CONVEYOR BELTS

Engineering manual

SIG

Theory for belt calculation and selection

Introduction

Rubber conveyor belts are the most important system to convey bulk materials. Mines, quarries, cement works, power stations, ports, iron works are typical users of these handling systems;however, everywhere it is possible to see short and long conveyor belts in the most heterogeneous fields of application. Also where in the past tracks were used, at the present is normal to see rubber conveyor belts at work.
There are many and different reasons for this evolution and success as for example: very high conveying capacity, low maintenance, low power absorption and low price of installation.
Characteristics of conveyor belts have been incresed by the evolution of materials used for their construction: so, kilometers of center to center distance, thousand of tons per hour are usual at the present.
In particular, the high elongation of materials used for weft yarns permitted to increase the idler inclination and consequently to achieve high charge sections.
Also the problem of cut and impact, that in the past destroyed many kilometers of belts, now has been solved using particular electronics systems or steel reinforcements.

Typology of conveyor belts

Two types of belts are considered in this manual: textile and steel cord, both composed by a carrying carcass and two rubber covers. Rubber is used both to protect the plies and to guarantee the best adhesion between them.
According to the specific use, it is possible to build conveyor belts with many types of rubber in order to assure high abrasion resistance, heat resistance, oil resistance or autoextinguish and antistatic characteristics.

Both textile and steel cord belts are used for standard conveyor, elevator and tubular conveyor systems.

Textile belts are armored with two or more plies of particular synthetic materials as polyester and polyamide (nylon), in order to obtain the best compromise between different characteristics: low elongation, high tensile strength, good flexibility, low thickness and weight.

Steel cord belts are made with steel cables in the warp to achieve high tensile strength with very low elongation; it is also possible to have high elongation steel cords in the weft in order to assure very good cut and impact resistance and in the maintime high characteristics of trough-ability.

Manual purpose

Our last publications gave great importance to technical matters as calculation of belt tensile strength, belt dimension, pulley diameters, transition distance and minimum curves radius.
From this experience, we understood that many of our customers are interested also to know the basical theory for a preliminary personal calculation of the most important characteristics of a conveyor belt. For this reason, SIG technical department though to divulge its knowledge with this manual.
As belt conveyor calculation is easy to be understood but not very fast to be executed, SIG is in condition to give to everyone who needs frequent calculation, its up to date and well tested software package.
However, for particular application, it is necessary to make more careful investigations not included in this manual. For example, when conveyor belts are very long, with many vertical curves or when drive pulleys are more than one, we suggest to ask our technical department for the solution of each problem.

Section 1

1.1 Conveyor belt theory

The motion transmission from a pulley to a belt is only due to the wrapping friction existing between these two elements. Intuitively, friction coefficients and material characteristics being equals, as greater the arco of contact (wrap) is, as greater the friction will be; moreover, as for other type of motion transmission (cars, trains, lifts, cableways, etc.), as greater the friction is, as greater the motion transmission will be. In other words, the motion transmission is impossible without friction.

However, to observe the friction effects, a good belt tensioning is necessary in order to allow a suitable pressure on the pulley; on the contrary, if the tensioning is not enough to guarantee the minimum pressure for the motion transmission, belt slippings on the drive pulley could occour with power waste and increase of temperature and belt cover deterioration.

For example, this is the same problem of a car that tries to accelerate or to brake on the sand: as greater the friction (due to rubber cleats on the wheel surface) and the weight (wheel pressure on the sand) are, as easier the movement will be. Obviously, also for conveyor belts, the worst situation is at starting, when the greatest power is involved.

Another example to explain the importance of tensioning is the engine V belt: if it is too long and consequently has too low tension, the motion transmission could not be perfect and the alternator running could be very difficult.

So, we can understand the importance of "tension at the run off point of the drive pulley" also named "pretension". Pretension is a term which refers to a tension that must be applied to the belt by something of extern like a counterweight (Fig.1) or a screw take-up. To better understand why the pretension refers to the run off point of the drive pulley, we suggest to carefully consider the equations here below explained and moreover the following paragraph.

Parameters here above mentioned (pretension, friction coefficient and wrap) are strictly connected each other by the Eytelwein limit equation: for a conveyor belt with wrap α and friction coefficient μ, the relation between T_1 and T_2, respectively the tension at run on and run off point of the drive pulley, is

$$\frac{T_1}{T_2} \leq e^{\mu\alpha} \qquad (1.1)$$

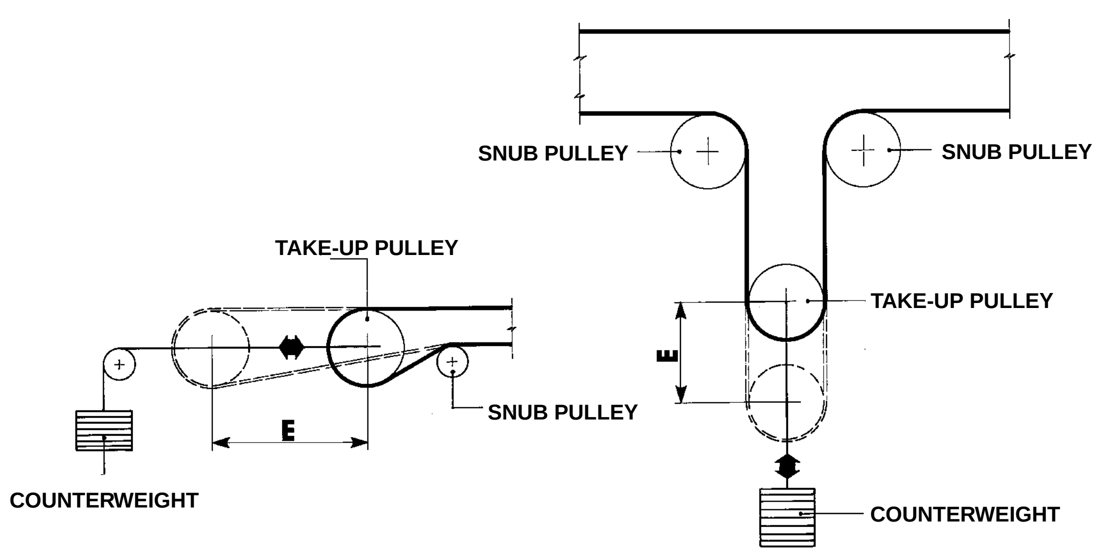


Fig. 1

At limit, when the friction coefficient μ is used to the limit of slipping, the equation becomes

$$\frac{T_1}{T_2} = e^{\mu\alpha} \qquad (1.2)$$

From this equation we understand that the maximum tension T_1 necessary to move the whole system is proportional to the tension T_2 at the other side of the pulley. So, known the T_2 value, the maximum value of T_1 is also known and depends on the coefficients μ and α.
If the motion characteristics require
$T_1 > T_2 \cdot e^{\mu\alpha}$, the power transmission is not possible and slippings occour on the drive pulley. These concepts will be developed in the next paragraph.
Moreover, the difference between T_1 and T_2 is the force F that the motor transmicts to the belt

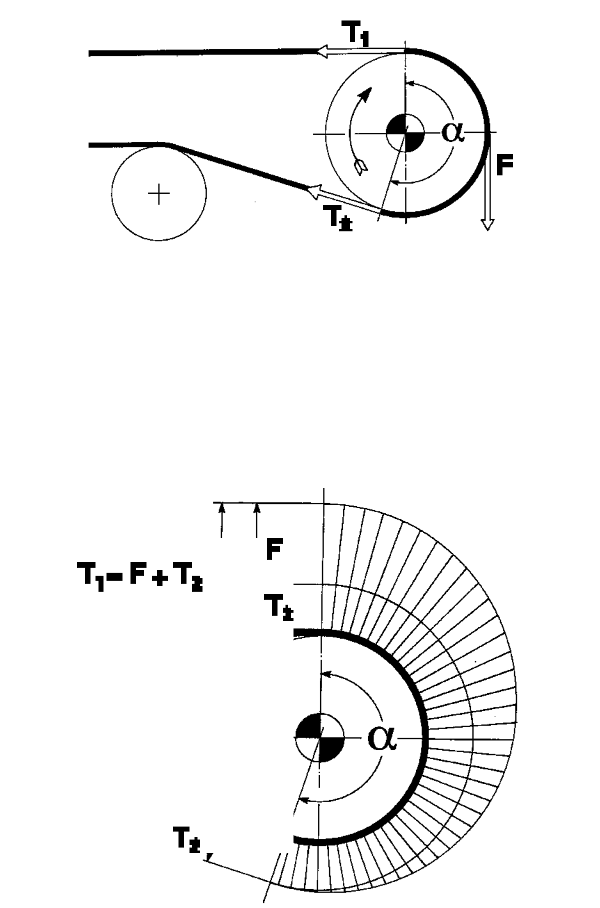


Fig. 2

and is such as to balance all the frictions along the belt and to permit the belt movement (Fig.2):

$$F = T_1 - T_2 \qquad (1.3)$$

F could be also named “friction force” as it has the responsability of the motion transmission; in fact, $F = 0$ or $T_1 = T_2$ means no motion transmission: the drive pulley is running but the belt is slipping on it.
Combining the equations 1.1 and 1.2 we have

$$\begin{cases} T_2 = F\left(\dfrac{1}{e^{\mu\alpha} - 1}\right) = KF \\ T_1 = F\left(1 + \dfrac{1}{e^{\mu\alpha} - 1}\right) = (K + 1)F \end{cases} \qquad (1.4)$$

So, knowing the friction factor

$$K = \frac{1}{e^{\mu\alpha} - 1} \qquad (1.5)$$

and the peripheral force F, it is possible to calculate the tension T_1 that the belt must stand and the tension T_2 that must be furnished to the belt by a take-up in order to guarantee the right motion transmission.

1.2 Pretension

As above explained, from equation 1.3 we understand the need to limit T_2, as adding itself to F, it defines the maximum tension T_1 which must obviously be reduced as much as possible. In the main time, T_2 must not have too low values such to compromise the motion transmission. For these reasons, it is fundamental to select the suitable T_2 as the best compromise between opposite requirements.
The meaning of the Eytelwein equation can be understood with a simple example showing the behaviour of a little winch. A rope handled by a man with a tension T_2 passes through the winch and is loaded with a tension T_1 at the other end (Fig. 3).

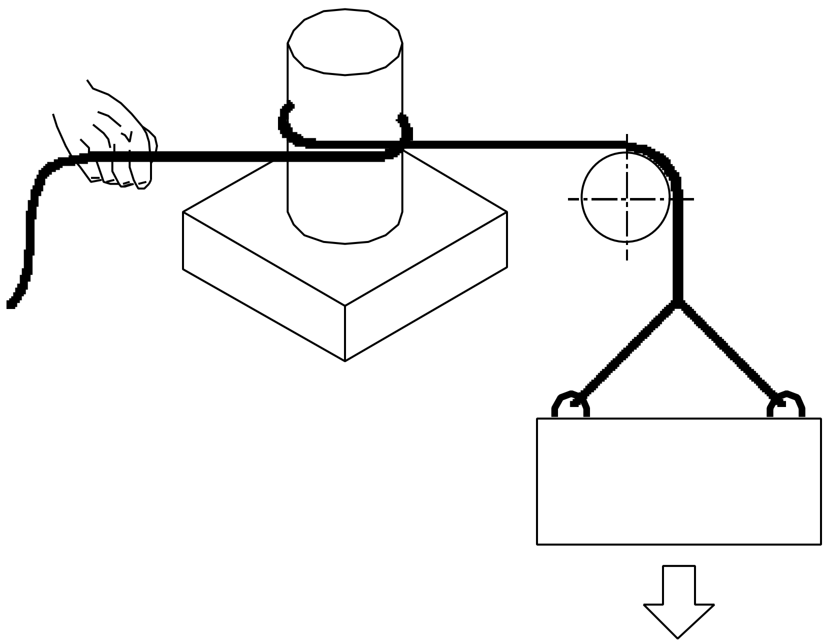


Fig. 3

The tension T_2 is as lower as higher the exponential $e^{\mu c}$ is and in particular as higher the wrap is. At the limit, if the rope makes several turns around the pulley, the tension T_2 could be extremely low.

A reduction of T_2 ($T_1/T_2 > e^{\mu\alpha}$) causes rope slippings on the pulley whereas an increment of it ($T_1/T_2 < e^{\mu\alpha}$) allows to take up a greater load T_1.

We note that the second case is better than the first because there are not slippings; however, the tension on the whole rope is uselessly increased. For example, this principle explains how a seaman can hold a big ship only with its own hands.

Coming back to the conveyor belts and considering that $T_2 = KF$, where F depends only on the total belt friction and for this reason it is not possible to modify it, when the tension $T_1 > T_2 \cdot e^{\mu\alpha}$ is growing up over evident tolerable values, it could be necessary to act in one of the following directions in order to increase

$$K = \frac{1}{e^{\mu\alpha} - 1}:$$

1. Increasing the friction coefficient μ, for example by clothing the drive pulley with striped rubber or by eliminating water or moisture presence.
2. Increasing wrap α by using snub pulleys or, if it is necessary, by the adoption of two or more drive pulleys.

On the contrary, if it is impossible to increase the friction coefficient μ, i.e. because of the moisture or it is not convenient to increase the wrap α, the only solution is increasing the pretension T_2. This fact will produce an increase af T_1 and, consequently, it will be necessary to use a stronger belt.

1.3 Pretension generation

As described at the point 1.1, the pretension T_2 must generally be produced by a suitable external device in the most convenient point of the conveyor. There are various type of take-up device: it is necessary to study the problem for each conveyor. Generally, automatic take-up are always the best theorical choice but they need a lot of space and are much more expensive than screw take-up.

1.3.1 Screw take-up (fix device)

Screw take-up are used for little center distance and low capacity conveyors, where elongations are not very important in the description of the running characteristics of the belt.
The principle of the pretension generation by screw take-up is based on the spreading apart of the two pulleys until the peripheral force is well transmitted without slippings; however, as great strenghts produce elongations, to have always the same right pretension it would be necessary to have a variable center distance; this fact means to spead apart the pulleys when the belt is elongated. So, when the center distance is fixed it is necessary to have, in empty running conditions too, a pretension value such as to take in consideration the elongation at worst conditions. As it is not possible to preview the necessary pretension with high math precision, the belt tensioning by screw take-up occurs empirically; this determines excessive tensions in the most of the running conditions and, particularly, at repose.

1.3.2 Counterweight (automatic take-up device)

Counterweight on the return section of the conveyor guarantees the length compensation that is impossible with a screw take-up; as the elongation phenomenon is not negligible, this kind of arrangement must always be present for conveyor with center distance greater than 80 meters. Also for lower center distance a counterweight could be suggested if the running conditions are particularly strong: high capacity, frequent full loading start-up. Only with a counterweight the pretension have always the desired value, for high belt elongation too.
When the counterweight displacement is difficult or dangerous or its value is so big that could produce negative inertia effect, it is better to generate the pretension by a winch take-up drived by a tension control system. The behaviour of a winch take-up is the same of a counterweight only if it has suitable dynamic characteristics; on the contrary, if winch take up gives only a fixed tension for every charge conditions, it doesn’t guarantee the length compensation as a screw take up.

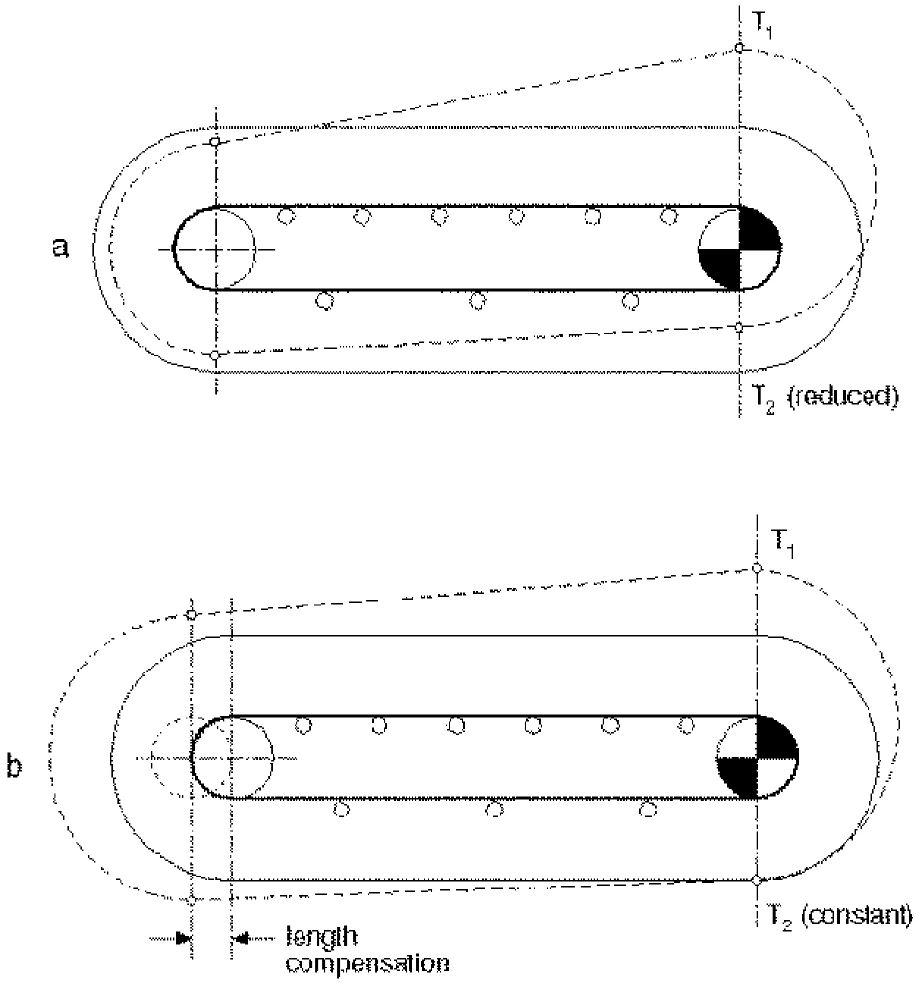


Fig. 4

1.3.3 Comparison between screw take-up and conterweight

In order to well understand the advantage of a counterweight in comparison with a screw take-up, we consider the previous drawing (Fig. 4).
The continuous line represents the tension profile in still conditions: all the belt sections theorically have the same tension. In the first case the tension is supplied by a screw take-up and it is impossible to calculate its value; in the second one it is produced by a counterweight displaced at run off point of the drive pulley and it is always equal to half of the counterweight value.
When the belt is running (dotted lines), particularly if it is full loaded, it presents elastic elongations caused by higher tensions; in the case of screw take-up (Fig.4a), the tension at run off point of the drive pulley is lower than in still conditions and it could become insufficient to guarantee the right transfer of motion from drive pulley to the belt.
On the contrary, with automatic devices (Fig. 4b), the belt elongation is compensated by a lowering of the counterweight i.e. by an increase of the center distance (automatic distance compensation); in the maintime, the pretension is always equal to the still value.
In conclusion, with a center distance compensation it is possible to supply the minimum pretension enough to allow the power transmission; obviously, in this case the pretension could be much lower than in the use of a screw take-up.

1.4 Pretension application point

As general rule, the pretension T_2 should be generated by counterweight or winch take-up with value equal to $F_V = 2 \cdot T_2$ (Fig. 5) in the point where it is necessary, i.e. at run off point of the drive pulley. In this ideal case, the pretension value T_2 is applied to the belt without any type of inertia phenomenon and whatever the running situation.

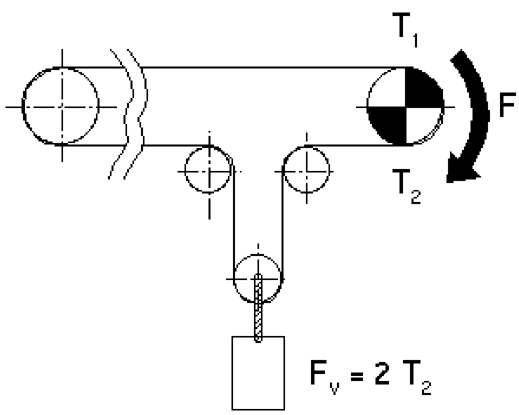


Fig. 5

On the contrary, if the counterweight is applied far from the drive pulley, the elongation at starting conditions is not istantly compensated as it is necessary waiting the tensioning of the belt in the return section. For raising conveyor belts the problem is less evident because the belt weight in the return section creates an increasing of the pretension at the drive pulley.

Now, we analize the behaviour of a belt when the pretension is not enough or the take-up device is not well displaced. If the starting pretension has too low values, belt slips on the drive pulley and consequently waves in the return belt section are generated; so, the motor has not any type of load and increases its rate. For this reason, when the counterweight effect is enough to transmit the motion, a peripheral force much greater than the standard values is suddenly released on the belt; in some particular and heavy case this phenomenon could also determine serious belt damagings.

This problem is as more evident as greater the elongation characteristics of the belt are:so, because of its high elastic modulus, a metallic belt will certainly have less problems than a textile one.

In addition to these problems of inertia, it is usually advisable to apply the counterweight near the drive pulley also to reduce its value at the minimum; in fact, in the section between the counterweight and the drive pulley, friction resistances are generally produced during the running:so, as far from the drive pulley the counterweight is, as greater these frictions are and consequently as greater the counterweight value must be.

Consequently, if the minimum counterweight has value

$$F_{V_{min}} = 2T_2 \qquad (1.6)$$

using a take-up at the return pulley, it must be decidedly greater i.e.

$$F_V = 2\,(T_2 + T_{W_r}) \qquad (1.7)$$

in order to guarantee the right pretension T_2 at the drive pulley, where T_{W_r} is the tension necessary to balance the frictions in the return belt section (Fig. 6).

As already mentioned, only with raising conveyors it could be advisable to apply the take-up at the tail because the belt weight in the return section could determine a tension greater than the friction force T_{W_r} i.e. favour the belt pretensioning. In this case we have

$$F_V = 2\,(T_2 + T_{W_r} - W_b H) \qquad (1.8)$$

where

W_b = belt weight [Kg/m]

H = difference in height between drive and take-up [m]

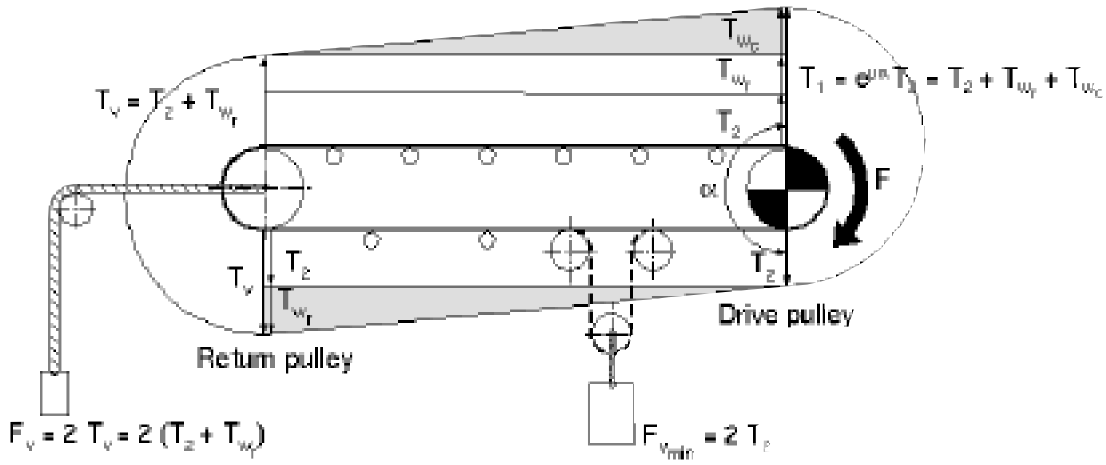


Fig. 6

So, if $W_bH > T_{w_r}$ it is better to apply the counterweight at the tail of the conveyor belt in order to reduce its value.

Finally, in cases of drive pulley at the tail of the conveyor, the counterweight is applied at the head of the conveyor, i.e. the nearest admissible position. For this reason it must have a value

$$F_V = 2\,(T_2 + T_{W_c}) \qquad (1.9)$$

where T_{W_c}, analogously to T_{W_r}, is the tension necessary to balance the frictions in the carrying section. Both for the presence of the load on the belt and because of a greater weight of the moving part of the carrying idlers, T_{W_c} is always bigger than T_{W_r} and for this reason the counterweight value is necessarily greater. As the pretension effect is reflected on the whole belt, a greater average stress and consequently a greater average elongation occours.

With the exception of particular requirements, drive pulleys at the tail are theorically advisable only for descent conveyors, in particular when the belt and the loading weight create an effect that reduces the tension T_{W_c} due to the frictions, at values lower than T_{W_r}. You have

$$T_v = T_2 + T_{W_c} - (W_b + W_m)\,H \qquad (1.10)$$

where:

W_m = weight of the load for linear meter [Kg/m].

1.5 Compound drive

If the peripheral force that has to be transmitted to the belt is too high or it is not possible to guarantee a reasonable friction coefficient between drive pulley and belt, it is advisable to reduce the pretension by increasing the arc of contact (wrap); over 210° this means to adopt two drive pulleys (Fig. 7).

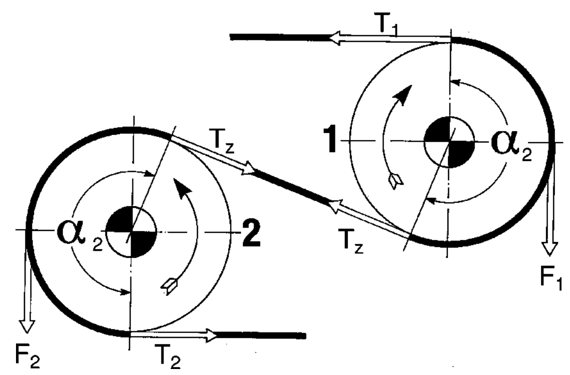


Fig. 7

Neglecting belt elongations along the drive pulleys and combining the Eytelwein equations for the two drive pulleys, we have:

a) Pulley nr. 1

$$\frac{T_1}{T_Z} = e^{\mu\alpha_1} \qquad (1.11)$$

$$T_1 - T_Z = F_1 \qquad (1.12)$$

b) Pulley nr. 2

$$\frac{T_Z}{T_2} = e^{\mu\alpha_2} \qquad (1.13)$$

$$T_Z - T_2 = F_2 \qquad (1.14)$$

It follows, as for only one drive pulley

$$F = F_1 + F_2 = T_1 - T_2 \qquad (1.15)$$

Moreover, multiplying member to member the Eytelwein equations we obtain:

$$\frac{T_1\ T_Z}{T_Z\ T_2} = e^{\mu\alpha_1}\ e^{\mu\alpha_2} \qquad (1.16)$$

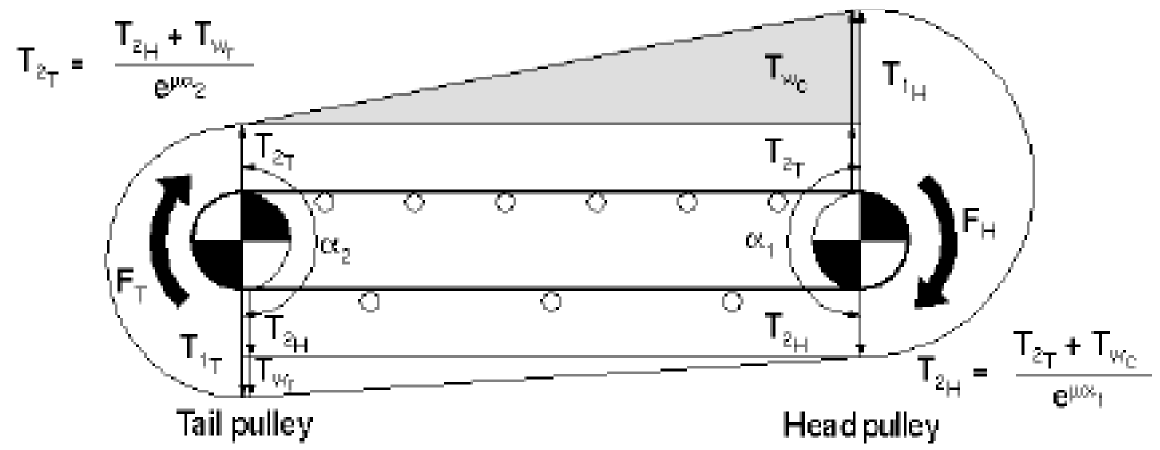
and so

$$\frac{T_1}{T_2} = e^{\mu\,(\alpha_1 + \alpha_2)} = e^{\mu\alpha} \qquad (1.17)$$

where $\alpha = \alpha_1 + \alpha_2$ is the total wrap.
So, for the tension calculation of this drive system, we can operate as we have only one drive pulley with wrap equal to the sum of each single wrap.
For practical design of a compound drive, the subdivision of the total motor power is made according to this simple consideration: the power applied to the drive pulley nr. 2 is such as to guarantee the minimum tension necessary to avoid slippings at the drive pulley nr. 1; so, the counterweigh will have a value equal to T_2 instead of T_Z. In other words, the drive pulley nr. 2 amplifies the conterweight value from T_2 to T_Z.

1.6 Head-tail drive

The same consideration has to be taken in order to subdivide the motor power between head and drive pulleys (Fig. 8). The only difference is that friction compounds between the two drive pulleys has to be considered; for this reason it is not possible to consider the total wrap as the sum of the two single ones.
Intuitively, as for compound drive, the power at the tail pulley must be such as to generate a tension T_{1_T} different from T_{2_H} for the friction values along the return section of the conveyor T_{W_r}. So, the counterweight applied at the tail pulley will be equal to T_{2_T} instead of T_{2_H}. If the motor power is subdivided in the right manner, we have great advantages from this tipology of driving as the average tensions along the conveyor will decrease.



$T_{2_T} = \frac{T_{2_H} + T_{W_r}}{e^{\mu\alpha_2}}$
T_{W_C}
T_{1_H}
T_{2_T}
T_{2_T}
F_H
α_2
α_1
F_T
T_{2_H}
T_{2_H}
T_{1_T}
T_{W_r}
$T_{2_H} = \frac{T_{2_T} + T_{W_C}}{e^{\mu\alpha_1}}$
Tail pulley
Head pulley

Fig. 8

Section 2

2.1 Design of a conveyor belt

The peripheral force F that the drive pulley transmits to the belt, must overcome all the resistances which oppose to the motion; F can be considered as sum of the terms exposed in the following paragraphs.

2.1.1 Forces necessary for movement of empty belt and carrying idlers

$$F_1 = CfL\left(2q\frac{B}{1000}\cos\beta + \frac{q_r'}{a'} + \frac{q_r''}{a''}\right)$$

where

C = length coefficient calculated by an exponential formula that translates the Graph 2.

f = idler friction coefficient (Tab. 9)

L = center to center distance [m]

q = belt weight for square meter [Kg/m²] (Tab.13)

B = belt width [mm]

β = average belt slope [°]

qr', qr'' = weight of moving parts of idlers along carrying and return section [Kg/cad] (Tab. 11)

a', a'' = distance between carrying and return idlers [m] (Tab. 12)

In the case of Flexobord®, (belts with rubber cleats and edges) it is advisable to increase F_1 of 20%, in order to consider the particular running conditions.

2.1.2 Forces necessary for the translation of the load

$$F2 = CfL\frac{Q}{3,6\ v}\cos\beta \qquad (2.2)$$

where

Q = capacity [Ton/h]

v = speed [m/sec]

2.1.3 Forces necessary for the elevation of the load

$$F_3 = \frac{QH}{3,6\ v} \qquad (2.3)$$

where

H = elevation of the conveyor [m]

2.1.4 Auxiliary forces

Auxiliary forces for particular applications must be considered each time depending on the characteristics of the application.

2.1.5 Total force and motor power calculation

The total peripheral force F necessary to transmit to the belt is

$$F = F_1 + F_2 + F_3 + F_4 \qquad (2.4)$$

The theorical motor power P_a, necessary to transmit F to the belt, is:

$$P_a = Fv/102\ \ [kW] \qquad (2.5)$$

Considering the mechanical efficiency coefficient μ for the transmission (Tab. 8), we can find the required motor power P_m

$$P_m = \frac{Fv}{102}\Big/\eta \qquad (2.6)$$

2.1.6 Tension calculation

Now, we go on to the calculation of the belt tensions; first of all, we calculate the friction factor K

$$K = \frac{1}{e^{\mu\alpha} - 1} \qquad (2.7)$$

from which the nominal tensions to the drive pulley respectively at run off (T_{2_n}) and run on point (T_{1_n}) are coming:

$$T_{2_n} = FK$$
$$T_{1_n} = F\,(K + 1) \qquad (2.8)$$

According to the paragraph 1.3, if the conveyor has a simple screw take up instead of a counterweight, we suggest to increase the friction factor of 40% in such way as to take in consideration a possible overtensioning.
As we have already considered at point 1.4, summing the eventual effect (only if > 0) of the overtension T_V [daN] due to the counterweight value F_V (see next paragraph for its calcualtion)

$$T_v = \frac{F_v}{2} - L_1\left(q\frac{B}{1000} + \frac{q_r''}{a''}\right)f + L_1 q\frac{B}{1000}\operatorname{sen}\beta - T_{2_n} \qquad (2.9)$$

we obtain the following effective tensions to the drive pulley

$$T_2 = T_{2_n} + T_V$$
$$T_1 = T_{1_n} + T_V \qquad (2.10)$$

Known T_1 you can find the minimum tensile strength CR_m of the belt by calculating, first of all, the working tension CL

$$CL = \frac{10T_1}{B} \quad [N / mm] \qquad (2.11)$$

and following multiplying for the required safety factor f_s (usually 10 for textile and 8 for metallic belts)

$$CR_m = CL \cdot f_s \quad [N / mm] \qquad (2.12)$$

Once the belt style CR is chosen, it is possible to verify the effective safety factor f_s' using the inverse of the previous relation

$$f_s' = CR/CL \qquad (2.13)$$

2.1.7 Take-up calculation

Now we operate the calculation of the take-up F_V, necessary to give the suitable pretension to the belt in such way as to:

2.1.7.1 Guarantee the motion transmission without slippings.

If the counterweight is near the drive pulley its value is twice the T_2 value. Often, the counterweight is in the return section of the conveyor far from the drive pulley (Fig. 9a) but it is not unusual to find it on the tail pulley (Fig. 9b); for this reason it is necessary to take in consideration that the counterweight effect is reduced because of the presence of frictions in the return section.

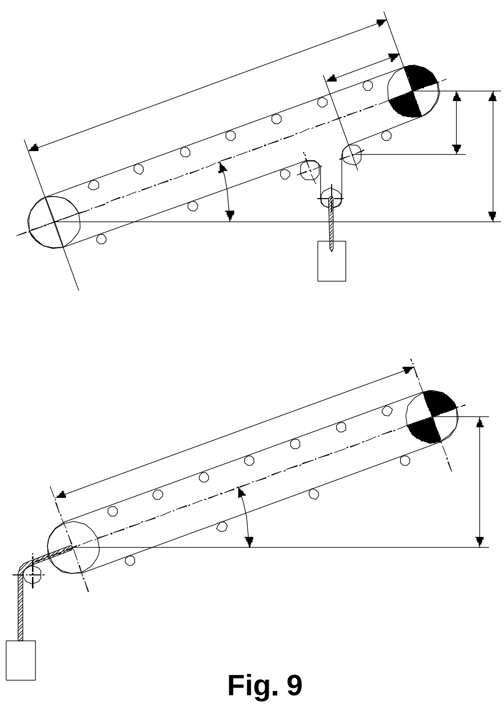


Fig. 9

Moreover, if the conveyor is not horizontal, the same belt weight determines a further increas or reduction of the belt tension, respectively if the counterweight has a position lower or higher than the drive pulley. So, the take-up must generate a tension equal to

$$T_{v_1} = T_{2_n} + L_1\left(q\frac{B}{1000} + \frac{q_r''}{a''}\right)f - L_1 q\frac{B}{1000}\operatorname{sen}\beta \qquad (2.14)$$

where
L_1 = distance between the counterweight and the drive pulley along the belt [m].

2.1.7.2 Guarantee a minimum belt sag

The belt and the load weight determine a belt bending between the idlers as greater as higher the distance between the idlers and the applied weight are.
In the hypothesis that the bending of the belt between two idlers is similar to a catenary, the minimum tension necessary to apply to the belt is:

$$T_{sup} = \frac{a'}{8S_1}\left(q\frac{B}{1000} + \frac{Q}{3{,}6v}\right)$$

for the carrying section (2.15)

$$T_{inf} = \frac{a''}{8S_2} q \frac{B}{1000}$$

for the return section

where
S_1, S_2 = maximum allowable ratio between the sag and the idler distance.
Usually S_1, S_2 = 1 ÷ 2%. For greater values, an excessive increase of the peripheral force is necessary in order to move the belt and the material; moreover the material running could have problems of instability. On the contrary, to have a too small sag very high and useless pretension values are necessary.
In all, the counterweight value is twice the maximum among the three a.m. tensions:

$$F_{V_{min}} = \max \{T_{V_1}, T_{sup}, T_{inf}\} \cdot 2 \qquad (2.16)$$

The same system calculation is used in the case of screw take-up too.

2.1.8 Check according to the installed moto power

On the basis of the real installed motor power, it is always better to verify that the maximum peripheral force F_t that the motor could transmit to the belt is bearable by the belt, over all in the starting conditions.

$$F_t = N\eta\ 102/v \qquad (2.17)$$

from whom we derive the maximum tension $T_{1_{max}}$ that the belt could bear if the motor released on it its whole power

$$T_{1_{max}} = F_t\,(K + 1) \qquad (2.18)$$

to which we add, as known, the term

$$T_v = \frac{F_v}{2} - L_1\left(q\frac{B}{1000} + \frac{q_r''}{a''}\right)f + L_1 q \frac{B}{1000}\,\mathrm{sen}\,\beta - F_t K \qquad (2.19)$$

(only if positive) taking in consideration the counterweight effect.

Now we analize the maximum starting tension T_a multiplying F_t for the starting factor w (Tab.7)

$$T_a = F_t\,(K + 1)\,w \qquad (2.20)$$

to which we add the counterweight effect

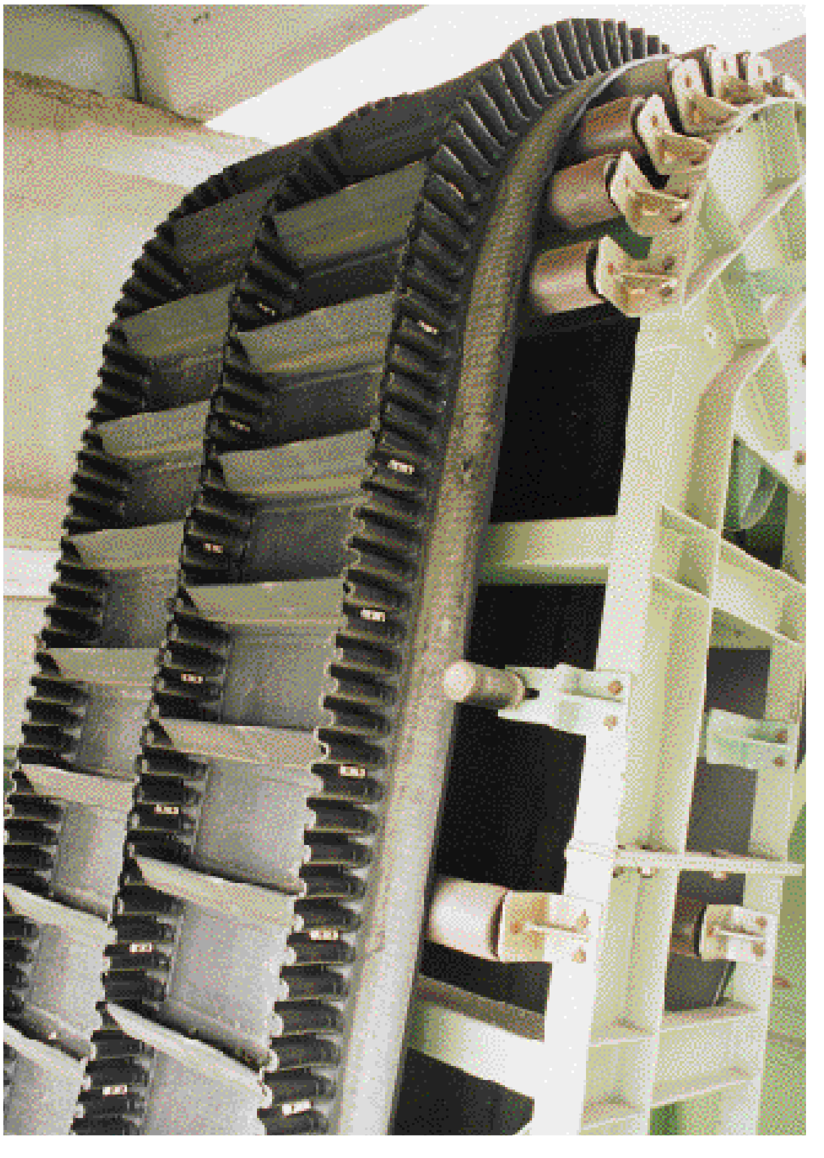
$$T_{v_a} = \frac{F_v}{2} - L_1\left(q\frac{B}{1000} + \frac{q_r''}{a''}\right)f + L_1 q \frac{B}{1000}\,\mathrm{sen}\,\beta - F_t K w \qquad (2.21)$$

Chosen the belt style CR, we can verify the starting safety factor

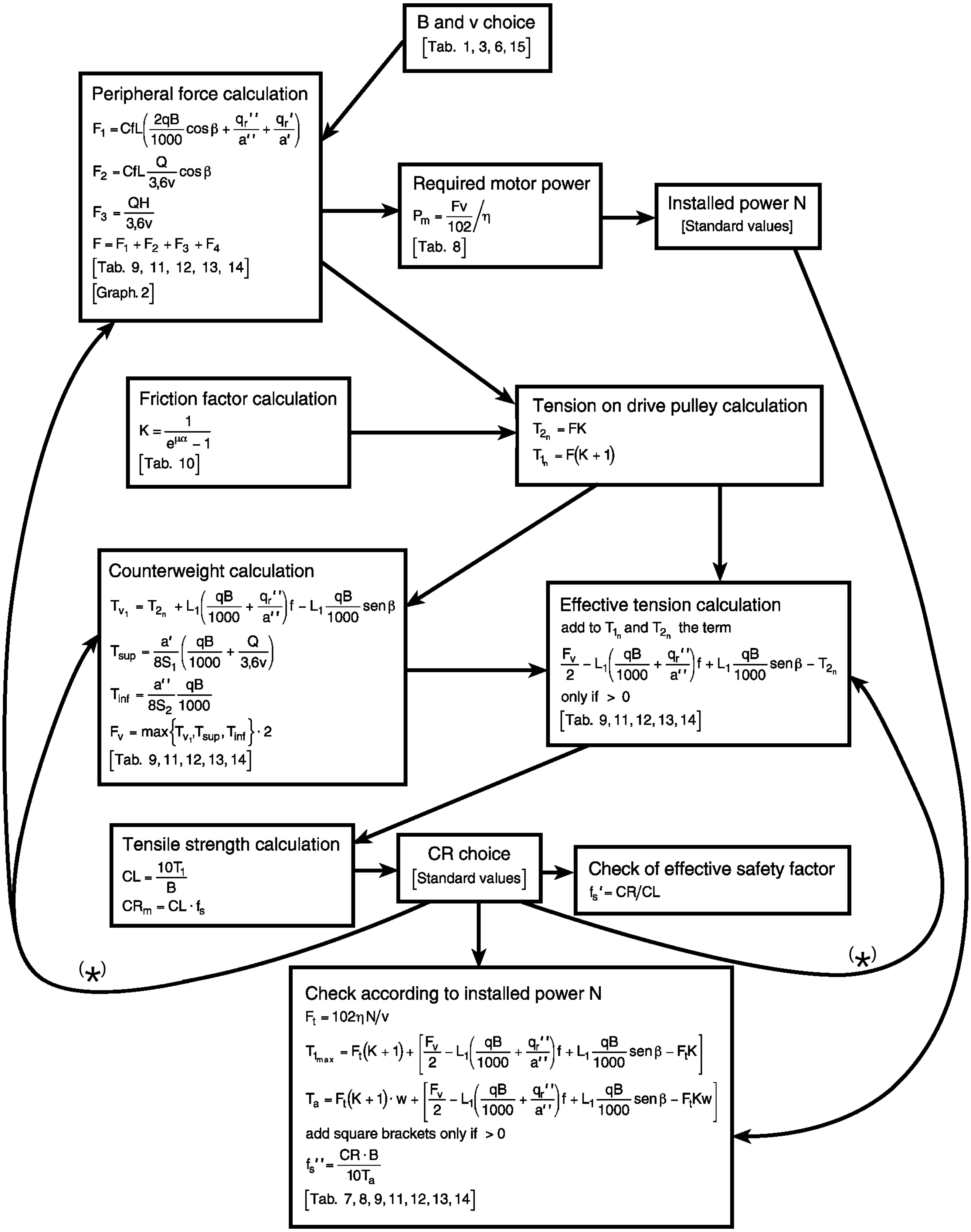
$$f_s'' = \frac{CR \cdot B}{10 T_a} \qquad (2.22)$$

that must not be less than 8 for textile belts and 6 for steel cord belts, in particular if the plant is characterized by several and frequent startings.





2.2 Resumptive scheme for conveyor belt calculation



(*) Chosen the tensile strength CR, we operate a new calculation with the right belt weight “q”

2.3 Symbols

B	[m]	= belt width
v	[m/sec]	= speed
C		= length coefficient
f		= friction coefficient between belt and idlers
L	[m]	= center to center distance
H	[m]	= elevation
q	[Kg/m²]	= belt weight
β	[°]	= average belt inclination
Q	[Ton/h]	= capacity
F_1	[daN]	= force necessary for empty belt movement
F_2	[daN]	= force necessary for load horizontal movement
F_3	[daN]	= force necessary for load elevation
F_4	[daN]	= auxiliary forces
F	[daN]	= total peripheral force
P_m	[KW]	= required motor power
η		= transmission efficiency
N	[KW]	= applied motor power
μ		= friction coefficient between drive pulley and belt
α	[°]	= wrap
K		= friction factor
T_{1_n}	[daN]	= nominal max tension
T_{2_n}	[daN]	= nominal min tension
T_{V_1}	[daN]	= min tension to allow the motion transmission

T_1	[daN]	= real max tension
T_2	[daN]	= real min tension
L_1	[m]	= distance between take-up and drive pulley
qr'	[Kg/cad]	= weight of moving parts along the carrying section
qr''	[Kg/cad]	= weight of moving parts along the return section
a'	[m]	= carrying idler pitch
a''	[m]	= return idler pitch
S_1		= max sag along the carrying section
S_2		= max sag along the return section
F_V	[daN]	= counterweight value
T_V	[N/mm]	= take-up overtension
CL	[N/mm]	= working tension
CR_m	[N/mm]	= minimum required tensile strength
CR		= belt tensile strength
fs		= required safety factor
fs'		= effective safety factor
fs''	[daN]	= effective safety factor at starting
T_{1max}	[daN]	= max tension at standard conditions
F_t	[daN]	= max peripheral force at standard conditions
T_a	[daN]	= max tension at starting
w		= starting motor coefficient

2.4 Example of calculation

On the basis of the previous calculation sketch we work out an example of an ascent conveyor belt.

2.4.1 Datas

- Material: crushed limestone
- Lump size: max 500 mm sized
- Capacity Q = 1500 Ton/h
- Center distance L = 300 mm
- Elevation H = 30 m
- Max slope $\beta_{max} = 6{,}5°$
- Wrap $\alpha = 210°$
- Distance between drive pulley and counterweight $L_1 = 80$ m
- Idler inclination = 45°
- Double return idlers
- Idler diameter = 133 mm
- Triple reduction
- Squirrel cage motor with fluid coupling
- Rubber coated pulley with wet ambient
- Standard running conditions
- Antiabrasive cover rubber (CL type)
- Max sag $\beta \leq 1\%$

2.4.2 Considerations about material

From Tab. 2 we have the following considerations
- Density 1,5 Ton/m^3
- Angle of repose 38°
- Angle of surcharge 25°
- Max conveying angle 18° (less than β_{max})

Depending on the material lump size from Tab. 3 we can obtain the minimum belt width B_{min} = 1200 mm. From Tab. 6 it follows a max speed of 2,6 m/sec.

2.4.3 Capacity calculation

From Section 7 it follows that the capacity at the speed of 1 m/sec for horizontal belts is: 743,1 m^3/h.
It is necessary to multiply this value for the dip factor k (Graph. 1) equal to 0,98. In conclusion we have 682,4 m^3/h that means 1023,6 Ton/h.

To obtain a capacity of 1500 Ton/h, we need a speed of

$$v = \frac{1500}{1023{,}6} = 1{,}47 \text{ m/sec.}$$

If we choose a speed of 1,5 m/sec, we satisfy both the required capacity and the condition about the maximum suggested speed.
As the required capacity has been achieved, we can adopt the minimum belt width B = 1200 mm; on the contrary, if it had not been possible to guarantee the capacity, it would have been necessary to increase the belt width and to recalculate the new value of capacity in order to obtain the minimum required speed.

2.4.4 Peripheral force calculation

At the beginning of this calculation, it is necessary making an hypothesis about the belt style: we suppose to use a 1600/4 belt whose carcass weight of 11,9 Kg/m^2 is available in Tab.14.
In order to take into consideration the top cover thickness we use Tab. 15: as antiabrasive rubber CL is required, we choose 8 mm for top cover and 4 mm for bottom cover. Globally we have

$$(8 + 4) \text{ mm} \cdot 1{,}2 \text{ Kg} / (\text{m}^2 \cdot \text{mm}) + 11{,}9 \text{ Kg} / \text{m}^2 =$$

$$= (14{,}4 + 11{,}9) \text{ Kg} / \text{m}^2 = 26{,}3 \text{ Kg} / \text{m}^2$$

At the end of the calculation, if we had to adopt a style belt different than the one chosen at the beginning, it would be necessary to check its own weight in order to operate a new calculation of the peripheral force F; this is really necessary only if the weight differences are remarkable. With our spreed sheet it is very easy to operate new calculations in order to improve the precision of the result; however, we also observe that the effect of empty belt weight doesn't modify substantially the F value.
The necessary coefficient for the calculation are:

- Length coeff. C = 1,30 (Graph. 1)
- Idler friction coeff. f = 0,020 (Tab. 9)
- Weight of carrying idlers (tern of idlers) q_r' = 30,3 Kg (Tab. 11)
- Weight of return idlers (couple of idlers) q_r'' = 26,9 Kg (Tab. 11)
- Carrying idler pitch a' = 0,9 m (Tab.13)
- Return idler pitch a'' = 3 m (Tab. 13)
- Average conveyor slope

$$\beta = \text{tg}\,\frac{H}{L} = 5{,}74°$$

It follows

$$F_1 = CfL\left(2\frac{qB}{1000}\cos\beta + \frac{q_r'}{a'} + \frac{q_r''}{a''}\right) =$$

$$= 1{,}30 \cdot 0{,}02 \cdot 300 \cdot (2 \cdot 26{,}3 \cdot \frac{1200}{1000} \cdot 0{,}995 +$$

$$+ \frac{30{,}3}{0{,}9} + \frac{26{,}9}{3}) = 820 \text{ daN}$$

$$F_2 = CfL\frac{Q}{3{,}6v}\cos\beta =$$

$$= 1{,}30 \cdot 0{,}02 \cdot 300 \cdot \frac{1500}{3{,}6 \cdot 1{,}5} \cdot 0{,}995 = 2150 \text{ daN}$$

$$F_3 = \frac{QH}{3{,}6v} = \frac{1500 \cdot 30}{3{,}6 \cdot 1{,}5} = 8333 \text{ daN}$$

$$F = F_1 + F_2 + F_3 = 11303 \text{ daN}$$

2.4.5 Power calculation

When we know F value, it is possible to calculate the required motor power for its transmission: from Tab. 8 we have the transmission efficiency coefficient η, so

$$P_m = \frac{Fv}{102}/\eta = \frac{11303}{102} \cdot 1{,}5/0{,}94 = 176{,}83 \text{ KW}$$

In order to respect the standard, we have to choose 200 KW of motor power.

2.4.6 Tension calculation

Side by side, we can calculate the friction factor K using the friction coefficient μ between belt and pulley (Tab.10)

$$K = \frac{1}{e^{\mu\alpha} - 1} = \frac{1}{e^{0{,}35 \cdot 3{,}66} - 1} = 0{,}384$$

Known K, we calculate the nominal tension on the drive pulley

T_{2_n} = FK = 4337 daN

T_{1_n} = F (K + 1) = 15640 daN

2.4.7 Take-up calculation

The pretension F_v due to the take up must eventually be added to the tensions T_{1_n} and T_{2_n}. Now we calculate the F_V value.

Known the distance L_1 between the counterweight and the drive pulley, we calculate the minimum tension necessary to transmit the motion without any belt slipping.

$$T_{V_1} = T_{2_n} + L_1\left(\frac{qB}{1000} + \frac{q_r''}{a''}\right)f +$$

$$-L_1 q\frac{B}{1000} \cdot \text{sen}\beta = 4337 +$$

$$+80 \cdot \left(26{,}3 \cdot \frac{1200}{1000} + \frac{26{,}9}{3}\right) \cdot 0{,}02 +$$

$$-80 \cdot \frac{1200}{1000} \cdot 26{,}3 \cdot 0{,}1 = 4149 \text{ daN}$$

Moreover, with a maximum sag S of the belt between the idlers equal to 1% of their pitch, the minimum tension required must be, respectively for the carrying and the return section:

$$T_{min} = \frac{a'}{8S_1}\left(qB + \frac{Q}{3{,}6v}\right) =$$

$$\frac{0{,}9}{8 \cdot 0{,}01}\left(26{,}3 \cdot 1{,}2 + \frac{1500}{3{,}6 \cdot 1{,}5}\right) = 3480 \text{ daN}$$

$$T_{inf} = \frac{a''}{8S_2} qB =$$

$$= \frac{3}{8 \cdot 0{,}01} \cdot 26{,}3 \cdot 1{,}2 = 1184 \text{ daN}$$

Now we choose the maximum tension among these just calculated values:

$$F_{V_{min}} = 2\,(T_{V_1}, T_{sup}, T_{inf}) = 8298 \text{ daN}$$

If we choose F_v = 9500 daN, we have an over-tension T_V, in comparison with T_{2_n}, equal to

$$T_v = \frac{F_v}{2} - L_1\left(\frac{qB}{1000} + \frac{q_r''}{a''}\right)f + L_1\frac{qB}{1000}\operatorname{sen}\beta +$$

$$-T_{2_n} = \frac{9500}{2} - 80 \cdot (26{,}3 \cdot \frac{1200}{1000} +$$

$$+\frac{26{,}9}{3}) \cdot 0{,}02 + 80 \cdot \frac{1200}{1000} \cdot 26{,}3 \cdot 0{,}1 - 4337 =$$
$$= 601 \text{ daN}$$

Thanks to this calculation, the real tensions on the drive pulley are:

$T_1 = T_{1_n} + T_V = 15640 + 601 = 16241$ daN

$T_2 = T_{2_n} + T_V = 4337 + 601 = 4938$ daN

2.4.8 Tensile strength calculation

Known T_1, we calculate the working tension CL dividing for the belt width B

$$CL = \frac{10T_1}{B} = \frac{10 \cdot 16241}{1200} = 135{,}34 \text{ N/mm}$$

and the breaking tension CR_m

$CR_m = CL \cdot f_s = 135{,}34 \cdot 10 =$

$= 1353{,}4$ N /mm

Chosen the belt style 1600 N/mm, we verify the effective safety factor f_s'

$$f_s' = \frac{1600}{135{,}34} = 11{,}8$$

2.4.9 Check according to the installed motor power

Finally, on the basis of the applied motor power, we check the starting safety factor f_s'' using the properly starting coefficient w (Tab.7).
First of all, we calculate the peripheral force F_t that the motor could transmit to the belt

$F_t = 102\eta N/v = 102 \cdot 0{,}94 \cdot 200/1{,}5 =$

$= 12784$ daN

so

$T_{1_{max}} = F_t\,(K + 1) = 12784 \cdot (0{,}384 + 1) =$

$= 17689$ daN

Now, we check if the counterweight produces an effect higher than the tension F_tK at run off point of the drive pulley.

$$T_{v_a} = \frac{F_v}{2} - L_1\left(\frac{qB}{1000} + \frac{q_r''}{a''}\right)f +$$

$$+L_1\frac{qB}{1000}\operatorname{sen}\beta - F_tK = \frac{9500}{2} +$$

$$-80 \cdot \left(26{,}3 \cdot \frac{1200}{1000} + \frac{26{,}9}{3}\right) \cdot 0{,}02 +$$

$$+80 \cdot \frac{1200}{1000} \cdot 26{,}3 \cdot 0{,}1 - 12784 \cdot 0{,}384 =$$

$$= -566 \text{ daN}$$

An overtension T_{v_a} negative means that the tension F_tK is higher than the one produced by the counterweight; so, the value of $T_{1_{max}}$ does not change. It follows

$T_a = T_{1_{max}}\, w = 17693 \cdot 1{,}3 = 22996$ daN

and

$$f_s'' = \frac{CR \cdot B}{10T_a} = \frac{1600 \cdot 1200}{10 \cdot 22996} = 8{,}3$$

CUSTOMER:

Example of calculation

Rdo:

Determination of fundamental belt dimensions

according to ISO 5048

Belt reference

GENERAL FEATURES						
Belt width (Tab. 3)	mm	B =	1200	0	0	0
Capacity (Tab. 1)	Ton/h	Q =	1500,00	0,00	0,00	0,00
Speed (Tab. 6)	m/sec	v =	1,50	0,00	0,00	0,00
Center distance	m	L =	300,00	0,00	0,00	0,00
Elevation	m	H =	30,00	0,00	0,00	0,00
Average belt slope "β"	deg		5,74	0,00	0,00	0,00
Wrap (arc of contact) "α"	deg		210,00	180,00	180,00	180,00
Carrying idlers weight (Tab. 11)	Kg	qr' =	30,30	0,00	0,00	0,00
Carrying idlers pitch (Tab. 12)	m	a' =	0,90	1,00	1,00	1,00
Return idlers weight (Tab. 11)	Kg	qr" =	26,90	0,00	0,00	0,00
Return idlers pitch (Tab. 12)	m	a" =	3,00	3,00	3,00	3,00
Belt weight (Tab. 13)	Kg/m²	q =	26,30	0,00	0,00	0,00
Length coeff.		C =	1,30	0,00	0,00	0,00
Frict. coeff. idlers (Tab. 9)		f =	0,020	0,022	0,022	0,022
Frict. coeff. drive pulley/belt (Tab. 10)		μ =	0,35	0,35	0,35	0,35
Take up.: Auto. 1,0/Man. 1,4			1,00	1,00	1,00	1,00
Wrap factor		K =	0,384	0,499	0,499	0,499
Counterweight	daN	Fv =	9500	0	0	0
PERIPHERAL FORCE ON THE DRIVE PULLEY						
Empty belt running	daN	F1 =	820	0	0	0
Material conveyance	daN	F2 =	2150	0	0	0
Load elevation	daN	F3 =	8333	0	0	0
Special resistances	daN	F4 =				
Total strength	***daN***	***F =***	***11303***	***0***	***0***	***0***
TENSION ON THE DRIVE PULLEY (HEAD)						
Max rated tension	daN	T1n =	15640	0	0	0
Min rated tension	daN	T2n=	4337	0	0	0
Take-up overtension	daN	Tv=	601	0	0	0
Tight-side tension	***daN***	***T1=***	***16241***	***0***	***0***	***0***
Slack-side tension	daN	T2 =	4938	0	0	0
BELT STRENGTH FEATURES						
Working tension	KN/m	CL =	135,34	0,00	0,00	0,00
Required safety factor		fs =	10,0	10,0	10,0	10,0
Min breaking load	***KN/m***	***CR =***	***1353,4***	***0,0***	***0,0***	***0,0***
Effective safety factor		**fs' =**	**11,8**	**0,0**	**0,0**	**0,0**

REQUIRED MOTOR POWER						
Required power	kW	Pa =	166,22	0,00	0,00	0,00
Drive efficiency (Tab. 8)		η =	0,94	0,95	0,95	0,95
Min motor power	**kW**	**Pm =**	**176,83**	**0,00**	**0,00**	**0,00**
Effective motor power	**kW**	**N =**	**200,0**	**0,0**	**0,0**	**0,0**
CHECK ACCORDING TO THE INSTALLED MOTOR POWER						
Tangential tension	daN	Ft =	12784	0	0	0
Max belt tension	daN	T1max =	17689	0	0	0
Working tension	KN/m	CL =	147,4	0,0	0,0	0,0
CR_{min} according to starting cond.	KN/m	CR_m =	1474,1	0,0	0,0	0,0
Belt tensile strength	***KN/m***	***CR =***	***1600***	***0***	***0***	***0***
Starting motor coeff. (Tab. 7)		w =	1,30	1,30	1,30	1,30
Max starting tension	daN	Ta =	22996	0	0	0
Starting safety factor		**fs" =**	**8,3**	**0,0**	**0,0**	**0,0**
TAKE-UP CALCULATION						
Min tension to guarantee right motion transmission						
Distance from drive pulley	m	L_1 =	80	0	0	0
Min tension	daN	Tv_1 =	4149	0	0	0
Min tension to reduce max sag along carrying side						
Required max sag		S_1	0,01	0,01	0,01	0,01
Min tension	daN	T_{sup}	3480	0	0	0
Min tension to reduce max sag along return side						
Required max sag		S_2	0,01	0,01	0,01	0,01
Min tension	daN	T_{inf}	1184	0	0	0
Min take-up value	daN	Fv_{min}	8298	0	0	0
Real applied take-up	daN	Fv	9500	0	0	0
				Belt type		
			1600/5			

2.5 Regenerative belts

For descent conveyors it is possible that the presence of the load allows the movement of the belt without any external power supply; in this case it is necessary to provide for a brake system that controls the belt speed.

To understand if a loaded conveyor is regenerative, it is enough to operate the calculation of the peripheral force F: a negative result means that the weight of the load along the whole conveyor generates a force greater than the total friction.

A negative value for F means also that the tension T_1 at run on point of the drive pulley is lower than the tension T_2 at run off point as the belt is not dragged but, on the contrary, it drags the drive pulley.

At the limit, F near to the zero means that the load can move itself at a costant speed without any need of external motor power. In other words, the load weight generates a force about equal to the total frictions.

For the calculation, we consider a friction coefficient between drive pulley and belt from 0,012 up to 0,016 (about 40% less than for non regenerative belts); so, if we obtain for F a negative value from the first “standard” calculation, it is necessary to repeat the whole calculation of F with friction coefficent as the a.m. ones.

However, when the belt runs in empty conditions, it is important not to forget that, in any case it needs an external motor power that guarantees its movement; as this motor power could be higher than the brake power necessary during loaded running conditions, we must repeat the calculation in this new situation: empty belt without regeneration and friction coefficient as usual (from 0,016 to 0,030).

At the end of these parallel calculations, two different values are obtained for the breaking load CR_m, for the counterweight F_V and for the required motor or brake power; obviously, the most adverse values must be chosen.

Usually, regenerative conveyor belts have the drive pulley at the tail and the counterweight at the head. In this way you obtain the best benefits from the belt in order to have the right tensioning with a counterweight as lighter as possible.

2.6 Analysis of tension along a conveyor belt

In particular cases, when the profile of the conveyor shows several critical points, it is convenient to know the tension distribution along the whole conveyor instead of the only values at the drive pulley. In this way it is possible to check the maximum strength point in the belt, as it could not be necessarily localized at run on point of the drive pulley.
The calculation is quite long but teorically very easy: starting from the counterweight, we proceed along the whole belt in the motion direction adding the total friction in the interested section to the tension in the starting point. It means that if we know the tension in a generic point of the belt, and we add the friction contributions, we obtain the tension in the following point.
Contributions in every single section with length L_i and lift H_i are:
1) Friction due to the movement of the empty belt and of the idler moving parts

$$Cf\left(WMP \cdot L_i + q\frac{B}{1000}\sqrt{L_i^2 - H_i^2}\right) \qquad (2.23)$$

where:
WMP = weight of moving parts [Kg/m] (see eq. 2.1)
L_i = length of the considered section [m]
H_i = elevation of the considered section [m]
B = belt width [mm]

2) Empty belt elevation qBH_i (2.24)

3) Horizontal load movement

$$Cf\frac{Q}{3{,}6v}\sqrt{L_i^2 - H_i^2} \qquad (2.25)$$

where:
v = speed [m/sec]

4) Load elevation $\frac{QH_i}{3{,}6\ v}$ (2.26)

5) At the drive pulley, it is necessary to add the peripheral force F computed on the basis of the total amount of the frictions along the conveyor belt

$$T_{run\ off} = T_{run\ on} - F \qquad (2.27)$$

If there are not mistakes, the last calculation point must have the same value of the starting point, i.e. the take-up point, where it is necessary to find half of the counterweight value, that is the tension used at the beginning of the calculation. In fact, along the belt we can find terms mutually exclusives (i.e. the belt elevation) and others that globally define a value that is equal to the peripheral force F applied to the drive pulley (see points 2.1.1 to 2.1.4).
The counterweight must be chosen according to the equations of paragraph 2.1.7.

Section 3

3.1 Vertical curves design

One of the greatest worry of a conveyor belts builder is the right running of the belt on the carrying idlers, in particular along the concave curves, where particular tensions could produce belt raising from the idlers. However, it is very important not to forget that a variation of inclination determines anomalous tension distributions in the carrying section of the belt, as higher as shorter the radius of curvature is.
This problem only happens because it is necessary to maintain the belt edges inclined also during the curves in order not to reduce the loading section; this situation produces a different path between edges and belt center and consequently different tensions in the section: lower along the shortest path, with the risk of excessive and dangerous slackages which could produce loosing of handled material, and greater along the longest one, producing anomaluos overtensioning.
In order to avoid these many and variuos drawbacks it is necessary to pay attention at the choice of such radius of curvature, in particular relatively to the tension values in these critical points.
With steel cord belts the situation is much more critical because of their particular elongation characteristics, lower than textile belts ones.

3.1.1 Concave curves

As already shown, a wrong choice of the radius of curvature of concave curves (Fig. 10), produces a raising of the belt from the carrying idlers, over all without load and in particular at starting.
To guarantee a right running in every conditions, it is necessary to design the curve in the worst situation, that means the starting phase with the belt loaded only until the beginning of the curve and empty after the same curve.
The main condition that have to be respected for the radius of curvature R is the following:

$$R \geq \frac{T_c}{W_b \cos\sigma}\, w \qquad (3.1)$$

where:
T_c = tension at the beginning of the curve with the a.m. load characteristics [daN]
W_b = belt weight [Kg/m]
w = starting coefficient (Tab. 7)
σ = angle of the curve [°]

Often, the minimum radius of curvature becoming from this calculation is too high to be accettable, generally for space requirements; in this case we advise almost to guarantee the respect of the following condition:

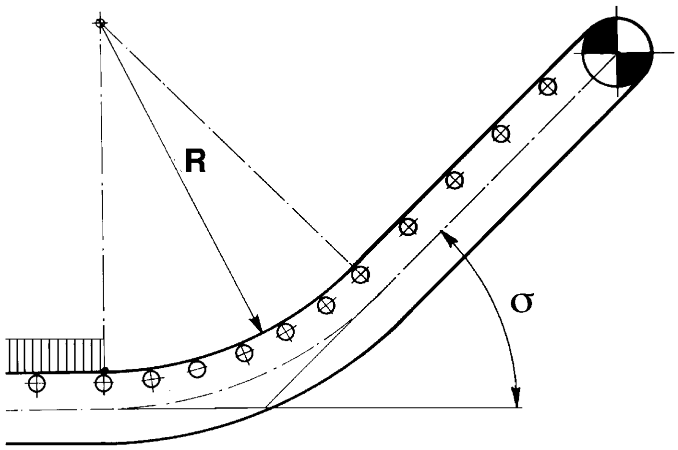


Fig. 10

$$R \geq \frac{T^*}{\left(W_b + \frac{Q}{3{,}6v}\right)\cos\sigma} w \qquad (3.2)$$

where

Q = capacity [Ton/h]

v = speed [m/sec]

Q/(3,6 · v) = weight of the material for linear meter [Kg/m]

where a favourable starting condition is considered i.e. with full loading belt, as the material weight produces a remarkable raising reduction of the belt from the carrying idlers.

Indipendently to this macroscopical problem, it is indispensable to avoid that the center of the belt suffer excessive strength and the edges are slackened or have foldings, that could reduce the loading section with consequent loosing of load. So, the conditions which must be respected are

$$R \geq \frac{\operatorname{sen}\beta \cdot CR \cdot A \cdot B/1000}{4{,}5 \cdot \left(10\frac{T_c}{B} - 4{,}4\right)} \qquad (3.3)$$

to avoid foldings on the edges and

$$R \geq \frac{\operatorname{sen}\beta \cdot CR \cdot A \cdot B/1000}{4{,}5 \cdot \left(\frac{CR}{f_s} - 10\frac{T_c}{B}\right)} \qquad (3.4)$$

to avoid over stress on the center.

where

B = belt width [mm]

β = angle of the carrying idlers [°]

CR = belt tensile strength [N/mm]

A = unitary elastic modulus of the belt (Tab. 16)

f_s = belt safety factor

The radius of curvature that must be chosen, is the highest of the three above calculated values.

3.1.2 Convex curves

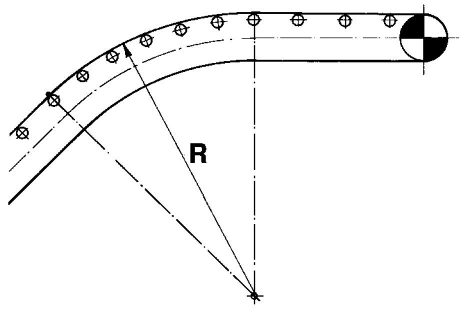


Fig. 11

Convex curves (Fig.11) have an opposite behaviour in comparison with concave ones. In particular, they tend to press down the belt on the carrying idlers; so, the only problems for the designer are the overstress of the belt edges and the slacking of the belt center. In every case, the argumentations are very similar to the case of concave curves:

$$R \geq \frac{\operatorname{sen}\beta \cdot CR \cdot A \cdot B/1000}{9 \cdot \left(\frac{CR}{f_s} - 10\frac{T_c}{B}\right)} \qquad (3.5)$$

to avoid overstress on the belt edges.

It is important not to forget the risk of capacity reduction because of longitudinal foldings of the belt center. So

$$R \geq \frac{\operatorname{sen}\beta \cdot CR \cdot A \cdot B/1000}{9 \cdot \left(10\frac{T_c}{B} - 4{,}4\right)} \qquad (3.6)$$

to avoid foldings on the belt center.

If this last value is higher than the previous, we advise to increase the pretension of the belt.

3.1.3 Calculation of tension T_c at the beginning of the curve

In order to define T_c value at the beginning of each vertical curve, we suggest to operate the tension calculation of an imaginary full loading belt represented by the belt section that comes before the interested curve; the maximum tension that comes from this calculation is the required T_c value (Fig. 12).
As this calculation does not respect the reality but it is only a portion of the calculation executed for the whole conveyor, it is necessary to maintain the same friction and lenght coefficients of the original calculation.

3.1.4 Pipex® belts curves

Pipex®, are conveyor belts for tubular conveyor systems; previous paragraph are not appliable at Pipex® belts because of the particular distributions of the strengths in the transversal section.So, it is usual to consider, for both vertical and horizontal curves the following formula:

$$R \geq 300 \text{ x tube diameter} \qquad (3.7)$$

This equation is valid for horizontal curves only if the angle is less than 45°. On the contrary the minimum radius must be the double.

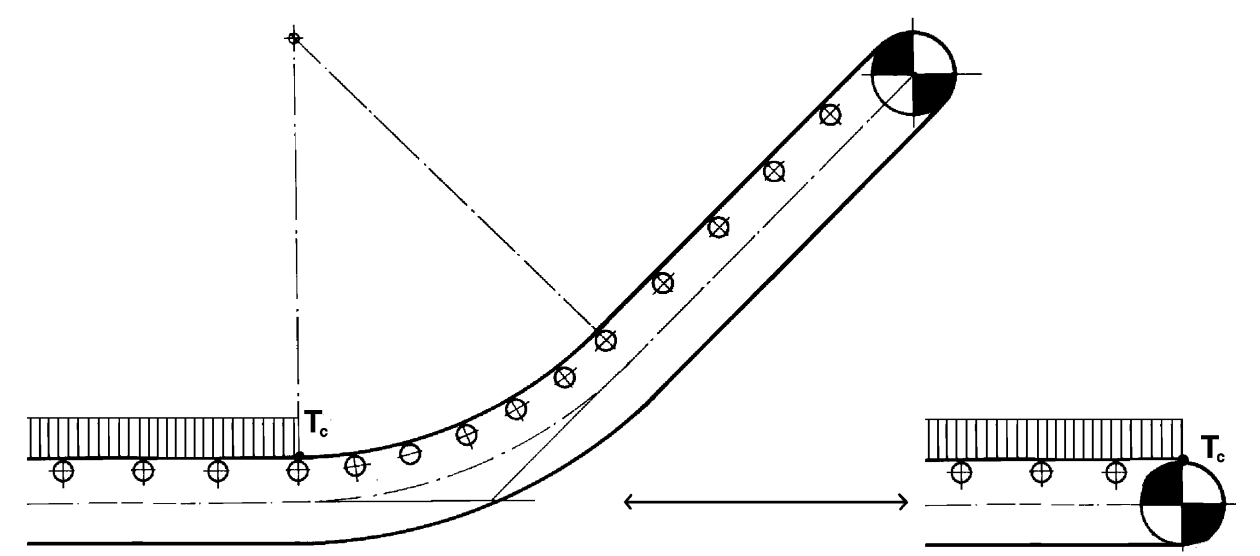


Fig. 12

4.1 Feeder belt design

Feeder belts have generally short center distance and no elevation but must bear strength greater than other conveyor belts even of higher length because of the material weight in the hopper. The forces that determine the tensile strenght CR and the required motor power are principally due only to the following two effects:

1) Translation of the load

$$F_1 = CfL \frac{Q}{3,6\ v} \cos\beta \qquad (4.1)$$

2) Weight of the material in the hopper.

Because of the friction on the hopper, for the most part of the materials it is possible to suppose that the effective height of load taking effect on the belt is about two times the loaded belt width; so, the effective load weight that burdens on the belt is

$$2000 \cdot B_t^2 \cdot L_t \cdot \rho \quad [daN] \qquad (4.2)$$

where:

B_t = hopper opening width [m]

L_t = hopper opening length [m]

ρ = material density [Ton/m³]

from whom it is possible to obtain

$$F_2 = 2000 \cdot \mu_0 B_t^2 \cdot Lt \cdot \rho \quad [daN] \qquad (4.3)$$

where:

μ_0 = hopper friction factor depending on the type of the belt support

$$\mu_0 = \begin{cases} 0,4 & \text{Idlers} \\ 0,6 & \text{Sliding surface} \\ \text{up to } 1 & \text{Heavy conditions} \end{cases}$$

So, with good approximation, the peripheral force F that is transmitted by the drive pulley to the belt is:

$$F = F_1 + F_2 \qquad (4.4)$$

Only for long feeder belts with light charging conditions, it could be necessary to consider the effect of the empty belt movement too (see eq. 2.1).

When F value is available, it is possible to start for the tension calculation as shown for standard belts (Section 2).

Section 5

5.1 Elevator belt design

The tension T [daN] that an elevator belt must bear is due to the following causes:
1) Weight of the belt for linear meter P_1 [Kg/m]
2) Weight of buckets P_2 [Kg/cad]
3) Weight of the load P_3 [Kg/cad]

Further tensions occour because of:
4) Friction in the carter at charging point
5) Minimum starting tension necessary to guarantee the motion transmission without slippings.

Here below the a.m. terms are analized:

1) Tension T_1 due to the belt weight [daN]

$$T_1 = P_1 \cdot H \qquad (5.1)$$

2) Tension T_2 due to the weight of the buckets applied on the belt [daN]

$$T_2 = P_2\ Hn/p \qquad (5.2)$$

3) Tension T_3 due to the weight of the material held in every single bucket [daN]

$$T_3 = P_3\ Hn/p\ (*) \qquad (5.3)$$

4) Tension T_4 due to the friction on the carter [daN]

$$T_4 = DJT_3/H \qquad (5.4)$$

5) Starting tension T_5 necessary to guarantee the right motion transmission without slippings [daN]

$$T_5 = MAX\ (T_a, T_b, T_c) \qquad (5.5)$$

where

$T_a = 0{,}2B$ [daN] with B = belt width [mm]

$$T_b = K\ (T_3 + T_4) - (T_1 + T_2) \quad [daN] \qquad (5.6)$$

$T_c = F_V/2$ represents the take-up and the lower pulley weight effect.

H = elevation [m]
n = number of bucket row
p = bucket pitch [m]
K = Friction factor on drive pulley: 0,84 for bare pulleys, 0,5 for lagged.
J = Friction factor on the carter: generally 8; for big lump size 12; for continuous buckets without friction at charging point 6.

(*) Note for T_3 calculation:
capacity Q and weight of the handled material for each bucket P_3, are connected by the following relation:

$$P_{3calc} = Q\ \frac{p}{3{,}6\ v}\ \cos\beta \qquad (5.7)$$

If there are inconsistency between these two values, in the calculation of T_3 it is necessary to use the greatest one between the data P_3 and the value P_{3calc} coming from the capacity data.

We obtain the maximum tension by summing the here above explained value:

$$T = T_1 + T_2 + T_3 + T_4 + T_5 \qquad (5.8)$$

For the calculation of the minimum tensile strength CR_m it is necessary to take into consideration that the useful belt width B_u [mm] is lower than the geometrical width B because of the hole executed in order to apply the buckets on the belt:

$$B_u = B - d_f n_f \qquad (5.9)$$

where

B_u = useful width [mm]

d_f = hole diameter [mm]

n_f = maximum number of hole in a transversal belt section

If at least one of these datas are unknown, we suggest to use a safety factor $f_s = 12$ instead of 10 in the calculation of CR_m.

So, the working tension CL of the belt is

$$CL = \frac{10T}{B_u} \quad (5.10)$$

Multiplying for the safety factor f_s, we find the minimum tensile strenght CR_m

$$CR_m = \frac{10T}{B_u} f_s \quad (5.11)$$

Chosen a tensile strenght CR greater or equal to the here above calculated value CR_m, it is possible to verify the effective safety factor f_s':

$$f_s' = \frac{CR \cdot B_u}{10T} \quad (5.12)$$

The motor power P_a necessary for an elevator belt, must balance T_3 e T_4 because the tensions T_1, T_2 and T_3 produce auto-compensative effects along the whole lenght of the conveyor

$$P_a = \frac{T_3 + T_4}{102} v \quad (5.13)$$

Introducing the mechanical efficiency η of the transmission (Tab. 8) and a power surplus of 20%, the minimum motor to apply to the conveyor belt must be

$$P_m = 1{,}2P_a / \eta \quad (5.14)$$

Chosen the real motor power N, we suggest to verify that, at starting with full loaded belt, a too great motor power is not transmitted to the belt as it could compromise its structure. For this reason, starting from N, we calculate the tension $(T_3 + T_4)_a$ with the following revers formula:

$$(T_3 + T_4)_a = 102 \frac{N}{v} \quad (5.15)$$

Known this value we calculate again the tension T_{b_a} necessary to transmitt the peripheral force:

$$T_{b_a} = K (T_3 + T_4)_a - (T_1 + T_2) \quad (5.16)$$

So

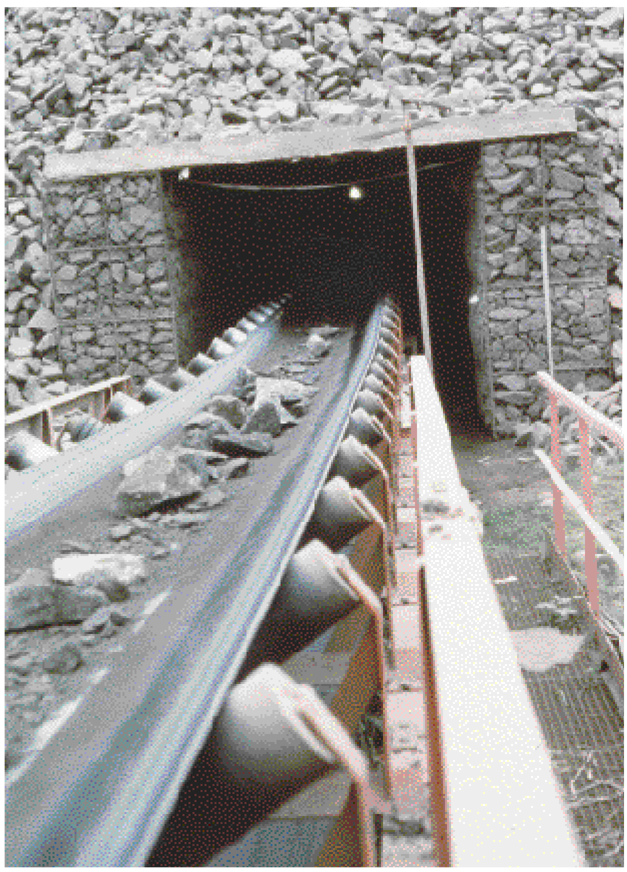
$$T_{5_a} = MAX (T_a, T_{b_a}, T_c) \quad (5.17)$$

Taking into consideration the motor and trans-mission characteristics and using the starting coefficient w (Tab. 7), the maximum starting tension T_a is

$$T_a = T_1 + T_2 + w (T_3 + T_4)_a + T_{5_a} \quad (5.18)$$

From this value, it is possible to calculate the starting safety factor f_s'' that must not be less than 6, in particular for conveyor with frequent startings.

$$f_s'' = CR \frac{B_u}{10T_a} \cdot \quad (5.19)$$



Section 6

6.1 Transition distance

The transition distance is the section between drive or return pulley and first tern of idlers (Fig. 13); it is a particular zone where belt over-tensioning are possible because the belt edges follow a path longer than the belt centre; international standards suggest to maintain the over-tensioning lower than 30%.

The following equation shows the minimum value L_t [m] for the transition distance when the belt tension is the maximum admissible (100% RMBT "Recommended Maximum Belt Tension" $= CL \dfrac{CR}{f_s}$ (6.1) i.e. the ratio between the working tension CL and the maximum belt tension CR/f_s).

$$L_t = 0{,}707 \cdot V \cdot \sqrt{\frac{10A}{0{,}3}} \qquad (6.2)$$

where:

V = difference in level of belt edges when they pass from the first tern of idlers to the pulley [m]

A = unitary belt modulus (Tab. 16)

In case of Pipex® belt, it is not possible to apply the previous formula; so, we suggest to use the following relation:

$$L_t = 25 \text{ x tube diameter} \qquad (6.3)$$

If the belt is not used at 100% of RMBT, it is possible to reduce the transition distance according to the following table:

% RMBT	90%	80%	70%	≤ 60%
L_t multiplier factor	0.93	0.82	0.74	0.71

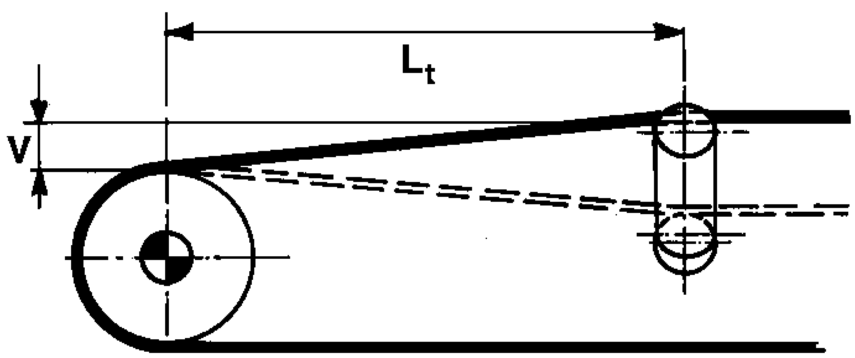


Fig. 13

6.2 Minimum pulley diameter

Pulleys with too small diameter could reduce the belt life because of anomalous overtensioning in the carcass. For this reason, the international standard suggests a simple relation between the carcass thickness (or the steel cable diameter) "e" and the minimum pulley diameter D:

$$D = e \cdot C \qquad (6.4)$$

where C is a coefficient depending on the elastic modulus of the carrying material

$$C = \begin{cases} 90 & \text{polyammid} \\ 108 & \text{polyester} \\ 145 & \text{steel cord} \end{cases}$$

Pulley diameters are standardized according to the following classes:
100, 125, 160, 200, 250, 315, 400, 500, 630, 800, 1000, 1250 mm.

When the belt is not used at 100% of RMBT (Recommended Maximum Belt Tension), it is possible to adopt classes lower than the ones becoming from the previous formulas, according to the following rule:

1. from 30% up to 60% of RMBT:
   1 class less;
2. up to 30% of RMBT:
   2 classes less.

Starting from the diameter obtained with the a.m. considerations, it is possible to choose further lower values in consideration of the position along the conveyor:

1. low tension pulleys (return and counterweight pulleys):
   1 class less;
2. snub pulleys (wrap up to 30°):
   2 class less.

We suggest not to reduce the diameter of tripper pulleys because of high tensions that could be present when it is near to the drive pulley.
For specific values of minimum pulley diameters, see catalogues of our products.



Section 7

7.1 Capacity of conveyor belts

Volumetric capacity of a conveyor belt can be easily worked out starting from its transversal section S with the following formula:

$$Q = 3600 \cdot Svk \qquad (7.1)$$

where
Q = belt capacity [m³/h]
S = belt charging section [m²]
v = belt speed [m/sec]
k = dip factor (see Graph. 1)

Belt charging sections could be calculated with the following formula according to the international standard, for various conditions and for tern of idler with length of each idler equal to 0,35B, where B is the belt width in meters:

$$S = \left[0{,}35B + (0{,}55B - 0{,}05)\cdot \cos\lambda\right]^2 \cdot {}^{tg\theta}\!/_{6} +$$

$$+\left[0{,}35B + \frac{0{,}55B - 0{,}05}{2}\cdot \cos\lambda\right]\cdot \qquad (7.2)$$

$$\cdot\left[\frac{0{,}55B - 0{,}05}{2}\cdot \text{sen}\,\lambda\right]$$

where
λ = idler inclination
θ = surcharge angle = 0,75 x angle of repose.

In case of one or two idler sets, the previous formula is semplified as follow:

$$S = \frac{(0{,}9B - 0{,}05)^2}{12}\cdot \cos\lambda \cdot \qquad (7.3)$$

$$\cdot(2\cos\lambda \cdot tg\theta + 3\,\text{sen}\,\lambda)$$

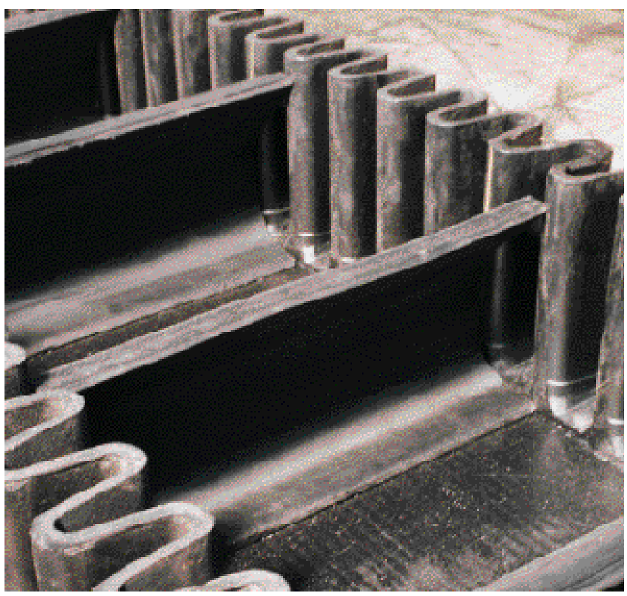
As the charging section changes with the inclination of the conveyor belt, it is necessary to multiply it for the dip factor k according to the maximum belt inclination.
In Table 1 the capacity worked out with the previous formula in case of tern of idler is shown.

7.2 Capacity of Pipex® belts

For Pipex belts, the charging section is approximatively the internal circular area of the tube; so, taking in consideration that usually the charging area is only the 75% of the theorical one, we have:

$$S = \frac{3}{16}\cdot \pi \cdot D_1^2 \qquad (7.4)$$

where D_i = nominal tube diameter.



7.3 Capacity of a Flexobord® belt

Flexobord® are belts with rubber cleats and edges designed in such a way to obtain buckets with a particular volume. Capacity of Flexobord® must be worked out with different geometrical considerations, as here below explained.
The charging section of each cleat is due to 3 different components:

1. Phisical cleat section S_a: only depending on cleat dimensions and shape (Tab. 17).
2. Filling section S_b: also depending on maximum plant inclination.
3. Friction section S_c: depending on material angle of repose and cleats height.

Now, we analyze points 2 and 3 (Fig. 14) using simple trigonometry knowledge.
The “filling height” is

$$h_1 = l \cdot tg\,(90 - \alpha)\ [mm] \qquad (7.5)$$

where
α = max belt inclination [°]
l = useful height of the cleat [mm] (Tab. 17).

It follows

$$S_b = \frac{l \cdot h_1}{2 \cdot 10^4}\ [dm^2] \qquad (7.6)$$

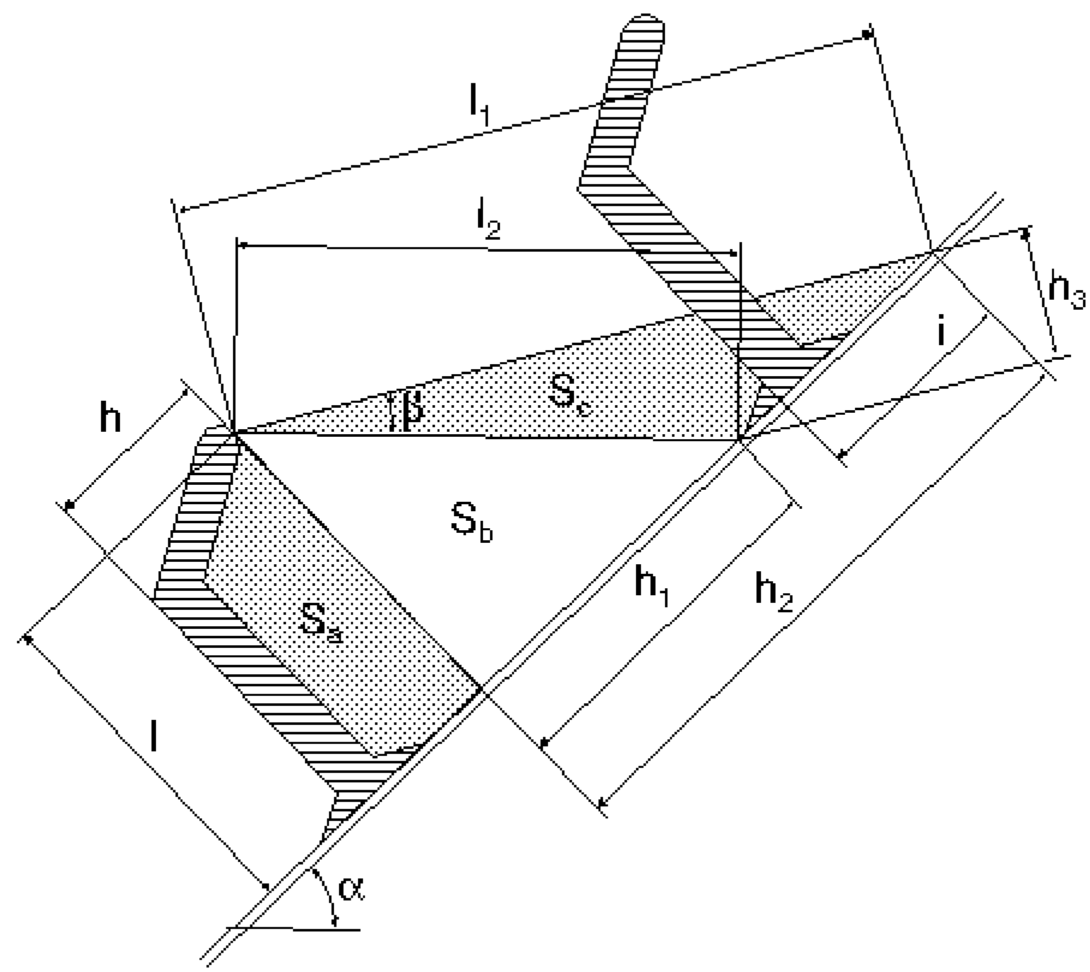


Fig. 14

Analogously, the "friction height" h_2 is

$$h_2 = l \cdot tg\,(90 - \alpha + \beta) \qquad (7.7)$$

where

β = surcharge angle [°] depending on the cohesion characteristics of the conveyed material (Tab. 2).

Moreover (see Fig. 14)

$$l_2 = \frac{l}{\cos(90 - \beta)}$$

$$l_1 = \frac{l}{\cos(90 - \alpha + \beta)} \qquad (7.8)$$

from whom $h_3 = l_2 \cdot sen\beta$ and

$$S_c = \frac{h_3 \cdot l_1}{2 \cdot 10^4}\ [dm^2] \qquad (7.9)$$

So, the section of every single cleat is

$$S = S_a + S_b + S_c\ . \qquad (7.10)$$

In order to obtain a required belt capacity it is necessary to choose a right pitch "p" between cleats, taking in consideration that for small values an interference between cleat sections could happen; in this case a reduction of the global loading section would occur.

Using a simple proportion the useful section S_u for every single cleat is

$$S_u = S \cdot \left(1 - (i/h_2)^2\right) \qquad (7.11)$$

where

$i = h + h_2 - P$ interference [mm] between cleats, only if positive otherwise null (Fig. 14)

p = cleat pitch [mm]

h = cleat thickness [mm] (Tab. 17)

Known the useful section S_u, we find the useful volume V for each bucket multiplying for the useful belt width B_u

$$B_u = B - 2 \cdot (b_1 + b_2)\ \ [mm] \qquad (7.12)$$

where

b_1 = edge width [mm] (Tab. 18)

b_2 = free lateral space width (Tab. 19) [mm].

So

$$V = S_u \cdot B_u / 100\ \ [dm^3] \qquad (7.13)$$

The theorical belt capacity is

$$Q_0 = \frac{V}{p}\ v \cdot 3600\ [m^3/h] \qquad (7.14)$$

This value must be multiplied for the filling factor (usually 75%). It follows

$Q_{eff} = 0{,}75Q_0$.

Known the dimensions of Flexobord® belt according to the capacity requirements, the design of the belt carrying characteristics follows the standard calculation (Section 2). We suggest to increase up to 20% the force value necessary to move the empty belt in order to consider the particular frictions which occur in the Flexobord® belts.

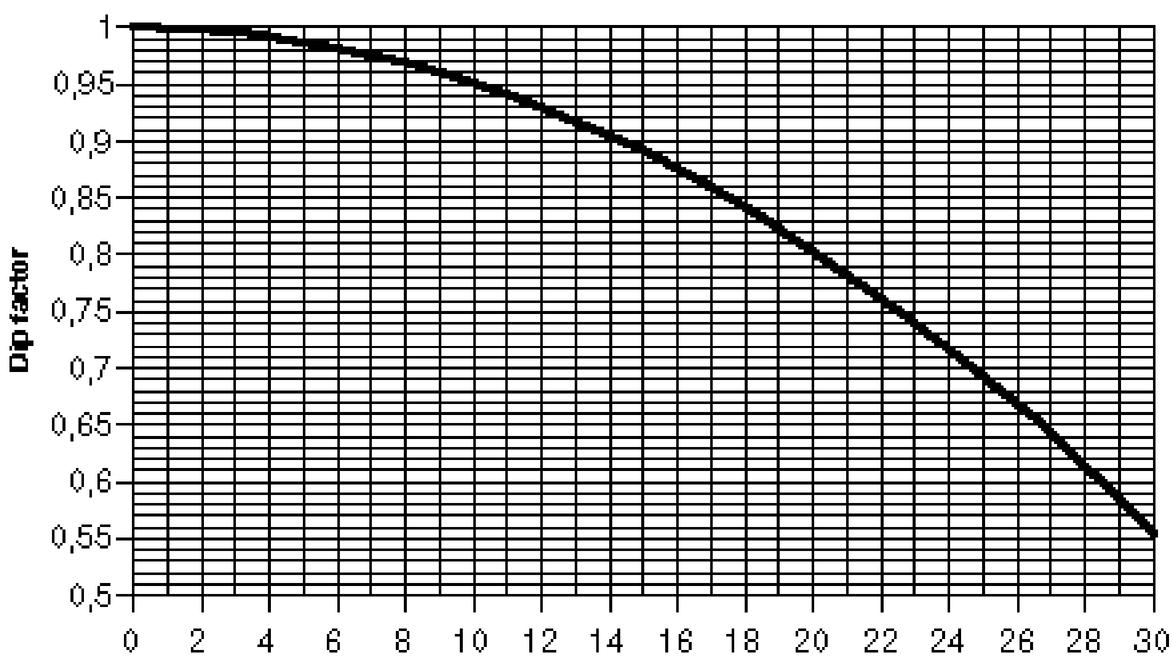
Take also note that for, the right running of a Flexobord®, it is necessary to use belts with particular characteristics of transversal stiffness (Texrigid® or Crossrigid® belts).

Capacity [m³/h] at speed of 1 m/sec for horizontal belts (Tab.1)

WIDTH mm	20°	25°	30°	35°	45°	20°	25°	30°	35°	45°
	Angle of surcharge 30°					***Angle of surcharge 25°***				
400	54	58	61	64	67	48	52	56	59	62
500	91	97	103	107	112	81	88	93	98	104
650	163	175	184	192	200	145	158	168	177	187
800	256	275	290	302	314	228	248	264	278	294
1000	413	443	467	486	507	368	400	426	448	474
1200	607	651	687	715	745	541	588	627	659	696
1400	839	899	949	987	1028	748	812	866	910	962
1600	1107	1188	1253	1304	1357	988	1073	1144	1201	1270
1800	1414	1516	1600	1664	1732	1261	1370	1461	1533	1620
2000	1757	1884	1988	2068	2152	1567	1702	1816	1906	2014
2200	2138	2292	2419	2516	2618	1907	2071	2209	2319	2450
	Angle of surcharge 20°					***Angle of surcharge 15°***				
400	43	47	51	54	58	37	42	46	49	54
500	71	79	85	90	97	63	70	77	83	91
650	129	142	153	162	175	113	127	139	149	163
800	202	223	241	256	275	178	200	219	235	257
1000	327	360	389	412	443	288	323	354	379	415
1200	481	530	572	607	652	424	475	520	558	610
1400	664	732	790	838	900	585	657	719	771	843
1600	878	967	1044	1107	1189	774	868	950	1018	1113
1800	1120	1235	1333	1413	1517	988	1108	1213	1300	1421
2000	1393	1535	1657	1757	1886	1229	1378	1508	1616	1766
2200	1695	1868	2016	2137	2295	1495	1677	1835	1967	2149

IDLER INCLINATION applies to the column headers above.

For inclined belts, multiply this data of capacity for the dip factor (Graph. 1)



Graph. 1

Material characteristics (Tab. 2)

Material	Density [Ton/m³]	Angle of repose	Max conveying angle	Angle surcharge
Anthracite (coal) fines	1,0	35	18	25
Anthracite (coal) sized	0,9	27	16	10
Ashes of kiln	0,9	35	20	25
Bagasse	0,1/0,2	45	25	30
Bauxite dry fine	1,0/1,4	35	18	25
Beet pulp dry	0,2/0,3	31	20	20
Beet pulp wet	0,4/0,7	31	20	20
Beets whole	0,75	35/40	20	25
Cement dry bulk	1,5	39	20	30
Clay 0-75 mm	1,0/1,2	35	18/20	25
Clinker	1,2/1,5	30/40	18/20	20/25
Coffee fresh bean	0,5	25	10/15	10
Coke	0,4	30/45	18	20/30
Concrete	2,25	–	20/22	–
Copper	1,6/2,4	30/44	18/20	20/30
Corn	0,7/0,8	20/25	12	10
Galena	3,2/4,3	30	15	20
Glass	1,3/1,6	30/45	20/22	25
Granite 0-10 mm	1,3/1,4	40	20	30
Granite 10-150 mm	1,4/1,5	35	18	25
Gravel dray (washed)	1,4/1,6	35	16	25
Gravel sized pebbles	1,4/1,6	30	12	20
Gravel wet	1,5/1,7	32	20	20
Gypsum	1,0/1,4	30/40	15/20	30
Heard clyed wet	1,7/1,8	45	22	30
Heart clyed dry	1,2/1,3	35	20	25
Heart wet	1,25/1,5	35	20/23	25
Iron ore	1,6/3,2	35	18/20	25
Iron ore pellets	2,5/2,9	20	12	10
Lignite (coal borwn)	0,7/0,9	38	18	25
Lime fine crushed pebble	0,9	30	17	20
Limestone crushed	1,4/1,5	38	18	25
Manganese ore	2,0/2,2	39	20	25
Marble crushed 0-10 mm	1,2/1,5	30/44	15	20/30
Nickel cobalt sulphate ore	1,3/2,4	30/40	20	20/30
Oats	0,4	21	10	10
Phospate acid fertilizer	0,9	26	13	10
Pyrites crusced	2,5/3,4	35	16	25
Pyrites pellets	2,0/2,3	42	18	30
Quarze sized	1,3/1,5	35	18	25
Rice polished	0,6/0,8	20	8	10
Rock crushed	1,5/1,7	35	16/18	25
Salt	0,7/0,9	–	18/22	–
Sand dry	1,5/1,7	35	16/18	25
Sand of foundry	1,3/1,4	32	20	20
Sand wet	1,8/2,0	45	20	30
Sandstone crushed	1,3/1,5	40	18	30
Sawdust	0,2	36	22	25
Shale	1,3/1,6	37/40	18/22	25
Slate dust	1,1/1,3	35	20	25
Soybean whole	0,7/0,8	21/28	12/16	10
Starch	0,5/0,7	24	12	10
Sugar	0,8/1,1	37/45	15/22	20/30
Wood chips dry	0,2/0,4	45	23/25	30
Zinc ore	1,8/2,6	38	22/25	25

If the conveyor dip is higher or near to the max conveying angle for the particular material, let's value the possibility to adopt a SPINATEX® belt, with rubber chevrons.

Conveyor belt minimum width (Tab. 3)

Min width mm	Sized material mm	Unsized material mm
400	64	100
450	75	125
500	100	150
600	110	200
650	125	225
750	145	275
800	160	300
900	180	350
1000	200	400
1050	215	425
1200	250	500
1400	275	600
1600	325	700
1800	350	800
2000	400	900
2200	440	1000

The selection of the belt width depends on the required conveying capacity but it is important not to forget the relation between belt width and material lump size as using too narrow belts it is possible to have serious problems of material instability. See Tab. 2 for dimension of conveyed material.

Maximum conveyor belt speed

Generally, a speed increase produces a reduction of the belt tension; however, because of stability problems, it is not possible to choose speed higher than a maximum value depending in particular on the belt width and on the material lump size characteristics.
Using following tables (Tab. 4, 5, 6), we obtain a speed factor (A+B) that allows to choose the maximum speed according to the belt width. For higher safety, we suggest to adopt a speed from 0,5 up to 1,5 m/sec lower than the limit shown.

Lump size factor A (Tab. 4)

Fine grain to dust	Less than 10 mm	0
Granular	Less than 25 mm	1
Sized and unsized	Less than 20% of max permissible	2
Sized only	Less than 60% of max permissible	3
Sized and unsized	Max permissible with reference to the belt width	4

Abrasiveness factor B (Tab. 5)

Cereal, wood chips and pulp, flue dust, lime, sand, loam	Non abrasive	1
Gravel, slate, coal, salt, sandstone	Midly abrasive	2
Limestone, pellets, spar, concret	Abrasive	3
Ores, glass, granite, pyrite, coke, rock, sinter	Very abrasive	4

Max suggested speed [m/sec] (Tab. 6)

A + B	Width mm				
	up to 500	600 to 650	750 to 800	900 to 1050	1200 to 2000
1	2,5	3,0	3,5	4,0	4,5
2	2,3	2,8	3,2	3,7	4,1
3 - 4	2,0	2,4	2,8	3,2	3,6
5 - 6	1,7	2,0	2,4	2,7	3,0
7 - 8	1,5	1,8	2,1	2,4	2,6

Starting motor coefficient (Tab. 7)

Type of motor	w
Squirrel cage direct on line	2,20
Squirrel cage with starting clutch	1,60
Squirrel cage with fluid coupling	1,30
Slip ring induction motor with additional resistance	1,25
DC motor with electronic controller	1,25

Efficiency of transmission (Tab. 8)

Type of power train	η
Chain with open carter	0,93
Chain with close carter, oil lubrication	0,95
Worked gear	0,90
Melted gear	0,85
Single reduction	0,98
Double reduction	0,98
Triple reduction	0,94
Worm reduction unit with ratio ≤ 1:20	0,90
Worm reduction unit with ratio ≤ 1:60	0,70
Worm reduction unit with ratio ≤ 1:60	0,50
Suggested value without specific indications	0,90

Friction coefficient between belt and idlers (Tab. 9)

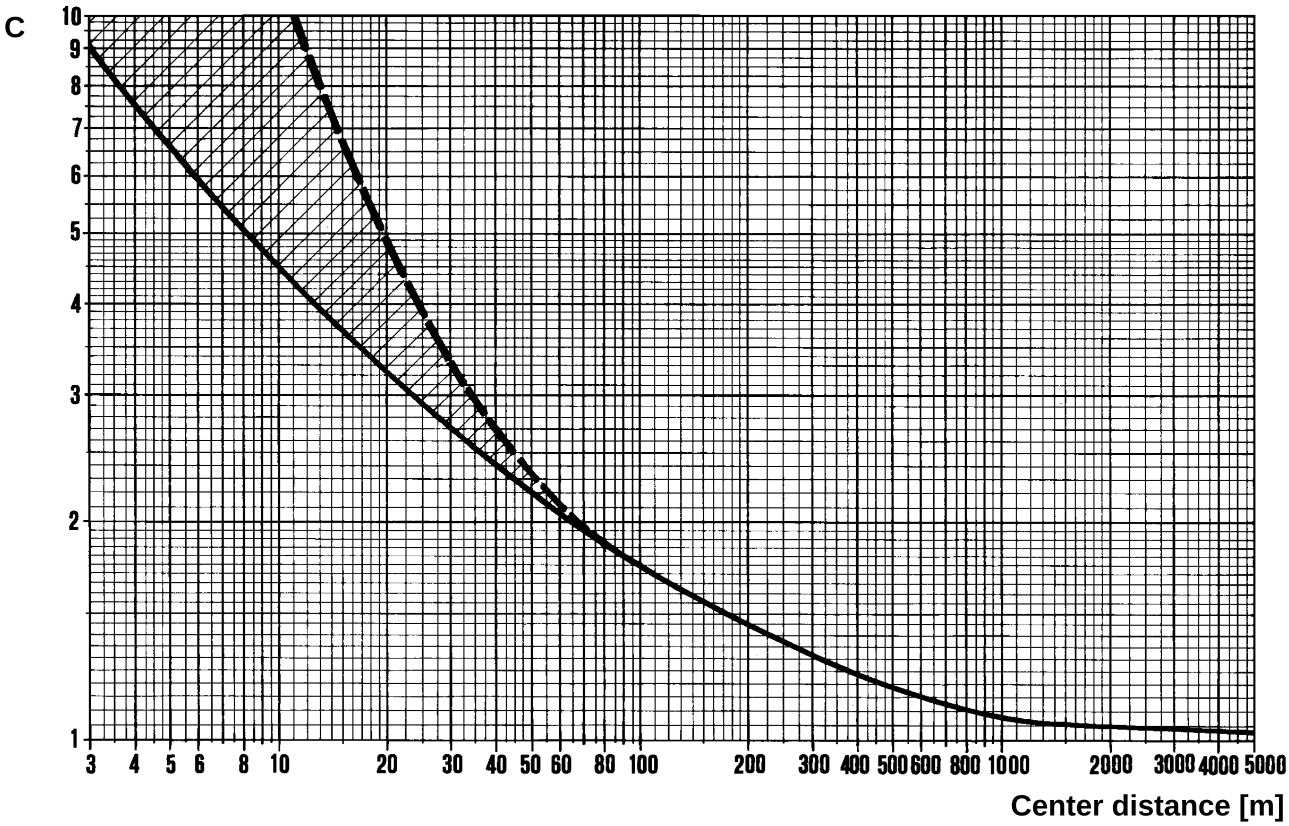
Working conditions	f
Good belt allignement, carrying idlers very sliding, low friction material, speed up to 5 m/sec.	0,017
Standard	0,020-0,022
Dusty atmosphere, low temperature, high friction material, overloading, speed over 5 m/sec.	0,025
Very low temperature but well running installations	0,035
Regenerative conveyor with standard running conditions	0,012
Regenerative conveyors with heavy running conditions	0,016
Sliding belt without bottom cover	0,3
Sliding belt with bottom cover	0,5
Pipex® belts for tubular conveyor systems	0,030-0,037

Friction coefficent between drive pulley and belt (Tab. 10)

Running conditions	Pulley surface			
	sleek steel	striped polyurethane	striped rubber	striped of sponge ceramics
Dry	0,35÷0,40	0,35÷0,40	0,40÷0,45	0,40÷0,45
Wet but clean	0,10	0,35	0,35	0,35÷0,40
Wet and dirty	0,05÷0,10	0,20	0,25÷0,30	0,35

Without clear indications we suggest to use the following values:
Steel surface: 0,25
Rubber surface: 0,35

Length coefficient C (Graph. 2)



For center distance less than 80 meters, the graphic doesn't consider only one value for C coefficient because it is not possible to neglect the uncertainty due to the effect of the localized resistances (discharge points, pulleys, skirts, ...).

Equivalent idler weight [Kg] (Tab. 11)

Idler diam. [mm]	Idler Disposition	Belt width [mm]											
		300	400	500	650	800	1000	1200	1400	1600	1800	2000	2200
63	Plain	2,2	2,7	3,3	4,0	4,8	–	–	–	–	–	–	–
	Couple	3,5	3,7	4,1	4,8	5,7	–	–	–	–	–	–	–
	Tern	–	4,4	4,7	5,5	6,5	–	–	–	–	–	–	–
89	Plain	–	–	5,5	6,2	6,9	7,8	–	–	–	–	–	–
	Couple	–	–	8,4	9,2	9,9	10,8	–	–	–	–	–	–
	Tern	–	–	11,1	11,8	12,5	13,4	–	–	–	–	–	–
108	Plain	–	–	7,2	8,2	9,2	10,4	16,7	–	–	–	–	–
	Couple	–	–	11,4	12,4	13,4	14,5	19,5	–	–	–	–	–
	Tern	–	–	15,2	16,2	17,2	18,5	21,6	–	–	–	–	–
133	Plain	–	–	–	10,0	11,3	18,8	23,3	26,2	27,8	–	–	–
	Couple	–	–	–	15,3	16,7	22,3	26,9	31,0	34,5	–	–	–
	Tern	–	–	–	20,0	21,3	25,0	30,3	34,6	39,8	–	–	–
159	Plain	–	–	–	–	–	–	30,2	33,4	37,4	41,2	44,7	–
	Couple	–	–	–	–	–	–	35,5	35,2	43,2	46,7	50,7	–
	Tern	–	–	–	–	–	–	39,9	44,3	47,7	51,2	55,8	–
191	Plain	–	–	–	–	–	–	–	–	–	58,0	63,0	68,5
	Couple	–	–	–	–	–	–	–	–	–	63,2	69,5	75,0
	Tern	–	–	–	–	–	–	–	–	–	75,5	80,5	86,5

Equivalent idler weight for sixtines of Pipex® belts [Kg] (Tab. 11a)

Idler diam. [mm]	Pipex® diameter [mm]								
	100	150	200	250	300	350	400	450	500
63	3,7	4,4	5,2	–	–	–	–	–	–
89	–	–	–	9,9	11,2	–	–	–	–
108	–	–	–	–	–	16,6	18,7	23,3	27,4

See Tab. 15 for further Pipex® informations.

Indicative idlers pitch [m] (Tab. 12)

B [mm]	Carrying idlers			Return idlers
	L	M	P	
300	1,4	1,2	1,1	3,0
400	1,4	1,2	1,1	3,0
500	1,4	1,2	1,1	3,0
650	1,3	1,1	1,0	3,0
800	1,2	1,0	0,9	3,0
1000	1,1	1,0	0,9	3,0
1200	1,1	1,0	0,9	3,0
1400	1,0	0,9	0,8	3,0
1600	1,0	0,9	0,8	3,0
1800	0,9	0,8	0,7	2,5
2000	0,9	0,8	0,7	2,5
2200	0,9	0,8	0,7	2,5

L = Light materials (careals)
M = Medium weight and lump size material (coal)
P = Heavy materials (ore)

Indicative idler pitch for Pipex® belts (Tab. 12a)

$\varnothing_{tube}$ [mm]	B [mm]	Pitch
100	450	1,0
150	600	1,1
200	800	1,3
250	1000	1,5
300	1200	1,6
350	1400	2,0
400	1600	2,4
500	1900	2,6

Weight of Texter® and Texnyl® textile carcasses (Tab. 13)

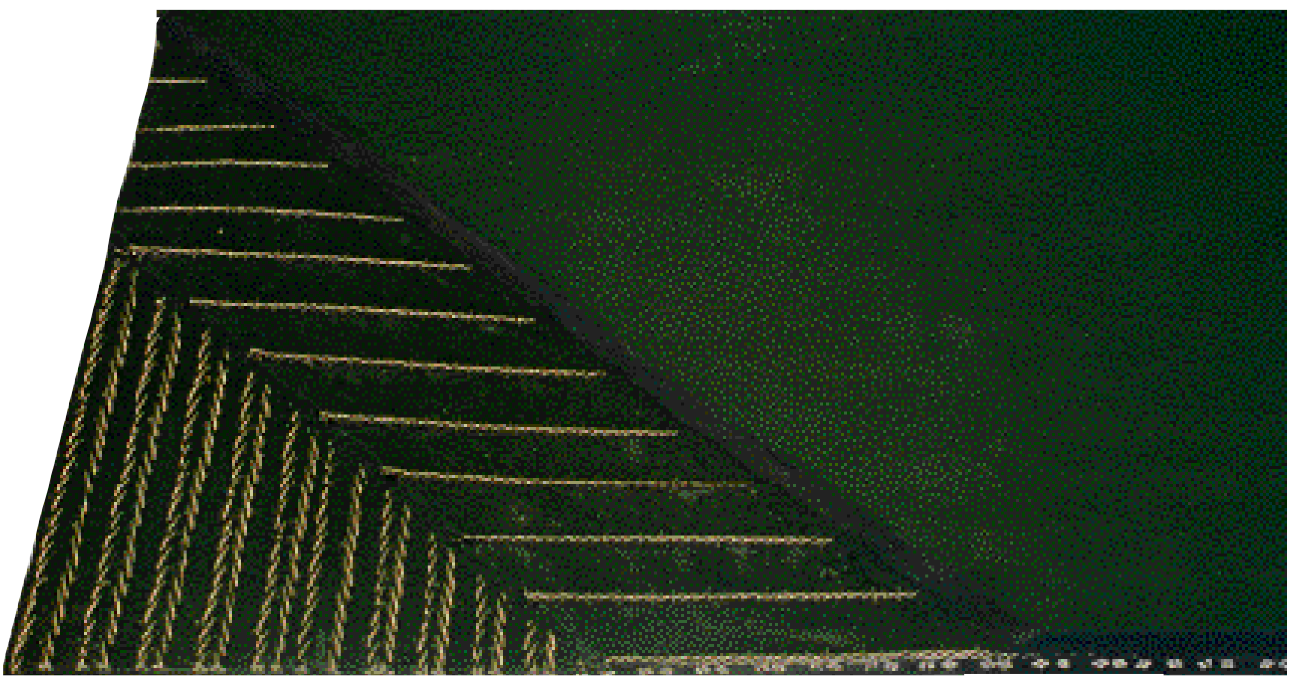
Carcass style	Carcass weight [Kg/m²]	Carcass style	Carcass weight [Kg/m²]	Carcass style	Carcass weight [Kg/m²]	Carcass style	Carcass weight [Kg/m²]	Carcass style	Carcass weight [Kg/m²]
250/2	2,2	630/3	4,9	1000/3	7,1	1250/5	10,0	2000/6	13,9
315/2	2,7	630/4	5,4	1000/4	8,0	1600/4	11,9	2500/4	17,0
400/3	3,3	800/3	6,0	1000/5	8,1	1600/5	11,8	2500/5	18,7
500/3	4,1	800/4	6,5	1250/3	9,0	2000/4	15,0	3150/5	22,3
500/4	4,4	800/5	6,7	1250/4	9,5	2000/5	15,0	3150/6	22,0

To obtain belt weight for particular cover thickness, we consider that the average weight of rubber is 1,2 Kg (m²*mm).
For example, 3 mm of cover for a belt 800 mm wide ha a weight of:
1,2 Kg (m²*mm) x 0,8 m x 3 mm = 2,88 Kg.

Weight of Siderflex® steel cord belts (Tab. 13a)

Belt style	Belt weight [Kg/m²]	Belt style	Belt weight [Kg/m²]	Belt style	Belt weight [Kg/m²]	Belt style	Belt weight [Kg/m²]
–	–	IW 350 6+3	15,9	ST 350 6+3	15,6	HE 350 6+3	17,8
–	–	IW 500 6+3	16,4	ST 500 6+3	16,3	HE 500 6+3	18,4
ID 630 6+4	20,6	IW 630 6+3	16,9	ST 630 6+3	16,6	HE 630 6+3	18,8
ID 800 6+4	22,1	IW 800 6+4	20,6	ST 800 8+4	22,1	HE 800 6+4	21,6
ID 1000 6+4	22,8	IW 1000 6+4	21,3	ST 1000 8+4	22,8	HE 1000 6+4	22,3
ID 1250 8+4	27,2	IW 1250 8+4	26,5	ST 1250 8+4	24,5	HE 1250 8+4	27,8
ID 1400 8+4	27,4	IW 1400 8+4	27,1	ST 1400 8+4	25,1	HE 1400 8+4	28,2
ID 1600 8+4	29,3	IW 1600 8+4	27,8	ST 1600 8+4	25,8	HE 1600 8+4	29,0
ID 1800 8+4	30,2	–	–	–	–	HE 1800 8+4	29,9
ID 2000 8+4	30,8	–	–	–	–	HE 2000 8+4	30,5
ID 2500 8+4	32,2	–	–	–	–	–	–
ID 3150 8+4	36,8	–	–	–	–	–	–

See Tab. 21 for the choice of the suitable type of Siderflex® according to the characteristics of the conveyed material.



Top and bottom cover thickness (Tab. 14)

Lump size mm	Cover quality	Belt cycles 2C/S min.	Materials: moderately abrasive (grain) min.	moderately abrasive (grain) max.	Abrasive (coal) min.	Abrasive (coal) max.	highly abrasive (limestone) min.	highly abrasive (limestone) max.	extremely abrasive (ore) min.	extremely abrasive (ore) max.
< 25	CL	< 0,5	3,0	5,0	3,0	7,0	4,0	10,0	6,0	12,0
		0,5 - 1,0	1,5	4,0	2,0	6,0	3,0	8,0	5,0	10,0
		> 1,0	1,5	3,0	1,5	4,0	3,0	6,0	4,0	8,0
	EC	< 0,5	2,0	4,0	2,0	6,0	3,0	7,0	4,0	10,0
		0,5 - 1,0	1,5	3,0	1,5	4,0	3,0	6,0	3,0	8,0
		> 1,0	1,5	2,0	1,5	3,0	3,0	5,0	3,0	6,0
25 to 125	CL	< 0,5	3,0	6,0	3,0	10,0	5,0	10,0	7,0	14,0
		0,5 - 1,0	3,0	5,0	3,0	6,0	4,0	10,0	6,0	12,0
		> 1,0	1,5	4,0	3,0	5,0	3,0	8,0	5,0	10,0
	EC	< 0,5	3,0	5,0	3,0	8,0	4,0	8,0	5,0	12,0
		0,5 - 1,0	3,0	4,0	3,0	6,0	3,0	7,0	4,0	10,0
		> 1,0	1,5	3,0	3,0	5,0	3,0	6,0	3,0	8,0
> 125	CL	< 0,5	5,0	8,0	6,0	12,0	8,0	14,0	10,0	16,0
		0,5 - 1,0	3,0	7,0	5,0	10,0	6,0	12,0	8,0	14,0
		> 1,0	3,0	6,0	5,0	8,0	6,0	10,0	6,0	12,0
	EC	< 0,5	4,0	7,0	5,0	10,0	6,0	12,0	7,0	14,0
		0,5 - 1,0	3,0	6,0	4,0	8,0	5,0	10,0	6,0	12,0
		> 1,0	3,0	5,0	3,0	6,0	5,0	8,0	5,0	10,0

Note: We recommend to use EC rubber for sharp and abrasive materials. Experience suggests to choice bottom cover thickness equal to half of the top cover thickness.
C = Center distance [m]
V = Speed [m/min]

Pipex® belt characteristics (Tab. 15)

Nominal tube diameter mm	Indicative belt width m	Min. transition distance m	Min. curve m	Capacity at 1m/sec	Max lump size	Max speed	Max inclination	Max curves angle
100	450	2,5	30	21,1	30	2,2	30°	45°
150	600	4,0	45	46,9	45	2,2		
200	800	5,0	60	84,7	60	2,5		
250	1000	6,5	75	132,4	75	2,5		
300	1200	7,5	90	190,7	95	3,0		
350	1400	9,1	105	256,2	110	3,5		
400	1600	10,0	120	339,1	125	4,0		
500	1900	12,4	150	529,7	150	4,5		

Belt modulus (Tab. 16)

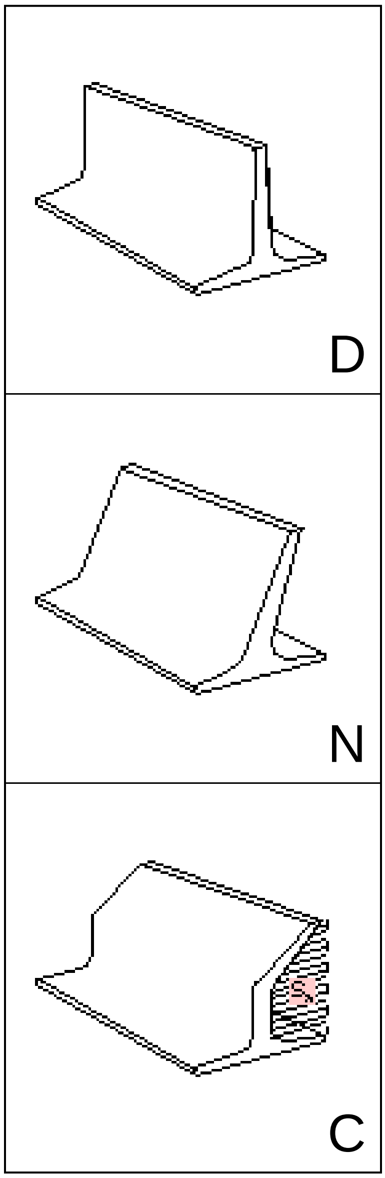
Ply breaking load	Unitary modulus A: EP	Unitary modulus A: NN or PP
100/125	10,5	6,6
160	11,0	6,8
200	11,5	7,5
250/315	12,0	8,0
400/630	12,5	8,5
SIDERFLEX® ID	25	
SIDERFLEX® IW, HE, ST	50	

To obtain the belt modulus (tension that must be applied to the belt in order to obtain 100% of elongation), multiply the unitary modulus A for related tensile strength. For example:
EP 630/3   11,5 x 200 = 2300 KN/m
ST1250   50 x 1250 = 62500 KN/m

Indicative Flexobord® cleat characteristics (Tab. 17)

Cleat type and height	Useful height l [mm]	Thichness h [mm]	Cleat section S_a [dm²]	Weight [Kg/m]
D110	100	–	–	2,5
N55	50	30	0,03	1,1
N75	70	40	0,05	1,8
N110	105	65	0,18	2,7
C75	70	40	0,12	2,2
C110	105	65	0,32	3,2
C145	135	85	0,43	4,4
C180	175	90	0,92	6,9
C230	220	100	1,29	10,1
C280	270	100	1,65	23,0
C330	320	100	2,95	24,5

For symbols see fig. 14 Section 7



Indicative Flexobord® edges characteristics (Tab. 18)

Edge heigh [mm]	Width [mm]	Weight [Kg/m]	Wave pitch [mm]
60	50	1,3	40
80	50	1,5	50
120	50	2,0	50
160	75	4,6	60
200	75	5,7	60
250	75	8,0	60
300	75	9,5	60
350	110	17,0	80

Free lateral space for Flexobord® belts [mm] (Tab. 19)

Belt width [mm]	400	500	650	800	1000	1200	1400	1600
Free lateral space width [mm]	60	60	75	100	125	150	175	175

Minimum pulley diameter for Flexobord® belts (Tab. 20)

Edge heigh [mm]	Drive pulley [mm]	Return pulley [mm]	Deflection wheel [mm]
80	200	200	350
120	315	315	500
160	400	400	650
200	500	500	800
240	630	630	1000
300	800	800	1200

Values coming from this table must be compared with diameters worked out in case of standard belts.
Maximum value has to be chosen.

Characteristics of Siderflex® steel cord belts (Tab. 21)

Tensile strength	KN/m	350	500	630	800	1000	1250	1400	1600	1800	2000	2500	3150
Siderflex® ID		Regular warp steel cords and high elongation weft steel cords											
Carcass thickness	mm	–	–	5,0	5,6	5,6	6,4	6,4	7,2	7,2	7,2	8,9	9,6
Mass	Kg/m²	–	–	3,45	4,35	5,20	6,40	6,90	7,90	9,10	9,80	11,65	13,76
Warp cord pitch	mm	–	–	14,00	15,00	12,00	14,00	13,00	15,00	13,00	12,00	14,00	15,00
Warp cord diameter	mm	–	–	3,00	3,60	3,60	4,40	4,40	5,20	5,20	5,20	6,90	7,60
Warp cord tensile strength	N	–	–	9700	13500	13500	19800	19800	26700	26700	26700	41200	51300
Weft cord pitch	mm	–	–	14,0	14,0	14,0	14,0	14,0	14,0	14,0	14,0	14,0	14,0
Weft cord diameter	mm	–	–	2,00	2,00	2,00	2,00	2,00	2,00	2,00	2,00	2,00	2,00
Weft cord tensile strength	N	–	–	2900	2900	2900	2900	2900	2900	2900	2900	2900	2900
Siderflex® IW		Elongation warp steel cords and high elongation weft steel cords											
Carcass thickness	mm	3,2	3,2	3,2	4,5	4,5	6,0	6,0	6,0	–	–	–	–
Mass	Kg/m²	1,85	2,45	2,95	4,15	5,0	6,35	7,05	7,90	–	–	–	–
Warp cord pitch	mm	8,33	5,81	4,63	6,67	5,38	7,04	6,25	5,50	–	–	–	–
Warp cord diameter	mm	2,00	2,00	2,00	2,85	2,85	3,70	3,70	3,70	–	–	–	–
Warp cord tensile strength	N	3075	3075	3075	5600	5600	9600	9600	9600	–	–	–	–
Weft cord pitch	mm	17,5	17,5	17,5	20,0	20,0	20,0	20,0	20,0	–	–	–	–
Weft cord diameter	mm	1,52	1,52	1,52	2,02	2,02	2,40	2,40	2,40	–	–	–	–
Weft cord tensile strength	N	1720	1720	1720	2900	2900	3775	3775	3775	–	–	–	–
Siderflex® HE		Elongation warp steel cords and double high elongation weft steel cords											
Carcass thickness	mm	2,4	4,7	4,7	5,4	5,4	7,1	7,1	7,1	7,1	7,1	–	–
Mass	Kg/m²	2,00	2,60	3,15	4,10	4,95	6,30	7,00	7,85	8,70	9,25	–	–
Warp cord pitch	mm	8,33	5,81	4,63	6,67	5,38	7,04	6,25	5,50	5,00	4,65	–	–
Warp cord diameter	mm	2,00	2,00	2,00	2,85	2,85	3,70	3,70	3,70	3,70	3,70	–	–
Warp cord tensile strength	N	3075	3075	3075	5600	5600	9600	9600	9600	9600	9600	–	–
Weft cord pitch	mm	12,5	12,5	12,5	12,5	12,5	15,0	15,0	15,0	12,5	12,5	–	–
Weft cord diameter	mm	1,52	1,52	1,52	1,52	1,52	2,02	2,02	2,02	2,02	2,02	–	–
Weft cord tensile strength	N	1720	1720	1720	1720	1720	2900	2900	2900	2900	2900	–	–
Siderflex® ST		Elongation warp steel cords an nylon weft cords											
Carcass thickness	mm	3,2	3,2	3,2	4,1	4,1	4,9	4,9	4,9	–	–	–	–
Mass	Kg/m²	1,50	2,15	2,65	3,60	4,45	5,60	6,30	7,15	–	–	–	–
Warp cord pitch	mm	8,33	5,81	4,63	6,67	5,38	7,04	6,25	5,50	–	–	–	–
Warp cord diameter	mm	2,00	2,00	2,00	2,85	2,85	3,70	3,70	3,70	–	–	–	–
Warp cord tensile strength	N	3075	3075	3075	5600	5600	9600	9600	9600	–	–	–	–
Weft cord pitch	mm	15,0	15,0	15,0	15,0	15,0	15,0	15,0	15,0	–	–	–	–
Weft cord diameter	mm	1,2	1,2	1,2	1,2	1,2	1,20	1,20	1,20	–	–	–	–
Weft cord tensile strength	N	800	800	800	800	800	800	800	800	–	–	–	–

Appendix A

Trade mark of S.I.G. S.p.A.

Texter®
conveyor belts with textile carcass made with polyester in the warp and polyamide (nylon) in the weft. Suitable for standard use.

Texnyl®
conveyor belts with textile carcass made with only polyamide both in the warp and in the weft.Suitable for particular applications as Pipex®.

Siderflex® ID
conveyor belts with regular warp steel cord according to DIN22131 and weft steel cord to increase cut and impact resistance. It can be joined with standard steel cord.

Siderflex® IW
conveyor belts with high elongation warp steel cord and weft steel cord to increase cut and impact resistance. The best compromise between ruggedness, lightness and low elongation.

Siderflex® HE
conveyor belts with high elongation warp steel cord and two different weft steel cord to achieve very high cut and impact resistance. Suitable for heavy use.

Siderflex® ST
conveyor belts with high elongation warp steel cord and polyamide (nylon) weft cords to have good cut resistance and low weight.

Pipex®
conveyor belts for tubular conveyor systems.

Spinatex®
conveyor belts suitable for inclined conveyor made with rubber chevrons vulcanized on the belt.

Flexobord®
conveyor belts with rubber cleat and edges, suitable for the conveyance of big quantity of material with very high inclinations and reduced dimensions.

Eletex®
conveyor belts with suitable textile carcass for elevator systems with steel or nylon buckets.

Elemet®
conveyor belts with suitable steel cord carcass for elevator systems.

Texrigid®
conveyor belts with high transversal stiffness, suitable for Flexobord® construction.

Crossrigid®
conveyor belts with textile carcass and two steel breaker used to achieve extremely high characteristics of transversal stiffness. Suitable to cover conveyor belts with moving discharge.

Rugotex®
conveyor belts with rough surface, suitable for the conveyance of material in vrac with high inclinations.

Separ®
rubber belts with vulcanized rubber cleats for magnetic separators.

Sicol®
solution for cold splicing and reparations.

Sicot®
rubber solution for hot splicing and reparations.

Sicop®
unvulcanized rubber sheets for hot splicing and reparations.

Technical data sheet for conveyor belt calculation

Customer: .... Date: ....
Required belt: ....

MATERIAL CHARACTERISTICS

Material .... Temperature Surcharge angle: .... °
Density: .... Ton/m³ Average: .... °C Abrasiveness
Lump size: .... mm Max: ....°C Low ❍ Medium ❍ High ❍

CONVEYOR DATA

Center distance: .... m Design capacity .... Ton/h Speed: .... m/sec
Width: .... mm Average capacity .... Ton/h Elevation: .... m
Radius of curve (if present): .... *If more than one, please enclose quoted drawing.*
Max tension (T_1): .... KN/m Min tension (T_2): .... KN/m

DRIVE UNIT

Position of drive pulley(s) Head ❍ Tail ❍ Return side ❍ Total wrap: .... °
Drive pulley surface Steel ❍ Rubber ❍ Ceramic ❍
Applied power: .... KW Start device: ....

IDLERS

	Inclination	Pitch	Diameter
Carrying side	.... °	.... mm	.... mm
Return side	.... °	.... mm	.... mm

Sliding plane ....

PULLEY DIAMETER AND TRANSITION DISTANCE

Drive pulley	Head pulley	Tail pulley	Counterweight pulley	Tripper pulley
.... mm	.... mm	.... mm	.... mm	.... mm

Transition distance at head .... mm at tail .... mm

TAKE-UP

Screw ❍ Teke-up travel: .... m
Couterweight ❍ Applied counterweight: .... Kg
Winch ❍ Position: ....

SPLICING

Vulcanized ❍ Mechanical fasteners .... ❍
Type: ....

DISCHARGE

Tail ❍ Lateral ❍ Tripper ❍ Tripper elevation: .... m

PREVIOUS BELT

Type	Tensile strength	N° of piles	Cover thickness	Quality	Width
....	.... KN/m	....	.... +.......... mm	....	.... mm

Producer: .... Life: ....
Cause of failure: ....

General index

(*) If You need frequent calculations, please do not exitate to ask our program Calnas. It permit working out the tensile strength of standard conveyor belts, elevator belts, feeder belts and also the capacity of Flexobord® belts. For its use it is only necessary to have the program Excel 6.0 in Your PC and this manual in Your hands.

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