Metric units: $\rho$ [kg/m^3], $b_i$ and $r_0$ [m].

6. The force $F_{ab}$ required to accelerate the belt mass.

$$F_{ab} = m_b \cdot a$$

The belt mass $m_b$ is obtained from the specific belt weight $w_b$ and belt length and width.

$$m_b = \frac{w_b \cdot L \cdot b}{g}$$

U.S. units: $F_{ab}$ [lb], $m_b$ [lb*ft^2/ft], $a$ [ft/s^2], $w_b$ [lb/ft^2], $L$ and $b$ [ft], $g = 32.2$ ft/s^2.
Metric units: $F_{ab}$ [N], $m_b$ [kg], $a$ [m/s^2], $w_b$ [N/m^2], $L$ and $b$ [m], $g = 9.81$ m/s^2.*

Thus for linear positioners, $T_e$ is expressed by:

$$T_e = F_a + F_l + F_w + W_s + [F_{ai}] + [F_{ab}]$$

Note that the forces in brackets can be calculated by estimating the belt mass and idler dimensions. In most cases, however, they are negligible and can be ignored.

---

Step 2. Select Belt Pitch

Use Graphs 2a, 2b, 2c or 2d to select the nominal belt pitch $p$ according to $T_e$. The graphs also provide an estimate of the required belt width. (For H pitch belts wider than 6" (152.4mm) and T10 pitch belts wider than 150mm, use Graph 1).

Step 3. Calculate Pulley Diameter

Use the preliminary pulley diameter $d$ desired for the design envelope and the selected nominal pitch $p$ to determine the preliminary number of pulley teeth $z_p$.

$$z_p = \frac{\pi \cdot d}{p}$$

Round to a whole number of pulley teeth $z_p$. Give preference to stock pulley diameters. Check against the minimum number of pulley teeth $z_{min}$ for the selected pitch given in Table 1 or Table 2.

Determine the pitch diameter $d$ according to the chosen number of pulley teeth $z_p$.

$$d = \frac{p \cdot z_p}{\pi}$$

---

Step 4. Determine Belt Length and Center Distance

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

---

Fig. 3
3. The friction force $F_{fv}$ resulting from vacuum in vacuum conveyors.

$$F_{fv} = \mu \cdot P \cdot A_v$$

where: $P$ = pressure (vacuum) relative to atmospheric
$A_v$ = total area of vacuum openings

U.S. units: $F_{fv}$ [lb], $P$ [lb/ft$^2$], $A_v$ [ft]$^2$
Metric units: $F_{fv}$ [N], $P$ [Pa], $A_v$ [m]$^2$

The formula above assumes a uniform pressure and a constant coefficient of friction.

4. The friction force $F_{fa}$ between the slide and the linear rail is determined experimentally, or from data from the linear bearing manufacturer. Other contributing factors to the friction force are bearing losses from the yolk, piston and pillow blocks (see Table 1A).

$$F_{fa} = (\mu + \mu_a) \cdot w_{ma} \cdot L_a \cdot \cos \beta$$

where: $L_a$ = accumulation length
$\mu_a$ = friction coefficient between accumulated material and the belt (see Table 1A)
$w_{ma}$ = material weight per unit length over the accumulation length

U.S. customary units: $L_a$ [ft], $w_{ma}$ [lb/ft].
Metric units: $L_a$ [m], $w_{ma}$ [N/m].

5. The inertial force $F_a$ caused by the acceleration of the conveyed load (see linear positioning).

6. The friction force $F_{fb}$ between the belt and slider bed caused by the belt weight.

$$F_{fb} = \mu \cdot w_b \cdot b \cdot L_c \cdot \cos \beta$$

where: $w_b$ = specific belt weight
$b$ = belt width
$L_c$ = conveying length

U.S. customary units: $w_b$ [lb/ft$^2$], $b$ [ft], $L_c$ [ft].
Metric units: $w_b$ [N/m$^2$], $b$ [m], $L_c$ [m].

For initial calculations, use belt width which is required to handle the size of the conveyed product.

Thus for conveyors, $T_e$ is expressed by:

$$T_e = T_1 + T_2 + F_{iv} + F_{fa} + F_a + (F_{fb}) + \ldots$$

$F_{fb}$ can be calculated by estimating the belt mass. In most cases, this weight is insignificant and can be ignored.

Note that other factors, such as belt supporting idlers, or accelerating the material fed onto the belt, may also account for some power requirement. In start-stop applications, acceleration forces as presented for linear positioning, may have to be evaluated.

**Linear Positioning**

$T_e$ for a linear positioning application is primarily the sum of the following six factors (see Fig. 3).

1. The force $F_a$ required for the acceleration of a loaded slide with the mass $m_s$ (replace the mass of the slide with the mass of the package in conveying).

$$F_a = m_s \cdot a$$

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

$$a = \frac{v_f - v_i}{t}$$

where: $v_f$ = final velocity
$v_i$ = initial velocity
$t$ = time

U.S. customary units: $F_a$ [lb], $a$ [ft/s$^2$], $v_f$ and $v_i$ [ft/s], $t$ [s].

The mass is derived from the weight $W_s$ [lb] and the acceleration due to gravity $g$ ($g$ = 32.2 ft/s$^2$):

$$m_s = \frac{W_s}{g} = \frac{W_s}{32.2} \left[ \frac{lb \cdot s^2}{ft} \right]$$

U.S. customary units: $F_a$ [lb], $a$ [ft/s$^2$], $v_f$ and $v_i$ [ft/s], $t$ [s], $m_s$ [kg].

2. The friction force $F_f$ between the slide and the linear rail is determined experimentally, or from data from the linear bearing manufacturer. Other contributing factors to the friction force are bearing losses from the yolk, piston and pillow blocks (see Fig. 3).

3. The externally applied working load $F_w$ (if existing).

4. The weight $W_s$ of the slide (not required in horizontal drives).

---

*If working in US units, $w_b$ found in the belt specifications must be converted to the units lb/ft$^2$. If working in metric units, $w_b$ must be converted to the units N/m$^2$.**

![Diagram of conveyor system](image-url)
Belt Sizing Guide

For equal diameter pulleys:

\[ z_b = 2 \cdot \frac{C}{p} + z_p \]

For unequal diameter pulleys: (See Fig. 4)

\[ z_b \approx 2 \cdot \frac{C}{p} + \frac{Z_{p2} + Z_{p1}}{2} + \frac{p}{4C} \cdot \left( \frac{Z_{p2} - Z_{p1}}{\pi} \right)^2 \]

Choose a whole number of belt teeth \( z_b \). If you have profiles welded to the belt, consider the profile spacing while choosing the number of belt teeth.

Determine the belt length \( L \) according to the chosen number of belt teeth.

\( L = z_b \cdot p \)

Determine the center distance \( C \) corresponding to the chosen belt length.

For equal diameter pulleys:

\[ C = \frac{L - \pi \cdot d}{2} \]

For unequal diameter pulleys:

\[ C = \frac{Y + \sqrt{Y^2 - 2 \cdot (d_2 - d_1)^2}}{4} \]

where: \( Y = L - \pi \cdot \frac{(d_2 + d_1)}{2} \)

Step 5. Calculate The Number of Teeth in Mesh of the Small Pulley

Calculate the number of teeth in mesh \( z_m \) using the appropriate formula.

For two equal diameter pulleys:

\[ z_m = \frac{z_p}{2} \]

For two unequal diameter pulleys:

\[ z_m = z_{p1} \cdot \left( 0.5 - \frac{d_2 - d_1}{2\pi \cdot C} \right) \]

Step 6. Determine Pre-tension

The pre-tension \( T_p \), defined as the belt tension in an idle drive, is illustrated as the distance between the belt and the dashed line in Figs. 1, 2, and 3. The pre-tension prevents jumping of the pulley teeth during belt operation. Based on experience, timing belts perform best with the slack side tension as follows:

\[ T_2 = (0.1,\ldots,0.3) T_e \]

Drives with a fixed center to center distance

Drives with fixed center distances have the position of the adjustable shaft locked after pre-tensioning the belt (see Figs. 1 and 3). Assuming tight and slack side tensions are constant over the respective belt lengths, and a minimum slack side tension in the range of the above relationship (uni-directional load only), the pre-tension is calculated utilizing the following equation:

\[ T_1 = T_2 + T_e \cdot \frac{L_1}{L} \]

where: \( L = \text{belt length} = L_1 + L_2 \)

\( L_1 = \text{tight side belt length} \)

\( L_2 = \text{slack side belt length} \)

U.S. units: \( L_1 \) [ft], and \( L_2 \) [ft].

Metric units: \( L_1 \) [m], and \( L_2 \) [m].

Drives with a fixed center to center distance are used in linear positioning, conveying and power transmission applications. In linear positioning applications, the maximum tight side length is inserted in the equation above.

The pre-tension for drives with a fixed center distance can also be approximated using the
following formulas:

Conveying  
(see Figs. 1 and 2)  
\[ T_i = (0.45,\ldots,0.55) T_e \]

Linear Positioning  
(see Fig. 3)  
\[ T_i = (1.0,\ldots,1.2) T_e \]  
\[ T_i = (1.0,\ldots,2.0) T_e \] \( \Rightarrow \) for ATL series only

Drives with a constant slack side tension  
Drives with constant slack side tension have an adjustable idler, tensioning the slack side, which is not locked (Figs. 2 and 5). During operation, the consistency of the slack side tension is maintained by the external tensioning force, \( F_e \). Drives with a constant slack side tension may be considered for some conveying applications, they have the advantage of minimizing the required pre-tension.

The minimum pre-tension can be calculated from the analysis of the forces at the idler in Fig. 5:
\[ T_i = T_2 = \frac{F_e}{2 \sin \theta_e} \]

where \( \theta_e \) is the wrap angle of the belt around the back bending idler (Fig. 5).

Step 7. Calculate Tight Side Tension and Slack Side Tension

Conveying  
(see Figs. 1 & 2)  
The tight side tension \( T_1 \) and the slack side tension \( T_2 \) are obtained by:
\[ T_1 = T_i + 0.75 T_e \]  
\[ T_2 = T_1 - T_e \]

Linear Positioning  
(see Fig. 3)  
The maximum tight side tension \( T_{1\text{max}} \) is obtained by:
\[ T_{1\text{max}} = T_i + T_e \]

The respective minimum slack side tension \( T_{2\text{min}} \) is obtained by:
\[ T_{2\text{min}} = T_i - T_e \]  
for a fixed center distance.

Step 8. Calculate Belt Width

Determine the allowable tension \( T_{1\text{all}} \) for the cords of a 1" (or 25 mm) wide belt of the selected pitch given in Table 1 or Table 2. Note that \( T_{1\text{all}} \) is different for open end (positioning) and welded (conveying) belts. Determine the necessary belt width to withstand \( T_{1\text{max}} \):
\[ b \geq \frac{T_{1\text{max}}}{T_{1\text{all}}} \]

U.S. units: \( T_1 \) [lb], \( T_{1\text{all}} \) [lb/in], \( b \) [in].  
Metric units: \( T_1 \) [N], \( T_{1\text{all}} \) [N/25mm], \( b \) [mm].

Determine the allowable effective tension \( T_{e\text{all}} \) for the teeth of a 1" (or 25 mm) wide belt of the selected pitch from Table 1 or Table 2. Note that \( T_{e\text{all}} \) is different for open end (positioning) and welded (conveying) belts.

Use Table 3 (Tooth in Mesh Factor) that follows to determine the tooth-in-mesh-factor \( t_m \) corresponding to the number of teeth in mesh \( z_m \).
Determine the speed factor \( t_v \) using Table 4 (Speed Factor) that follows.

Calculate the width of the belt teeth \( b \) necessary to transmit \( T_e \) using the following formula:

\[
b \geq \frac{T_e}{T_{eall} \cdot t_m \cdot t_v}
\]

U.S. units: \( T_e \) [lb], \( T_{eall} \) [lb/in], \( b \) [in].
Metric units: \( T_e \) [N], \( T_{eall} \) [N/25mm], \( b \) [mm].

Select the belt width that satisfies the last two conditions, giving preference to standard belt widths. However, belts of nonstandard widths are also available.

The factors \( t_m \) and \( t_v \) prevent excessive tooth loading and belt wear.

The forces contributing to \( T_e \) which in Step 1 were estimated, can now be calculated more accurately. Evaluate the contribution of these forces to the effective tension and, if necessary, recalculate \( T_e \) and repeat steps 6, 7 and 8.

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

**Step 9. Calculate Shaft Forces**

Determine the shaft force \( F_{s1} \) at the driver pulley:

For angle of wrap \( \theta = 180^\circ \):
\[ F_{s1} = T_1 + T_2 \]

For angle of wrap around the small pulley \( \theta < 180^\circ \) (unequal diameter pulleys):
\[ F_{s1} = \sqrt{T_1^2 + T_2^2 - 2T_1 \cdot T_2 \cos \theta} \]

where \( \theta = 2 \cdot \pi \cdot \left(0.5 - \frac{d_2 - d_1}{2 \cdot \pi \cdot C}\right) \)

Determine the shaft force \( F_{s2} \) at the idler pulley:

For angle of wrap \( \theta = 180^\circ \):
\[ F_{s2} = 2 \cdot T_2 \] when load moves toward the driver pulley, and
\[ F_{s2} = 2 \cdot T_1 \] when load moves away from the driver pulley.

For angle of wrap around the small pulley \( \theta < 180^\circ \) (unequal diameter pulleys):
\[ F_{s2} = T_1 \cdot \sqrt{2 (1 - \cos \theta)} \] when the load moves away from the driver.

**Step 10. Calculate the Stiffness of a Linear Positioner**

The total stiffness of the belt depends mainly on the stiffness of the belt segments between the pulleys. In most cases, the influence of belt teeth and belt cords in the tooth-in-mesh area can be ignored. Calculate the resultant stiffness coefficient of tight and slack sides \( k \), as a function of the slide position (Fig. 6).

\[ k = c_{sp} \cdot b \cdot \frac{L}{L_1 \cdot L_2} \]

where:
- \( L_1 \) = tight side length
- \( L_2 \) = slack side length
- \( c_{sp} \) = specific stiffness (Table 1).

U.S. units: \( k \) [lb/in], \( c_{sp} \) [lb/in], \( b \) [in], \( L \) [in].
Metric units: \( k \) [N/mm], \( c_{sp} \) [N/mm], \( b \) [mm], \( L \) [mm].

Note that \( k \) is at its minimum when the tight and slack sides are equal.

Determine the positioning error \( \Delta x \) due to belt elongation caused by the remaining static force \( F_{st} \) on the slide:
\[ \Delta x = \frac{F_{st}}{k} \]

In Fig. 6, for example, \( F_{st} \) is comprised of \( F_f \) and \( F_w \) and is balanced by the static effective tension \( T_{est} \) at the driver pulley.

Note that \( \Delta x \) is inversely proportional to the belt width. If you want reduced \( \Delta x \), increase the belt width or select a belt with stiffer cords and/or with a larger pitch.
Graph 1

### Tooth In Mesh Factor

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**Table 3**

### Speed Factor

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</table>

**Table 4**
Pitch Selection – Linear Positioning (Open Ended) Belts

Belt Width [mm]

Effective Tension (lb)

Effective Tension (N)

0 1 2 3 4 5 6

0 100 200 300 400 500 600 700 800 900 1000 1100 1200 1300 1400 1500 1600 1700

XL
L
H
H-HF
XH

Pitch Selection – Conveying (Welded) Belts

Belt Width [mm]

Effective Tension (lb)

Effective Tension (N)

0 1 2 3 4 5 6

0 50 100 150 200 250 300 350 400 450 500 550 600 650 700 750 800 850 900 950 1000 1050 1100 1150 1200 1250 1300 1350 1400

Graph 2a
Pitch Selection – Linear Positioning (Open Ended) Belts

Belt Width [in]

Effective Tension (N)

Pitch Selection – Conveying (Welded) Belts

Belt Width [in]

Effective Tension (lb)
Pitch Selection – Linear Positioning (Open Ended) Belts

Pitch Selection – Conveying (Welded) Belts
Technical Design Tools

Graph 2d
Belt Sizing Examples

**Conveying**

- **v** = 120 ft/min (Speed)
- **W** = 60 lb (Box weight)
- 18” x 12” (Box bottom size)
- **C** = 28 ft (336 in) (Center distance)
- **b** = 15° (Conveyor angle of incline)
- **d** = 3.5” (Pulley outside diameter)

Slider bed made of steel, belt teeth covered with nylon fabric.

Considering only the box size, a belt width of approximately 12” would be necessary. Instead of using one 12” wide belt, however, we decide to build a conveyor with two parallel running belts. The minimum belt width will be determined.

**Step 1**

The boxes are carried lengthwise on 2 ft centers.

Weight distribution over conveyor length: **w** = 30 lb/ft.

Friction force:

\[ F_f = \mu \cdot w_m \cdot L_m \cdot \cos \beta \]

\[ F_f = 0.3 \cdot 30 \text{ lb/ft} \cdot 28 \text{ ft} \cdot \cos 15° \]

\[ F_f = 243.4 \text{ lb} \]

Gravitational load:

\[ F_g = w_m \cdot L_m \cdot \sin \beta \]

\[ F_g = 30 \text{ lb/ft} \cdot 28 \text{ ft} \cdot \sin 15° \]

\[ F_g = 2174 \text{ lb} \]

Effective tension:

\[ T_e = F_f + F_g \]

\[ T_e = 243.4 \text{ lb} + 2174 \text{ lb} \]

\[ T_e = 460.8 \text{ lb} \]

**Step 2**

Selected belt tooth profile =>H (Graph 2a)

An effective tension of 460.8 lb can be transmitted by either L or H belt. We choose H tooth profile (0.5”).

The minimum belt width to transmit the load will be approximately 2.5 inches.

**Step 3**

Approximate number of pulley teeth:

\[ z_p = \frac{\pi \cdot d}{p} \]

\[ z_p = \frac{\pi \cdot 3.5 \text{ in}}{0.5 \text{ in}} \]

\[ z_p = 21.99 \]

Chosen number of teeth:

\[ z = 22 \]

(Chosen number of teeth is greater than the recommended minimum number of pulley teeth for H tooth profile belt [z_{min} = 14] given in Table 1)

Pulley pitch diameter:

\[ d = \frac{p \cdot z_p}{\pi} \]

\[ d = \frac{0.5 \text{ in} \cdot 22}{\pi} \]

\[ d = 3.501 \text{ in} \]

**Step 4**

Preliminary number of belt teeth:

\[ z_b = 2 \cdot \frac{C}{b} + z_p \]

\[ z_b = 2 \cdot \frac{336 \text{ in}}{0.5 \text{ in}} + 22 \]

\[ z_b = 1366 \]

Chosen number of belt teeth:

\[ z_b = 1366 \]

Belt length:

\[ L = z_b \cdot p \]

\[ L = 1366 \cdot 0.5 \text{ in} \]

\[ L = 683 \text{ in} \]

**Step 5**

Number of teeth in mesh:

\[ z_m = \frac{z_b}{2} \]

\[ z_m = 22 \]

\[ z_m = 11 \]

**Step 6**

Pre-tension:

\[ T_i = 0.5 \cdot T_e \]

\[ T_i = 0.5 \cdot 460.8 \text{ lb} \]

\[ T_i = 230.4 \text{ lb} \]

**Step 7**

Tight side tension:

\[ T_1 = T_i + 0.75 \cdot T_e \]

\[ T_1 = 230.4 \text{ lb} + 0.75 \cdot 460.8 \text{ lb} \]

\[ T_1 = 576 \text{ lb} \]

Slack side tension:

\[ T_2 = T_1 - T_e \]

\[ T_2 = 576 - 460.8 \text{ lb} \]

\[ T_2 = 115.2 \text{ lb} \]

**Step 8**

Allowable belt tension (from Table 1):

\[ T_{i\text{all}} = 245 \text{ lb/in} \]

Belt width b to withstand \( T_{1\text{max}} \):

\[ b \geq \frac{T_{1\text{max}}}{T_{i\text{all}}} \]

\[ b \geq \frac{576 \text{ lb}}{245 \text{ lb/in}} \]

\[ b \geq 2.35 \text{ in} \]

Allowable effective tension (from Table 1):

\[ T_{e\text{all}} = 330 \text{ lb/in} \]

Tooth in mesh factor:

\[ t_m = 0.92 \]

Speed factor (from Table 4; for \( v = 120 \text{ ft/min} \)):

\[ t_v = 1 \]

Belt width to transmit \( T_e \):

\[ b \geq \frac{T_e}{T_{e\text{all}} \cdot t_m \cdot t_v} \]

\[ b \geq \frac{460.8 \text{ lb}}{330 \text{ lb/in} \cdot 0.92 \cdot 1} \]

\[ b \geq 1.52 \text{ in} \]

Chosen belt width—boxes will be conveyed on two belts 1.5” wide each.

(Note that each belt is loaded by half of the calculated forces)
## Belt Sizing Examples

### Step 9
Shaft force at driver
\[ F_{s1} = T_1 + T_2 \]
\[ F_{s1} = 576 \text{ lb} + 115.2 \text{ lb} \]
\[ F_{s1} = 691.2 \text{ lb} \]

Shaft force at idler
\[ F_{s2} = 2T_2 \]
\[ F_{s2} = 2 \times 115.2 \text{ lb} \]
\[ F_{s2} = 230.4 \text{ lb} \]

### Linear Positioning

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Formula</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Speed</td>
<td>( v = 3.5 \text{ m/s} )</td>
<td></td>
</tr>
<tr>
<td>Slide acceleration</td>
<td>( a = 20 \text{ m/s}^2 )</td>
<td></td>
</tr>
<tr>
<td>Slide mass</td>
<td>( m_s = 30 \text{ kg} )</td>
<td></td>
</tr>
<tr>
<td>Friction force</td>
<td>( F_f = 50 \text{ N} )</td>
<td></td>
</tr>
<tr>
<td>Positioning error</td>
<td>( \Delta x \leq 0.1 \text{ mm} )</td>
<td></td>
</tr>
<tr>
<td>Pulley diameter</td>
<td>( d_p = 50 \text{ mm} )</td>
<td></td>
</tr>
<tr>
<td>Center distance</td>
<td>( C = 3000 \text{ mm} )</td>
<td></td>
</tr>
<tr>
<td>Travel</td>
<td>( S = 2500 \text{ mm} )</td>
<td></td>
</tr>
</tbody>
</table>

### Step 1
Force to accelerate the slide
\[ F_a = m_s \times a \]
\[ F_a = 600 \text{ N} \]

Friction force
\[ F_f = 50 \text{ N} \]

Effective tension
\[ T_e = F_a + F_f \]
\[ T_e = 650 \text{ N} + 50 \text{ N} \]

### Step 2
Selected belt tooth form =>AT5 (Graph 2c)
For linear positioning, belts of the AT series are preferred, because of the higher cord and tooth stiffness.

### Step 3
Approximate number of pulley teeth
\[ \tilde{z}_p = \frac{\pi \times d_p}{5 \text{ mm}} \]
\[ \tilde{z}_p = 31.4 \]

Chosen number of teeth
\[ z_p = 32 \]
(greater than the recommended minimum number of pulley teeth for an AT5 belt \([z_{\text{min}} = 12]\) given in Table 1)

Pulley pitch diameter
\[ d = \frac{5 \text{ mm} \times 32}{\pi} \]
\[ d = 50.93 \text{ mm} \]

### Step 4
Preliminary number of belt teeth
\[ \tilde{z}_b = 2 \times \frac{C}{p} + z_p \]
\[ \tilde{z}_b = 1232 \]

Chosen number of belt teeth
\[ z = 1232 \]

Belt length
\[ L = z_p \times p \]
\[ L = 6160 \text{ mm} \]

(inc. 160mm over the slide)

### Step 5
Number of teeth in mesh
\[ z_m = \frac{z_p}{2} \]
\[ z_m = 16 \]

### Step 6
Belt pre-tension
\[ T_i = 1.1 \times 650 \text{ N} \]
\[ T_i = 715 \text{ N} \]

### Step 7
Maximum tight side tension
\[ T_{1\text{max}} = T_i + T_e \]
\[ T_{1\text{max}} = 1365 \text{ N} + 650 \text{ N} \]

Maximum slack side tension
\[ T_{2\text{max}} = T_{1\text{max}} - T_e \]
\[ T_{2\text{max}} = 1365 \text{ N} - 650 \text{ N} \]

### Step 8
Allowable belt tension
(from Table 1)

<table>
<thead>
<tr>
<th>Allowable effective tension</th>
</tr>
</thead>
<tbody>
<tr>
<td>( b \geq \frac{650 \text{ N}}{1270 \text{ N} \times 1 \times 0.96} )</td>
</tr>
<tr>
<td>( b \geq 13.3 \text{ mm} )</td>
</tr>
</tbody>
</table>

### Step 10
Belt stiffness
\[ k = \frac{c_{sp} \times b \times L_1 + L_2}{L_1 \times L_2} \]
\[ k = \frac{17600 \times \frac{N}{\text{mm}} \times 50 \text{ mm} \times 6000 \text{ mm}}{3290 \text{ mm} \times 2710 \text{ mm}} \]
\[ k = 592.2 \text{ N/mm} \]

Slide displacement
\[ \Delta x = \frac{F_{st}}{k} \]
\[ \Delta x = \frac{50 \text{ N}}{592.2 \text{ N/mm}} \]
\[ \Delta x = 0.084 \text{ mm} < 0.1 \text{ mm} \]

Static load on the slide \( F_{st} \) is equal to the friction force \( (F_{st} = F_f = 50 \text{ N}) \)